INDENT REMOVAL AND EVALUATION IN ALUMINUM SPRINKLER IRRIGATION TUBING

Thesis for the Degree of M. S.

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Dale Earl Kirk

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This is to certify that the

thesis entitled

Indent Removal and Evaluation
in Aluminum Sprinkler Irrigation Tubing

presented by

Dale E. Kirk

has been accepted towards fulfillment
of the requirements for

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Ernest H. Kidder
Major professor

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INDENT REMOVAL AND EVALUATION IN ALUMINUM SPRINKLER IRRIGATION TUBING

By

Dale Earl Kirk

A THESIS

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INTRODUCTION

Sprinkler-type irrigation with the use of portable, surface pipe has been practiced on a commercial scale in various parts of the United States since the early 1930's (10). Until after World War II various types of steel surface pipe were used. Steel boiler tubing was used to a limited extent in the earlier installations, but the greater share of the pipe was either spiral or straight seam welded steel with wall thicknesses from 16 to 10 gage. Protection against rusting was generally provided by galvanizing, although some tubing was asphalt coated. While this tubing was much lighter than standard pipe of comparable diameter, it was still quite resistant to damage from denting and kinking. Deterioration was generally due to rusting of the base metal where the protective zinc or asphalt had been loosened or lost.

At the end of World War II the war-born aluminum reduction and extrusion plants began producing an extruded thin-wall aluminum tubing which rapidly became the dominant material used for sprinkler irrigation. The rapid increase in the production of aluminum irrigation tubing can be seen from the figures in Table I.

TABLE I

U. S. PRODUCTION OF ALUMINUM SPRINKLER IRRIGATION TUBING (13)

Year	Miles of Tubing	Year	Miles of Tubing
1946	230	1950	4800
1947	400	1951	4800
1948	2000	1952	7000
1949	2300	1953	9000

The availability of production potential in the field of aluminum products at the close of the war was a very timely stimulus to the expansion of the sprinkler irrigation practice. The advantages of sprinkler irrigation under many conditions had already been well demonstrated in many parts of the United States but the labor involved in moving the heavier steel tubing was a dominant factor in deterring farmer acceptance of this method of water application. The light weight and apparent general corrosion resistance of extruded aluminum tubing made it a comparatively ideal material and its application brought about a very rapid expansion in the acreage covered annually by sprinkler irrigation.

Future expansion should not be hampered by material shortages since aluminum is the third most abundant material on the earth's crust (2).

Virtually all the aluminum sprinkler tubing produced prior to 1952 was extruded and designated as alloy 63S-T6. Since 1952 at least one manufacturer in the state of Washington has been marketing a fabricated welded tubing from a harder alloy which is somewhat more resistant to field damage.

Due to the lower stiffness of aluminum and the thinness of the tubing wall, aluminum irrigation tubing is much more susceptible than steel to field damage in the form of dents and kinks. With a modulus of elasticity and a density each approximately one-third that of steel, the deflection of an empty section of aluminum tubing supported at the ends will be approximately equal to the deflection of a steel tube of the same dimensions. When loaded with water the deflection of the aluminum is much greater.

The types of damage most common are dents due to radial impact and kinks due to excessive bending. Much of the serious damage from dents is caused by accidentally driving over the tubing with automobiles, trucks, tractors and other farm implements. Dents of various sizes occur when the sprinkler systems are left in the field with horses or cattle. Smaller dents are often incurred in the handling of the tubing when moving from one irrigation set to the next. A large dent or flattened area will cause the pipe to kink or be off-set unless the ends are restrained.

Damage due to kinking may occur also when long sections loaded with water are lifted abruptly or are allowed to drop and strike a support at one or both ends with no support near the center.

These various indents in the original shape of the tubing always cause some change in the normal pattern of flow of water through it. More apparent to the farmer is the inconvenience of handling kinked sections or sections sufficiently dented to cause a bend or an offset in the tube. These sections tend to twist in the hands when carried, will not lie in the proper position for coupling, and will not stack up well with the other sections in a moving wagon or truck rack.

A repair may be effected by cutting out the damaged section and fitting a new coupler to one of the cut ends. This repair is expensive and leaves two shorter sections which are again a problem to handle with the other uniform length sections. For a three inch diameter line the cost of a coupler is approximately equivalent to the cost of six to ten feet of tubing.

Personal interviews with representatives of two west coast regional distributors of irrigation tubing indicated their cognizance of the dent problem. Their attempts to repair dented sections employed exterior malleting of the tube while a projectile-shaped mandrel or "mouse" was forced against the dent on the inside. This process is slow and costly and necessarily involves excessive bending and stretching of the metal near the regions malleted.

From random field observations, tears and cracks in the tubing were very uncommon. It was believed that this type of damage could best be handled by cutting out the damaged portion and installing a coupler. In a few cases repair by welding might be practical where aluminum welding service is available. This type of repair is not practical on the average farm.

The major problem in restoring damaged tubing to its full degree of usefulness involved the repair of sections damaged by dents and kinks or bends.

REVIEW OF LITERATURE

The deformation of circular cylindrical shells within the elastic limits of the material has been analyzed by Timoshenko (15) for concentrated diametral loading as well as for the general case of deformation. His analysis employs the theory of inextensional deformation of the center line of the shell cross sections. This theory, credited to Lord Rayleigh, by Hermes (8) assumes that the strain along an axial line is zero at a point equidistant from the ends of the shell, and that shear across this line at this point is also negligible. Hermes used electrical resistance strain gages to check the strains in steel tubing subjected to concentrated diametral loading. His measurements taken along an axial line 90° from the point of loading showed reasonable conformance with the inextensional theory of right circular cylindrical shells. While this inextensional theory method of analysis may be used to predict strains along the shell when it is loaded within the elastic limits of the material, it appears to be of little value in predicting other than border strains when deflections have entered the plastic range.

Timoshenko (14) discusses the buckling of thin cylindrical shells under the action of uniform external lateral pressure and various types of axial loading including eccentric loading to induce bending. While the buckling failure appeared to result in damage similar to that inflicted by a straight edged tool impressed tangentially to the tube, no mention was made of the equivalent external force or energy involved in making such an indentation. No references were found that treated the problem of predicting the amount of elastic and plastic deformation of a thin-walled cylinder under a concentrated load or the amount of force required to restore the metal back to its original position.

Partial yielding and plastic flow is discussed by Nadai (12) for thick-walled hollow cylinders subjected to internal pressures. His discussion of the yielding of a die-cast aluminum tube under internal pressure gives an indication of the general behavior of the metal stressed slightly beyond the yield stress in a smoothly generated cylinder but is of little value in predicting internal pressures required to restore a dented cylinder to its original shape.

Experimental determination of the behavior of copper tubing subjected to various combinations of internal pressure and axial loading are given by Espey (5). His investigation of the various load conditions showed that a thin-walled tube subjected either to "pure" internal pressure (plane strain) or to balanced biaxial tension will fail by instability at a considerably smaller circumferential strain than the longitudinal strain which occurs on necking of a tensile test specimen made from the same metal. A tube subjected to pure internal pressure will become unstable at only approximately one half of the circumferential strain as a tube subjected to pure longitudinal tension. His results further indicated that the longitudinal strain at which a circumferential neck occurs should decrease as the internal pressure superimposed on the longitudinal tension increases.

While experimental work may have been reported in various trade literature, the only references found in the formal literature which dealt directly with indent removal was an article in Engineer (3). The article described a machine which employed a tapered plug fitted to a piston with a ten-inch stroke. Dents were removed from light alloy tubes or spools by forcing the plug through the tubes by pneumatic power.

OBJECTIVES

The objectives about which this study centered are listed as follows:

1. Evaluate the hydraulic power losses in aluminum sprinkler irrigation tubing due to dents.
2. Develop means for satisfactory dent removal.
3. Determine the extent of recovery or efficiency of the dent removal method. This involved a study of the recovery of both hydraulic characteristics and recovery of the strength of the tube in bending.

EVALUATION OF HYDRAULIC POWER LOSSES

Scope of the Study

Observations of hydraulic power losses were limited to three-inch tubing. This is the size commonly used for lateral lines in sprinkler irrigation. Since these lines are moved once or twice each day they are the most susceptible to damage from handling. The flow rates observed ran from 39 gpm to 229 gpm which covered the range of flow rates consistent with sound engineering design practice as indicated by Gray (7) for this size tubing.

Equipment

A steady supply of water was obtained with a four-inch turbine type pump rated at 250 gpm when discharging against a 22 foot head. The pump discharged directly into a three and one-half foot length of four-inch standard pipe which was in direct alignment with the sections to be tested. Attachment to the test section was made by use of two specially designed couplers. These couplers were made to permit rapid and easy attachment of the test sections. They also tightly sealed the ends of badly oblated sections as well as sections retaining their circular shape. The seal was formed by placing a four and one-half inch long by one-quarter inch diameter rubber sewing machine belt over the tubing near each end as shown in Figure 1.

As the tubing was forced into the tapered socket in one end of the coupler the rubber belt was rolled and compressed between the tube and the coupler forming a tight, flexible seal. Longitudinal thrust on the couplers due to water pressure was taken up by a light chain which was hooked to the fittings at each end of the tube. C-type clamps fitted with a nipple and a rubber gasket seal were used to attach the manometer hoses to the sections being tested. A closed-top or inverted U-tube manometer using water as the indicator fluid was used for checking static differential heads up to 42 inches. The general coupler, U-tube, and C-clamp arrangement is shown in Figure 2. For heads beyond 42 inches a mercury U-tube manometer was used as shown in Figure 3. The 90° V-notch weir shown in Figure 4 was used for determining rates of flow.

