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The closure of thin wall pressure vessels

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THE CLOSURE OF THIN WALL PRESSURE VESSELS

BY

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A

THESIS

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Approved by Aaron J. Miles
Professor of Mechanical Engineering

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I wish to also thank Professor R. F. Davidson for the use of the mechanics laboratory.

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INTRODUCTION

The object of this thesis is to find various methods of closing thin wall aluminum cylinders. Such a closure must be strong enough, that when internal pressure is applied, ultimate failure will occur along a longitudinal element of the cylinder and not at the closure. Moreover, such a method must not only be mechanically sound, but it must be readily adaptable to mass production.

The materials, which were used in the resulting investigation, are 24S-T3 aluminum tubing and 61S-T6 aluminum tubing. 24S-T3 tubing has an ultimate tensile strength of 64,000 psi and a yield point of 42,000.[1] This alloy is a hard, brittle material with a per cent elongation of from 10 to 16 per cent, depending upon the thickness.[1] 61S-T6 aluminum

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(1) Aluminum Company of America, ALCOA Structural Handbook, 1950, p. 168.

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is a softer alloy with an ultimate tensile strength of 42,000 psi and a yield point of 35,000 psi.[2]

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(2) Aluminum Company of America, ALCOA Structural Handbook, 1950, p. 169.

Most of the tests are conducted on the 24S-T3 tubing as it is the stronger of the two.

An important characteristic which must be considered when working with aluminum alloys is the minimum bend radius. With most metals, the thickness of relatively thin plates has little to do with the minimum bending radius, but such is not the case with aluminum alloys. Each temper of an aluminum alloy establishes a specific workability characteristic that must be considered when bending it. Table I, below, lists the minimum bending radii for the two alloys under consideration.

TABLE I.

MINIMUM BENDING RADIUS			
Material	Plate Thickness (t)	Minimum Bend Radii	
24S-T3	0.032 in.	2t-4t (3)	1/8 in. (4)
	0.065 in.	3t-5t	1/4 in.
	0.125 in.	4t-6t	1/2 in.
61S-T6	0.032 in.	0-1t	1/16 in.
	0.065 in.	0.5t-1.5t	1/8 in.
	0.125 in.	1t-2t	1/4 in.

(3) Reynolds Metals Company, Aluminum Structural Design, 1951, p. 81.

(4) Reynolds Metals Company, Aluminum Forming, 1952, pp. 36-37.

The next thing to be considered is the analysis of tube failure under internal pressure. The ultimate force any material can withstand is given by the formula:

$$F = SA \qquad 1.$$

where F is the force in pounds, S is the ultimate resisting stress of the material in psi, and A is the resisting area of the material in square inches. The force exerted by an internal pressure is equal to

$$F = PA_p \qquad 2.$$

where F is the force in pounds, P the internal pressure in psi, and A_p is the projected area in square inches upon which the pressure is applied.

To obtain a formula for the maximum internal pressure that a tube can withstand before failing along a longitudinal seam, the following analysis holds. The resisting area of the tube is equal to 2tL, where t is the wall thickness in inches and L is the tube length in inches. The ultimate resisting stress of the tube material is equal to the ultimate tensile strength, S_t. Substituting in equation 1, the total longitudinal holding force of the tube is obtained.

$$F = 2tLS_t \qquad 3.$$

The projected area upon which the internal pressure acts, is D_i L, where D_i is the internal diameter of the tube in inches. Substituting in equation 2, the longitudinal force exerted by an internal pressure is obtained.

$$F = PD_i L \qquad 4.$$

Equating equations 3 and 4, the maximum internal pressure that a tube can withstand before failing longitudinally is obtained.

$$P_1 D_i L = 2tS_t L$$

$$\text{or} \quad P_1 = \frac{2tS_t}{D_i} \qquad 5.$$

To obtain a formula for the maximum internal pressure that a tube can withstand before failing transversally, a similar analysis is made. In this case, the resisting area is equal to $(D_o^2 - D_i^2)\,\pi/4$, where D_o is the outside diameter of the cylinder in inches. The projected area upon which the pressure is exerted is given by $D_i^2\,\pi/4$. Substituting in equation 1, we have

$$F = (D_o^2 - D_i^2)\,\frac{\pi S_t}{4} \qquad 6.$$

and substituting in equation 2, we have

$$F = \frac{PD_i^2\,\pi}{4} \qquad 7.$$

Equating these two forces, the maximum internal pressure that a tube can hold before failing transversally is obtained.

$$\frac{PD_i^2\,\pi}{4} = \frac{S_t\,\pi}{4}\,(D_o{}^2 - D_i{}^2)$$

$$\text{or} \quad P_t = S_t\,(D_o^2 - D_i^2)/D_i^2 \qquad 8.$$

A close approximation for the resisting transversal area can be obtained from $\pi D_i t$. Substituting this in equation 1, equation 8 now becomes

$$P_t D_i^2\,\pi/4 = S_t D_i\,\pi t$$

$$\text{or} \quad P_t = 4S_t t/D_i \qquad 9.$$

Dividing equation 9 by equation 5, we obtain a relationship between transversal bursting pressure and longitudinal bursting pressure.

$$\frac{P_t}{P_l} = \frac{4\ S_t t D_i}{2\ S_t t D_i} = 2$$

Thus it is seen that any tube is approximatly twice as strong transversally as it is longitudinally under internal pressure. Therefore, any closure need be only half as strong as the cylinder.

Table 2, on the next page, gives the longitudinal and transversal bursting pressure and the maximum force on a plug for all the different tubes used in the investigation.

TABLE II.

1	1	$1\frac{1}{2}$	2	2	Tube Diameter in in.
0.065	0.035	0.035	0.065	0.083	Wall Thickness in in.
9,570	4,820	3,140	4,450	3,800	Longitudinal Bursting Pressure in psi
20,600	10,150	6,600	9,360	8,000	Transversal Bursting Pressure in psi
5,690	3,275	5,040	12,220	10,030	Force on Plug in Pounds
24S-T3				61S-T6	Alloy

TABLE OF SYMBOLS

SYMBOL	UNITS	SIGNIFIGANCE
F	lbs	force
S	lbs in^{-2}	stress
A	in^{2}	area
t	in	wall thickness of the tube
L	in	tube length
S_t	lbs in^{-2}	ultimate tensile stress of the tube material
P	lbs in^{-2}	internal pressure
D_i	in	internal tube diameter
D_o	in	external tube diameter
P_t	lbs in^{-2}	internal pressure required to burst the tube transversally
P_l	lbs in^{-2}	internal pressure required to burst the tube longitudinally
A_p	in^{2}	projected area of tube

REVIEW OF LITERATURE

The problem of closing thin wall cylinders to withstand internal pressure is rather unique, and consequently, very little literature could be found relating directly to it. However, many books and references were consulted during the different stages of its development.

The strength and characteristics of the alloys used in this investigation were found in the publications from the Aluminum Company of America[1] and

(1) Aluminum Company of America, ALCOA Structural Handbook, 1950.

the Reynolds Metals Company.[2]

(2) Reynolds Metals Company, Aluminum Data Book, 1950.
Reynolds Metals Company, Aluminum Structural Design, 1952.
Reynolds Metals Company, Aluminum Forming, 1951.

Material on the holding power and installation of "O" rings was obtained from an "O" ring catalog.[3]

(3) Goshen Rubber Company, Catalog C.

Information on the stresses caused by internal pressure can be found in almost any text on the mechanics of materials.[4]

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(4) Laurson, Cox, Mechanics of Materials, Tenth printing, pp. 57-62.

Several books were consulted while threading was being considered as a possible solution to the problem. One of the most helpful, was the text currently being used in the machine design course.[5]

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(5) Faires, Design of Machine Elements, Fifteenth printing, pp. 62-88.

One other book that proved most helpful throughout this investigation was Kent's Mechanical Engineer's Handbook.[6]

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(6) Kent's Mechanical Engineer's Handbook, Design and Production Volume, Twelfth Edition.

DISCUSSION OF PROBLEM

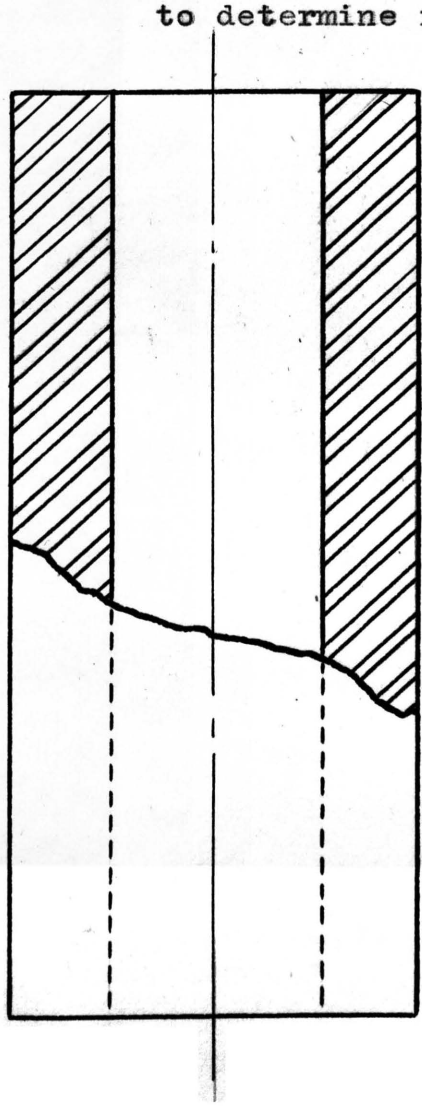
During the investigation to find various ways to seal thin wall cylinders to withstand internal pressure, a number of different solutions were suggested and tried. Among these were welding, crimping, upsetting, and threading.

The use of welding as a solution to the problem is discarded for several reasons. First, the welding process, itself, changes the metallurgical properties of the material adjacent to the weld in such a manner as to severely weaken the tube. Second, it is desirable to develope a method of closure that does not utilize heat in its formation as the substance to be sealed is highly inflammable. Finally, the tube material used in this investigation is not readily adaptable to welding.

Crimping, as a solution, had previously been considered. Messers Bersu and Cansever explored the holding power of an outside tube crimp. While they did not reach any conslusive results as to whether this could be used as a solution, it was decided to concentrate on other possible solutions to the problem. This does not mean, however, that an outside tube crimp will not work.

The first thing to be investigated, is the possibility of cold upsetting the end of a tube. Cold upsetting is the application of pressure on a material, utilizing no heat; such pressure causing the material to flow into the form of its surroundings.

Upon checking numerous references, nothing could be found on the upsetting of aluminum alloys and its effect on the properties of the material. It was then decided to run a test on the tubing to determine if it was possible to cold upset an



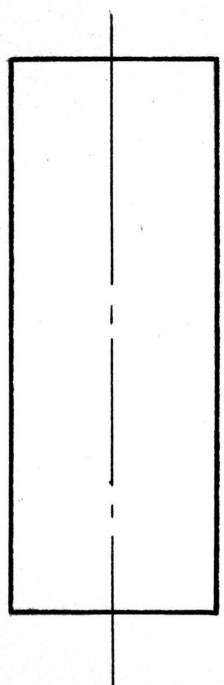


Figure 1a. Male and female upset dies.

