University of Minnesota

ST. ANTHONY FALLS HYDRAULIC LABORATORY

Project Report No. 160

FURTHER STUDIES OF FRICTION FACTORS FOR HELICAL CORRUGATED ALUMINUM PIPES WITH RE-CORRUGATED ANNULAR RINGS ON EACH END

by

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Prepared for

KAISER ALUMINUM AND CHEMICAL SALES, INC.

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A = area, ft^2

D = inside diameter of test pipe, ft

d = corrugation depth, in.

f = Darcy friction factor

g = acceleration due to gravity, ft/sec^2

h = head loss in pipe, ft

L = length of pipe, ft

n = Manning roughness coefficient

P = perimeter, ft

p = corrugation pitch, in.

Q = discharge, cfs

R_h = hydraulic radius = A/P, ft

R = radius, ft

R_e = Reynolds number = $\bar{V}D/\nu$

S = slope of hydraulic grade line = h/L

t = metal thickness, in.

$\bar{V}$ = average axial velocity = Q/A, fps

θ = helix angle of corrugation measured from axial direction, deg.

ν = kinematic viscosity, ft^2/sec

FURTHER STUDIES OF FRICTION FACTORS FOR HELICAL CORRUGATED ALUMINUM PIPES WITH RE-CORRUGATED ANNULAR RINGS ON EACH END

I. INTRODUCTION

The St. Anthony Falls Hydraulic Laboratory was engaged by Kaiser Aluminum and Chemical Sales, Inc., to determine the friction factors for fully developed flow in 24 and 12 in. helical corrugated pipes flowing full and to make qualitative observations of the pipe joint characteristics. These pipes are unique in that the pipes were manufactured by customary procedures for helical pipe and then the ends re-corrugated with four annular rings using a special machine. The pipe characteristics are given in Fig. 1 and the physical features are shown in Figs. 2 and 3. The purpose of the re-corrugated pipe ends is to make it more convenient to connect the pipes together under field conditions.

Previous studies were made at the St. Anthony Falls Hydraulic Laboratory in 1969 when friction factors for helical corrugated aluminum pipe were determined for pipe sizes of 12, 18, and 24 in. diameters (1)[*]. Again, in 1971, further studies of friction factors for corrugated aluminum pipes were made on both helical and annular corrugated pipes and for sizes varying from 12 in. to 66 in. in diameter (2).

Kaiser Aluminum and Chemical Sales, Inc., provided the laboratory with six 20 ft lengths of each size pipe, five annular bands of each size, and rolls of Inmont sealant tape for the joints. Two pipes of each size were provided with a flange on one end only to facilitate connecting the upstream pipe to the laboratory supply line and attaching a butterfly control valve to the downstream end of the test pipe. The seams of the factory-assembled pipe were sealed with neoprene beading during fabrication to minimize leakage, and these helical pipe seams did not leak during the test program. All six sections of the 24 in. pipe and five sections of the 12 in. pipe were installed for the experiments, resulting in test pipe lengths of about 120 ft (60 diameter) and 100 ft (100 diameter) respectively.

Upon receiving the pipe, close inspection revealed an undesirable joint connection that would occur if the bands were left full length as shown in

[*]Numbers in parentheses refer to the List of References on page 28.

Fig. 2, a photo of a typical joint with full length band, and in the upper sketch of Fig. 4. The annular band could not be extended over the helical corrugations; thus the pipes would have to be separated approximately 4 in. to fit the bands. As the sealant tape was only 8 in. wide it was felt this would make a fairly weak joint and also could affect the friction factors obtained. It was decided it was necessary to improve this joint arrangement. By cutting one corrugation off the annular band, the pipes could be brought together leaving only about a 1-1/4 in. gap as shown in Fig. 3 and the lower sketch in Fig. 4. The Inmont tape provided reasonable coverage of the joint and this arrangement was used on the test pipe.