Procedure

Dented tubing samples for the hydraulic study were obtained by crushing five foot long sections of three inch diameter tubing in a small hydraulic press and by driving over them with tractors and trucks. The hydraulic press was fitted with a die-block with a one-eighth inch radius

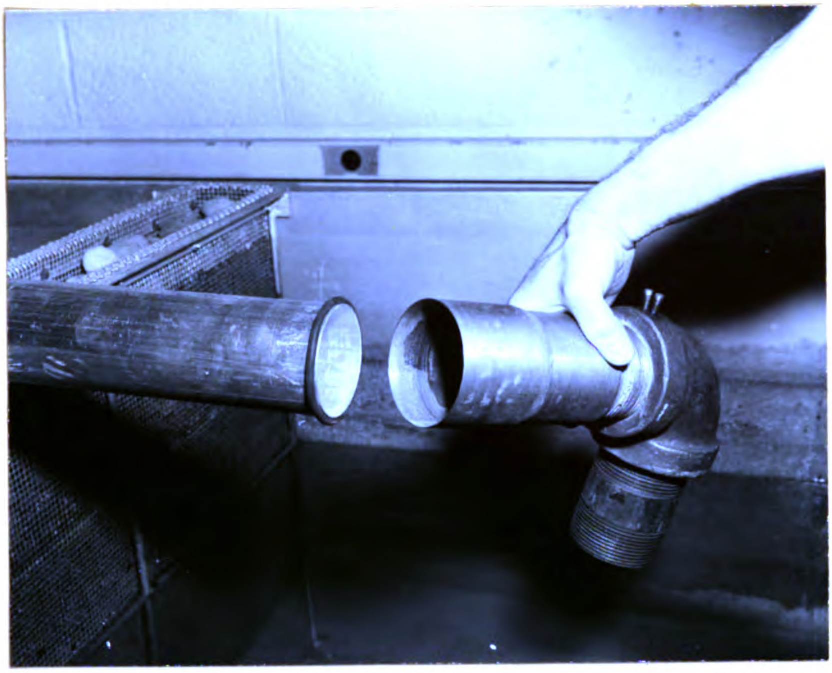

Fig. 1. Specially constructed quick coupler using a round rubber belt between mating sections for a seal.

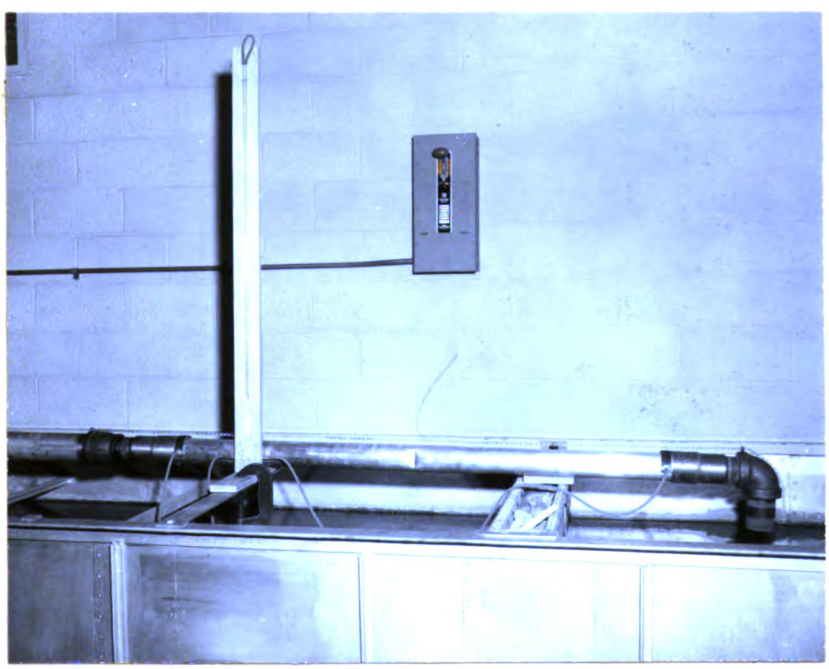

Fig. 2. A test section of tubing with couplers, manometer, and C-clamp taps in place. The tie chain was used for taking coupler end thrust.

Fig. 3. Dented section under test. Note the use of the mercury manometer in the foreground for measuring the higher friction heads.

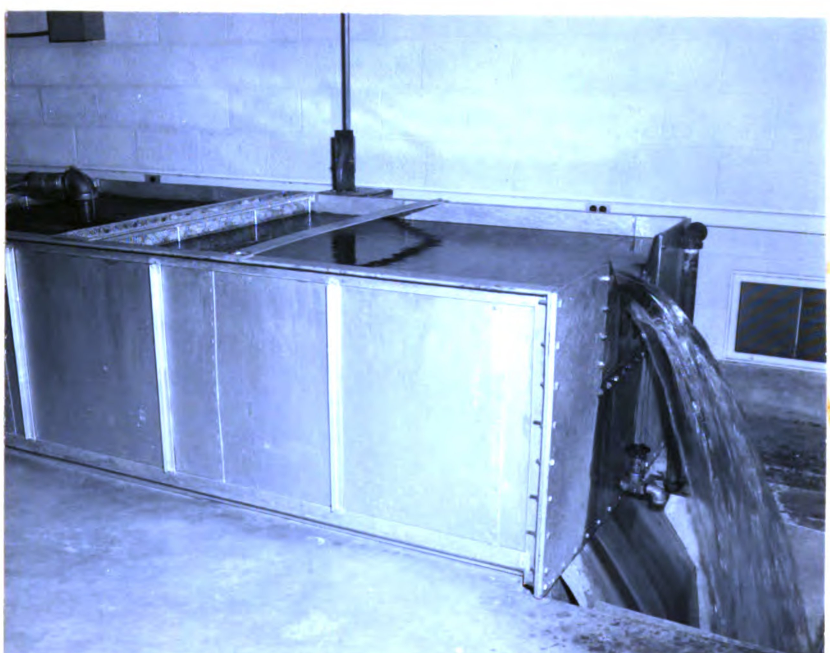

Fig. 4. Weir and stilling basin used for determining rates of flow through test section.

leading edge. This block was placed to form an impression in the tubing at right angles to its longitudinal axis as shown in Figure 5. For the shallower dents, the impression left by this edge roughly simulated a buckling failure of the tubing due to excessive bending load. A denting series consisted of five test sections each impressed a progressively greater amount with the die block as shown in Figure 6. Load and deflection data were taken at suitable increments while the sections were being dented.

Random samples of sections damaged by field equipment were obtained by driving over the test sections placed on a concrete floor. Figures 7 and 8 show the truck and one of the tractors used for denting or crushing the sections. The total denting force or weight of the wheel was measured by a highway scale as shown in Figure 9.

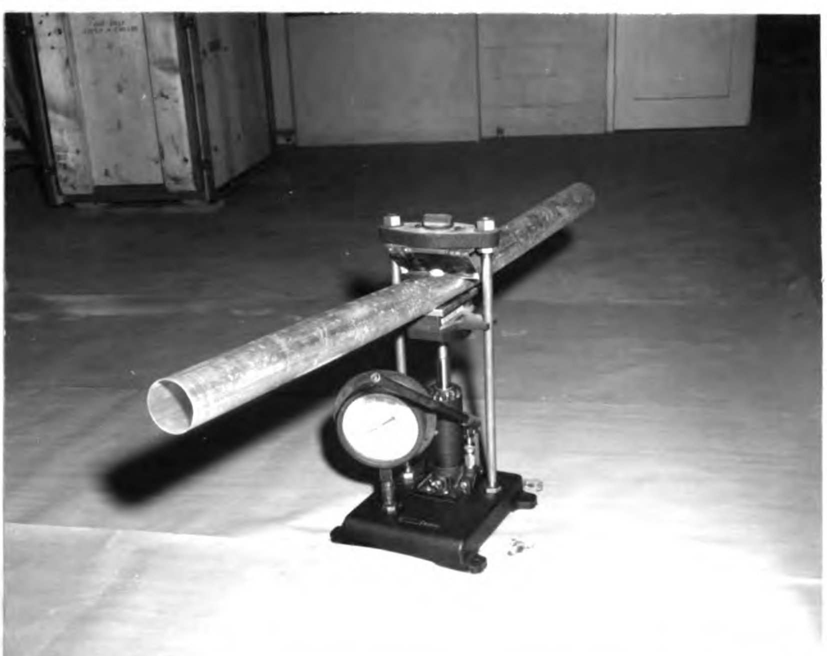

Fig. 5. Laboratory type hydraulic press used with one-eighth inch radius die block for transverse denting series. An 870 pound force was required to make this dent.

Fig. 6. Five sections dented in 0.6 inch increments with one-eighth inch radius die block.

Fig. 7. Truck exerting 1360 pounds on test section.

Fig. 8. Tractor exerting 1450 pounds on test section.

Fig. 9. Highway-type scale used for determining wheel weights.

A standard test length of 4.5 feet between manometer tap holes was selected for all test sections. For the tapping of pipes which have a much greater wall thickness Addison (1) suggests that a 3/32 inch tap hole is as small as can usually be recommended. He further states, however, that ideally the hole should be of the smallest diameter that will not get choked and that will not impose excessive damping. A one-sixteenth inch diameter hole was used and the burrs were carefully removed after drilling.

With the test section coupled to the pump, the clamps were fitted over the tap holes and the hoses scavenged to prevent errors in static head readings due to air inclusion. This was done by partially opening the pump discharge valve and allowing the water to run freely through the hoses which were disconnected from the manometer. When the mercury U-tube was required, the hoses were completely drained and air was used as the medium over the manometer fluid. Care was taken to run the water through each section of aluminum tubing in the same direction each time it was tested. This was done for two reasons. First, the turbulent flow pattern past the damaged area could possibly result in greater friction head in one direction than in the other. Secondly, the static head readings from the tap holes could be affected by the direction of flow if the hole directions or hole edges were slightly irregular.

The formula for discharge over a 90° V-notch sharp-crested weir is given by King (11) as $Q = 2.52H^{2.47}$ where Q is given in cubic feet per second and H is in feet of head. Barr (4) listed the formula as $Q = 2.48H^{2.48}$ for the same type weir. Differences in the equation constants result from differences in smoothness of the weir face and shape of the approaching channel. Since these conditions will be somewhat different in every case, each weir should be individually calibrated for most accurate results. Data taken by Gillette (6) on the M.S.C. Agricultural Engineering Department weir were used to calculate a discharge formula as $Q = 2.15H^{2.32}$. The formula was used in the form $Q = 965H^{2.32}$ where Q was given in gallons per minute. The weir zero was determined by clamping a temporary hook gage near the weir and adjusting it with a carpenter level until it was at the same elevation as the bottom of the weir notch. The tank was then filled to the level of the temporary hook gage and the reading on the vernier scale noted when the permanent hook gage in the stilling well was adjusted to the water level.

A minimum of five sets of readings were taken for determining the friction characteristics of each test section. A minimum of ten minutes was allowed between each valve setting and the time of reading to allow the flow system to reach an equilibrium.