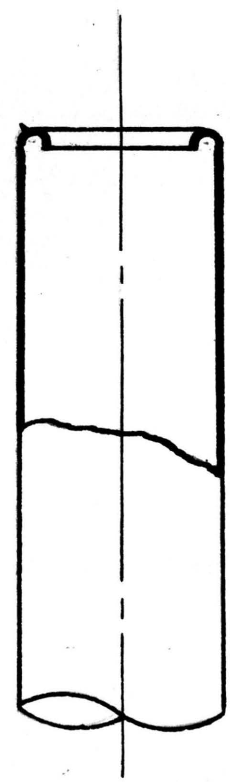


Figure 1b. Tube with rolled end after upsetting

aluminum tube. A simple set of dies were constructed similar to those shown in figure 1a. The two dies merely consisted of a female die to keep the tube from spreading and a male die upon which the force was applied. A one inch tube, 0.035 inches thick, was then placed in the female die and force was applied on the male die. Instead of upsetting, the tube had a tendency to roll over from outside to inside, forming a rounded end as is shown in figure 1b. Before this rolling process occurred, it was found that a portion of the end of the tube, shaped as a wedged ring, sheared from the outer edge. This shear ring then acted as a forming die in the development of the rolled end. It took considerable force to form this shearing ring, but once formed, the tube rolling force was much less. The amount of roll on the end of the tube was controlled by the final force on the male die.

After studying this rolling condition, it was decided to design a new die which would tend to facilitate this rolling action. This die was so designed that its radius was larger than the minimum bend radius of table 1. A diagram of the new die is shown in figure 2. The new die functioned very well and the forming force was greatly decreased as the sharp edges of the die practically eliminated the shearing ring. It was found that if the dies were well lubricated, the quality of the end crimp was greatly increased.

A plug was then made and tests were conducted in a tensile testing machine to determine whether the end crimp was strong enough to withstand the ultimate internal pressure needed to fracture the tube. A second male die was constructed and placed in the bottom of the female die so that both end crimps could be formed simultaneously. Tests conducted on three inch specimens of one inch tubing, 0.035 inches thick, showed that the force required to pull this end crimp out, ran anywhere from eighty to one hundred per cent of the calculated value listed in table 2.

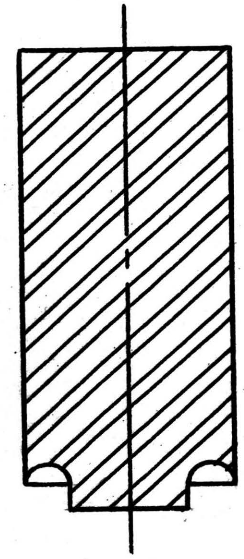


Figure 2. Upset die to facilitate the rolling action.

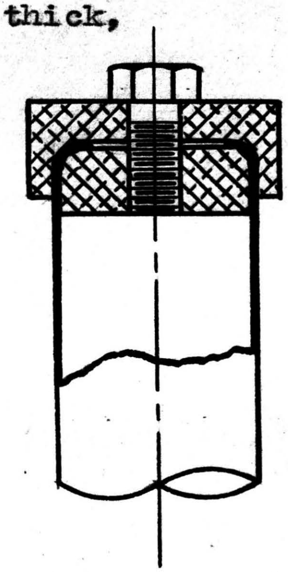


Figure 3. Rolled end crimp with outside cap.

It was then decided that the end crimp could probably be strengthened by using an outside cap like that shown in figure 3. Tests, however, showed that the use of a cap merely shifted the stress concentration in such a way as to cause the brittle, work hardened end to fail transversally. It was found that the use of a cap actually slightly decreased the strength of the crimp.

The next thing that was done in an effort to increase the strength of the rolled end crimp was to redesign the plug. The use of the flat plug seemed inadvisable because of its prying action on the inner lip of the roll. A new plug with a raised groove on the outside of its face was designed to fit the inside of the rolled surface. This plug did help the holding power of the crimp and a diagram of it is shown in figure 4.

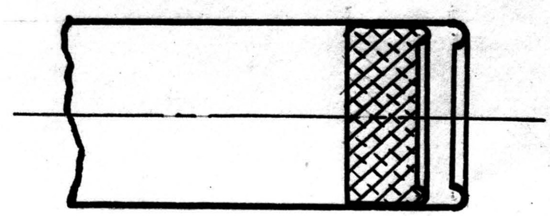
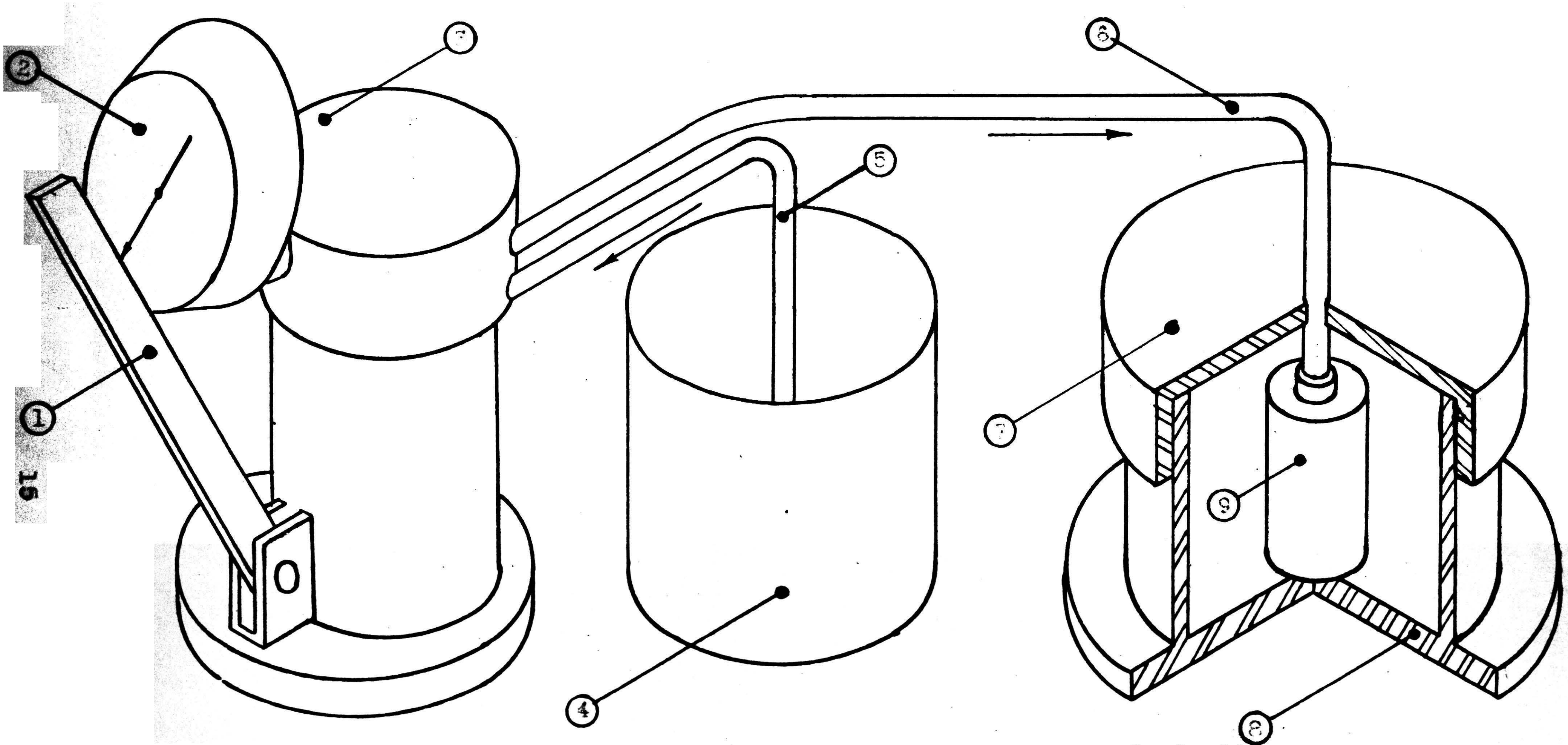


Figure 4. Improved plug with raised edge.

Exhaustive tests were then run on the testing machine with the results tabulated in table 3 and shown graphically in plates 1 and 2. The data was the average of several trials and was not too consistent. This data was obtained by forming each crimp at a constant rate. This made the holding power of the crimp not only a function of the total deflection, but also a function of the final forming force. This relationship was shown graphically in plate 2.

After the tests had been completed on the



1. Pump Handle
2. Pressure Gage
3. Pump

4. Oil Reservoir
5. Suction Line
6. Delivery Line

7. Basin Lid
8. Pressure Release Basin
9. Tube Specimen

Figure 5. Pressure System.

tensile testing machine, it was decided to test the crimp under internal pressure. The plugs were modified slightly to hold "O" rings which were used to make the cylinder pressure tight. A pressure applying system, similar to that shown in figure 5, was then constructed. The pressure was applied by a hydraulic hand pump using kerosene as the hydraulic fluid. This system attained pressure in excess of 9,000 psi.

Tests ran on the rolled end crimp showed that about seventy per cent of the one inch tubes held satisfactorily. On the two inch tubes, however, this percentage dropped to about thirty per cent and results were very erratic.

A diagram of the rolled end crimp and a table of its dimensions are given in figure 6 and table 4.

It was then decided that possibly the radius of the rolled end crimp was too small. To correct this, a pair of hemispherical dies like those shown in figure 7 were machined. It was thought that with these dies, the tube would seek its own rolling radius. The first dies

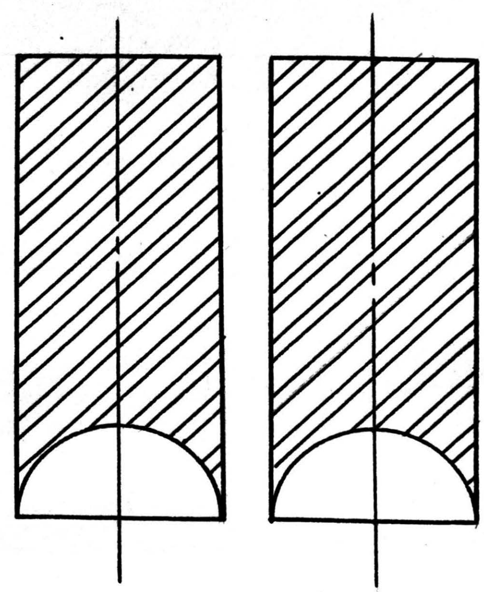


Figure 7. Set of hemispherical dies.

were made of hardened tool steel but proved unsatisfactory as they cracked and broke under pressure. The dies were then machined from a tough, nickle steel and these dies proved very satisfactory. When used, however, the tube did not seek its own rolling radius, but rather, formed a hemispherical end crimp.

A set of hemispherical plugs were then machined and tests were again run on the testing machine. These plugs proved satisfactory on thin wall, one inch tubing, but did not hold in other tubing. In the thick walled tubing, the hemispherical plug acted as a wedge which split the end of the tube longitudinally.

A new plug was then designed, having a fairly short radius on the outer rim. This plug proved very satisfactory with ultimate failing forces ranging up to 170 per cent of the calculated forces listed in table 2.

This plug had a tendency to move outward as the force is applied. This movement is not great, ranging from approximately one sixteenth of an inch in the small tubes, to one eighth of an inch in the larger tubes.

The results of the tests conducted on the hemispherical end crimp are tabulated in tables

5, 6, and 7. This crimp was once again formed at a constant rate making the holding power of the crimp a function of the final, total, forming force. The results of these tests are plotted graphically in plates 1, 2, 3, 4, 5, and 6. Once again, these graphs are averages of many runs made on the testing machine. The results with the hemispherical end crimp, however, are not nearly as erratic as that obtained on the rolled end crimp.

Some difficulty was encountered in keeping the two inch hemispherical die sharp. It was desirable to have the die as sharp as possible so that no shearing ring was obtained. A possible solution to this problem might be to make a new set of dies patterned after that shown in figure 8. This die is a combination of both the male and female dies.