The 24 in. pipe was installed first. When filled with water, a serious leakage problem occurred at all five pipe joints which was not acceptable for the pipe tests. The bands were tightened as much as possible with little effect in reducing the leakage. Typical joint leakage with a standard band and sealant tape is shown in Fig. 5. Close inspection of the re-corrugated ends reveal some roughness and imperfections in the annular corrugations so it was surmised the sealant was not thick enough or sufficiently fluid to fill all the gaps. The 12 in. pipe was considerably rougher on the ends than the 24 in. pipe. First attempts to seal the leaks by applying various sealants along the edges of the bands were unsuccessful. The only alternative was to remove the bands completely and start over. First, a wide tape was wrapped around the gap between the pipes to prevent sealant from being squeezed inside, the sealant tape wrapped around this, asphalt sealant applied on the pipes where the sealant tape did not reach, and finally the annular band installed and tightened as much as possible. Even with all these precautions some leakage still occurred, so rod hoops were obtained and four hoops placed on each annular band for reinforcement as shown in Fig. 6. This effectively reduced the leakage to practically nothing. The measured leakage for all joints of the 24 in. pipe was less than 0.0006 cfs with maximum shut-off static head. As the annular rings on the 12 in. pipe were rougher, sponge rubber was wrapped around the pipe before the metal band was installed in addition to the sealant. Each metal band on the 12 in. pipe was also reinforced by four rod hoops completely sealing the joints against all leaks. In view of the difficulties encountered in sealing the joints in the laboratory under ideal conditions, it is anticipated this problem would be more extreme under field conditions.

The study was under the immediate direction of Professor Edward Silberman and was conducted by Warren Q. Dahlin, Scientist. Contact with the sponsor was through Mr. David Thomas, who also visited the laboratory during the course of the experiments.

II. TEST PREPARATIONS AND PROCEDURES

Upon arrival at the laboratory the pipes were carefully inspected, especially on the inside, for any sealant that might have been squeezed through or any other abnormalities that could affect the flow through the pipe. The various pipe parameters were then measured, such as length, average diameter, helix angle, corrugation pitch, metal thickness, and depth of corrugations. All of these except the individual pipe lengths are given in Fig. 1. The inside pipe diameter is an important parameter in determining the friction factors and it was measured carefully. Personnel crawled through the 24 in. pipes and measured the inside diameter directly using two sliding bars equipped with verniers. Measurements were made about 4 and 8 in. from each end (annular section), 12 in. from each end, and at 2 ft intervals for the rest of the pipe in between (helical section). At each section the diameter was measured in the vertical and horizontal directions and the average computed. In addition, at each section the corrugation depths were measured at both vertical and horizontal positions with a vernier equipped depth gage and the average computed. For the 12 in. pipes the outside diameter and corrugation depths were measured again in the vertical and horizontal positions and at the same intervals along the pipe as for the 24 in. pipe, and the inside diameter computed. The section lengths of the 12 in. pipe were very close to 20 ft with the 24 in. pipes somewhat shorter.

It was decided to put two pairs of pressure taps in each pipe section, one pair 5 ft from each end, which would result in a tap spacing of about 10 ft through the test section as shown in Fig. 7. The arrangement may also be seen in Figs. 8 and 9. These pairs of taps were flush-mounted wall taps and interconnected to give an average pressure at the cross section (Fig. 7). The taps were placed at the bottom of the corrugations, that is the points of smallest diameter, similar to the arrangements that gave satisfactory results in previous tests. The tap was first located by drilling a 1/8 in. hole through the pipe. The inner surface of the pipe where the hole penetrated was carefully deburred. A 1/8 in. wire was placed in the hole, a round form placed around the wire, and a plastic casting material poured in the form. After the casting had hardened, the form was removed and the cast plug drilled and tapped for standard pipe fittings, (see Figs. 7 and 10). The piezometric tap positions were sometimes shifted a fraction of an inch so that they could be placed at the exact bottoms of the corrugations, which resulted in slight variations in the selected 10 ft tap spacing.

The pipes were installed in the laboratory flow system as outlined in Fig. 7 and shown in Figs. 8 and 9. Mississippi River water from the laboratory supply channel entered the test pipe through a shut-off valve and three right-angled guide vane bends. As the inlet pipe was also 24 in., the 24 in. helical pipe could be bolted on directly. To connect the 12 in. helical pipe, a plate with a 12 in. hole in it was bolted to the lower 24 in. elbow. The pipe was then bolted to the plate. On the 24 in. test pipe taps were located on the four downstream pipe sections; these were preceded by two pipe sections (20 diameters) to provide entry length to produce fully developed flow. On the 12 in. test pipe, taps were located on the last three pipe sections with two upstream pipe sections (40 diameters) for proper flow development. This resulted in test pipe lengths of about 120 ft and 100 ft respectively, (Fig. 7).

The test pipe was laid on wood sleepers placed every 5 ft at a 0.0 percent slope, the slope being established with the pipe full of water after completing the installation. The pipe was aligned straight using a reference line and side braces placed about every 10 ft. The pipe was relatively free to expand laterally, and measurements made with the pipe full showed that it was slightly out of round.