Results

Due to the limited number of samples used and the method of selecting these samples, it could not be assumed that the observations recorded were representative of a normally distributed population of values. The values observed in the study did, however, appear to cover the range of restrictions to flow that might conceivably be encountered in the field for three inch tubing.

Sections number three and twelve were taken as being representative of the maximum damage that might possibly be tolerated by the farmer. A rate of flow of 170 gallons per minute was taken as a typical maximum rate of flow for three inch tubing consistent with good irrigation system design practice. System operation was assumed to be twenty hours per day and thirty days per month. With an overall pump and motor efficiency of fifty percent and a power cost of $0.02 per kilowatt hour, the additional power cost per month per dent was found to be $0.275 per month for a dent of number three order and a $0.23 per month for a dent of number twelve order. These costs were on the basis of no relief of restriction when operating pressure was applied.

DEVELOPMENT OF MEANS FOR SATISFACTORY DENT REMOVAL

The requirements for any methods or equipment developed for the removal of dents were set down as follows:

1. Must be adaptable to various lengths and diameters of tubing.
2. Must accommodate various types of loose and rigid couplers.
3. The ends of the tubing can not be restrained since longitudinal movement is required for straightening.
4. Equipment must be sufficiently portable to be readily carried to the farm.
5. The maximum cost of repair must be much less than the cost of a repair coupler for a damaged section.

The portability factor virtually eliminated the use of any outside restraining form that might be employed with either a mechanical expansion or hydraulic expansion of the tube from the inside. From the standpoint of simplicity, the hydraulic expansion method without a restraining form appeared to meet the best requirements set down above.

Hydraulic radial expansion of monotube gun barrels beyond the elastic limit without restraining forms is practiced by the United States Navy for pre-stressing the outer wall metal against the inner wall metal. Upon firing, this results in much less tube expansion since a greater thickness of metal is subjected to a more nearly uniform stress. Hydraulic radial expansion against an outer forming shell is used in the steel industry in the final forming of large sizes of fabricated steel tubing.

To determine the maximum pressures that might be used on aluminum tubing, the yield and ultimate pressure values were computed. The yield and ultimate stress values of alloy 63S-T6 aluminum are listed as 31,000 and 35,000 pounds per square inch respectively by Hoyt (11). Since the yield strength value was based on two percent permanent elongation or set, a value of 30,000 pounds per square inch was chosen for calculating internal pressures within the elastic limit. The computed pressures to produce these stresses in a thin walled cylinder are listed for the various sizes of tubing in Table II. The values were computed from the equation

$$P = \frac{2St}{D}$$

where P is the gage pressure in pounds per square inch

S is the mean circumferential stress in pounds per square inch

t is the wall thickness in inches

D is the inside diameter in inches.

TABLE II

GAGE PRESSURES TO PRODUCE YIELD AND ULTIMATE STRESS IN ALUMINUM TUBING

Nominal Tubing Size O.D.	Wall Thickness	Inside Diameter	Pressure to cause 30,000 lb per sq in stress,	Pressure to cause 35,000 lb per sq in stress
in.	in.	in.	lb/sq. in.	lb/sq. in.
2	0.050	1.900	1579	1842
2.5	0.050	2.400	1250	1458
3	0.050	2.900	1034	1207
4	0.050	3.900	769	897
4	0.054	3.892	832	971
4	0.062	3.876	960	1120
5	0.052	4.896	637	743
5	0.062	4.876	763	890
6	0.062	5.876	633	739
7	0.072	6.856	630	735
8	0.083	7.834	636	742
8	0.094	7.812	722	842
9	0.094	8.812	640	747
10	0.109	9.782	669	780

For three inch diameter tubing the gage pressures should be restricted to less than 1000 pounds per square inch if the tubing is not to be permanently enlarged. The force on each end plug at this pressure would be 6636 pounds or more than three and one fourth tons. Corrosion of the metal with use, variation in wall thickness in manufacturing, and Espey's findings were factors considered in determining maximum working pressures to use for dent removal.

The requisites of the hydraulic expansion method for repairing sprinkler tubing include a ready water source, a set of proper end plugs, an accurate pressure gege, a small-volume high-pressure pump, and miscellaneous fittings.

The most critical item in the hydraulic expansion method is the end plug design. The main requisites of such a design were listed as follows:

1. Fast and easy to install and extract.
2. Should not damage the tubing.
3. Near perfect water seal.
4. Feed water in and out rapidly.
5. Provide for complete scavenging of air from the tubing.
6. Must work past all types of couplers.
7. Light weight.
8. Inexpensive.
9. Parts replaceable.

The two plugs, which were used, both embodied the same principle of operation but were of different type construction as shown in Figure 10. These plugs had been developed for preliminary investigation previous to the formation of this project. They were composed of a steel body with a slotted, tapered hole which received a tapered bronze expansion core. The core was fitted to an internal threaded drive for forcing it into the plug body. As the core was forced into the body, the segments of steel about the slotted tapered hole expanded radially against the inside of the tubing. The tubing was restrained from expanding by the use of an

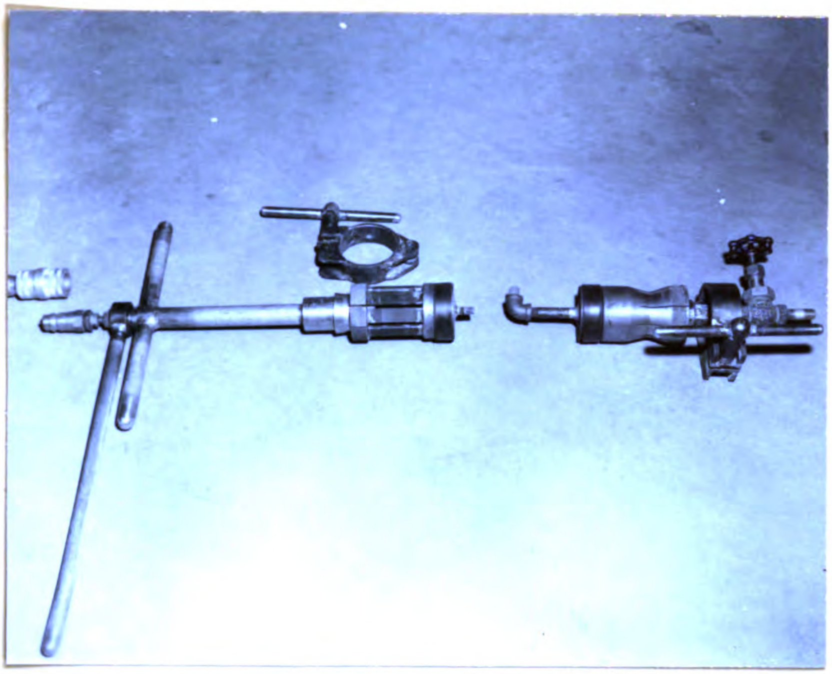

Fig. 10. Two types of plugs used for sealing the ends of three inch diameter tubing.

adjustable band or clamp on the outside. Water was admitted through a pipe in the center of one plug and air scavanged out through a pipe in the center of the other plug. A riser nipple bled off the air from the pocket formed near the top of the tubing as it was filled with water. The seal was made on each plug by a rubber cup which fit against the plug body. To insure positive expansion of the rubber cap against rough or slightly misshapened tubing, a steel mushroom core was fitted inside the cup to serve as an expander. This expander could be controlled from outside the tubing whenever the cup failed to seat itself readily.

The plugs used were constructed in only the three inch size and generally satisfied quite well the mechanical requirements previously set down. Their cost of construction was undetermined but it was believed that the general type of design changes which must always take place in adapting an experimental model to volume production would result in a satisfactory manufacturing cost. In preliminary destructive tests of tubing the plugs held pressures up to 1100 pounds per square inch. For three inch tubing with a 2.9 inch inside diameter this meant resisting an axial load of 7267 pounds force. To obtain sufficient wall friction to withstand this force, the threadlike grooves on the outer surface of the plug body segments were embedded slightly into the inside surface of the tubing as seen in Figure 11. These groove impressions were only a few thousandths of an inch deep and not considered detrimental to the tubing.

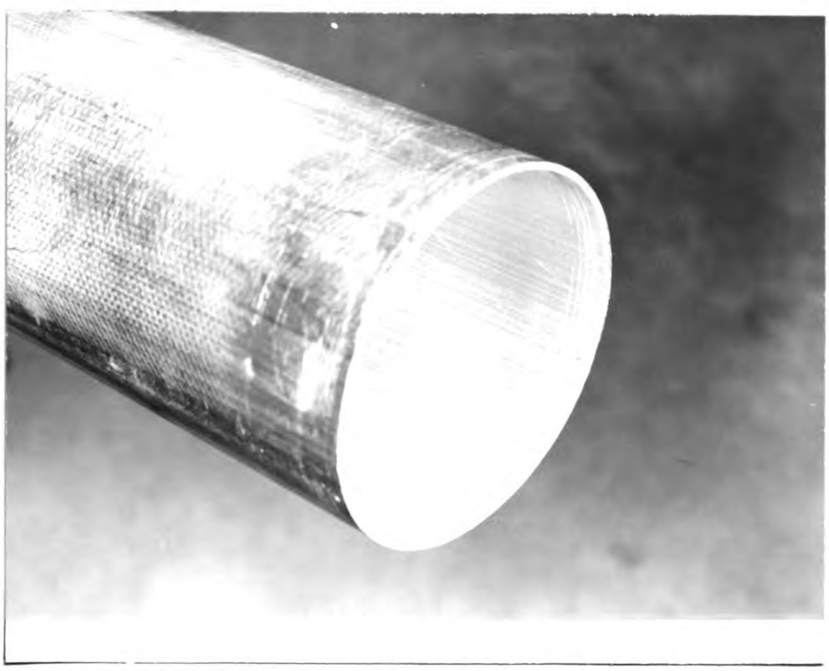

Fig. 11. End of tubing showing the groove impressions left by the end plug.

DETERMINATION OF RECOVERY OR EFFICIENCY OF REPAIR

The efficiency of the hydrostatic radial expansion method of repair was checked in three ways. First, the extent of recovery of the original tubing diameter was noted for several random types of dents. Second, the extent of recovery of the original hydraulic friction characteristics was measured. Third, a replicated experiment was set up for determining the extent of recovery of the original bending strength of the tubing.