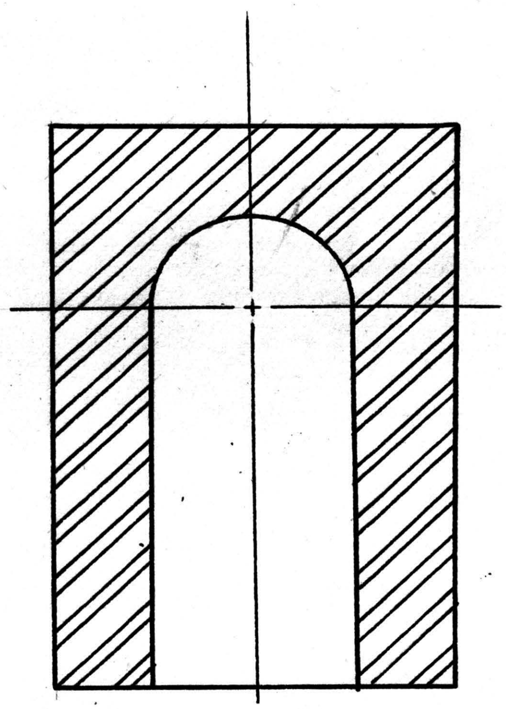


Figure 8. Proposed die for the hemispherical end crimp.

Once again, it was found that the use of a lubricant greatly increased the quality of the crimp and lowered the total forming force.

After the tests had been completed on the testing machine, the plugs were fitted with "O" rings and tests were conducted, using internal pressure to blow the tubes. The results under pressure were very gratifying with one hundred per cent failure along a longitudinal seam. Some of these failures using the hemispherical end crimp are shown in figure 9 in the appendix.

A diagram of the hemispherical end crimp and a table of its dimensions are shown in figure 10 and table 8.

Once tests had been completed on the end crimp closures, it was decided to investigate threading as a possible solution to the problem. It was proved in the introduction, that a tube is more than twice as strong transversally as it is longitudinally under internal pressure. Therefore, any thread can be cut half way through the tube wall and still not weaken the tube with respect to internal pressure. The difficulty of machining thin wall tubes, however, all but eliminated threading as a solution.

The first type of thread to be used was a special V shaped thread on the outside of the tube. Such a thread had to have twenty-six threads to the inch and machining a close fit was all but impossible.

To combat this problem of close fit, a tapered V thread was used. Such a thread was developed and

and a successful failure was obtained. To obtain sufficient strength, a half an inch of threads had to be in contact. Because of the difficulty of assembly and because of the machining problem, it was decided to abandon this method as a solution.

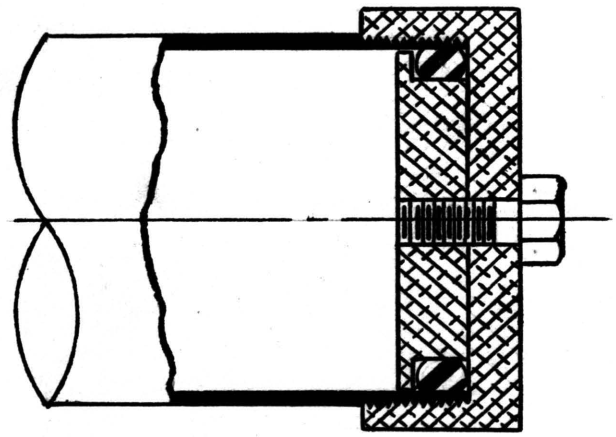


Figure 11. Outside V-Thread with cap.

A diagram of the method described above is shown in figure 11.

Another type of thread was then developed that proved satisfactory on all counts. This thread was a special tapered buttress thread having eight threads to the inch. Due to the machining problem, this thread was tried only on the two inch tubes, 0.065 inches thick.

A drawing of this plug with its dimensions are given in figure 12 and table 9. Photographs of the failure of this tube are shown in figure 13 in the appendix.

Another method of closure was proposed and developed by Mr. Norbert Newman. This method, like the threading method, was based on the fact that a tube, under internal pressure, is twice as strong transversally as it is longitudinally. Utilizing this fact, he suggested a partial cut method in which a number of rectangular flaps were cut in the tube itself and bent into the groove of a plug. These flaps then kept the plug from pulling out. The width and number of these flaps were so controlled that the sum of their width did not exceed one-half the circumference of the tube.

After running exhaustive tests on the tensile testing machine, during which he varied number, width, and angle of the flaps, a tube was developed that would hold under internal pressure.

A drawing of this plug with its dimensions are given in figure 14 and table 10. Photographs of the failure using this closure are shown in figure 15 in the appendix.

Somewhat along the same line, other methods were developed using pins and rivets to hold the plug. While these methods were mechanically sound, the assembly problem was such that it all but eliminated these methods as possible solutions.

TABLE III.

Tension tests on one inch, 24S-T3 tubing, 0.035 inches thick

ROLLED END CRIMP

Trial	Original Length in inches	Final Length in inches	Total Deformation in inches	Total Forming Force in pounds	Holding Force in pounds
1.	2.961	2.842	0.119	3,000	2,500
2.	3.037	2.913	0.124	3,500	2,670
3.	3.008	2.874	0.134	4,000	2,880
4.	2.964	2.823	0.141	4,500	3,045
5.	2.951	2.799	0.152	5,000	3,160
6.	3.014	2.844	0.170	5,500	3,325
7.	3.010	2.817	0.193	6,000	3,485
8.	3.027	2.807	0.220	6,500	3,520
9.	3.031	2.786	0.245	7,000	3,565

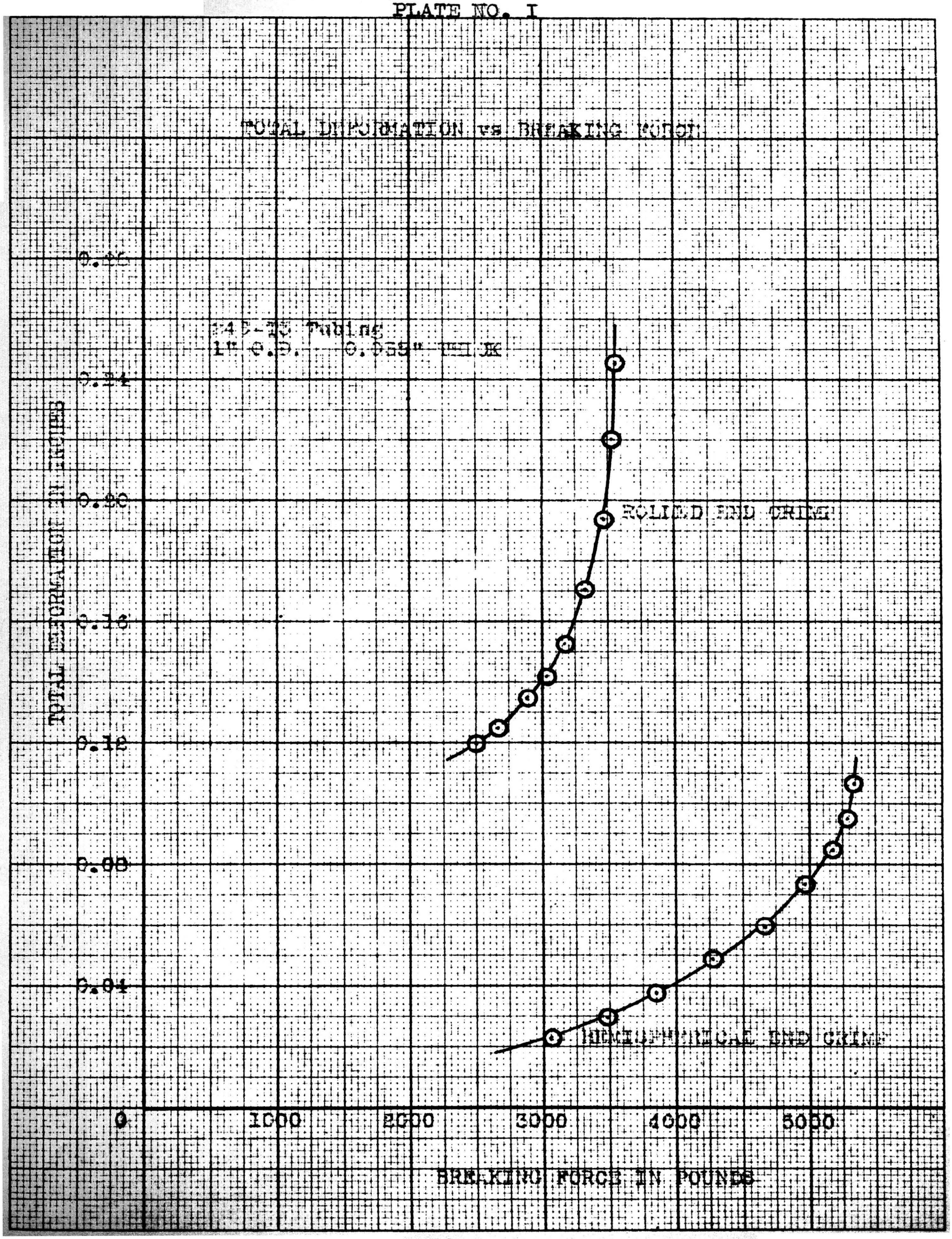
TABLE V.

Tension tests on one inch, 24S-T3 tubing, 0.035 inches thick

HEMISPHERICAL END CRIMP

Trial	Original Length in inches	Final Length in inches	Total Deformation in inches	Total Forming Force in pounds	Holding Force in pounds
1.	3.002	2.980	0.022	3,000	3,070
2.	3.003	2.974	0.029	3,500	3,485
3.	3.007	2.970	0.037	4,000	3,840
4.	3.010	2.962	0.048	4,500	4,280
5.	3.008	2.949	0.059	5,000	4,670
6.	3.007	2.934	0.073	5,500	4,960
7.	3.021	2.937	0.084	6,000	5,180
8.	3.030	2.936	0.094	6,500	5,290
9.	2.994	2.888	0.106	7,000	5,340

