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Crimp method of closure of small pressure vessels

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CRIMP METHOD OF CLOSURE OF SMALL PRESSURE VESSELS

BY

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A

THESIS

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INTRODUCTION

The object of this thesis is to design a means of closing the ends of thin aluminum alloy tubing in the 1" O.D. to 2" O.D. by .035 to .065 inch thick sizes, so that the closure will withstand internal hydrostatic pressure sufficient to burst the tube along the longitudinal element.

The aim is to approach the ultimate mechanical efficiency possible, and at the same time to keep actual production in mind.

The author being acquainted with the investigations made by Mr. Aydin Cansever on the topic of "The Holding Strength of Crimps on Tubes" came to mutual understanding with Dr. Aaron J. Miles to continue investigations from a different angle.

Due to the fact that the tubing used for this investigation is thin walled tubing, machining is out of the question; that is, even the smallest size of threads cut on the tubing will actually reduce its total efficiency considerably.

The 24S-T3 aluminum alloy tubing is heat treated. This heat treatment supplies an ultimate tensile strength of 64,000 psi and a yield point of 42,000 psi. In this investigation it is positively desired that those characteristics should not be changed; that is, the actual experimentation should be carried on with the 24S-T3 aluminum alloy, and not with something else. The T3 symbol of the 24S-T3 aluminum alloy indicates the actual heat treatment. So it can be concluded that no annealing or any other heat treatment is required. As a result welding is out of the question. Although welding is a handy engineering method, at the same time it is an air cooling process.

In spite of the fact that the above stated facts about welding make it unnecessary to carry the discussion any further, it can be added that weldings made on a thin tube whose weldability is only fair are more than questionable.

Another point which should be encountered in this investigation is the low unit elongation of the 24S-T3 aluminum alloy. Its high Brinell Hardness of 120 B.H., and its low unit elongation of 10 percent makes it unfavorable to cold working. The usual method used for this kind of alloys is to work them while hot, that is, by applying heat and forming them while in this state. This is not allowable in this investigation. So for every individual case, an individual method should be found.

It is evident that closed vessels under high hydraulic pressures are most favorable to leakage. The fluid at high pressures will try to deform weak spots and leak. So, it is not only a question of forming a fluid tight joint, but to form a joint tight under hydraulic pressures, that is, joints which are not favorable to the slightest deformations under hydraulic loads.

Summarizing the aims of this investigation, it can be said that the objectives of this investigation are:

1. To design ends for closed vessels.[1]
2. To form the designs.
3. To compare different closed vessel ends from the production and market point of view.
4. To try to find out the behavior and efficiency by analytical means and actual tests.

REVIEW OF LITERATURE

The subject of this investigation being an altogether industrial one, its public literature can be said to be almost nonexistant.

General information about the 24S-T3 alloy can be obtained from publications of the Reynolds Metals Company,[1]

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1. Reynolds Metals Company, The Aluminum Data Book, Aluminum Alloys and Mill Products. 1950.

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and Aluminum Company of America.[2]

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2. Aluminum Company of America, Alcoa Aluminum and Its Alloys, 1950.

The above mentioned publications give a general idea about the metal used in the investigation.

A fair idea about O-rings can be gained by looking in Kent's Handbook.[3] More thorough information can be obtained

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3. Kent's Mechanical Engineers' Handbook, Design and Production Volume, 12th Edition, paragraf-pressure seals, pp. 1803-1815.

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from Goshen Rubber Company's publication.[4]

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4. Goshen Rubber Co., Inc. - Catalog C.

Information about hydraulic loads can be obtained from any Hydraulics textbook.

In this investigation, for obtaining information about stresses under hydraulic loads, the Murphy's Advanced Mechanics of Materials was used.[5]

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5. Glenn Murphy, Advanced Mechanics of Materials, Chapter III, pp. 59-69, First Edition, 2nd Impression, 1946.

Detailed information about crimps and plugs under mechanical tensional loads can be found in Ayden Cansever's thesis under the heading of "The Holding Strength of Crimps on Tubes."

CHEMICAL AND PHYSICAL PROPERTIES OF 24S-T3 ALUMINUM ALLOY AND SOME INFORMATION ABOUT ITS PRODUCTION METHOD

The 24S series of aluminum tubing alloy consists of copper, magnesium, manganese and aluminum (approximately 4.5 percent Cu, 0.6 percent Mn, 1.5 percent mg, 93.4 percent Al).

The production method of the pipe is the well known method of forcing heated blooms through predetermined die openings equipped with mandrels interiorly concentric with the die. The mandrel and die diameters are of such dimensions as to produce the required wall thickness of the tube. By drawing the extruded tube through a bulb concentric to a die, the well known drawn tube is formed.

The 24S-T3 aluminum tube is a drawn tube, heat treated, naturally aged and cold worked. Its ultimate strength is 64,000 psi and its yield strength is 42,000 psi. The elongation percentage in tube form is approximately 10 percent. In sheet or plate form the minimum bend radius for 90 degrees bends is 2t[1] and for 180 degrees bends is 4t. It has excellent electrical resistance, machinability, and rural corrosion resistance. Its industrial, mineral, and sea water corrosion resistance can be said to be fair. Its arc weldability is fair. Its hardness in Brinell number is 120 (500 kg load - 10 mm ball).

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1. t = thickness

The main characteristics concerning the experiment in question in this thesis are the physical properties, mainly the strength, elongation, bend radii, and hardness. As a conclusion it can be said that, although the alloy in question has a high tensile and compressive strength, its high Brinell hardness, its high minimum bend radius, and low elongation percentage makes it a not so easily cold workable alloy. This was proven to be true while working on crimp formation of plugs. In many cases the radius of the crimp to the depth relation, and the locking effect of the rubber used, caused the pipe to shear.

PLUG FORMATION

In this chapter the formation of different plugs are discussed according to their production methods: 1. Mechanical plugs. 2. Rubber or Hydraulic plugs.

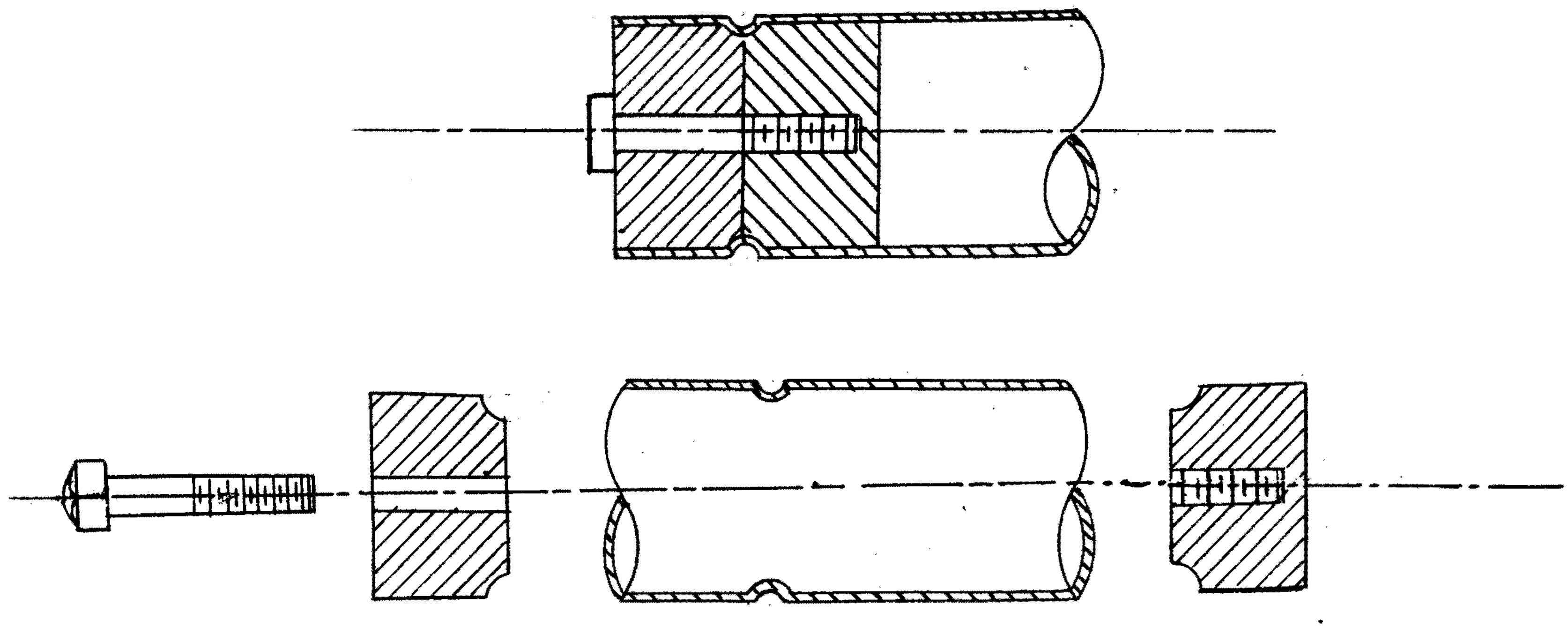
Mechanical Plugs

Mechanical plugs are plugs formed by mechanical means. Their commercial production in a wide scale makes them most desirable. Here four kinds of mechanical plugs are introduced. The fact that the four mechanical plugs introduced in this chapter can be produced separately piece by piece and then assembled together makes them most adaptable in any industry.

<u>Two Piece Assembly</u>

This plug consists of two pieces as shown in the diagrammatic sketch 1a. The formation of the plug itself is not a mechanical problem whatsoever. By means of a die the required crimp on the tube coinciding exactly with the plug can be formed, every piece necessary for the assembly of the two piece plug thus being formed separately. Then it can be assembled.

As is seen in the diagrammatic sketch, in order to avoid any leakage due to hydraulic pressure, the threaded half of the plug should not be threaded all the way through, and an oil seal should be used.



Sketch of the Two Piece Assembly
Mechanicol Plug

<u>Split Skirt</u>

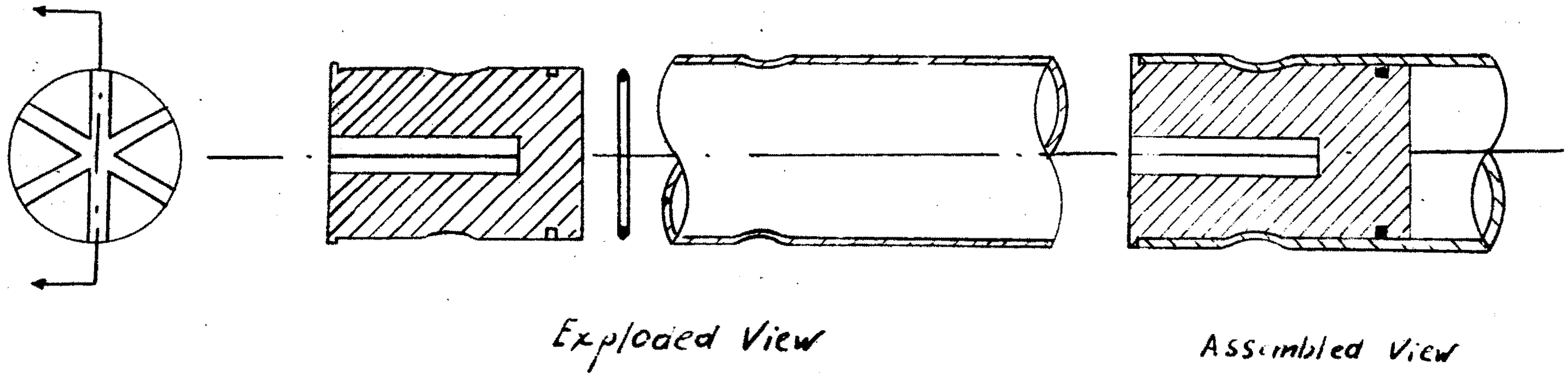
In this method the elasticity of the plug material should be considered. But due to the fact that there are not many restrictions as to the size of the radial grooves, this is not a problem whatsoever.