Diametral Recovery

Diametral recovery was observed on the transverse dent or kink series shown in Figure 6 and also on representative samples damaged by the truck wheel and the tractor wheel. The measurements taken represented the minimum setting which would allow an outside caliper to pass across any part of the dented area as the tubing was being straightened. Figures 12 and 13 plotted from the hydrostatic expansion data show the typical recovery pattern as hydraulic pressure is applied. For transverse dents or kinks complete recovery of the original outside diameter was not possible without the use of localized forces such as might

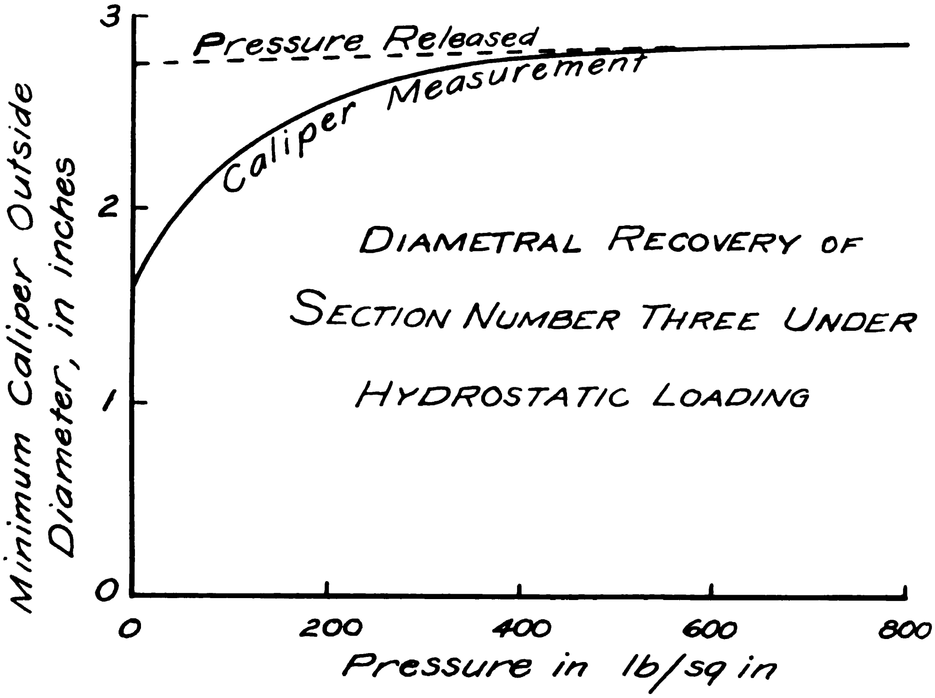

Fig. 12. Diametral recovery curve for section number three.

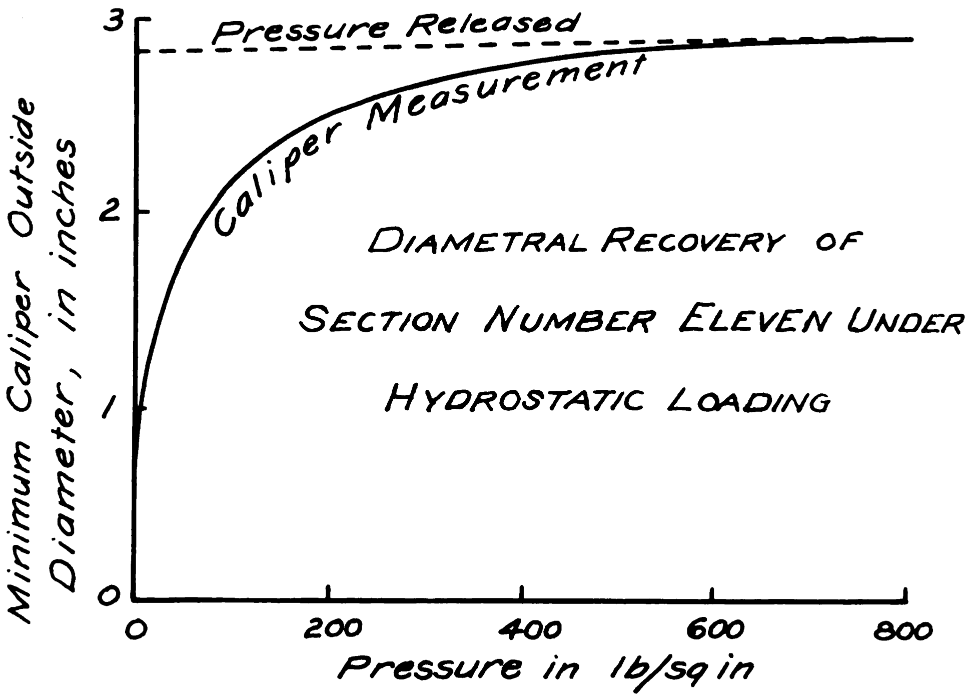

Fig. 13. Diametral recovery curve for section number twelve.

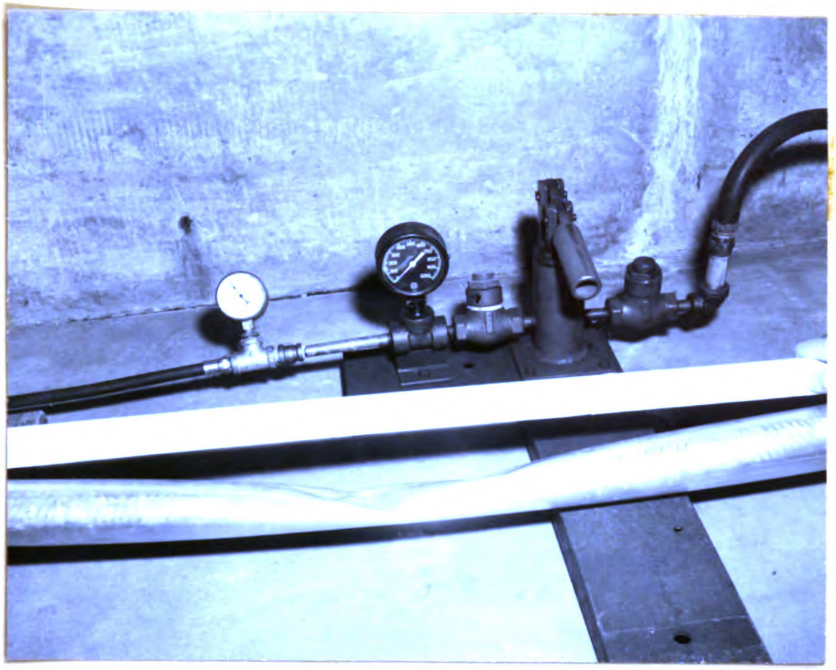

Fig. 14. Tractor-dented section number twelve shown with hand-operated water pump and pressure gage before pressure was applied.

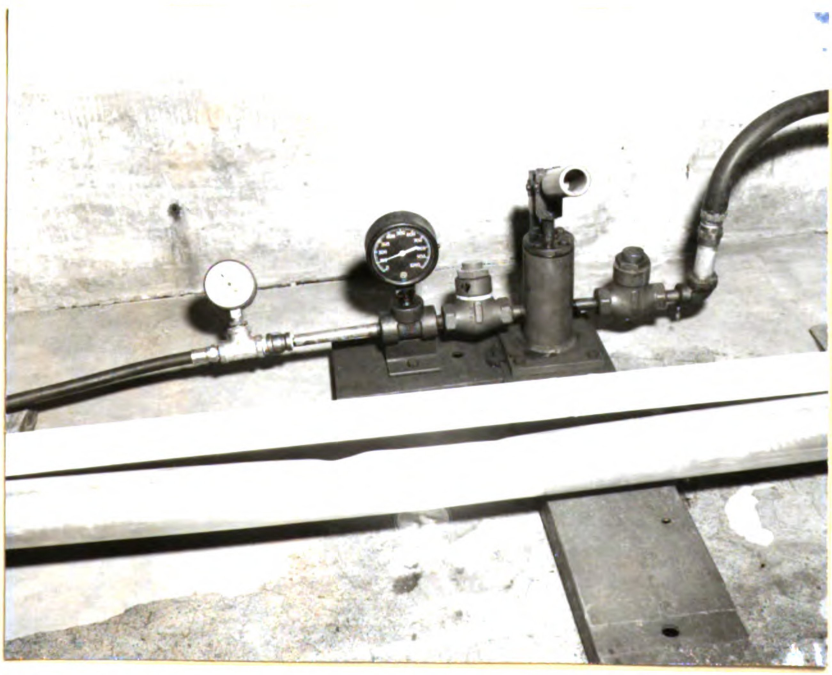

Fig. 15. Section number twelve with 100 pounds per square inch pressure applied.

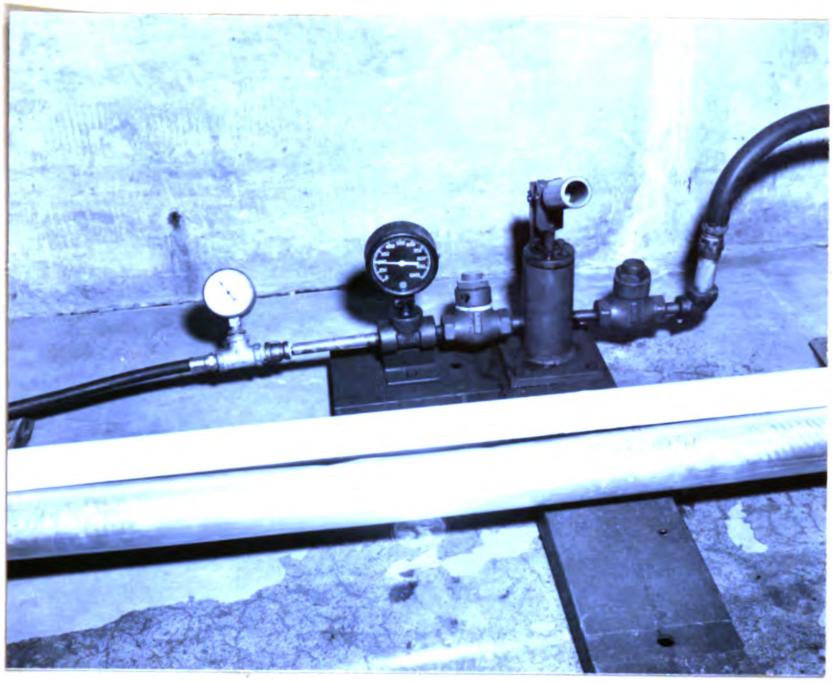

Fig. 16. Section number twelve with 200 pounds per square inch pressure applied.