PLATE NO. I



Tubing
0.20
0.12
0.08
0.04
0
1000
2000
3000
4000
5000
BREAKING FORCE IN POUNDS

PLATE NO. II

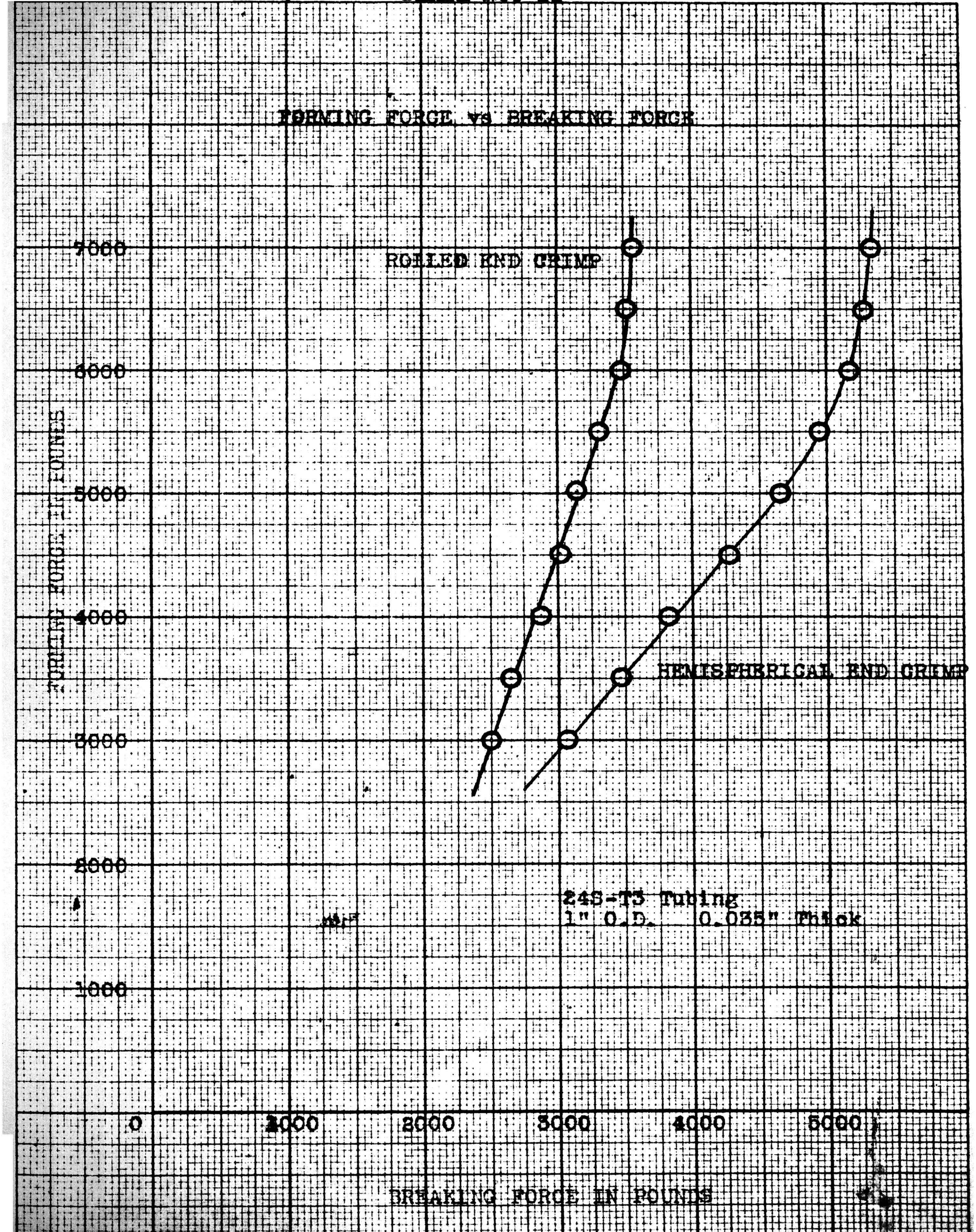

FORMING FORCE vs BREAKING FORCE
ROLLED END CRIMP
HEMISPHERICAL END CRIMP
24S-T3 Tubing
1" O.D. 0.035" Thick
FORMING FORCE IN POUNDS
7000
6000
5000
4000
3000
2000
1000
0
1000
2000
3000
4000
5000
BREAKING FORCE IN POUNDS

TABLE VI.

Tension tests on one and one-half inch, 24S-T3 tubing, 0.035 inches thick

HEMISPHERICAL END CRIMP

Trial	Original Length in inches	Final Length in inches	Total Deformation in inches	Total Forming Force in pounds	Holding Force in pounds
1.	3.968	3.907	0.061	6,000	5,050
2.	3.966	3.881	0.085	7,000	6,060
3.	3.976	3.870	0.106	8,000	7,060
4.	3.896	3.776	0.120	9,000	7,600
5.	4.018	3.874	0.144	10,000	7,880
6.	3.972	3.814	0.158	11,000	7,980
7.	4.039	3.841	0.198	12,000	8,080

PLATE NO. III

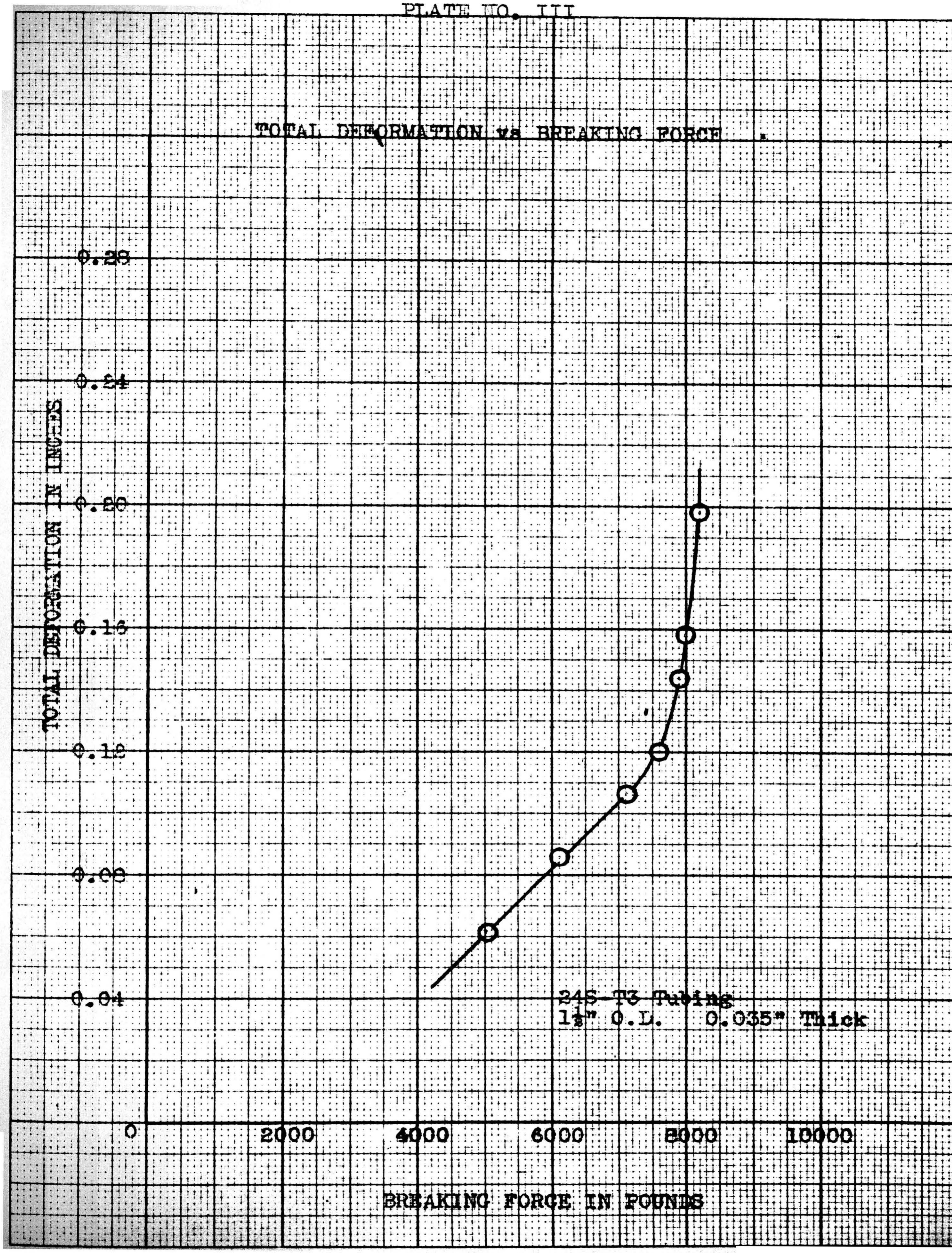

TOTAL DEFORMATION vs BREAKING FORCE
TOTAL DEFORMATION IN INCHES
0.28
0.24
0.20
0.16
0.12
0.08
0.04
0
2000
4000
6000
8000
10000
BREAKING FORCE IN POUNDS
24S-T3 Tubing
1⅛" O.D. 0.035" Thick

PLATE NO. IV

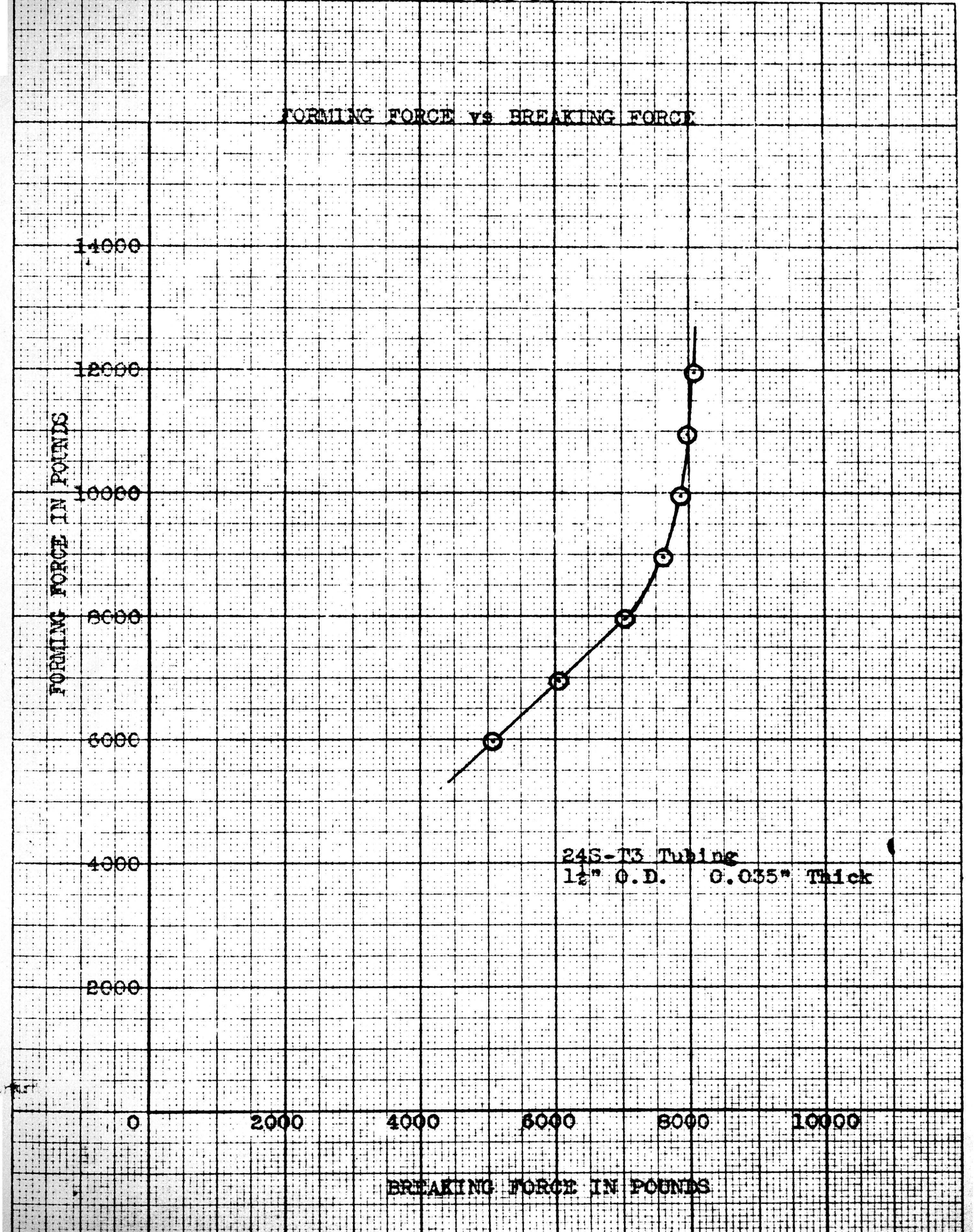

FORMING FORCE vs BREAKING FORCE
FORMING FORCE IN POUNDS
14000
12000
10000
8000
6000
4000
2000
0
2000
4000
6000
8000
10000
BREAKING FORCE IN POUNDS
24S-T3 Tubing
1½" O.D. 0.035" Thick

TABLE VII.