Special precautions were taken to connect the pipes together as described earlier. A control valve was placed at the downstream end of the test pipe to control the discharge through the pipe for most of the test runs (Fig. 9).

The pairs of piezometer taps were then interconnected; petcocks were provided as shown in Fig. 10 to bleed off accumulated air before each test. Pressure lines connected the taps to the manometer board as shown in Fig. 11.

The test pipe discharged into a channel from which the flow could be directed into either the laboratory weighing tanks for discharges up to 15 cfs or the laboratory volumetric tanks for discharges up to 300 cfs. These tanks are calibrated and measurements of flow rates are accurate within about 0.1 percent for the weighing tanks and about 0.5 percent for the volumetric tanks. All of the measurements on the 12 in. pipe were made in the weighing tanks and those on the 24 in. pipe in both facilities.

Experiments were first made with the downstream control valve in place, varying the valve setting to obtain a reasonable spacing of data points. As the valve and short section of pipe downstream of the valve cause some head loss, they were removed to obtain the maximum discharge through the pipe. Also the nature of the flow discharging from the end of the pipe could be observed. It is interesting to note that even with the annular corrugations on the end of the pipe the flow still emerges in a noticeable helical motion not much different from entirely helical pipe. This can be seen in Fig. 12 where dye is poured in the discharge from the 24 in. pipe, and in Fig. 13 where a cloth streamer is held at the exit of the 12 in. pipe.

At the beginning of a series of test runs, considerable care was taken to purge the pipe and piezometric pressure lines of any existing air. The upstream shut-off valve was opened to its maximum position and left in that position for all runs with the downstream control valve in place. After the flowing water had forced all air out of the pipe the downstream valve was closed, putting the pipe and pressure lines under the maximum head available. Most of the air in the piezometric pressure lines rose to the high point in the lines at the pressure tap locations. The air was bled off at this point through the petcocks provided for that purpose as shown in Fig. 10. This was done at these locations thus clearing the lines at each tap. Air could still be trapped in the lines near the manometer board and in the manometer board itself. To purge this air out, a waste line from the manifold on top of the manometer board was opened allowing the pressure in the pipe to force water through the taps, pressure lines, manometer tubes, and out through the waste line. The waste line was closed off and another dye supply line to the manifold opened. Red colored water, from a pressurized supply tank shown on the right in Fig. 11, was forced through the manifold into the manometer tubes which made reading of the pressure values much easier during a test run.

The downstream control valve was then opened to establish a given discharge and the necessary measurements made. For each test run the water temperature was measured, the pressure value for each tap read on the central manometer board (Fig. 11), and the discharge measured. The water level in the manometer tubes fluctuated considerably for some runs and the readings obtained were visual averages. To establish the discharge, the weight or volume of water was measured for a time of 7-10 minutes. A complete test run took from 30 to 45 minutes. Numerous test runs were made with the downstream valve in place. Several more runs were made later with the valve removed to obtain maximum discharges and Reynolds numbers. From this information the friction-factor-versus-Reynolds-number curve was developed.

III. DETERMINATION OF FRICTION FACTOR

The Darcy friction factor f, the Manning roughness coefficient n, and the Reynolds number R_e were computed from the test data for each run and the f and n values plotted against the R_e. The Darcy friction factor f is defined by the equation

$$h = f \frac{L}{D} \frac{\bar{\bar{V}}^2}{2g} \quad \text{or} \quad f = \frac{h}{L} \frac{2gD}{\bar{\bar{V}}^2}$$

with $S = \frac{h}{L}$, $f = \frac{2g\,DS}{\bar{\bar{V}}^2}$

This equation was used in computing f.

The Manning roughness coefficient n is defined by the equation

$$\bar{\bar{V}} = \frac{1.486}{n} R_h^{2/3} S^{1/2} \quad \text{or} \quad n = \frac{1.486\, R_h^{2/3}\, S^{1/2}}{\bar{\bar{V}}}$$

This equation was used in computing n and is in the English system of units.

The Reynolds number was computed from

$$R_e = \frac{\bar{\bar{V}} D}{\nu}$$

where $\bar{\bar{V}}$, the mean velocity in the three equations, was determined from $\bar{\bar{V}} = Q/A$, Q being the measured discharge. In all these computations the average measured inside diameter of the pipe was used for the diameter D. Using the measured water temperature, the kinematic viscosity ν was obtained from standard tables.