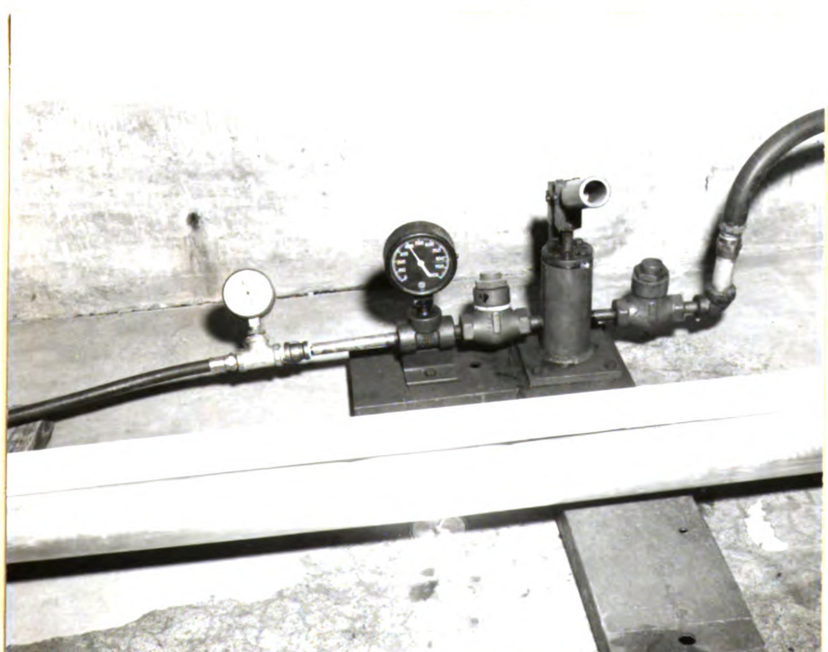

Fig. 17. Section number twelve with 400 pounds per square inch pressure applied.

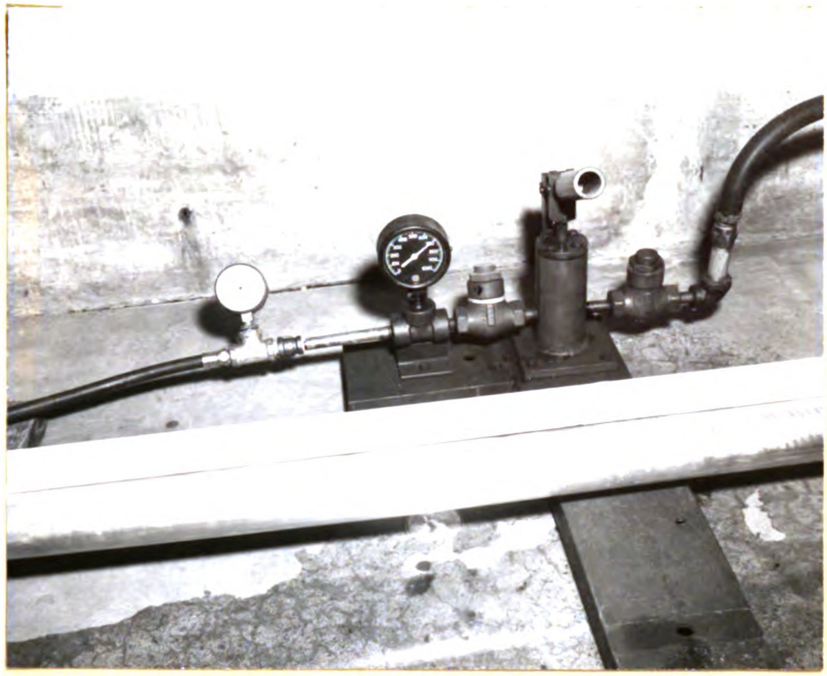

Fig. 18. Section number twelve with 700 pounds per square inch pressure applied. Note the slight amount of residual kink or offset as shown by the gage stick held along the top edge of the tubing

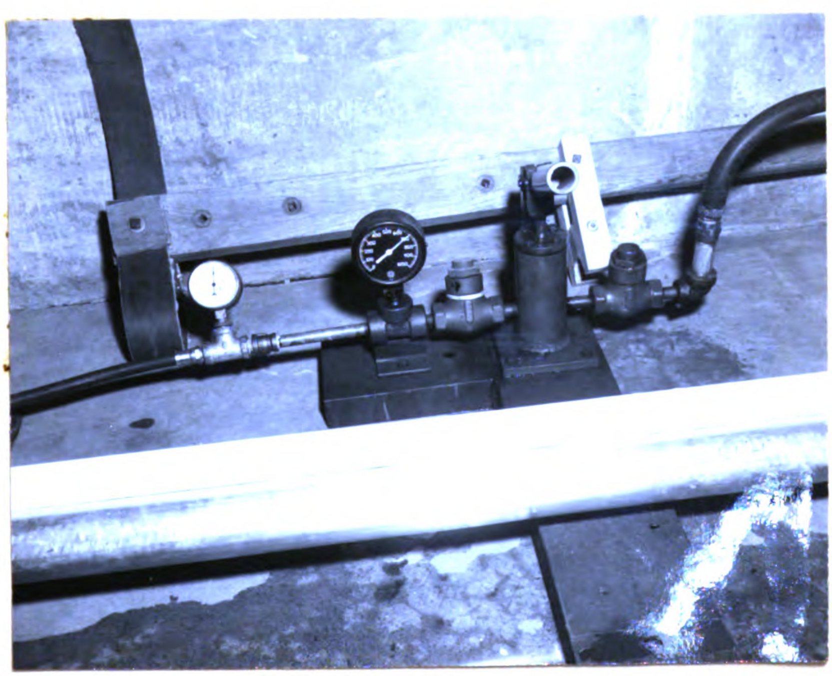

Fig. 19. Section number twelve after the residual kink has been removed under pressure by use of the bending bar shown in the background.

be obtained by malleting against a mandrel placed inside the tubing. Flatter types of dents with less severe bending at the edges were recovered virtually one hundred percent. Hydraulic expansion brought the dent out as shown in the series represented in Figure 14 through Figure 18. The slight offset or kink left as shown in Figure 18 was eliminated by the use of the crudely improvised bending bar shown in the background in Figure 19. By applying the bending bar and creating a bending moment on the tube while it was under a pressure of 700 psi, the metal on the convex or outer side of the bend could be stressed rather easily beyond the elastic limit without danger of buckling the metal on the concave side. This reduction of the bending moment required to permanently deform the tubing can be seen from the following computations:

$$M = \frac{S\,I}{C}$$

where M = bending moment in pound inches

S = stress on extreme fiber in pounds per square inch

I = moment of inertia of the tubing about its neutral axis in inches^4

C = distance from neutral axis to extreme fiber in inches.

Taking:

$$I = \frac{\pi}{64}\,(D^4 - d^4)$$

$$= \frac{\pi}{64} (3.00^4 - 2.90^4)$$

$$= 0.504 \text{ in.}^4$$

S $=$ 30,000 pounds per square inch yield stress

C $=$ 1.5 inches

Then:

$$M = \frac{(30,000)(0.504)}{1.5}$$

$$= 10,080 \text{ inch pounds}$$

By subjecting the tubing to an internal pressure of 700 pounds per square inch, the extreme fibers were prestressed axially as shown below.

$$S_p = \frac{P}{A}$$

where

S_p = prestress in pounds per square inch

P = total axial force acting in pounds

A = cross section area of stressed metal in square inches.

Taking:

P = (pressure) (projected area)

$$= (700) \left(\frac{\pi d^2}{4}\right)$$

$$= (700) \left(\frac{\pi}{4}\right) (2.9)^2$$

$$= 4624 \text{ pounds}$$

$$A = \frac{\pi}{4}(D^2 - d^2)$$

$$= \frac{\pi}{4} (3.00^2 - 2.90^2)$$

$$= 0.4635 \text{ square inches}$$

Then:

$$S_p = \frac{4624}{0.4635}$$

$$= 9976 \text{ pounds per square inch tensile prestress}$$

This prestress reduced the required bending moment as shown below.

$$M = \frac{(30,000 - 9976)(0.504)}{1.5}$$

$$= 6728 \text{ inch pounds}$$

10,080 - 6728 = 3352 inch pound reduction of necessary bending moment.

Besides giving this reduction of 33 percent in the necessary bending moment, the prestress in tension further protected the convex side from buckling under the compressive stresses as the bending bar was applied.

Hydraulic Recovery

The same equipment and procedure used in determining hydraulic power losses due to denting were used in determining the extent of hydraulic recovery after the sections were straightened.

Resistance to the flow of water in pipes may be expressed in the general equation

$$H = K\,Q^m$$

where

H is the friction head in feet of water

K is an empirical constant which includes pipe length and diameter

Q is the rate of flow in gallons per minute

m is an empirical constant.

By taking the logarithm of both sides of the equation, the exponent, m, may be expressed as a coefficient of the logarithm of Q.

$$\text{Log } H = \text{Log } KQ^m$$
$$= \text{Log } K - \text{Log } Q^m$$
$$= \text{Log } K - m\text{Log } Q \cdot$$

Thus by plotting the logarithms of the values of the equation on rectangular coordinates or by plotting the values directly on logarithmic paper the locus should describe a straight line.