Tension tests on two inch, 24S-T3 tubing, 0.065 inches thick

HEMISPHERICAL END CRIMP

Trial	Original Length in inches	Final Length in inches	Total Deformation in inches	Total Forming Force in pounds	Holding Force in pounds
1.	4.078	4.011	0.067	12,000	7,060
2.	4.048	3.946	0.102	14,000	9,300
3.	4.028	3.908	0.120	16,000	10,940
4.	4.098	3.964	0.134	18,000	12,500
5.	4.066	3.915	0.151	20,000	14,150
6.	4.082	3.911	0.171	22,000	16,130
7.	4.113	3.925	0.188	24,000	17,240
8.	4.082	3.865	0.217	26,000	18,250

PLATE NO. V

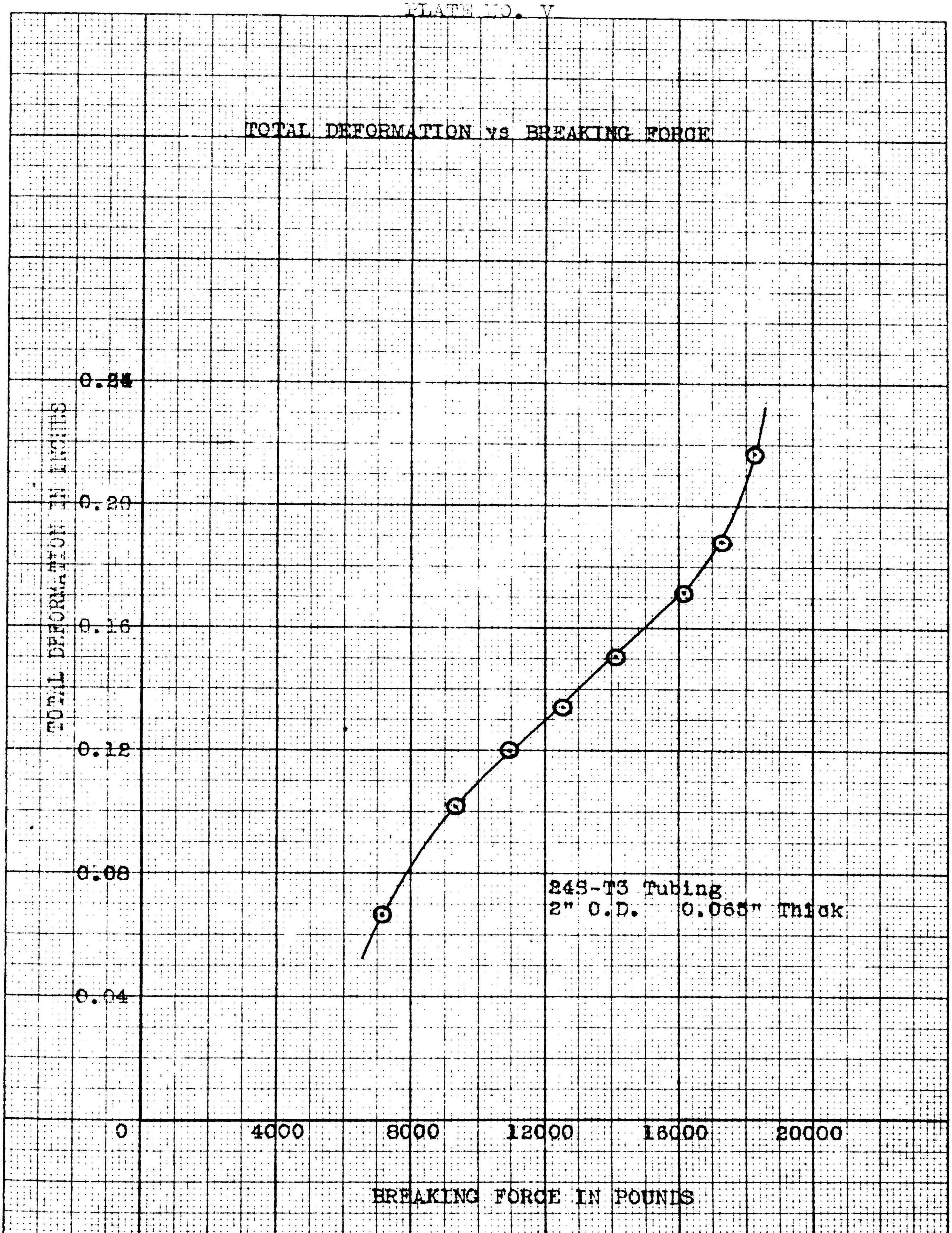
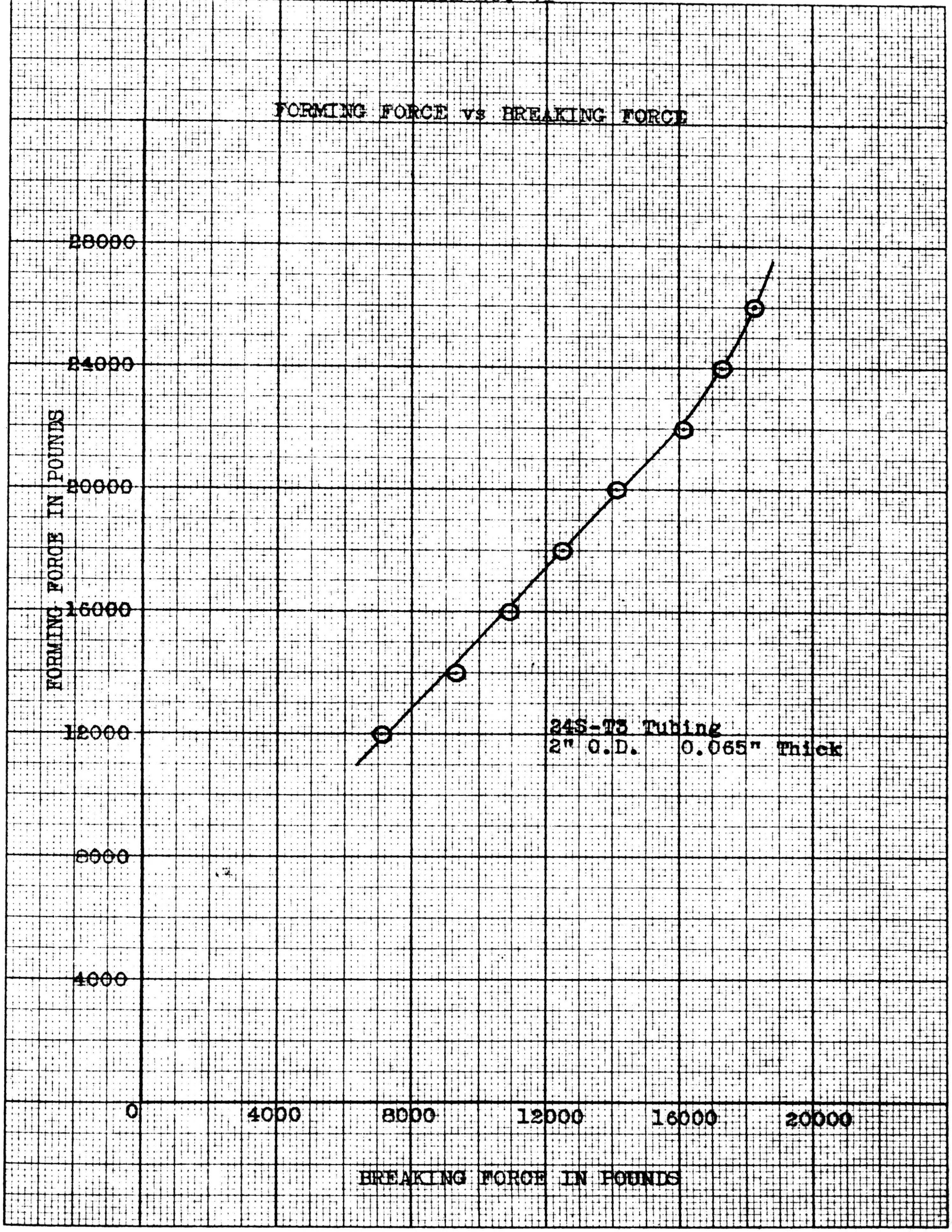

TOTAL DEFORMATION vs BREAKING FORCE
TOTAL DEFORMATION IN INCHES
0.24
0.20
0.16
0.12
0.08
0.04
0
4000
8000
12000
16000
20000
BREAKING FORCE IN POUNDS
24S-T3 Tubing
2" O.D. 0.065" Thick

PLATE NO. VI



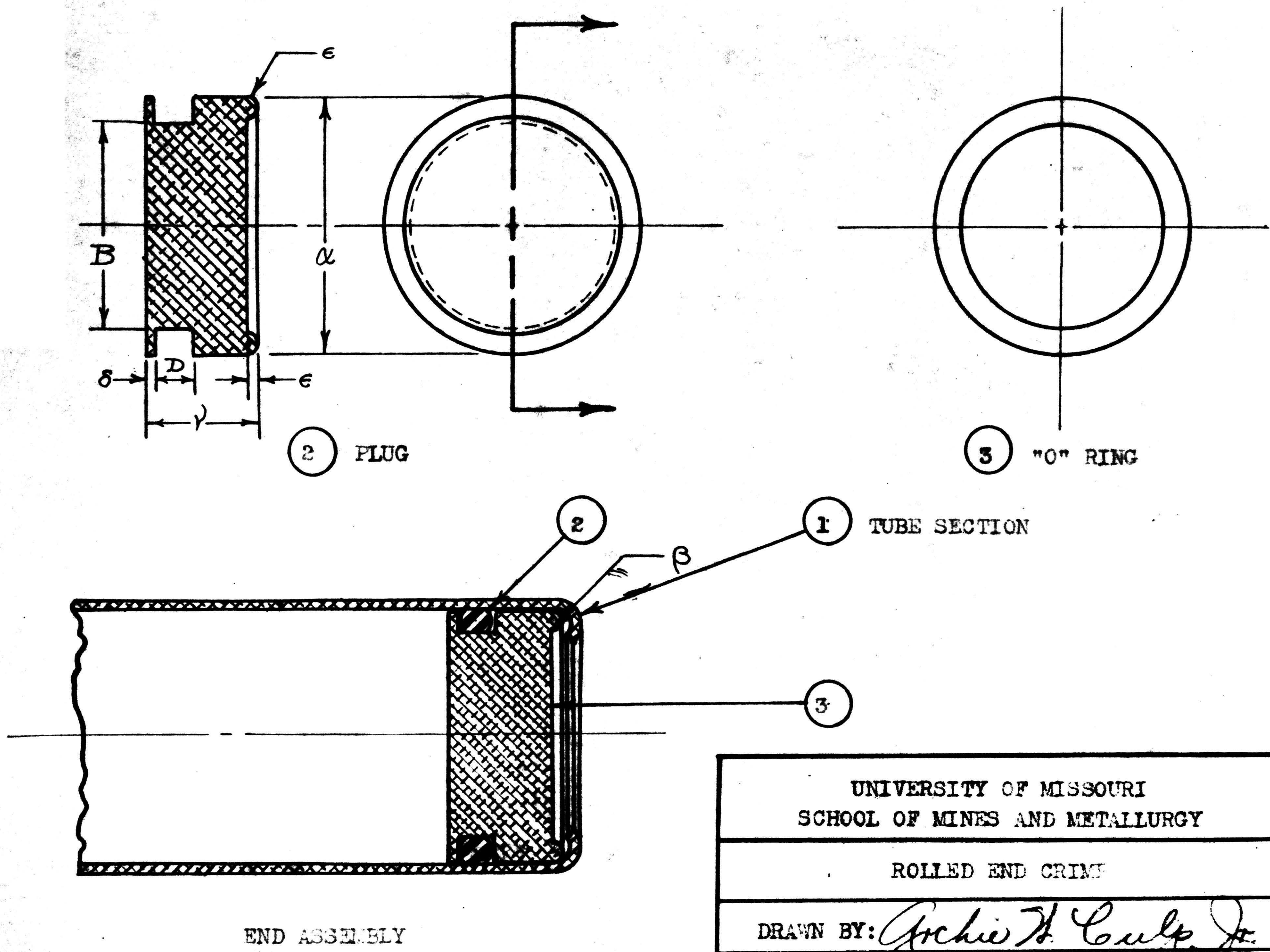
FORMING FORCE vs BREAKING FORCE
28000
24000
20000
16000
12000
8000
4000
0
4000
8000
12000
16000
20000
FORMING FORCE IN POUNDS
BREAKING FORCE IN POUNDS
24S-T3 Tubing
2" O.D. 0.065" Thick

TABLE IV.