Again as it is in the case of the two piece plug, the plug can be formed separately by a mass production method. The crimp on the tube is formed by means of a die which exactly coincides with the groove (or grooves) formed in the plug. Now it all remains by applying transverse pressure to the skirt of the plug to insert the plug through the previously formed crimp. Of course an oil seal in order to prevent any leakage due to high hydraulic pressures should be inserted.

<u>Tapered</u>

The tapered kind of plug consists of two parts: a female plug which is tapered as shown in Figure 1c and a male plug which is tapered with exactly the same angle of inclination as shown in No. 1, Figure 1c.

Only when forming part No. 2 should the thickness of the pipe be taken into consideration. Then the tube is pushed through the plug (No. 1, Fig. 1c). The only restriction is that the taper angle should not be of a size that will cause the tube walls to shear (an angle of 8 degrees was tried with the utmost satisfaction). As is seen, the production of the tapered plug is simple. There is no need for any special tools and it can easily be made in any



Sketch of Split Skirt Plug

small machine shop on a large scale. It is self locked from hydraulic leakages. The higher the pressure applied, the more it will help the plug to tighten, thus making the plug to tube connection leakproof. The fact is that the plug was tested by applying mechanical tension and proved itself 100 percent efficient.

As is well known, the smoother the surfaces of the plug are, the better its total mechanical efficiency will be. No oil seal is needed.

<u>Intruded</u>

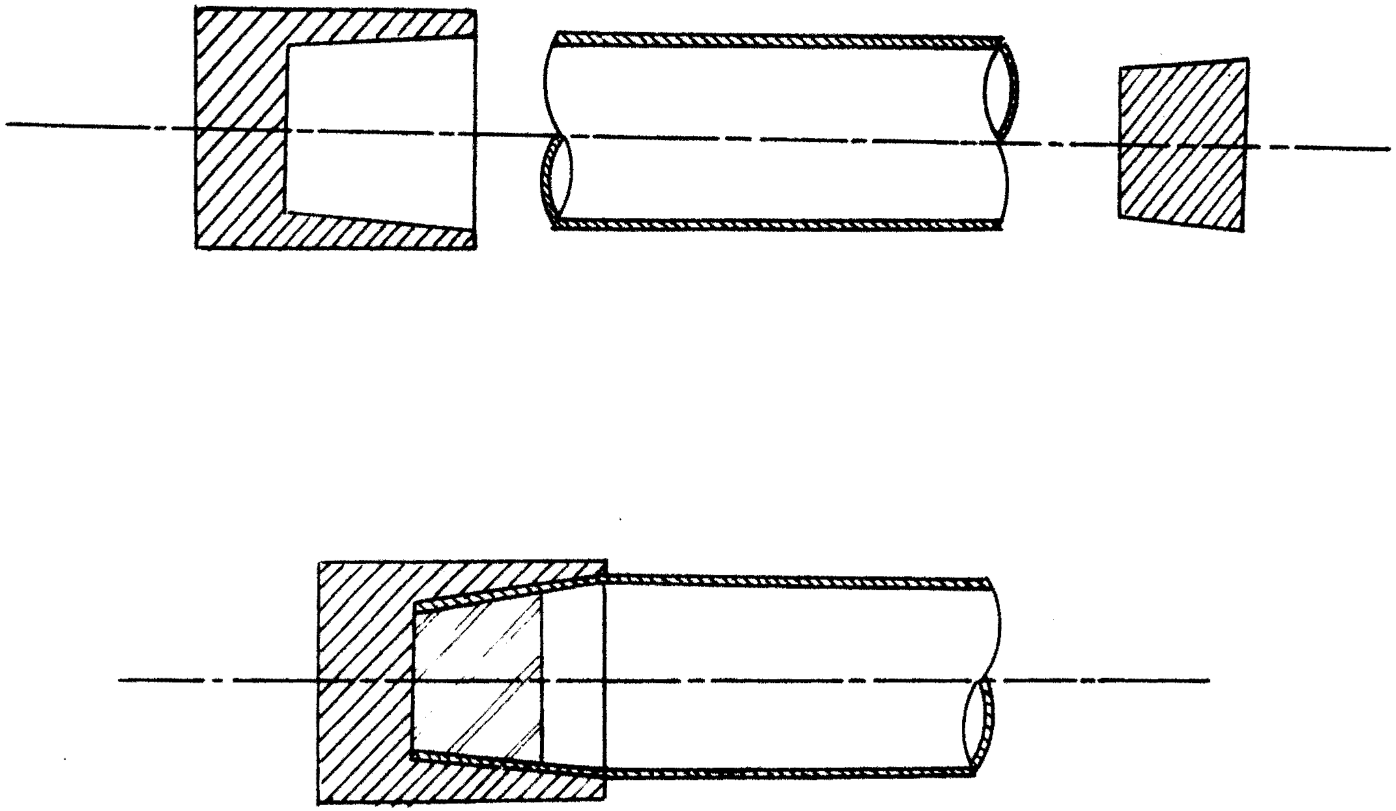
As shown in Figure 1d, the intruded plug consists of three parts:

1. Part No. 2, which is intruded into the pipe. The taper angle should not be a large one. Otherwise the pipe's unit elongation can pass the limit of ultimate unit elongation (10 percent). So the pipe will shear while intruding No. 2.

2. Part No. 1, which has the same taper angle as No. 2, and whose inside surfaces will exactly fit to the outside surface of the pipe as shown in Figure 1d.

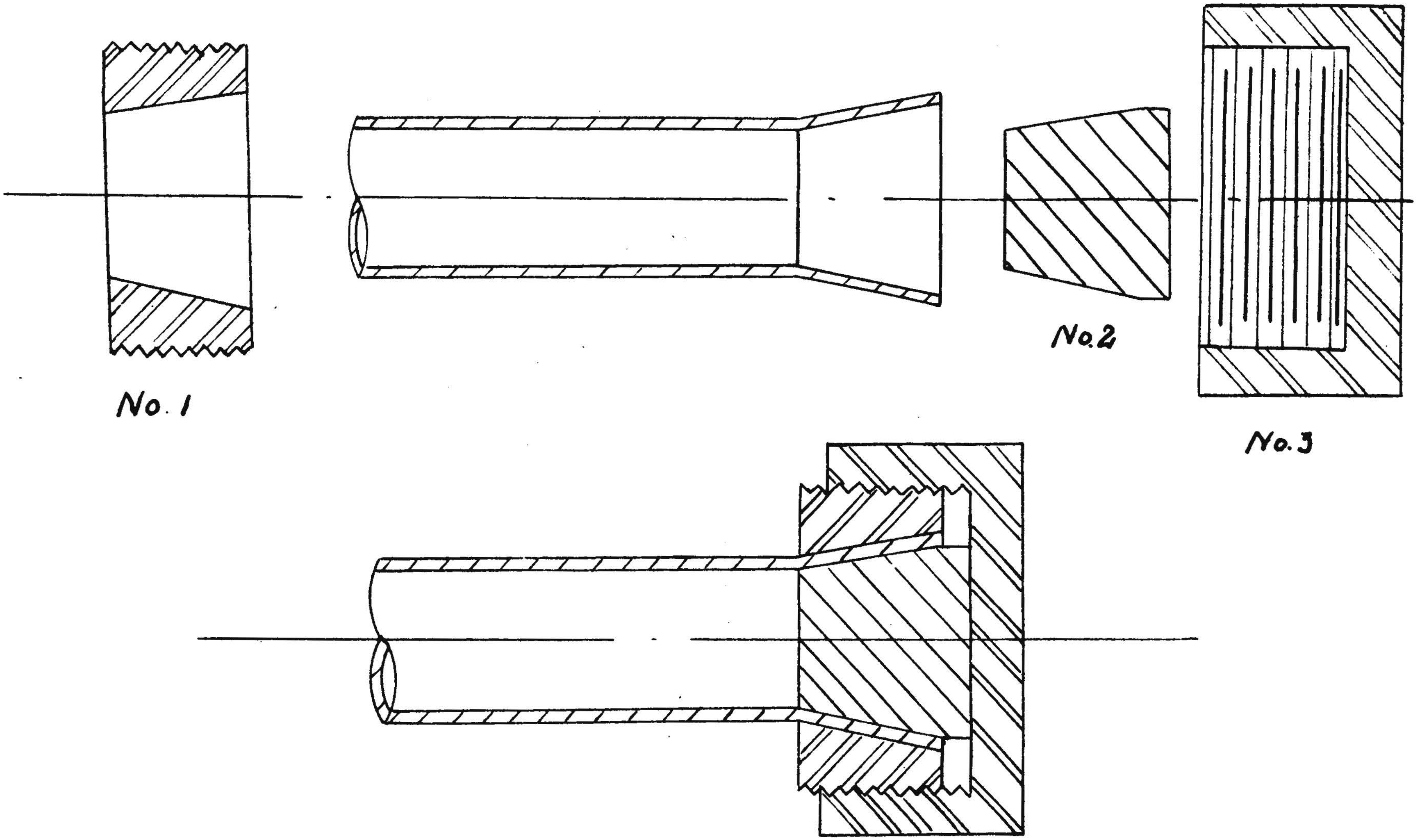
3. Part No. 3, which is bolted to No. 1.

It can very easily be mass produced. It can be used for both ends of closed vessels, and it can be made 100 percent efficient. What is more, it is ideal for hydraulic loads. By tightening the bolt to the nut, it can be made leakproof. It proved itself 100 percent efficient under mechanical tension.



Sketch of Tapered Mechanical Plug

Fig 1.C



Sketch of the Intruded Plug.

Fig 1.a

Rubber or Hydraulic Pressure Plugs.

In rubber or hydraulic pressure plugs the grooves on the plugs are cut, then the formation of crimps are made by applying hydraulic or rubber pressure. This method of plug formation requires different kinds of devices for each individual case. The locking effect of the wall surfaces of the pipe due to the hydraulic pressure restricting the pipe fluidity is an important factor which should be overcome. But once an applicable method is found, then for special cases this method can be applied to mass production. Here in this chapter two methods are discussed.

The plugs discussed here are classified according to their tube (pipe) to plug position; that is, when the plug is inside the tube, the plug is called an inside plug, and when the plug is outside the tube, it is called an outside plug.