The observations for friction head and rate of flow of the various sections of tubing were plotted on logarithmic paper. Figure 20 is a sample of the plot for one section. The constant, m, which becomes the slope of the line when the data is plotted on logarithmic coordinates, may be taken directly from the curve. Since the logarithm of one is zero, the ordinate value at an abscissa value of one on logarithmic coordinates corresponds with the zero intercept value of the curve on rectangular coordinates. Thus, the

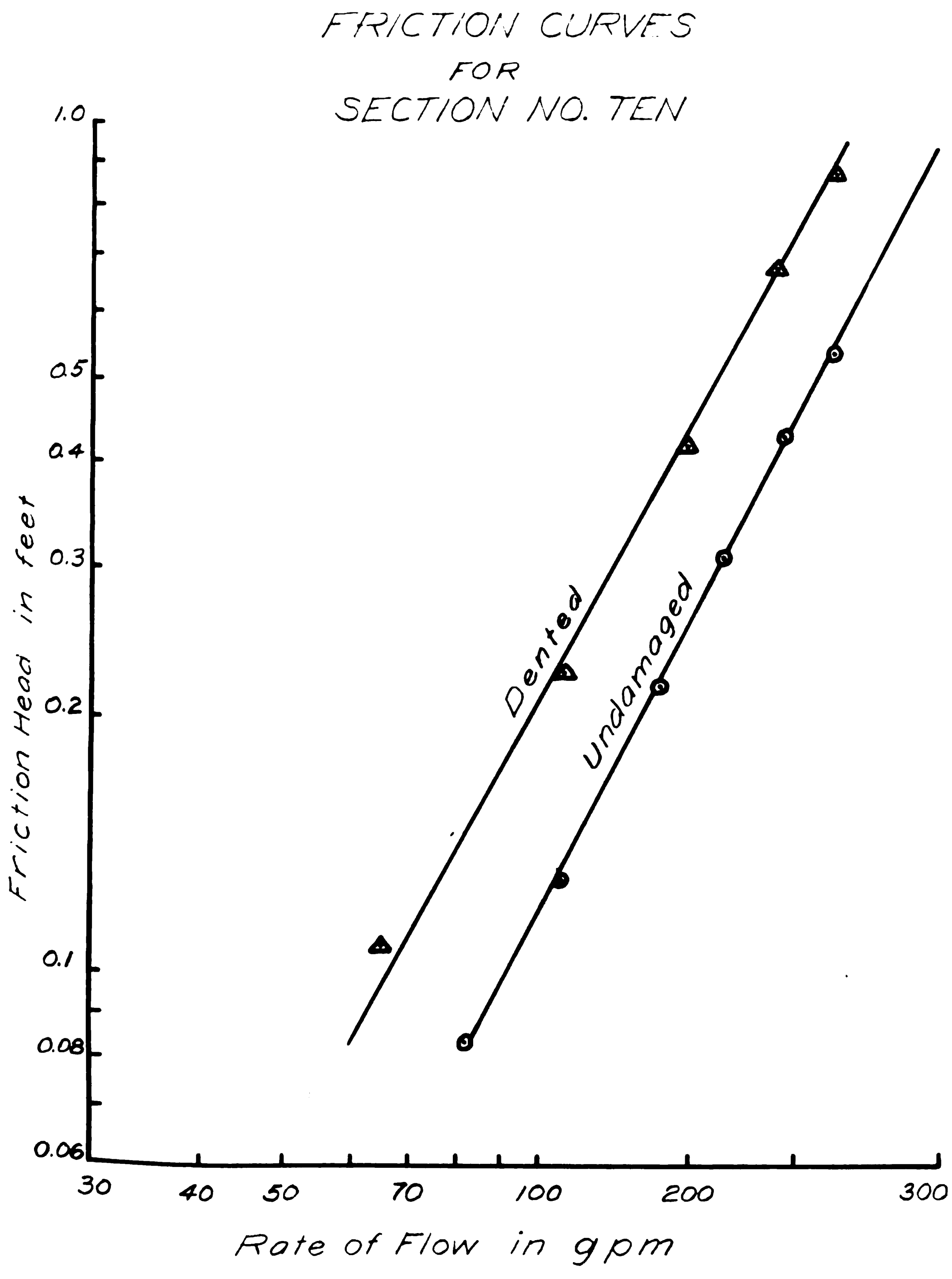

Fig. 20. Curve showing relationship of friction head and rate of flow for tubing section number ten.

value of H may be read directly from the curve at an abscissa value of one on logarithmic coordinates.

For the range of large flow rates and small friction heads involved, however, the value of H determined graphically would have to be obtained by projecting the line past two cycles on the logarithmic paper. Since this would greatly magnify any error made in choosing a line through the plotted points, the method of least squares of logarithms was used for determining the equation constants.

The exponent m was determined for each tubing section for each treatment by the following formula:

$$m = \frac{n\Sigma(\log Q \log H) - (\Sigma \log Q)(\Sigma \log H)}{n\Sigma(\log Q)^2 - (\Sigma \log Q)^2}$$

where n = number of observations of H and Q.

The values of K were determined by taking the antilog of the value found by the following formula:

$$\text{Log } K = \frac{\Sigma \log H}{n} - m \frac{\Sigma \log Q}{n}$$

Table III shows the emperical friction constants obtained from the above equations for the tubing before it was damaged and after it was damaged and straightened.

TABLE III

K AND m FROM GENERAL FRICTION EQUATION $H = KQ^m$
DETERMINED BY METHOD OF LEAST SQUARES

Section No.	Pipe Treatment					
	Original		Dented		Straightened	
	$Kx10^{-4}$	m	$Kx10^{-4}$	m	$Kx10^{-4}$	m
1	0.747	2.1020	1.840	1.9217	3.332	1.7993
2	3.597	1.7943	2.730	1.8844	2.915	1.8474
3	4.347	1.7544	5.306	1.8724	4.031	1.7622
4	2.333	1.8997	15.230	2.0038		
5	0.807	2.1048	31.410	2.0563		
7	1.191	2.0107	1.450	2.2580	1.007	2.0437
9	2.766	2.7153	2.029	1.8817	1.525	1.9521
10	2.468	1.8778	10.200	1.7012	1.834	1.9255
11	2.624	1.8734	2.529	2.3201		
12	1.058	2.0345	5.810	1.8324	1.237	1.9992

To give a clearer picture of the comparative friction of tubing sections under various treatments, a sample friction head value was computed from the above tabulated data for each 4.5 foot test section. Each friction head was computed on the basis of 150 gallons per minute water flow and was listed in inches of water as shown in Table IV.

TABLE IV

FRICTION HEAD VALUES COMPUTED FROM EQUATION
CONSTANTS IN TABLE III

	Computed Friction Head at 150 Gallons Per Minute, in Inches of Water		
Section	Original	Dented	Straightened
1	2.803	2.797	2.742
2	2.888	3.434	3.055
3	2.857	6.289	2.755
4	3.175	34.922	
5	3.071	93.696	
7	2.827	11.884	2.821
9	2.751	2.524	2.700
10	3.011	5.136	2.841
11	3.130	28.300	
12	2.830	5.645	2.772

A single classification analysis of variance showed that there was no significant difference between the friction values obtained for the original and the straightened sections.

Bending Strength Recovery

For any method of dent removal to be acceptable it should effectively restore the damaged tubing to near its original strength. Restoration of shape and hydraulic characteristics should not be considered sufficient if the reclaimed section must be given favored treatment in the field to prevent kinking at the originally damaged area.

A replicated experiment was set up for determining bending strength recovery of tubing sections damaged by a tractor wheel. Two groups of six samples each were used in the experiment. Each sample consisted of a section of new, three inch O.D. aluminum tubing five feet long.

The possibility was considered that there might be a significant difference in the mechanical properties of the two thirty-foot long parent sections of tubing from which the samples were cut. To bring out this difference, if any, and to compensate for its effect the six samples for each group were cut from one-half of each thirty-foot parent section. The six samples in one group were each crushed in as nearly the same manner as possible by driving a tractor rear wheel over them. Care was taken to start the tire tread contact in nearly the same position on each sample and to drive over each of them at a uniform rate of speed. The uniformity of the dent pattern obtained can be seen in Figure 21. Each of the dented sections was subjected to 750 pounds per square inch hydrostatic pressure and the residual offset of approximately 0.2 inch at the center was removed by the use of the bending bar. The slight residual oblateness of the tube in the dented region was removed by pressing the tubing between a pair of vise jaws lined with soft lumber. Figure 22 shows the sections ready for the bending test. The other group of six undented samples was used as a check. The comparative bending strength of the two groups was measured by third-point loading in a universal testing machine. To prevent collapse or premature buckling at the point of contact of the load, a pair of dumbbell-shaped wooden plugs was used. Figure 23 shows the wooden plugs and the saddle blocks which were used at the four points of load contact in the bending test.

Load and deflection readings were taken at deflection increments of 0.1 inch for each sample until it buckled. Figure 24 shows a sample loaded in the testing machine to near the point of buckling.

Figure 25, plotted from the bending test data, shows the comparative bending strengths of the three samples of undamaged tubing and the three samples of damaged and

Fig. 21. Six sections crushed by tractor rear wheel for bending strength test.

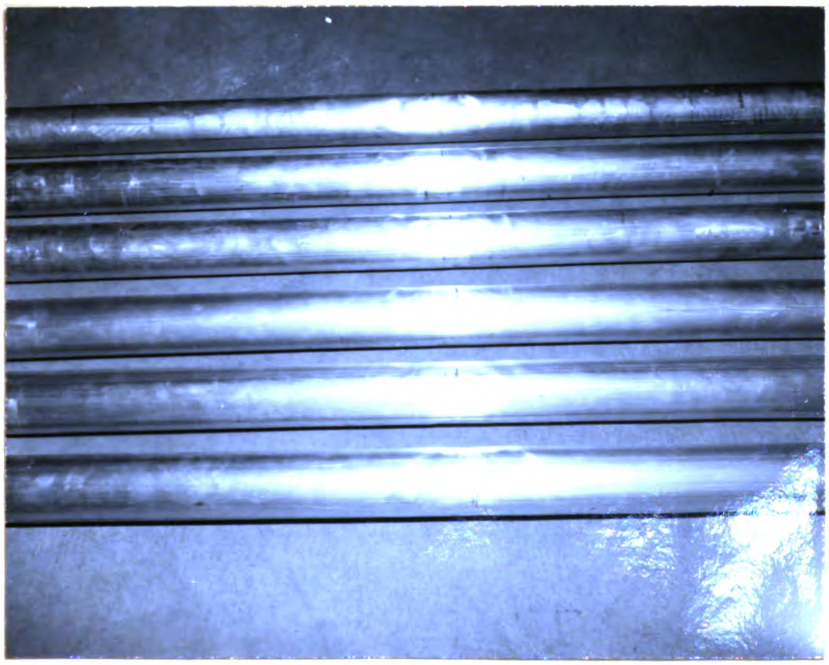

Fig. 22. Same six sections as shown above after straightening for bending strength test.