DIMENSIONS FOR ROLLED END CRIMP

DIMENSION FROM DRAWING	24S-T3 1" O.D. 0.035" Thick	24S-T3 1" O.D. 0.065" Thick	24S-T3 $1\frac{1}{2}$" O.D. 0.035" Thick	24S-T3 2" O.D. 0.065" Thick	61S-T6 2" O.D. 0.083" Thick
α	0.926"	0.866"	1.426"	1.866"	1.830"
B[1]	0.748"	0.685"	1.184"	1.497"	1.460"
δ	0.0625"	0.0625"	0.0625"	0.0625"	0.0625"
D[1]	0.140"	0.140"	0.187"	0.281"	0.281"
ϵ	0.0313"	0.0313"	0.0469"	0.0625"	0.0625"
γ	0.4375"	0.4375"	0.625"	0.875"	1.125"
β	0.125"	0.125"	0.1562"	0.1875"	0.1875"
"O" RING SIZE	AN6227-14	AN6227-13	AN6227-22	AN6227-28	AN6227-28

(1) Dimensions "B" and "D" for all tubes except the 2" O.D. 61S-T6, are taken from Catalog C, Goshen Rubber Co., p. 9.



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ROLLED END CRIMP

DRAWN BY: Archie H. Culp, Jr.

DATE: 7/29/53 SCALE: NO. 3

Figure 6.

TABLE VIII.

DIMENSIONS FOR HEMISPHERICAL END CRIMP

IMENSION FROM DRAWING	24S-T3 1" O.D. 0.035" Thick	24S-T3 1" O.D. 0.065" Thick	24S-T3 $1\frac{1}{2}$" O.D. 0.035" Thick	24S-T3 2" O.D. 0.065" Thick	61S-T6 2" O.D. 0.083" Thick
α	0.926"	0.866"	1.426"	1.866"	1.830"
B^1	0.748"	0.685"	1.184"	1.497"	1.460"
δ	0.0625"	0.0625"	0.0625"	0.0625"	0.0625"
D^1	0.140"	0.140"	0.187"	0.281"	0.281"
γ	0.5625"	0.5625"	0.687"	0.875"	1.125"
ω	0.125"	0.125"	0.1875"	0.250"	0.250"
β	0.500"	0.500"	0.750"	1.000"	1.000"
θ	58°	58°	57°	56°	58°
O" RING SIZE	AN6227-14	AN6227-13	AN6227-22	AN6227-28	AN6227-28

(1) Dimensions "B" and "D" for all tubes except the 2" O.D. 61S-T6, are taken from Catalog C, Goshen Rubber Co., p.9.

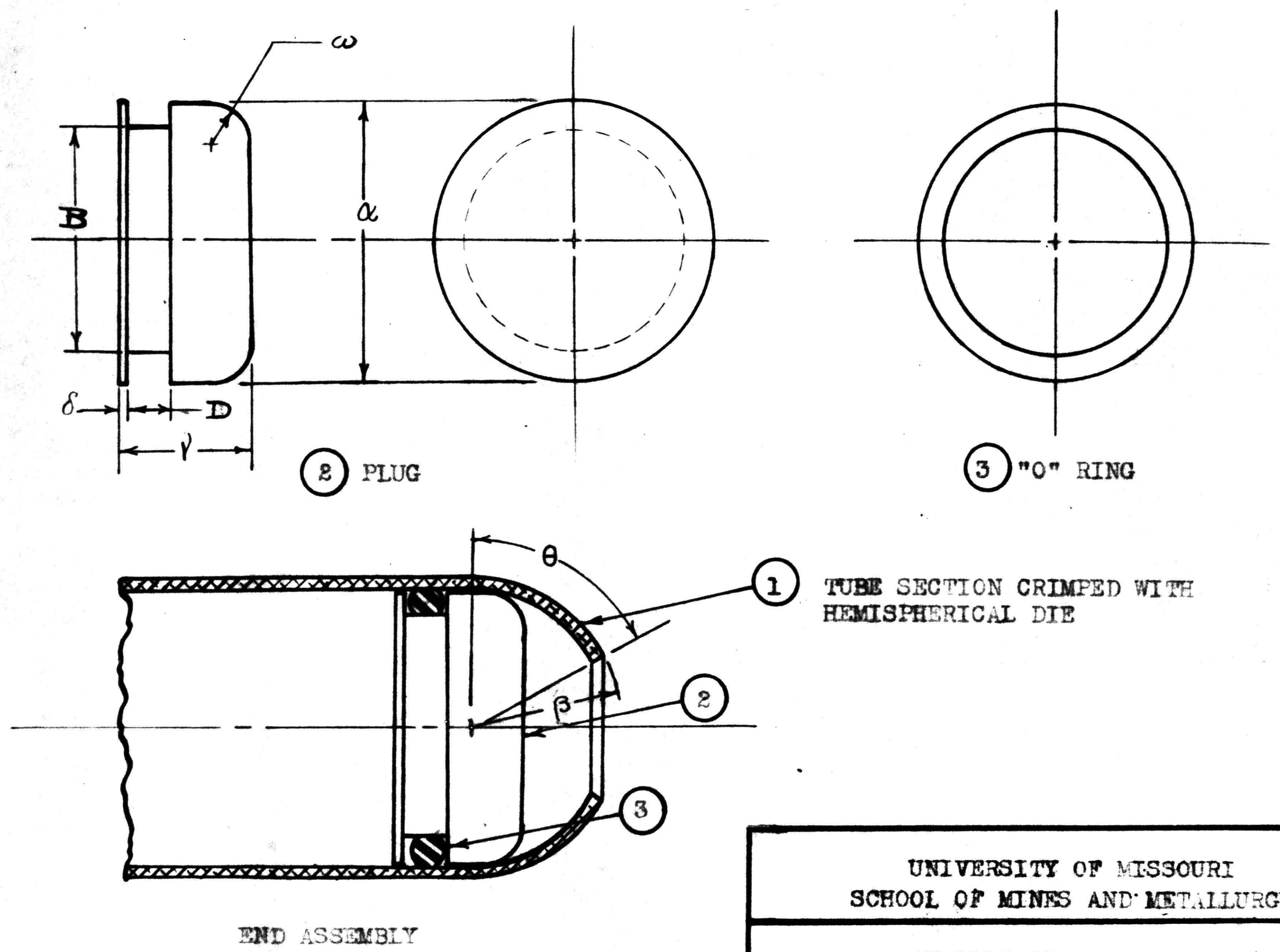

Figure 10.

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HEMISPHERICAL END CRIMP

DRAWN BY: Archie W. Culp Jr.

DATE: 7/28/53 | SCALE: | NO. 1

TABLE IX.

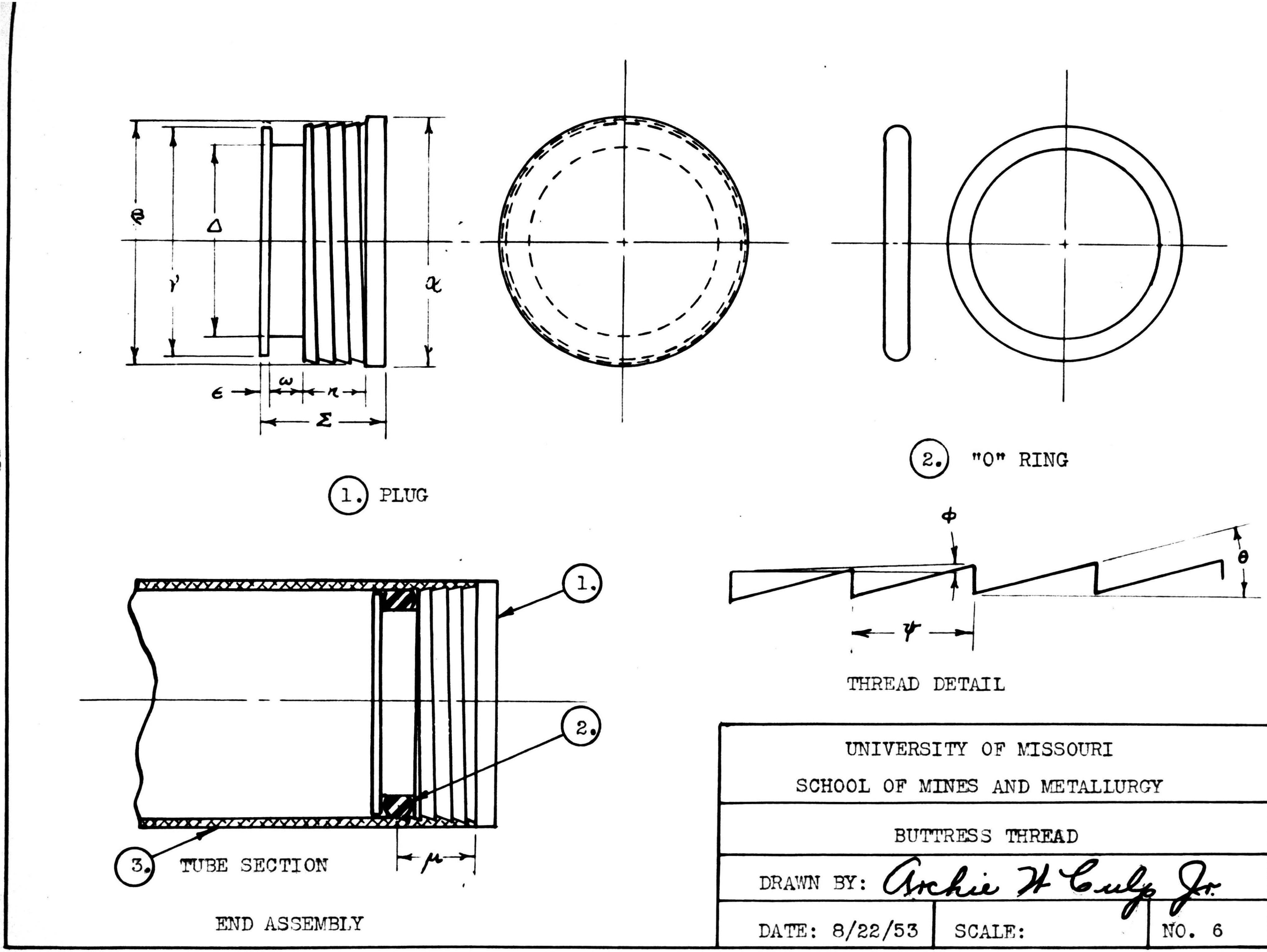
DIMENSIONS FOR THE TAPERED BUTTRESS THREAD [1]

α	2.000"
β	1.960"
γ	1.860"
$\Delta^{(2)}$	1.500"
Σ	1.000"
ϵ	0.0625"
$\omega^{(2)}$	0.281"
η	0.500"
μ	0.625"
ϕ	2°
θ	15°
ψ	0.125"

(1) These dimensions are for the 2" O.D., 0.065" tubing

(2) Dimensions and are taken from Catalog C, Goshen Rubber Co., p.9.