Inside Plugs

One method used successfully in forming the inside plugs is the following: Two concentric cylinders, as shown in Figure 2a, A and B. The outside diameter of B should be exactly equal to the inside diameter of A. The diameter C shown in Figure 2a should be exactly equal to the outside diameter of the tube used. The tube, cylinder A, and cylinder B are placed into each other as shown in the exploded view sketch of Figure 2a. Then a compressive load is applied. This load creates an hydraulic pressure on the rubber which forces the pipe to take the form of the plug grooves. The fact that the rubber pressure is applied only in the grooved section of the plug (or better, almost so) eliminates the so-called locking effect on the pipe. The crimps are formed by the tube metal sliding between the walls of cylinder A (or almost so), thus reducing its unit elongation to a figure far below its rupture point.

The use of a lubricant between the walls of cylinder A and the tube was found to be of great help.

Outside Plugs

Two almost identical methods have been used which proved to be failures mostly due to the locking effect. Both of them are shown in Figure 2b 1, and 2b 2. The reasons for their failure are the following:

2b 1. The rubber slipped out of the gap between No. 1, and No. 3(see Fig. 2b 1).

2b 2. The elasticity of the rubber makes it almost impractical for shaping No. 3 to fit No. 1 (see Fig. 2b 1) with the required negligible clearance (tube thickness taken into consideration).

2b 3. The result of 2b 1, and 2b 2 was: A. It was either that No. 3 (Fig. 2b 1) in travelling a distance further down in No. 1 (Fig. 2b 1) locked itself on the walls of the tube, or B. the rubber slipping around No. 3 (Fig. 2b 1) produced a locking effect on the walls of the pipe.

Then an alternation of the previous designs in Figures 2b 1 and 2b 2 was made as shown in Figure 2b 3. As is obvious this time the locking between cylinder and tube walls

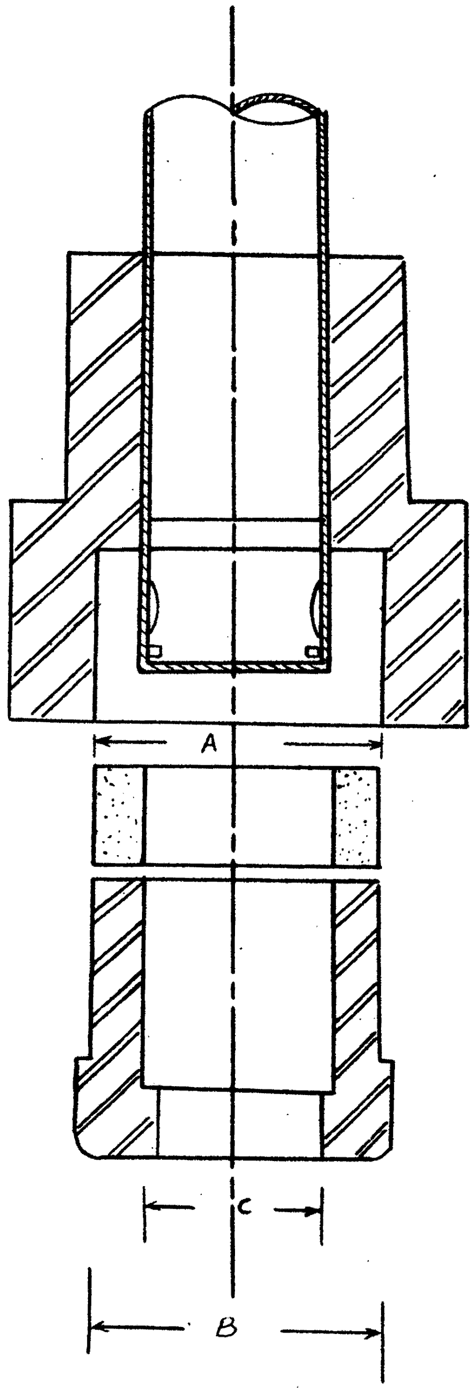
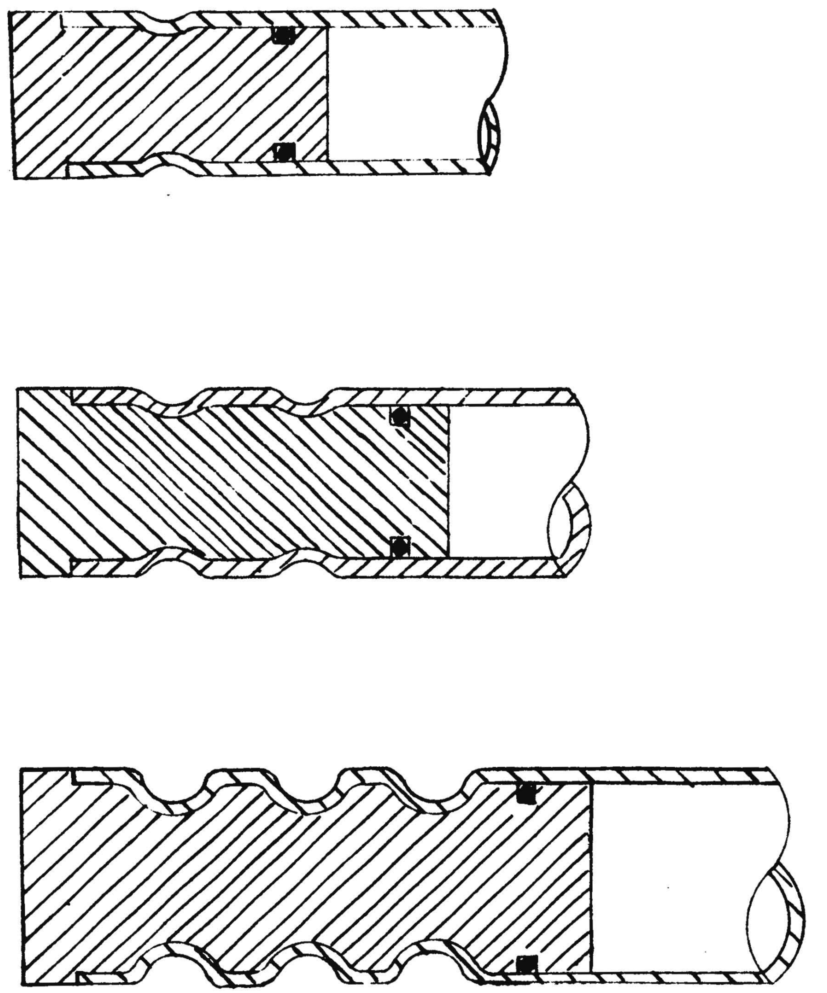
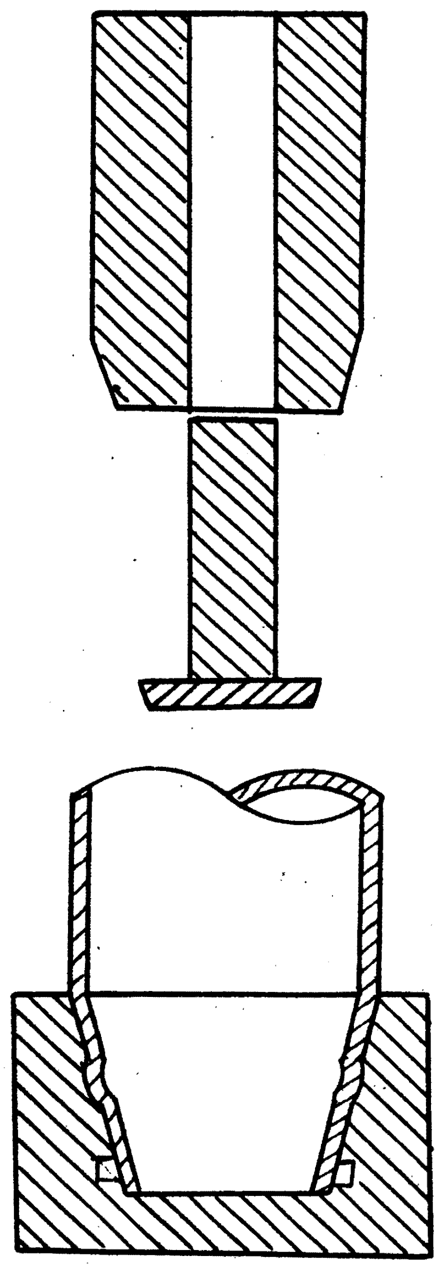


Fig 2.a

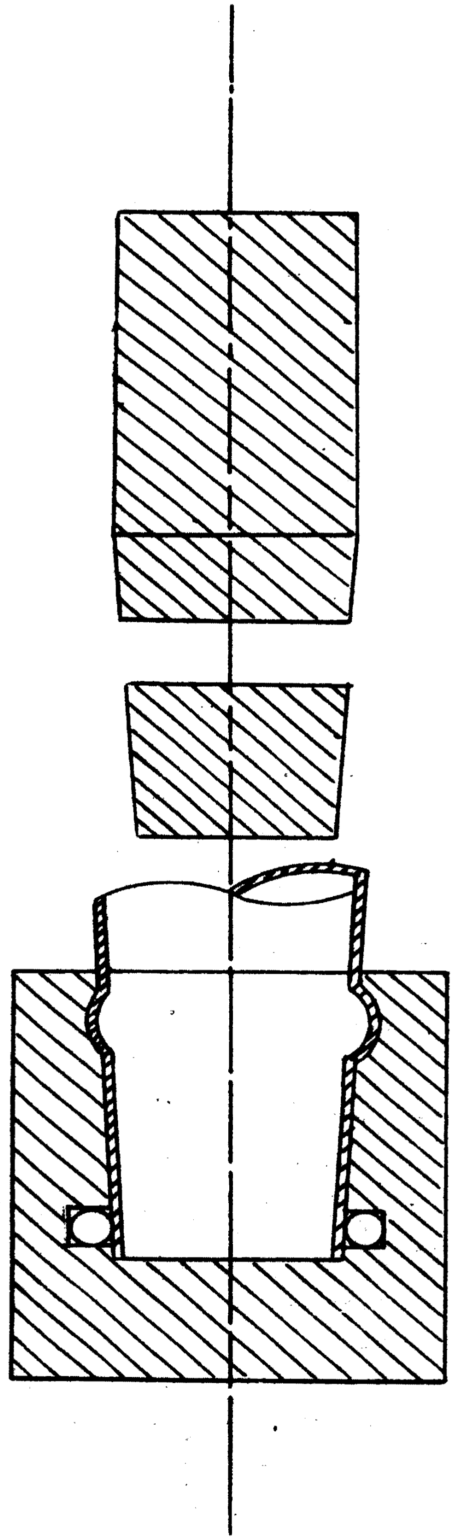
Device for Rubber Crimp Formation



Inside Plugs Formed by the Concentric Cylinders Shown in Fig 2a.