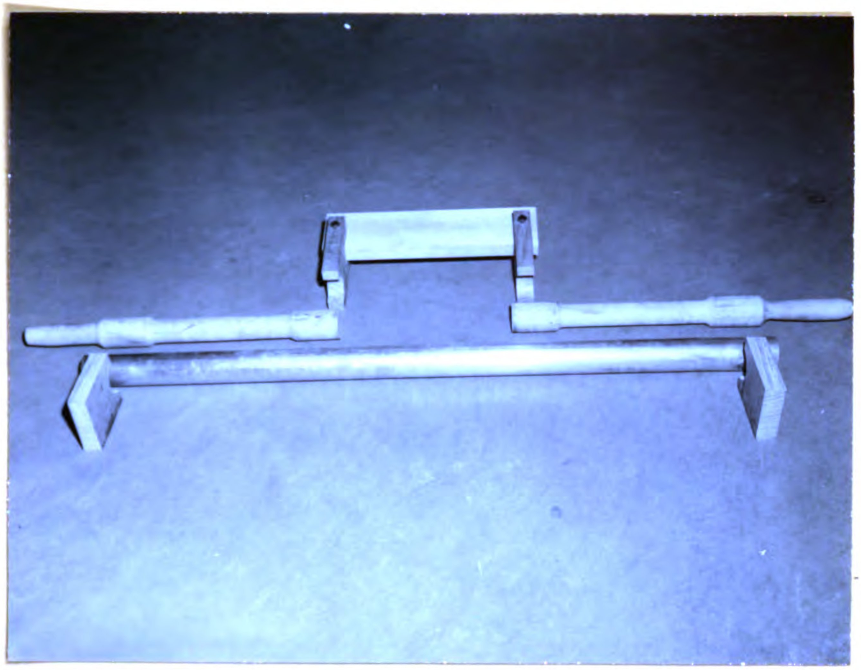

Fig. 23. Wooden plugs shown with saddle-type contact blocks for obtaining constant bending moment near the mid-section of the tubing.

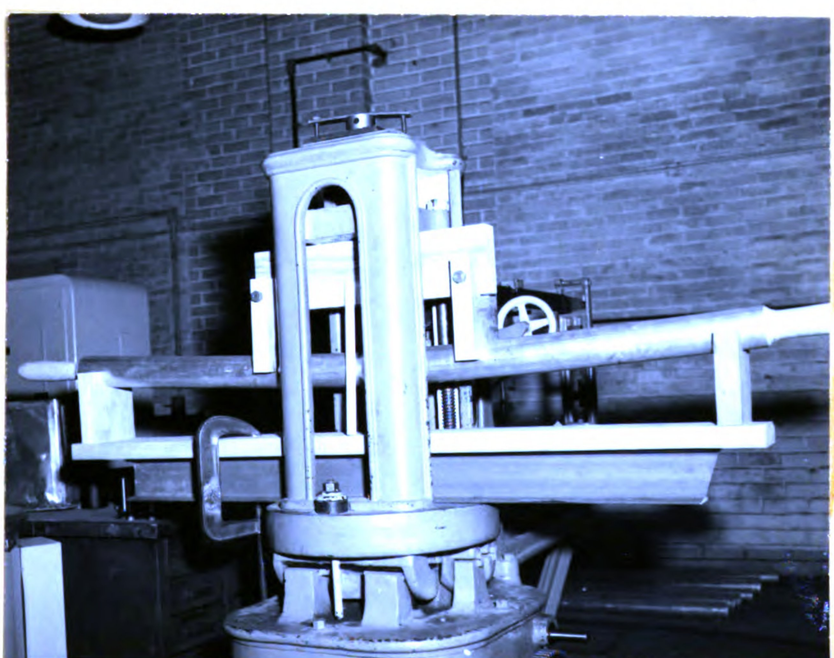

Fig. 24. Tubing sample bent to near the point of buckling in a universal testing machine. Note the wooden plugs inserted in the tubing to help prevent crushing.

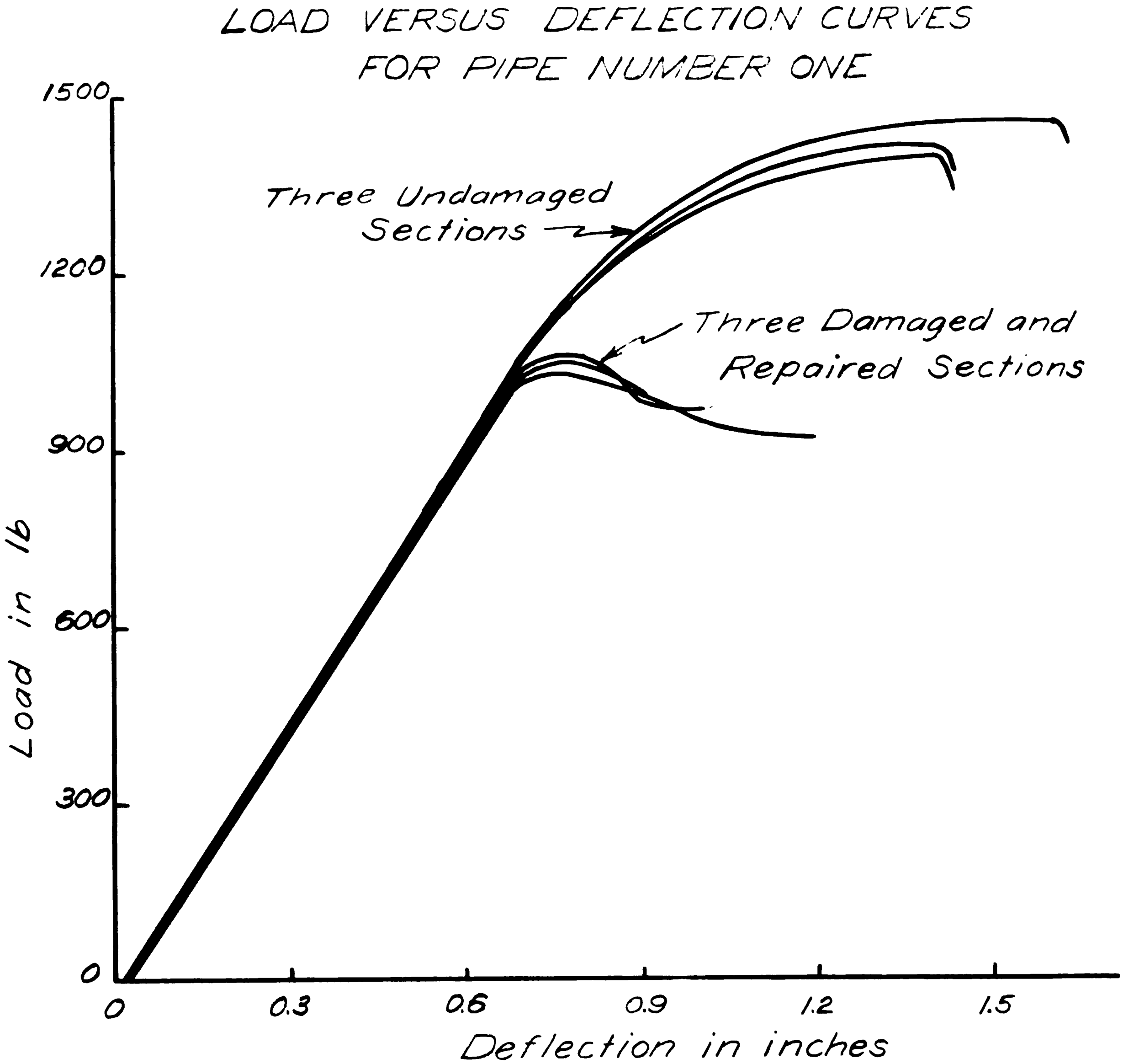

Fig. 25. Curves showing comparative bending strengths of three samples of damaged tubing and three samples of undamaged tubing taken from the first piece of parent material.