The above dimensions were arrived at, using the mechanical properties of 17S-T aluminum as the plug material.



1. PLUG
2. "O" RING
THREAD DETAIL
3. TUBE SECTION
END ASSEMBLY
UNIVERSITY OF MISSOURI
SCHOOL OF MINES AND METALLURGY
BUTTRESS THREAD
DRAWN BY: Archie W Culp Jr.
DATE: 8/22/53
SCALE:
NO. 6

Figure 12.

TABLE X.

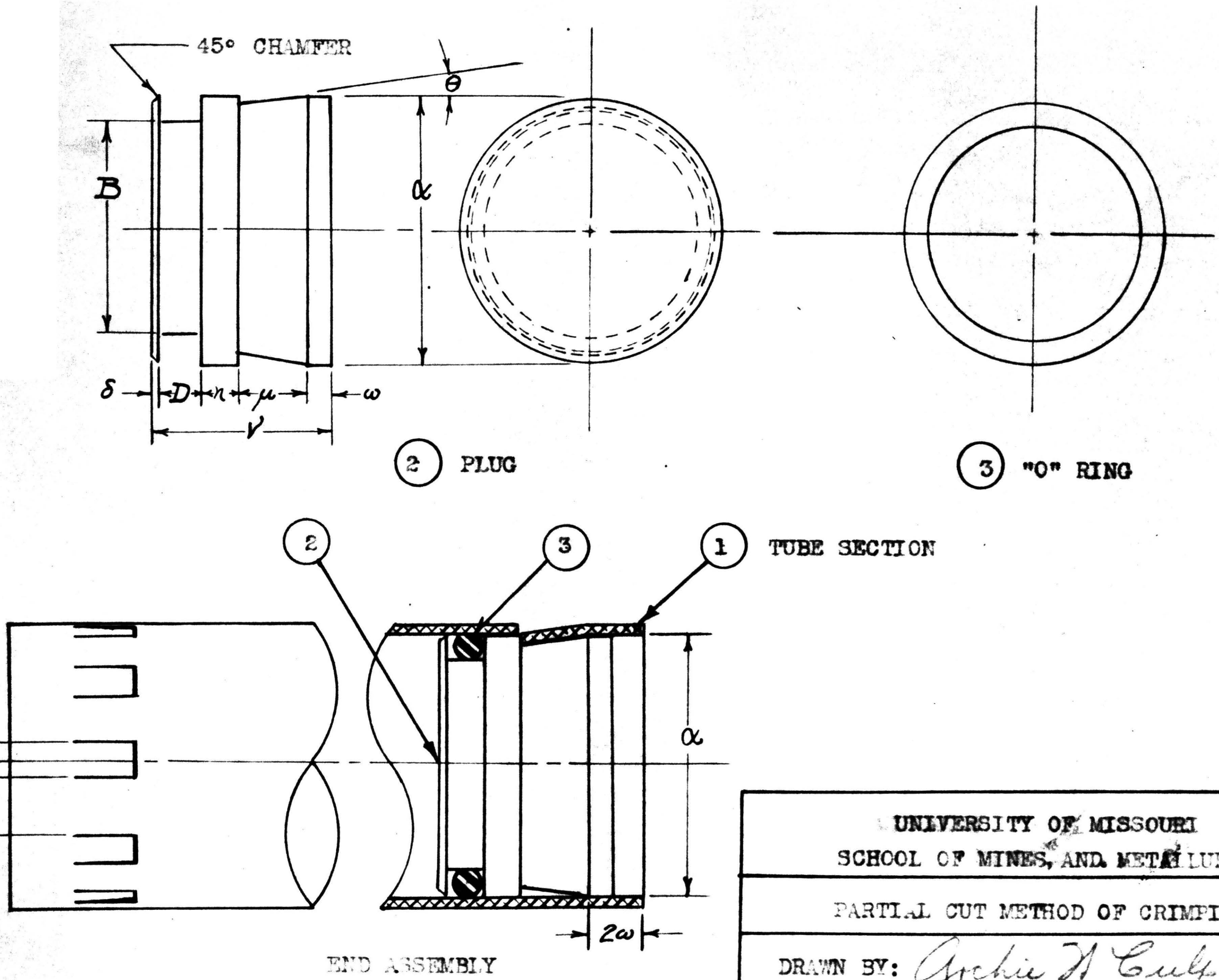
DIMENSIONS FOR THE PARTIAL CUT METHOD

DIMENSION FROM DRAWING	24S-T3 1" O.D. 0.035" Thick	24S-T3 1" O.D. 0.065" Thick	24S-T3 $1\frac{1}{2}$" O.D. 0.035" Thick	24S-T3 2" O.D. 0.065" Thick	61S-T6 2" O.D. 0.083" Thick
α	0.928"	0.868"	1.428"	1.868"	1.832"
B[1]	0.748"	0.685"	1.184"	1.497"	1.460"
δ	0.0625"	0.0625"	0.0625"	0.0625"	0.0625"
D[1]	0.140"	0.140"	0.187"	0.281"	0.281"
η	0.0625"	0.0781"	0.0937"	0.250"	0.125"
μ	0.375"	0.375"	0.375"	0.500"	0.500"
ω	0.0781"	0.0781"	0.0937"	0.1875"	0.125"
γ	0.7181"	0.7337"	0.8119"	1.281"	1.0935"
ϕ	0.125"	0.125"	0.1875"	0.250"	0.250"
ψ	0.1892"	0.1892"	0.2837"	0.3783"	0.3783"
θ	5.75°	10.5°	5.75°	7.5°	9.5°
"O" RING SIZE	AN6227-14	AN6227-13	AN6227-22	AN6227-28	AN6227-28

(1) Dimensions "B" and "D" for all tubes except the 2" O.D. 61S-T6 are taken from Catalog C, Goshen Rubber Co., p. 9.

The above dimensions were arrived at, using the mechanical properties of 17S-T aluminum as the plug material.

Figure 14.



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PARTIAL CUT METHOD OF CRIMPING

DRAWN BY: Archie W Culp Jr

DATE: 7/28/53 | SCALE: | NO. 2

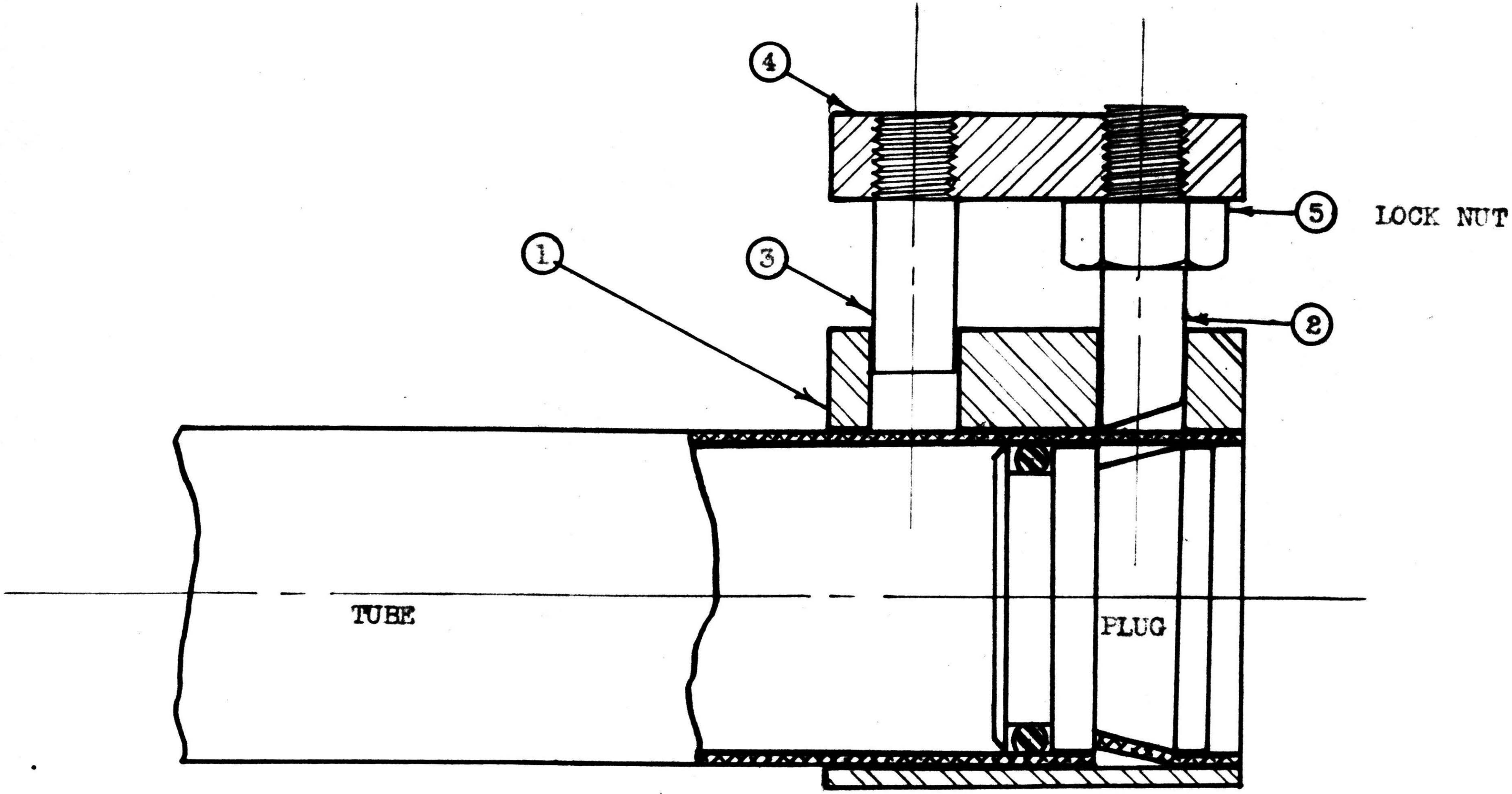

Figure 16.

UNIVERSITY OF MISSOURI SCHOOL OF MINES AND METALLURGY
ASSEMBLY VIEW OF DIE FOR PARTIAL CUT
DRAWN BY: Archie N. Culp Jr.
DATE: 8/5/53 · SCALE: · NO. 5

SUMMARY AND CONCLUSIONS

Before discussing the methods developed in this investigation, a few words should be said about "O" rings. The use of "O" rings was very successful in sealing the tubes for internal pressure. While it was recommended that "O" rings be carefully installed so as not to cut or injure the ring, little difficulty was encountered along this line. In fact, many of the "O" rings were used in several runs. In this investigation, the rectangular type installation was used and its dimensions were obtained from the "O" ring catalog.[1]

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(1) Goshen Rubber Co., Catalog C, p. 9.

This investigation brought about several suitable methods of closing thin wall aluminum cylinders. In some of the methods, such as the V thread and the riveted and pinned methods, the difficulty of assembly made them impracticable for mass production. Other methods, such as the rolled end crimp, were not mechanically sound. Three methods did come out of this investigation that appear to meet all the requirements of a good solution. These solutions were the hemispherical end crimp, the tapered buttress thread, and the partial cut method.

While the rolled end crimp is not entirely mechanically sound, it is easy to form and readily adaptable to mass production. For this reason, possibly it should not be discarded as a possible solution. Since a number of successful fractures were obtained, perhaps the rolled end crimp can be strengthened sufficiently by using an outside crimp or a similar device. A diagram of such a possible closure is shown in figure 17.