Sketch of Outside Plug
Fig 2b1

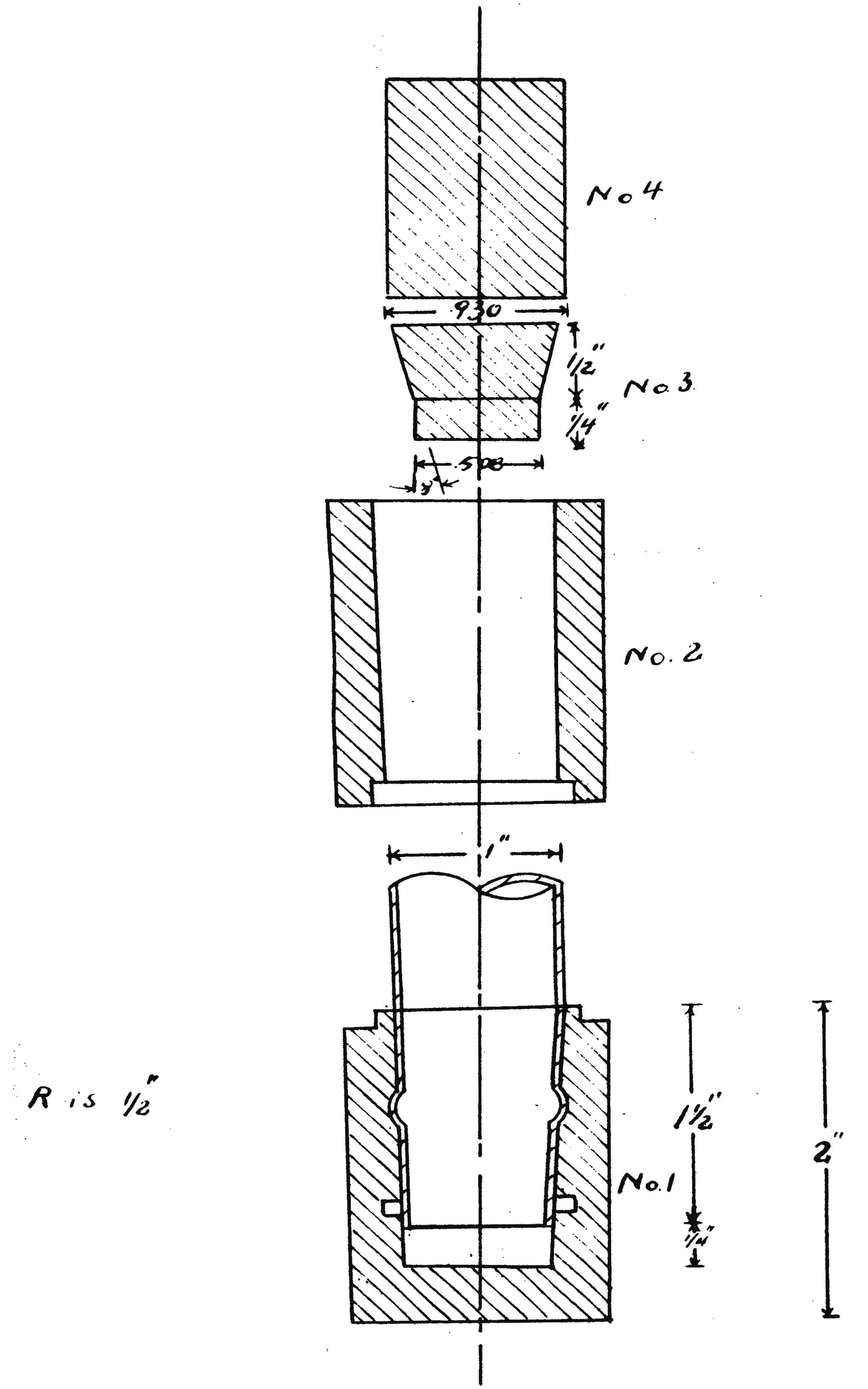


Diagmatic Sketch of Outside Plug
Fig 2.b.2

will not occur. But the sliding problem of the tube wall metal in order to form the required crimps, and the locking effect due to the applied rubber pressure to the walls of the tube, still exists. The part of the tube found above the crimps will be hydraulicly (by the applied rubber pressure) locked, but the part of the tube found below the crimps can be made to slide by using the part No. 2 (Fig. 2b 3). No. 3 should be made with great mechanical skill in order to have a negligible clearance which, although it will allow the portion of the pipe below the crimps to slide, will not allow the rubber to slip between the pipe walls and No. 3 causing an hydraulic locking effect in that portion of the pipe.

This kind of plug, although an efficient plug due to the fact that the applied hydraulic pressure will tend to act normal to the walls of the pipe, thus helping to seal the plug hydraulicly, is hard to make. But even from the mechanical efficiency point of view, it is not superior to the tapered and intruded plugs whose production is far more simple and easy.

In short, it can be summarized that out of the six methods of plug formation discussed in this chapter, the mechanical plugs are far more adaptable to market production, and, from their mechanical efficiency point of view, they are not inferior to the hydraulicly formed plugs (formed by the rubber method). Especially about the Tapered and Intruded mechanical plugs it can be said that they are the most easily formed ones, and that from the stress analysis point of view far more superior to the other four.



Outside Plug Fig 2.6.3

CRIMP FORMATION

The crimp formation on sheet metal depends on many factors; they are mainly:

1. Its workability at ranges above its elastic limits, that is, at plasticity ranges.
2. Its unit elongation.
3. Its ultimate strength.
4. Its thickness, especially when bending is in question.
5. Its minimum bending radii.
6. Its surface condition, that is, whether the surfaces are smooth, or have cracks which will change predicted results.
7. Methods used in forming the crimps.
8. The fact of whether or not any lubricants were used, and if so, the kind of lubricants used.

Taking all those factors which in a way are directly connected with crimp formation to a more narrow scale, that is, to 24S-T3 aluminum alloy, we survey the following facts:

1. The 24S-T3 aluminum alloy is a heat treated alloy. So if it is required that this alloy should carry its full characteristics, it can only be cold worked. It can clearly be seen that all the advantages of hot working while actually forming the crimp is out of the question.

2. Its unit elongation being 10 percent, it is evident that it is not a highly ductile material.

3. Its ultimate strength is 64,000 + psi. Although

this high figure seems to favor crimp formation at cold temperatures, it actually, due to its high Brinell hardness (120 B.H.) and low unit elongation, means just one thing; that is, while crimp forming, higher pressures should be used. From the crimp formation point of view, this cannot be counted as an advantage.

4. The thicknesses used were

   a. for 1" O.D. pipe, .035 inches and .065 inches

   b. for $1\frac{1}{2}$" O.D. pipe, .035 inches

   c. for 2" O.D. pipe, .065 inches.

5. For its minimum bending radii one can be specific and give the approximate figures of 2t[1] for 90° bends, and

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1. t = thickness

4t for 180° bends.

All the above from the crimp formation point of view cover the characteristics of the 24S-T3 tubing used. If it is added to this that the method of crimp formation was by means of rubber pressure, and that lubricant was used during formation, the summation of the characteristics of the 24S-T3 aluminum alloy which are directly connected with crimp formation are completed.

Thus having a general picture of the characteristics of the 24S-T3 aluminum alloy from the crimp formation standpoint, it can be concluded that it is not so easy to form small size crimps unless the sheet metal, or the pipe as it is in this case, forming the crimp is made to run from the sides, thus keeping its unit elongation below its ultimate unit elongation. In the next few paragraphs this subject is discussed in more detail.

If the pressure applied on the rubber is of such a nature as shown in Figure 1, then the unit elongation reduces

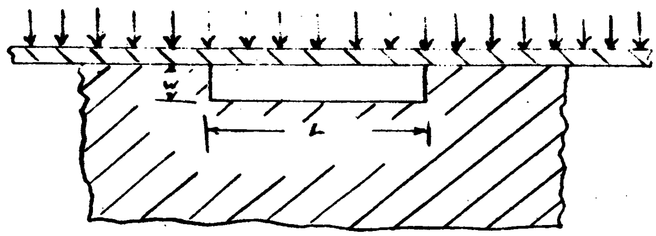

Fig. 1

to 2 W/L (or very near so) in which case unless W is very small compared ($e_u = \frac{L}{10}$) the unit elongation of the metal being 10 percent, it will most probably fail. Even in cases where e_u is pretty close to $\frac{L}{10}$ but does not exceed this limit, although it will not fail, the stresses produced due to the elongation will weaken the crimp efficiency as a whole. This is due to the fact that the uniform pressure is applied beyond the two edges of the grooved section of the die, so producing the so called locking effect.

In cases where the pressure applied extends to a relative infinite distance beyond the grooved section, this locking effect will reduce the unit elongation of the sheet metal to 2 W/L.

It can be stated that for each individual case a certain method of removing this locking effect should be found. One method used for sheet metals is shown in Figure 2.

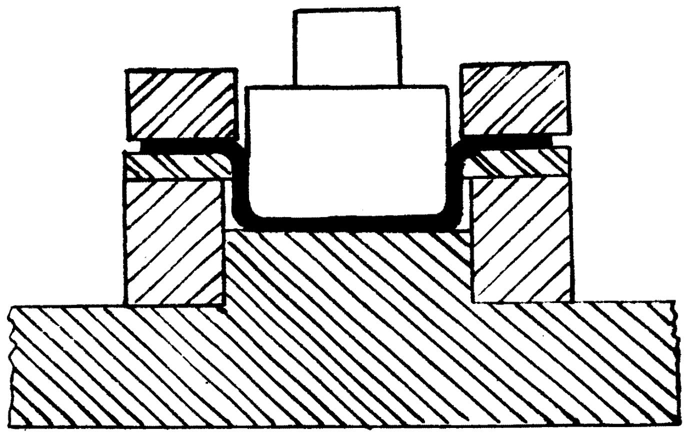

Fig. 2

As for the crimp formation for plugs which actually is one of the aims of the tests run, the method used depends on the position of the plug in relation to the pipe, and on the nature of the crimps themselves. Here the crimps used were of negative radii, that is, the grooved section was cut in the plug. As for the plug to the pipe position, they were two cases, either the plug was inside the pipe, or outside. These two individual cases are discussed in detail in the chapter of "Plug Formation."

The factor of minimum radii to be used in crimp formation is one of the utmost importance. As is known when forming radii in sheet metals, the outside section of the metal is under tension whereas the inside section is under compression. The outside section under this bending effect tends to crack whereas in the inside section of the metal internal stresses are created.

It is found for most metals that the minimum bending radii almost have nothing to do with the thicknesses of the metal. Exceptions are some aluminum alloys. The 24S-T3 aluminum alloy is one of them. The figures for minimum radii in regards to their thicknesses for the individual case of 24S-T3 aluminum alloy for 90 and 180 degrees bends are given below.[1]

1. Aluminum Structural Design, Reynolds Metal Company, Table 17, p. 81.

Thicknesses	.016"		.032"		.064"		.125"		.188"		.250"	
Bends in degrees	90	180	90	180	90	180	90	180	90	180	90	180
Minimum bend radii	1½t	3t	2t	4t	3t	5t	4t	6t	4t	6t	5t	7t

The 24S-T3 aluminum alloy can be formed over appreciably smaller radii immediately after quenching.[2]

2. Aluminum Structural Design, Reynolds Metal Company, Table 17, p. 81.

TENSIONAL LOADS ON CRIMPS

The crimps discussed in this chapter are under tensional loads, and their general formation is of a nature that they will stand such loads. The question as to whether any efficient crimps could be made which will stand torsional loads is beyond the scope of this thesis. The loads that are applied to the plug will tend to push the plug out, and the crimped pipe to plug connections will resist this load. If resistant to torsional loads crimps were needed, this could be done by tapering the traverse crimps, or by adding horizontal crimps. The thickness being kept constant, there are mainly three factors which in one way or another are connected with crimp resistance to axial loads:

1. the radius of the crimp
2. the depth of the crimp
3. the number of the crimps.

Before going into any details about these three factors stated above, the expected failures should be mentioned.