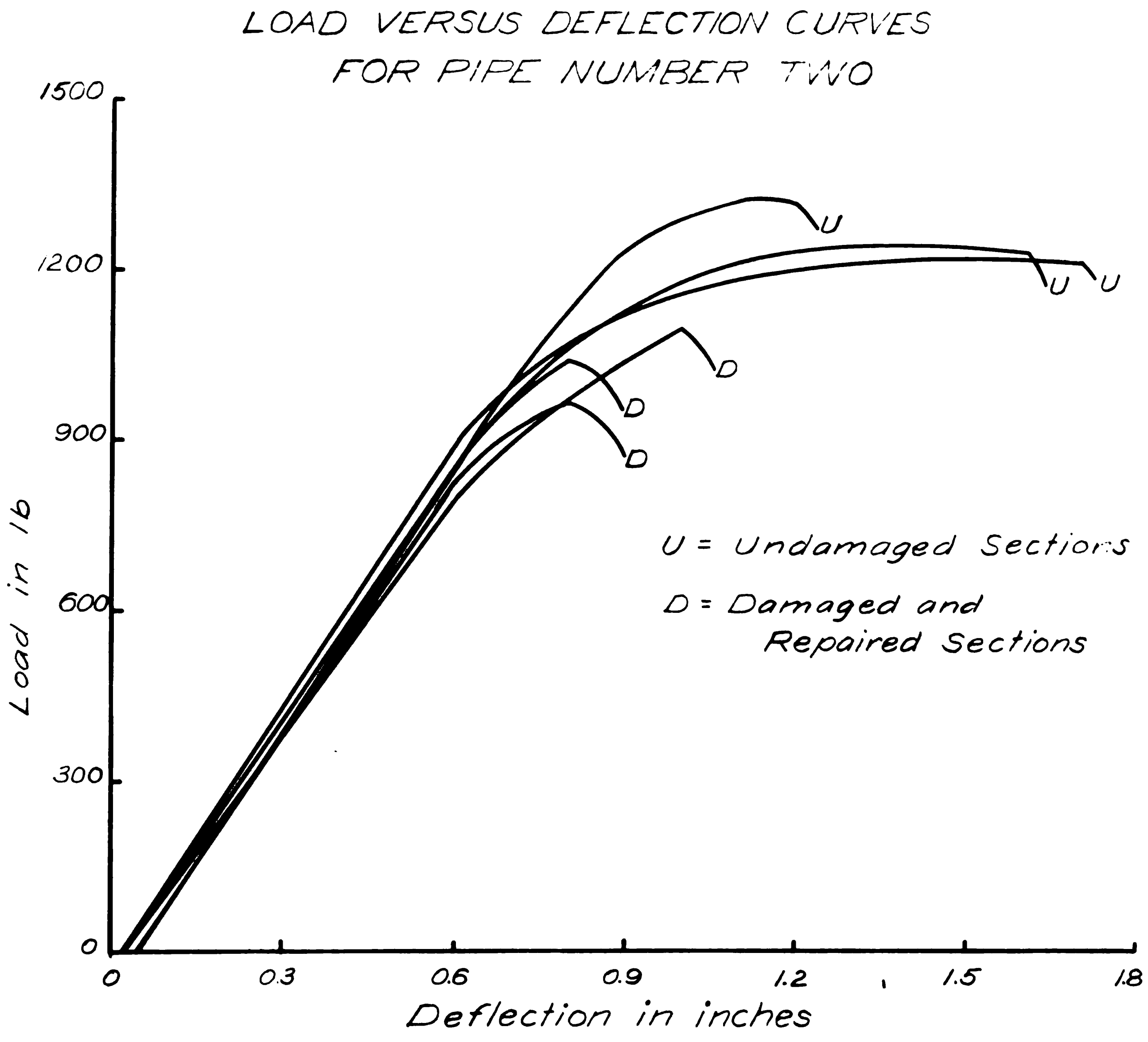

Fig. 26. Curves showing comparative bending strengths of three samples of damaged tubing and three samples of undamaged tubing taken from the second piece of parent material.

straightened tubing from one of the parent sections. Since the curves were plotted from the unadjusted data, they do not all fall through the origin. This displacement was due to slack in the linkage and the tendency for the various parts to shift slightly at the initial loading causing the first displacement reading to be somewhat erratic. It can be seen from the curves that the bending strength for the damaged sections was very near the strength of the undamaged sections up to the proportional limit of each. Beyond the proportional limit the damaged sections soon buckled while the undamaged sections were bent into a permanent curve to some extent before buckling occurred and the strength fell off drastically. The type of buckling failure can be seen in Figure 27 which shows all twelve samples after the bending test.

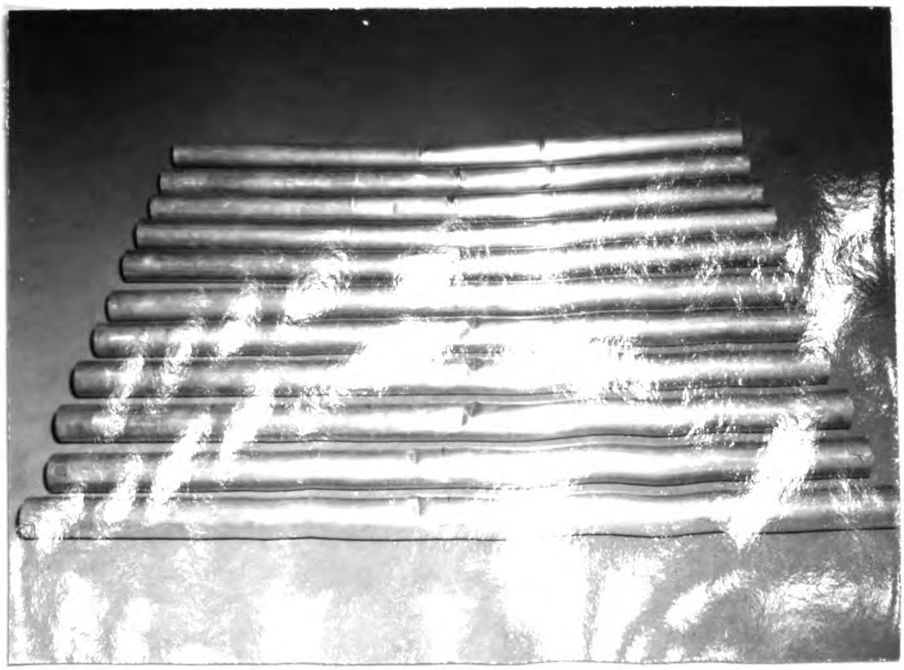

Fig. 27. The twelve sections of tubing after being buckled in the bending test. The six damaged and repaired sections are in the foreground.

Figure 26 shows approximately the same results for the samples taken from the second parent section of tubing. There was more dispersion of the loading curves but the strengths were, in general, again comparable up to the proportional limit of the sample from both treatments.

Table Va shows the maximum loading of each sample within the proportional limit as taken from the curves and Table Vb lists an analysis of variance for this data. The analysis was carried out, primarily, to show whether there was sufficient variation of the loadings within each treatment to make the data unreliable. The within or error term in the analysis is clearly insignificant as compared with the effect of the sample treatment and the sample parentage. Since the mean sum of squares for interaction was much larger than the mean sum of squares for variation of the samples within each group, the former was selected as the proper error term. Due to the small number of degrees of freedom for the interaction term, neither the effects of the dent nor the effects of the pipe showed up as being significant at the five percent level.

Tables VIa and VIb show the data and analysis of variance for the maximum loadings obtained for the samples. Here again the error within samples was small compared with the interaction term. Use of the interaction term with only one degree of freedom showed the effect of both the denting and the parentage to be insignificant.

TABLE Va

LOAD AT PROPORTIONAL LIMIT (in pounds) ON 4.5 FOOT SPAN OF THREE-INCH OUTSIDE DIAMETER ALUMINUM TUBING WITH THIRD POINT LOADING

	Dented	Undented	Pipe Totals
	lbs.	lbs.	lbs.
Pipe I	1007 1000 990	1040 1010 1010	6057
Pipe II	820 862 740	897 870 885	5074
Treatment Totals	5419	5712	11131, Grand Total

TABLE Vb

FORCE TO CREATE BENDING MOMENT AT PROPORTIONAL LIMIT ANALYSIS OF VARIANCE OF BENDING TESTS*

Source	D.F.	SS	MS	F	F 5% point
Total	11	98,797			
Dent	1	7,154	7,154	3.08	161
Pipe	1	80,524	80,524	34.70	161
Interaction	1	2,324	2,324		
Within	8	8,795	1,099		
Interaction Plus within	9	11,119	1,235		

* Notations and method of presentation follow examples in Goulden, Cyril H., "Methods of Statistical Analysis."

TABLE VIa

MAXIMUM COMBINED LOAD IN POUNDS ON 4.5 FOOT SPAN OF THREE-INCH ALUMINUM TUBING WITH THIRD POINT LOADING

	Dented	Undented	Pipe Totals
Pipe I	1012 1050 1063	1420 1402 1467	7414
Pipe II	984 1050 1098	1226 1244 1229	6831
Treatment Totals	6257	7988	14,245 Grand Total

TABLE VIb

COMBINED FORCE TO CREATE MAXIMUM BENDING MOMENT ANALYSIS OF VARIANCE OF BENDING TESTS

Source	D.F.	SS	MS	F	F 5% Point
Total	11	318,117			
Dent	1	249,696	249,696	8.42	161
Pipe	1	28,324	28,324	0.95	161
Interaction	1	29,701	29,701		
Within	8	10,396	1,299.5		
Interaction Plus Within	9	40,097	4,455.2		

It should be pointed out, however, that had the within term been taken as the error term, the effects of the dent and parentage would have been significant in both analyses. The strong effect of the parentage of the source of tubing from which the samples were cut was not anticipated in the design of the experiment. Had this been anticipated the experiment might have been modified to cut all the samples from one forty-foot section of tubing or to cut pairs of samples from many different sources of tubing.

The analysis of variance for the experiment as conducted does point toward two indications which are of importance. The small within mean sum of squares, (MS), indicated that the denting procedure, the straightening procedure, and the loading procedure all resulted in very little dispersion of data and that they should be capable of fairly accurate reproduction. The large mean sum of squares for the two sources of tubing or pipe indicated a rather wide variation of some of the mechanical properties of extruded aluminum tubing from the same manufacturer. This variation should definitely be considered if further work of this nature is anticipated.

From the practical standpoint, the bending strength within the nominal elastic range of undamaged tubing appeared to be a reasonable criteria for comparison. Loading beyond this point would cause permanent deformation whether the section was damaged or undamaged. For the type of damage inflicted in this test series, the repair procedure restored the proportional limit in bending to an average of 95 percent of the value for undamaged tubing.

SUMMARY AND CONCLUSIONS

The use of thin-walled aluminum tubing in sprinkler irrigation is increasing rapidly as the acreage covered by this method of irrigation is expanded annually. While an exact evaluation has not been made of the extent of the losses of materials and labor due to damage of this tubing in handling, these losses appear sufficiently significant for consideration.

Losses of power from pumping water past dents in sprinkler irrigation tubing appeared from the study to be generally too small to be of economic significance. It appears that the inconvenience of handling a badly mashed section would generally result in its being pulled out of service before power losses due to increased friction would become a problem.

Dents of the type normally caused by running over the tubing with rubber tired vehicles were removed successfully by the use of hydraulic radial expansion. By closing the ends of the tubing with suitable end plugs and applying hydrostatic pressure up to approximately 75 percent of the yield strength of the tubing, the dents were removed sufficiently for most practical purposes. Any slight residual kink left in the section was removed by exerting a bending moment at the dented section while the tubing was under pressure.

Tubing which had been dented and straightened showed essentially complete recovery of its original hydraulic characteristics. Bending strength recovery of dented and straightened tubing averaged 95 percent recovery up to the proportional limit of the metal. Bending strength tests showed that the damaged and straightened sections would not stand as much pure plastic bending as the undamaged sections without buckling.

While a cost study of the dent removing procedure was not a part of this project, it would appear that due to the limited amount of equipment required, the hydraulic radial expansion method of repair should be quite economical. A set of plugs and a small hand pump carried on the pick-up truck of an irrigation equipment salesman could likely be used to build customer good will besides netting the salesman a good wage from the straightening fee charged. With the cost of a length of three inch tubing, for example, running from fifteen dollars to thirty dollars depending upon the length of the section and the style of coupler used, there is an opportunity to show a considerable saving to the farmer with a few minutes work on each dented tubing section.

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