Typical hydraulic gradelines recorded on the manometer board are shown in Fig. 14 for the 24 in. pipe and Fig. 15 for the 12 in. pipe for about one-half of the total number of runs. The head loss h was determined from these hydraulic gradelines for the higher discharges, and for the lower discharges from gradelines plotted with an expanded vertical scale to obtain more accurate results. The slope $S = \frac{h}{L}$ was computed and used in the friction factor equations. For the 24 in. pipe some irregularities may be seen in the first four tap readings which give a slight indication that excessive joint loss might be occurring, but the last four tap readings were generally in a fairly straight line and the slope of this line was used to determine the head loss (Fig. 14).

The tap readings for the 12 in. pipe shown in Fig. 15 are quite irregular. Again most of the tap readings were plotted with an expanded vertical scale to determine the head loss more accurately. To examine the irregularities more extensively two of the runs plotted with the expanded vertical scale are shown in Figs. 16 and 17. The first two taps were located in the third section of pipe (taps 3 and 4 in Fig. 7), the middle two in the fourth section of pipe (taps 5 and 6), and the last two in the fifth section of pipe (taps 7 and 8). Examination of the plots in Figs. 16 and 17 show a definite pattern. Lines cc drawn through the two tap readings in each section of pipe have about the same slope. At the joints a relatively high loss occurs over a short distance. In analyzing the plot it was concluded that the hydraulic gradeline should not be drawn from tap 3 to 8 with the points divided on both sides of the line as would normally be done. If this were done the loss would include 10 ft more of helical pipe than it should and the proportioning of loss between the annular joints and the helical section would not be proper. It was concluded the proper slope should be through taps 3, 5, and 7 (line bb in Figs. 16 and 17) or taps 4, 6, and 8 (line aa in Figs. 16 and 17). Thus the slope is based on 40 ft of pipe over three sections and includes the average joint loss. For convenience, a solid line was drawn parallel to and between the dashed lines aa and bb. These solid lines are the hydraulic gradelines presented in Figs. 14 and 15 and used to determine the head loss.

The variation of the Darcy friction factor f with Reynolds number R_e is shown in Fig. 18 where both the new values and values from the 1971 tests are plotted. The variation of the Manning roughness coefficient n with Reynolds number R_e is shown in Fig. 19 for both tests. Summaries of friction measurements for the 24 in. pipe are presented in Table I and for the 12 in. pipe in Table II.

IV. SUMMARY OF FRICTION FACTOR RESULTS

The Darcy friction factor f and Manning roughness coefficient n values plotted in a reasonably consistent manner as shown in Figs. 18 and 19. Some scatter in values is noticeable, particularly for the 24 in. pipe. The scatter is more noticeable for the 24 in. helical pipe with the annular corrugated ends than the all-helical pipe tested in 1971. The data points for the 12 in. pipe plot in a well-defined curve as indicated, whereas for the 24 in. pipe it is not as well defined and a horizontal straight line was drawn averaging the points. For both pipes the scatter at lower Reynolds numbers may be attributed to the small head loss which is difficult to measure.