The applied axial load will be resisted by a reaction normal to the walls of the crimped section of the sheet metal, or tube. (See Fig. 1)

If the resistances between sheet metal and grooved metal are negligible, then:

1. The perpendicular force will tend to:
   a. straighten the crimp
   b. shear the crimp.

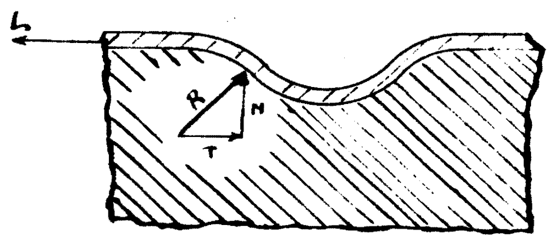

Fig. 1

Note: R is uniformly distributed.

2. Due to horizontal force T it fails by tension.

So summarizing the above, it can be said that there are two kinds of failures that will occur:

1. due to a combination of shear and tension stresses, the crimp will fail from rupture.

2. due to an upward force, the crimp will straighten.

Under applied loads, it is proven that bend or curved sections will fail first. So as it is in this case, no matter how perfect a crimp will be made, the failure point will be in the bend section, that is, in the crimp itself. So even under ideal conditions, under tensional loads it will be impossible to reach the required 100 percent efficiency. Of course, by using reinforcements, it is not impossible even to reach this limit. See Tables

Now the discussion of how this limit could be approached

ONE CRIMP PLUGS UNDER TENSIONAL LOADS

Tube O. D. 1", thickness .035

A. Crimp diameter 3/16

Depths	Loads Carried	Efficiency percent	Failure
.025"	1360 lbs.	19.3	Pulled out
.050"	1750 lbs.	24.8	Sheared
.075"	975 lbs.	13.8	Sheared

B. Crimp diameter 1/4

Depths	Loads Carried	Efficiency percent	Failure
.025"	920 lbs.	13.1	Pulled out
.050"	2445 lbs.	34.6	Sheared
.075"	830 lbs.	11.7	Sheared while forming

The ultimate load the pipe will stand

DtS = Load

Load = 1 x .035 x 64,000 = 7,050 lbs.

So the efficiencies were coevaluated on this basis.

from tensional loads standpoint of view could be carried.

For a fixed thickness and for one crimp the strength of the crimp is in some proportion to the radius of the crimp and to the depth of the crimp. For fixed number of crimps and fixed crimp radii, as the depth is increased the strength will increase up to a point where the crimp will start to shear instead of slipping. After this point is reached an increase in depth will weaken the strength of the crimp, due to sharper curves. So when a crimp reaches this point it means that it reached its ultimate strength.

Now keeping the radius and depth of this crimp of ultimate strength constant, and changing the number of crimps, it can be concluded:

That if all the crimps are made in perfection, with no machining, or forming, errors whatsoever, the crimps will carry a load equal to $\frac{\text{Total load}}{\text{number of crimps}}$. If the ultimate load carried per crimp is divided into the ultimate load applied, the number of crimps which will make the whole system 100 percent efficient can be found. Of course, this is an ideal case. There would be small machining errors, and small formation errors, so for 100 percent efficient plugs, the equation can be written:

$$\text{Number of crimps} = k \, \frac{\text{Total load}}{\text{ultimate strength per crimp}}$$

where k should be found for every individual system.

From the following table it can be seen that the efficiency for three grooved plugs is three times greater than

MULTICRIMPED PLUGS UNDER TENSIONAL LOADS

Tube thickness .035, Tube O.D. 1 inch.

Crimp diameter 1/4; depth .025

Number of crimps	Loads	Efficiency percent	Failure
1	920 lbs.	13.0	Pulled out
2	2495 lbs.	36.6	Pulled out
3	2745 lbs.	39.50	Sheared

for the one grooved plugs.

It should be added that in a case where there is not any precision, adding crimps would not help to increase the efficiency of the whole system whatsoever. The crimps due to bad formation will receive the load individually, thus fail one by one as it is in the case of one crimp.

Tapered and Intruded Plugs under Tensional Loads

The failure in mechanical tension of the mechanical tapered and intruded plugs for a 1" x .035 pipe was 8,100 and 7,900 respectively. The failure occurred in the pipe itself.

So both of those plugs proved that they are loo percent efficient to tensional loads.

O RINGS

The practical application of O rings in eliminating hydraulic leakages makes them very desirable in cases such as the one investigated in this thesis.

Their behavior under hydraulic loads is indicated in Figure 1.

When O rings are installed it is desired that the surfaces of the grooved sections should be smooth and rounded.

The required sizes of O rings should include the required minimum diametral squeeze allowance. Since in this case the diametral elongation of the pipe walls due to hydraulic pressure is expected, it is advisable that larger diametral squeeze allowances should be used.

The behavior of O rings under hydraulic pressure is shown in Figure 1.[1]

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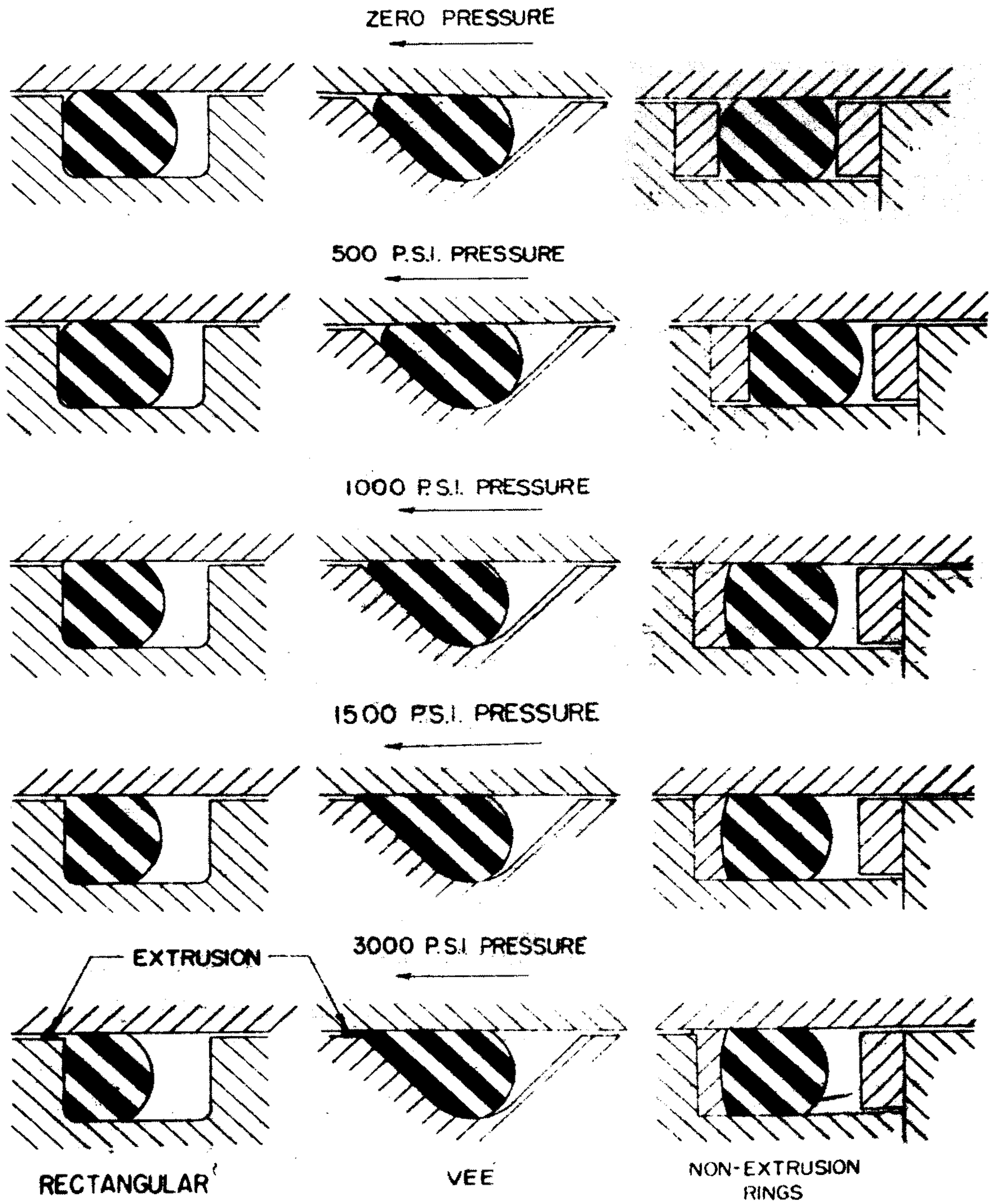
1. Reproduced from Goshen Rubber Company's Catalog C, page 2.

ACTION OF "O"- RING PACKINGS & GASKETS IN RECTANGULAR & VEE SHAPED GROOVES



PIPE TO PLUG CONNECTIONS UNDER HYDRAULIC LOADS

Under hydraulic loads the behavior of the pipe to plug connections are completely different from what they are under tensional loads.

When a closed vessel is under hydraulic pressure there are two kinds of reactions which should be encountered:

1. As it was in the case of tensional loads the horizontal force H. (Fig. 1.)
2. And the traverse Force T. (Fig. 1.)

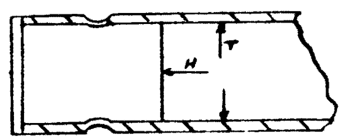


Fig. 1

A. Effects of Hydraulic Pressures on Inside Crimped Plugs.

The horizontal hydraulic force as it was discussed in the chapter under the heading Tensional Loads on Crimps will try to push the plug out, and the reactions will be just the same as they were discussed.

The traverse forces due to the hydraulic pressure will tend to elongate the inside diameter of the pipe, thus producing a clearance between the pipe and plug walls. This clearance will be equal to $\frac{PD^2(2-\mu)}{4tE}$. (See Fig. 2.)

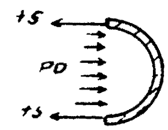


Fig. 2

The actual elongation will not just be a result of the unit elongation due to traverse load, but a summation of the horizontal and traverse loads. That is the horizontal load will actually influence the total traverse elongation. By taking a section out of the walls of the tubing the following results can be observed. See Figure 2b.

$$Su = \frac{PD}{2t}$$

$$\epsilon u = \frac{PD}{2tE}$$

$$Sv = \frac{PD}{4t}$$

$$\epsilon v = \frac{PD}{4tE}$$

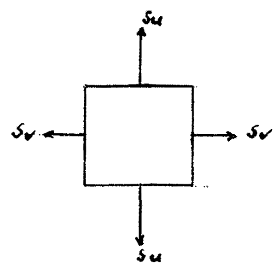


Fig. 2.b.

The actual elongation will be the summation of the traverse load minus the horizontal load times the Poison's constant

$$\epsilon_u' = \frac{PD}{2tE} - \mu \frac{PD}{4tE} = \frac{PD\,(2-\mu)}{4tE}$$

and the diametral total elongation will be:

$$u = \frac{PD^2 (2-\mu)}{4tE}$$

where:

P = the inside hydraulic pressure in psi.

D = the inside diameter in inches.

t = the thickness of the pipe in inches.

E = the modulus of elasticity in pounds per square inch.