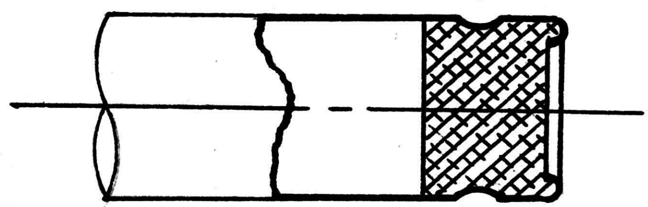


Figure 17. Proposed plan to strengthen the rolled end crimp with an outside crimp.

One inherent property that is common to both end crimps, is the dilation of the end of the tube at the crimp. This dilation was measured with a micrometer and was found to range from 0.004 to 0.008 of an inch along the diameter. However, this characteristic can be controlled by controlling the diameter of the female die.

The hemispherical end crimp is mechanically sound and easily formed. It does have one disadvantage, however, in that, when pressure is applied, the plug moves outward slightly. This movement is not great but the resulting increase in volume will tend to lower the internal pressure, especially if an incompressible fluid is used. This closure has the advantage that should the "O" ring fail, the plug is inherently self sealing. This method is very strong with holding power ranging up to 170 per cent of the required value.

It will be noted in plates 1, 2, 3, 4, 5, and 6, that the holding power of the end crimp during the tensile test, increased rapidly to a point and then the increase becomes less and less. This characteristic could easily be noted by examining the type of failure of each crimp. Before this saturation point was reached, failure occurred by the edge of the tube end splitting along a longitudinal seam. Above this point, failure occurred by a ring at the tube end failing transversally.

Perhaps another end crimp could be formed which would combine the properties of the rolled end crimp and the hemispherical end crimp. Such a crimp would have a larger radius than that used in the rolled end crimp. This crimp would probably be stronger than the rolled end crimp and yet might eliminate the axial movement of the plug common to the hemispherical end crimp.

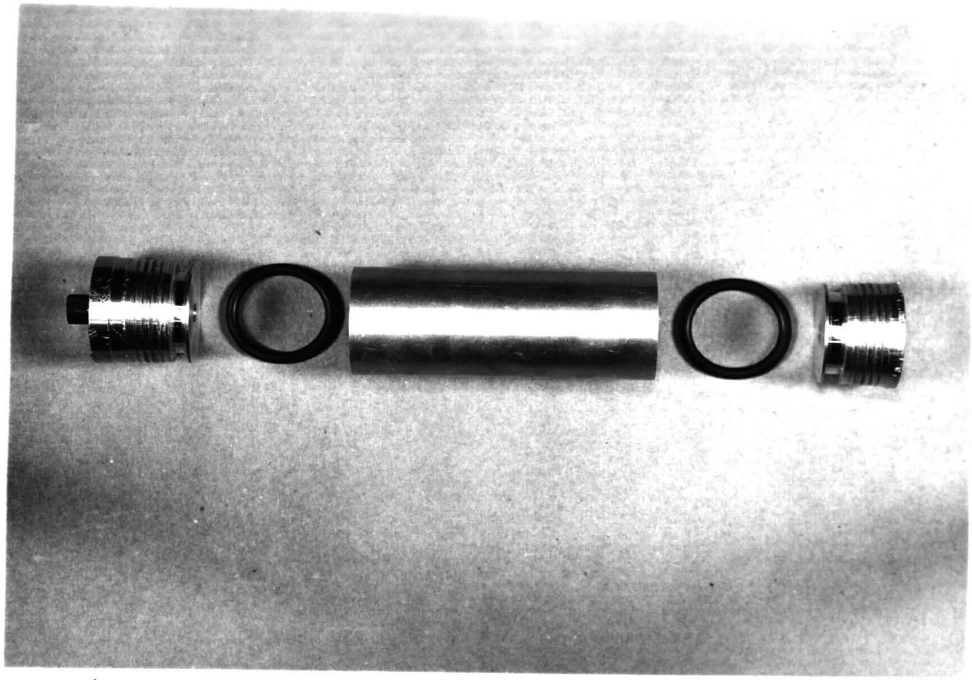
The buttress type thread appears to meet all the requisites of a good closure. It is mechanically sound and can be easily adapted to mass production. The strength of this method can be more readily controlled than that of the previous methods. The only disadvantage of this method is its application to very thin walled tubes. Satisfactory results are obtained with tubing having a wall thickness of 0.065 inches, but to be used on thinner walled tubing, careful alignment and cutting is necessary.

The partial cut method, like the threaded closure, appears to be satisfactory on all counts. Moreover, it can be easily used on all sizes of tubing. Figure 16 shows the apparatus used to form the partial cut, where 1 is the jig, 2 the punch, 3 the dowel, and 4 is the punch holder. As can be seen from this drawing, careful alignment is also a critical requirement in this method.

Comparing these different solutions, it would be difficult to say which is the best as each has its own advantages and disadvantages. However, any one of the above methods should be very satisfactory as a solution to the problem.

APPENDIX

Figure 9. Photograph of hemispherical end crimp, before and after failure.



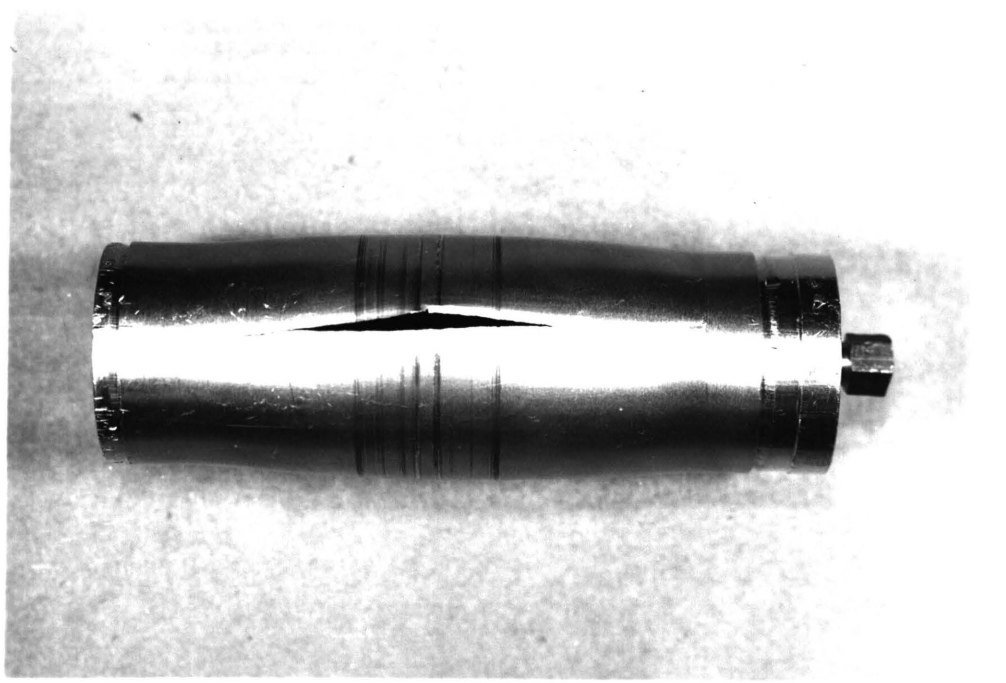


Figure 13. Photograph of buttress thread, before and after failure.

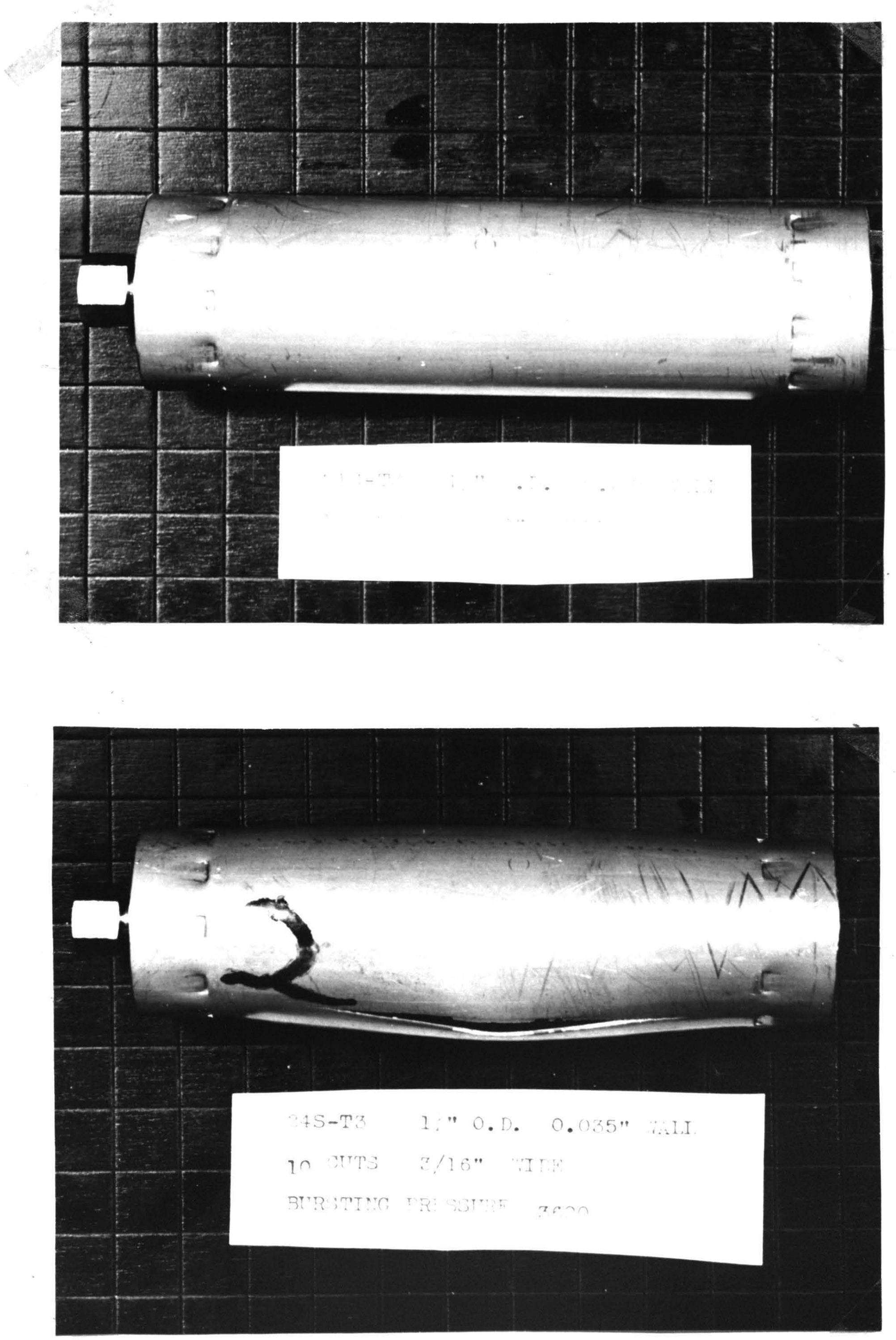


Figure 15. Photograph of the partial cut method, before and after failure.

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VITA

The author was born January 20, 1931, at St. Joseph, Mo., the only son of Mr. and Mrs. Archie W. Culp.

His early education was received in several grade schools in St. Joseph, and in 1948, he graduated from Lafayette High School of that city. He then entered the St. Joseph Jr. College, where he completed two years of study. In September, 1950, he transferred to the Missouri School of Mines and Metallurgy, where he graduated in June, 1952, with the degree B. S. in Mechanical Engineering. To date, the author has continued graduate work while serving as a graduate assistant in the Mechanical Engineering Department.