μ = Poison's ratio

So as it can be seen actually the horizontal load will help to reduce the diametral clearance. Thus taking into consideration that a small clearance δ due to machining and forming errors produces a total clearance of:

$$\text{Clearance total} = \delta + \frac{PD^2 (2-\mu)}{4tE}$$

Taking the ideal case where δ is zero, then the $\frac{PD^2 (2-\mu)}{4tE}$ clearance as long as P is not zero will be there producing the following effects:

1. Due to the difference in pressure between the two sides of the plug, leakage will occur.

2. As a result of this leakage the liquid under hydraulic pressure will try to push the pipe apart from the plug. In other words, it will try to straighten the crimps (Fig. 3). This will help the horizontal force to push out the plug.

The case being as described, the discussion in the pre-

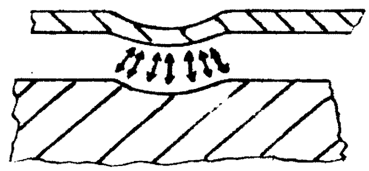


Fig. 3

ceeding chapter that 100 percent efficient inside plugs can be formed when the number of crimps equal to the axial load divided by ultimate strength per crimp (that is, when k = 1) does not hold any longer. The leakage will occur just the same no matter how many crimps there are. Of course, the length of the plugged part of the pipe will help to form frictional reactions, but that is all.

In order to overcome this leakage, hydraulic seals should be used as shown in Figure 4. But even in that case, the fact whether the seals used could close the gap of $\frac{PD^2\ (2-\mu)}{4tE}$ should be investigated.

In actual experiments two inside plugs, a single crimped one and a double crimped one, were used. The result was just the same. Those two plugs were of the same radius and depths, so the only variable was the crimp number. Although the one with two crimps was expected to hold a higher pressure, both of them failed from leakage at 900$\pm$ psi. The efficiency was as low as 10 percent ±.

If it is required to draw a conclusion out of this experiment it can be stated that although the inside crimped plugs can be made efficient enough for purely tensional loads, they can hardly be made efficient when hydraulic pressures are used. At this point it should be added that further experimentation should be carried out from the hydraulic pressure point of view.

B. Effects of Hydraulic Pressures on Outside Crimped Plugs.

The longitudinal force trying to pull out the plug will be reduced by a factor $\frac{d^2}{D^2}$. That is, the smaller the d/D ratio will be, the less the pushing out force will be. (See Fig. 4) That is a positive advantage over the inside plug.

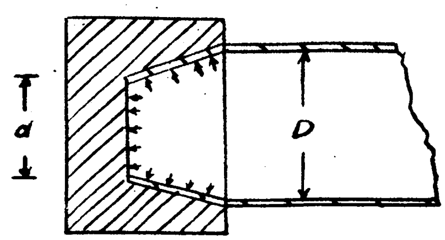


Fig. 4

What is more, this time the u factor will work in advantage, thus reducing the clearance. That is, total clearance $= -u + \delta$. In the case where the factor of $\frac{PD^2 (2-\mu)}{4tE}$ is larger than δ then the resultant force will produce frictional reactions which will help to hold the plug in against the longitudinal forces.

As is seen, the ends are open, so seal should be used.

C. Tapered and Intruded Plugs.[1] (See Fig. 1c and 1d)

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1. See Figures 1c and 1d in the chapter of Plug Formation.

Both of these plugs carry almost the same characteristics. Both of them are leakproof. The pushing out force helps to make them even more leakproof. There can be no failure within the plug itself. The weak spot is that the failure will occur just at the tapered part due to the traverse reaction. This part can be protected by prolonging the skirt of the plug beyond this weak spot, thus reinforcing it. See Figure 5.

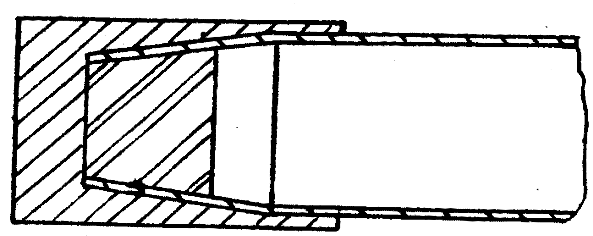

Fig. 5

Both of those plugs proved themselves 100 percent efficient under tensional loads, and both of them passed the limit of 2500± psi under hydraulic loads.[2] It is felt sure

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2. Due to failure in the pressure applying system, it was not possible to go to higher pressures.

that those systems could be proven 100 percent efficient. Some more experimentation in order to confirm this should be made.

PRESSURE APPLYING SYSTEM

The pressure applying system consists of the parts shown in Figure 1.

1. A pressure applying pump which has a dial indicating the pressures. This pump actually is an injector testing pump.

2. The system shown in Figure 2.

The pump was designed by the manufacturer to stand 15,000 psi of hydraulic pressure.

The computations for resistive strength are shown below: (See Fig. 2.)

The 1 inch in diameter, .035 inches thick, pipe will stand an internal pressure:

$$P = \frac{2tS}{D} = \frac{2 \times .035 \times 64{,}000}{(1.000 - .0070)} = 4500 \text{ psi.}$$

In Figure 2 part No. 1 will stand an internal pressure of:

$$P_1 = \frac{2 \times \frac{.840 - .1875}{2} \times 40{,}000}{.1875} = 140{,}000 \text{ psi.}$$

$$P_2 = \frac{2 \times \frac{.675 - .125}{2} \times 40{,}000}{.125} = 17{,}600 \text{ psi.}$$

In Figure 2 part No. 3 will stand a pressure of:

$$P = \frac{2 \times \frac{.930 - .675}{2} \times 40{,}000}{.675} = 15{,}000 \text{ psi.}$$

The safety factor of the weakest point is:

$$S.O.F. = \frac{15{,}000}{4500} = 3.35$$

Actually this is much higher than 3.35 but for practical reasons it can be stated that the system is with a

S.O.F. of 3.35.

As can be seen from Figure 1, the system was connected to the pump through an L connection. This connection is a cast iron connection which will have a F.O.S. An ultimate of 10,000 psi is assumed. Actually this is maleable cast iron which will stand up to 40,000 psi.

$$P = \frac{2\ \underline{(3.35 - .840)}\ /\ 2}{.840}\ 10{,}000 = 29{,}000$$

So the F.O.S. of the L connection is:

$$F.O.S. = \frac{29{,}000}{4500} = 6.4$$

Although the system is a safe one, in order to avoid any explosions in case some air traps are present, the whole system was put in the cover shown in Figure 1.

The system did not prove itself efficient enough from the hydraulic leakage point of view. The maximum pressures obtained were 2,500 psiI. At this limit the connection of part 1 to part 3 started leaking. Thus no matter what pressure was applied, it was impossible to pass this limit.

It is felt sure that by using more skillful machining the system can be made to stand higher pressure without leaking. But the system has the following two defects:

1. The pump being an injector testing pump cannot pump large amounts per stroke. As a result, a small leakage cannot be overcome. That is, after a leakage occurs, then the leakage overcomes the amount of liquid pumped, thus reducing the pressure.

As it was in the case of the inside plug, where in that case although no leakage occurred in the system itself, the small leakage in the plug was enough to keep the pressure constant. Thus the plug could not be blown away.

2. The system has not an adequate means of disposing air traps.

It is felt sure that this system can be bettered, but again it is felt sure that a much better system can be designed.

For instance, the system shown in Figure 3 has the following advantages:

1. It is simple, thus eliminating many assembling difficulties.

2. From air traps point of view, it is safe.

3. Large amounts of liquid can be pumped at a time.

Design:

a. the diameter of the plunger is taken as $\frac{1}{2}$ inch.

b. it is designed to carry a maximum load of 10,000 psi. This load actually is double the maximum load used in experimentation.

c. the metal used should be steel. That is, its ultimate tensile strength should be 40,000.

d. the wall thickness using a F.O.S. of 4

$$t = \frac{5000 \times \frac{1}{2} \times 4}{2 \times 40{,}000} = \frac{1}{2} \text{ inch}$$

e. the outside diameter of the cylinder:

$$d = 1\tfrac{1}{2} \text{ inches}$$

f. the plunger should be an airtight fit to the cylinder.

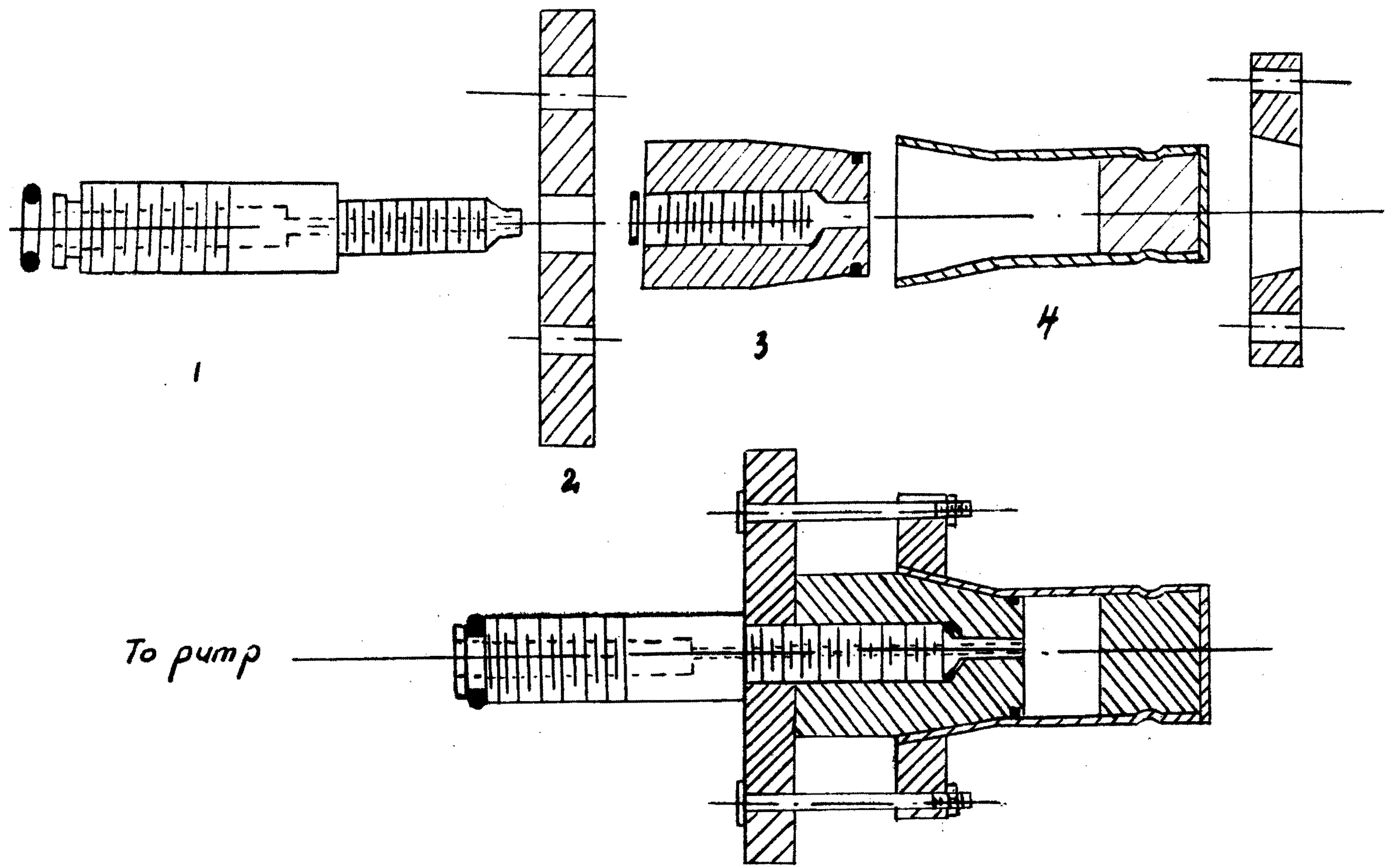
g. the total length should be at least six inches.

h. in order to make sure that no air is trapped in the system, a hole as shown in Figure 3 should be opened.

Now since the cross sectional area is r^2, any load applied by means of a press on the plunger can be reduced to its equivalent pressure.

Due to the fact that the maximum loads applied would not exceed the 5 tons, any press with a five ton capacity, and a dial indicator, will ideally perform this job.

Figure 1. Pressure Applying Unit



Sketch of pressure Applying Device
Fig 2.

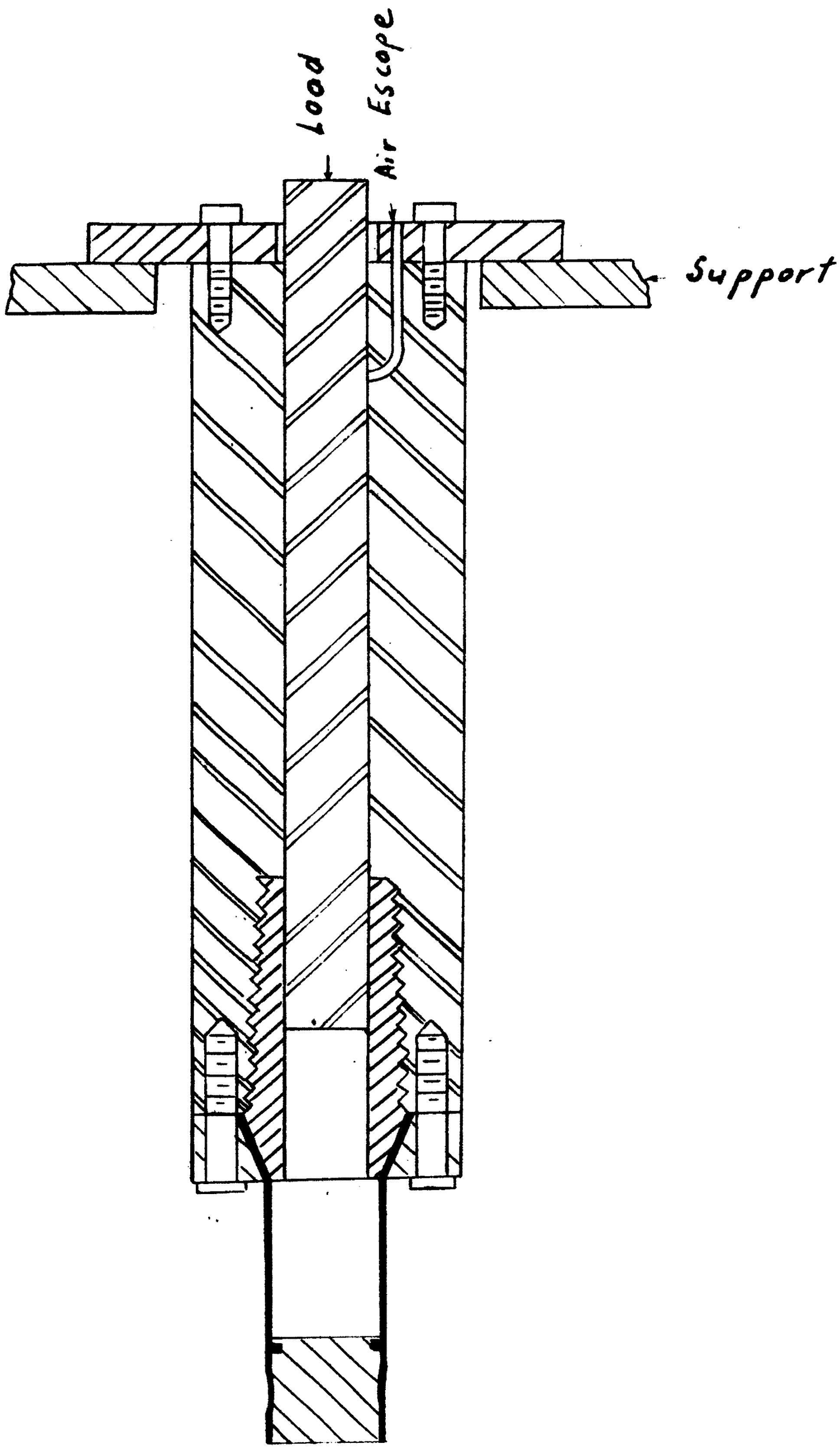


Fig. 3

Pressure Applying System

ADDITIONAL MINOR DETAILS

1. In order to minimize leakage in internal plugs, oil rings are used. In spite of the fact that the manufacturer states that those oil rings should not fit tight between the two walls, it is recommended that this tightness should be attained. In order to attain this tightness the oil rings' outside diameter should be, even if infinitesimally small, larger than the plug diameter. Then in order to intrude the plug into the pipe, the below illustrated method was used.

1a. Into the pipe a tapered cylinder is intruded. Figure 1.

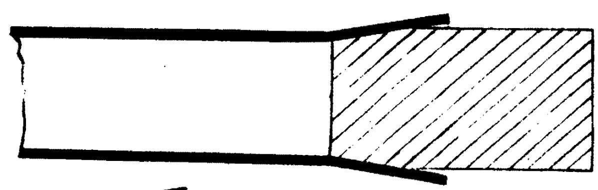


Fig. 1

1b. The intruded cylinder is removed and the actual plug is put in. Figure 2.

1c. The pipe pulled through the tapered ring. Figure 3.

As a result the seal is squeezed between the pipe and walls.

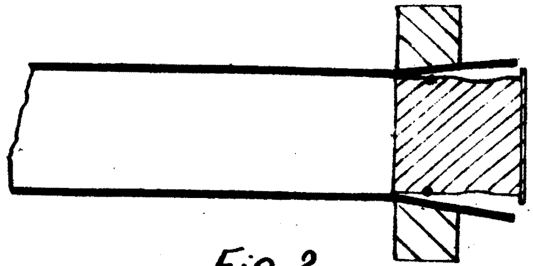


Fig. 2

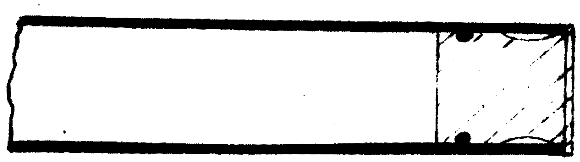


Fig. 3

One thing to which some attention should be paid is not to taper the pipe much. There is a possibility that it will shear due to its low unit elongation, and besides, the more tapered it is, the harder it is to pull it back.

2. In the case of the outside plugs, it is one of the hardest jobs to form the crimps by the rubber method. So a mechanical means should be used. But again, to use a com-

bination of the tapered and the inside plug will be ideal, that is, if the whole advantage of d^2/D^2 ratio, and of the traverse reactions is maintained while a means to hold the plug against longitudinal minor reactions is found that would be ideal. If a metal whose thermal unit elongation at low temperature is used, it should not exceed the critical temperature of the 24S-T3 aluminum alloy. See Figure 4.

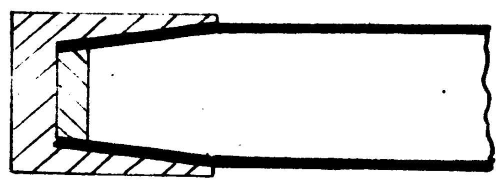


Fig. 4

3. In the case of the outside plugs, to close both ends is not possible. But for the case of the tapered, the method shown in Figure 5 can be used.

The tapered part of the plug inside the pipe can be pulled in to the pipe. Then the bolt can be screwed out.

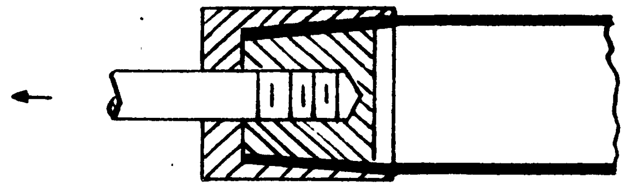


Fig. 5

CONCLUSIONS

1. For market and production purposes mechanical plugs are much superior to those made by rubber or hydraulic pressure.

2. Although by changing the crimp radius, crimp depth, and number of crimps it is possible to form 100 percent efficient inside plugs to tensional loads, due to the total diametral elongation it is doubtful whether it will be possible to reach anywhere near this limit when hydraulic loads are used.

3. Tapered and intruded plugs are inherently leakproof plugs. They are ideal to hydraulic loads.

4. To form the crimps of the outside plugs by rubber pressure is almost impossible due to the locking effect.

5. The outside plugs from stress analysis to hydraulic loads point of view are suitable, but from production and general practical efficiency point of view they are not recommendable.

Out of the investigations made it can be concluded that the tapered and the intruded plugs are much superior to the rest. From production and stress analysis to hydraulic loads point of view they are ideal. The object of this investigation being to find closed ends to thin walled vessels, it is felt definitely certain that they are recommendable for this function.

About the internal plugs it can be said that for small pressures, that is as ends for closed vessels containing liquids, they are very practical. In other words for vessels which are a mere container of liquids those closed ends find a very practical application.

Appendix

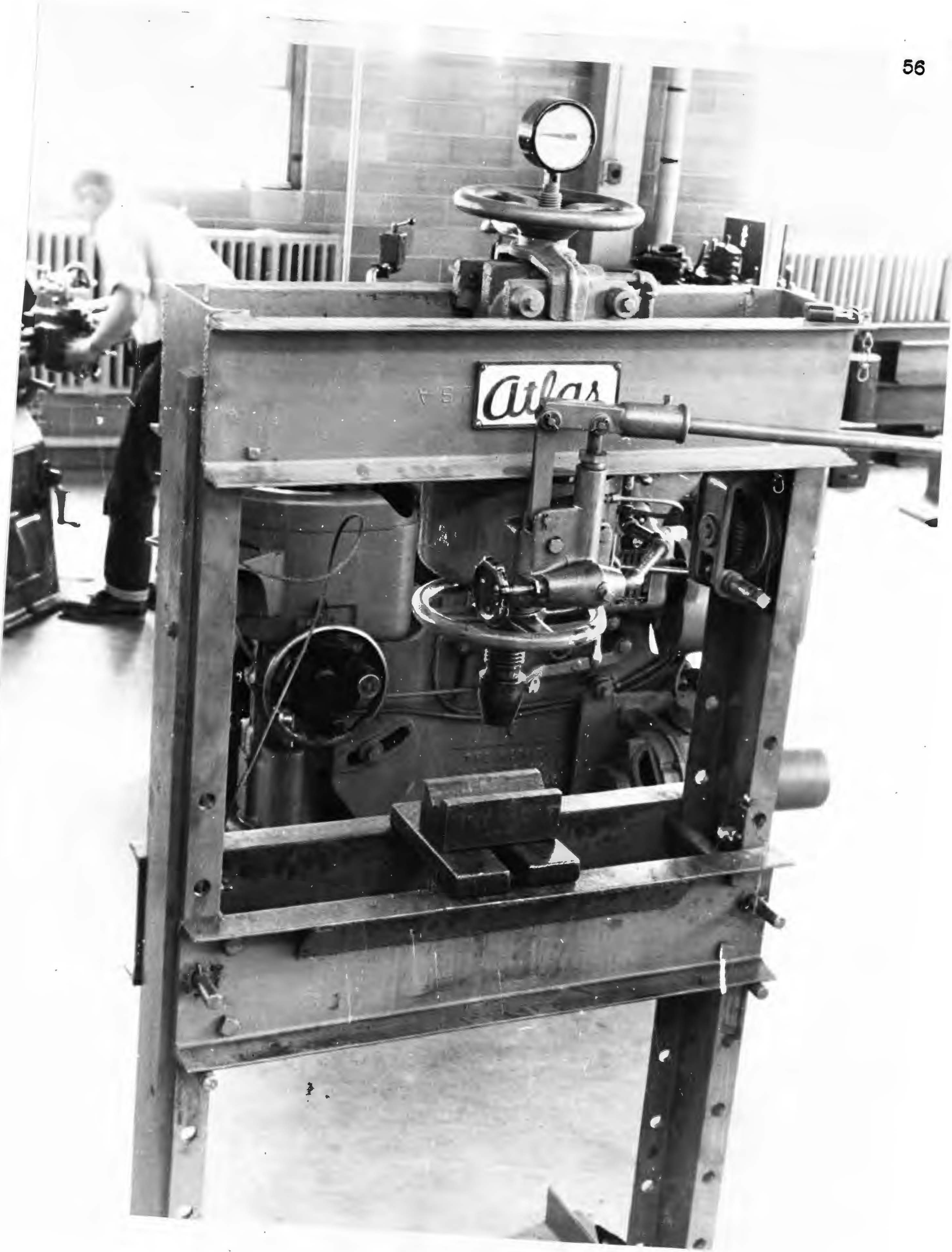


Figure 1. Fifty Ton Hydraulic Press

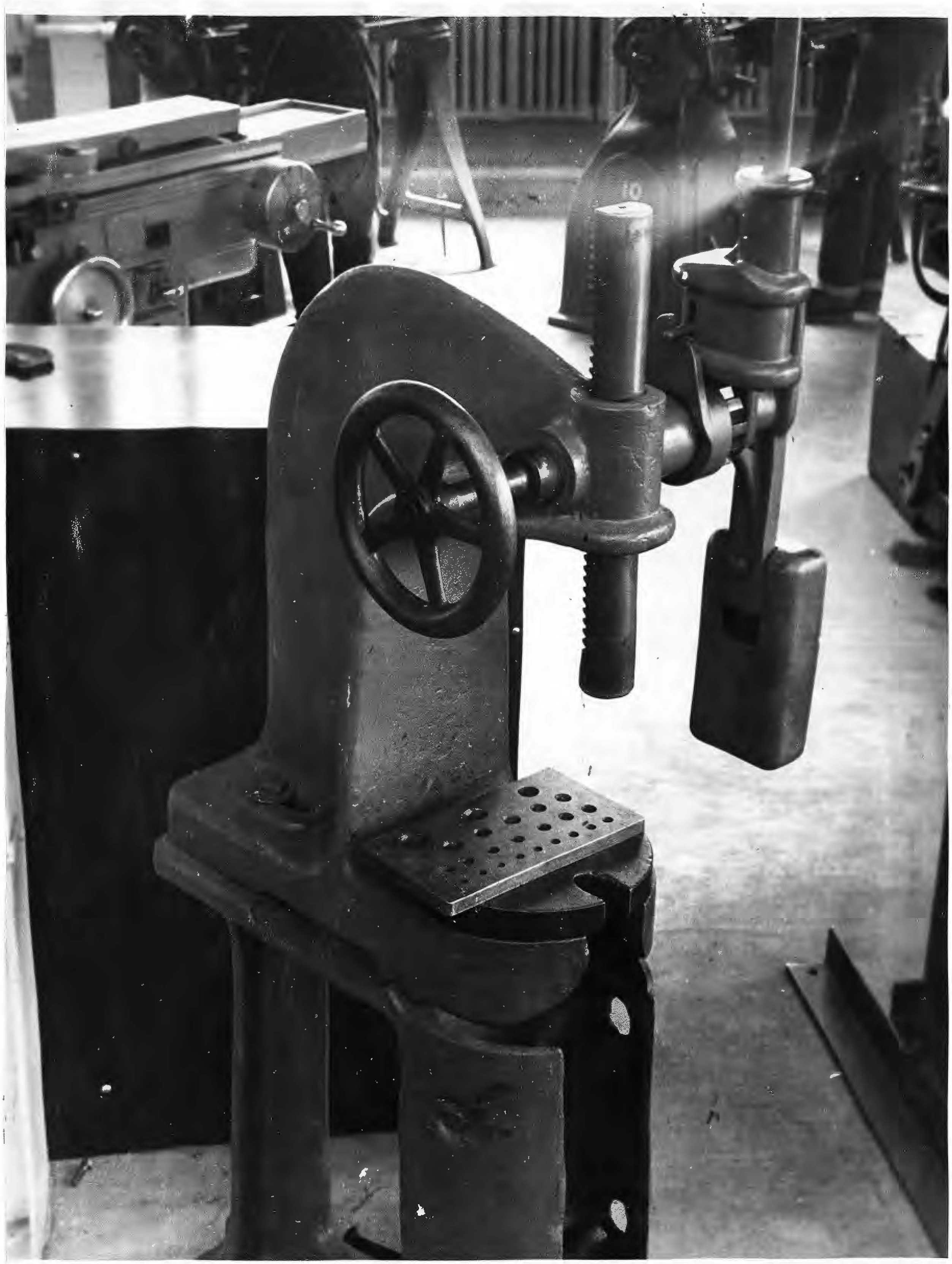


Figure 2. Hand Press

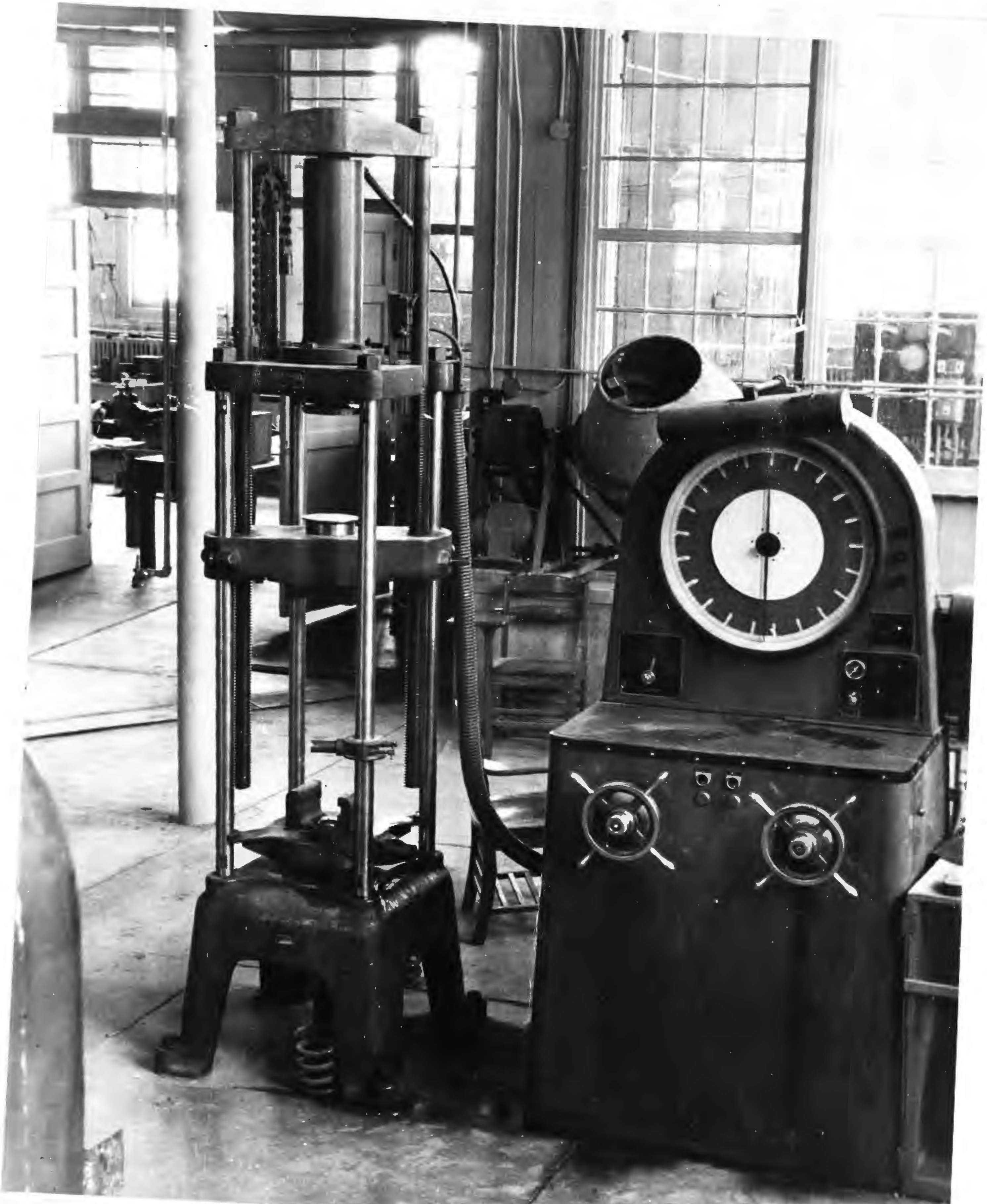


Figure 3. Tension Machine

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VITA

The author, Zafiri Bersu, was born on September 28, 1925, in Nausa, Macedonia. His family moved to Istanbul, Turkey, when he was one year old and settled there. His elementary schooling was received in Istanbul. He entered Robert's College of Istanbul, Turkey, in 1939. In 1944 he selected the field of mechanical engineering and in June, 1949, he graduated with the degree of Bachelor of Science in mechanical engineering. After his graduation he was employed by the High-Ways Department of Mersin, Turkey, as a Mechanical Engineer of the 52nd Asphalt Project.

In January, 1950, he was drafted in the army, and after eight months in the Armored Section of the Reserve Officers' Academy of Turkey, in October 1950 he graduated with the rank of 2nd Lieutenant. In January, 1951, he was sent to Korea, and there he served in the first lines of the Eighty Army. He got his promotion to 1st Lieutenant in April, 1951, and in September, 1951, he was rotated back to Turkey. His active reserve service thus being completed, he was transfered to inactive reserve.

In October, 1951, he joined the High-Ways Department of Turkey, and was employed in the Spare Parts Section in Ankara. When he was a student in the Reserve Officers' Academy, he took an examination given by the Institute of International Education, and thus gained the privilege of a return trip to the United States. He came to the United

States in April, 1952, and since he has continued his post graduate studies in the Missouri School of Mines and Metallurgy, Rolla, Missouri.