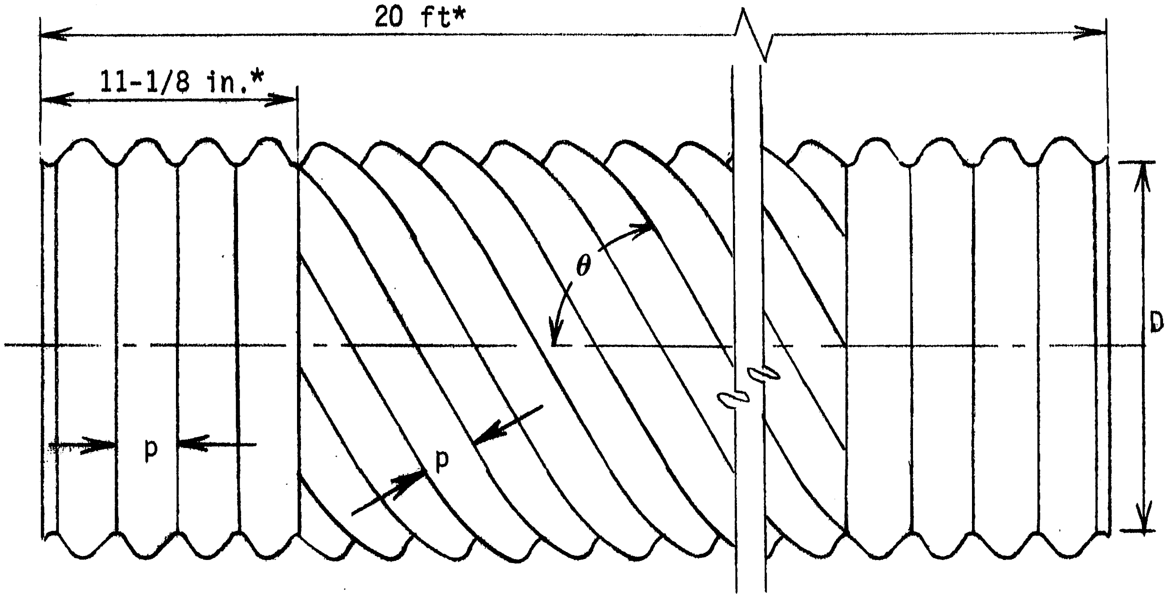
For the 24 in. pipe the Darcy friction factor values plotted as a horizontal line at a value of 0.0465 from a Reynolds number of about 1.3 million down to 170,000. This value is 10.2 percent higher than the value of 0.0422 for the all-helical pipe. For the 12 in. pipe the Darcy friction factor is horizontal at a value of 0.0253 from Reynolds numbers of about 900,000 down to 800,000 and then curves upward to an f value of .0286 at a Reynolds number of 100,000. Comparing the horizontal sections of the curve, this is 10.5 percent higher than the f value of 0.0229 for the all-helical pipe.

The Manning roughness coefficients plot in a similar manner, although the variations are less pronounced. For the 24 in. pipe the coefficient is 0.0179 for Reynolds numbers of 1.3 million down to 170,000. This is 5.9 percent higher than the n value of 0.0169 for the all-helical pipe. For the 12 in. pipe the n value is 0.0117 for Reynolds numbers of 900,000 to 700,000 and then increases to an n value of .0125 at 100,000. In the horizontal regime the n value of 0.0117 is 5.4 percent higher than the n value of 0.0111 for the all-helical pipe.

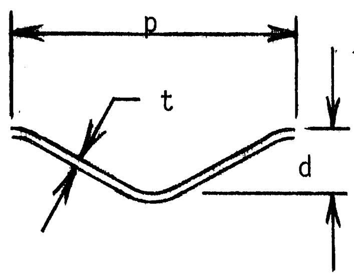
The higher friction factors in the present study compared to the earlier one can be attributed to the joints. Joint loss is particularly noticeable for the 12 in. pipe as pointed out in Figs. 16 and 17 and only somewhat noticeable for the 24 in. pipe. It is interesting to note that friction factors f_{cc} and n_{cc} which were computed from S values based on the slope of lines cc as illustrated in Figs. 16 and 17 are less than the average friction factors f_{av} and n_{av} for both this and the earlier study with helical joints because the joint loss is not included. The larger friction factors in the current tests are a result of both the annular form and poor mating at each joint and also are associated with the fact that there are twice as many joints in the present tests (every 20 ft as compared to every 40 ft in the earlier tests). The proportion of the joint loss caused by each factor cannot be estimated.

Since both the pipe sizes tested show increases in f of about 10 to 10.5 percent and in n of about 5.5 to 6 percent over the test results in 1971, it is believed that f and n values for intermediate and larger sizes of helical pipes can be estimated by adding the above percentages to the values given by the formulas on page 14 of the 1971 report. This procedure would probably not be applicable to diameters smaller than 12 in.

In the laboratory tests it should be noted that the friction factor values obtained are for fully developed flow with the pipe flowing full, test pipe level, and straightly aligned. Great care was taken to seal the joints minimizing the leakage, and to prevent any sealant from being squeezed into the pipe. This may not be possible in many field installations so that additional head losses could occur and should be considered to determine the total head loss. The re-corrugated annular rings on each end appeared somewhat rough, particularly for the 12 in. pipe. This may be inherent in the fabrication process, but anything that can be done to improve on this condition would probably make it more convenient to connect and seal the joints under field conditions.



Corrugated Pipe
*Dimensions vary slightly for different pipes



Form of Corrugation

	Measured Values				
Nominal Pipe Dia. in.	Average Dia. D-in.	Helix Angle θ - Deg.	p-in.	t-in.	d-in.
24	24.370	71	2.67	0.063	0.498
12	11.576	50	2.67	0.062	0.454

Fig. 1 - Pipe Details

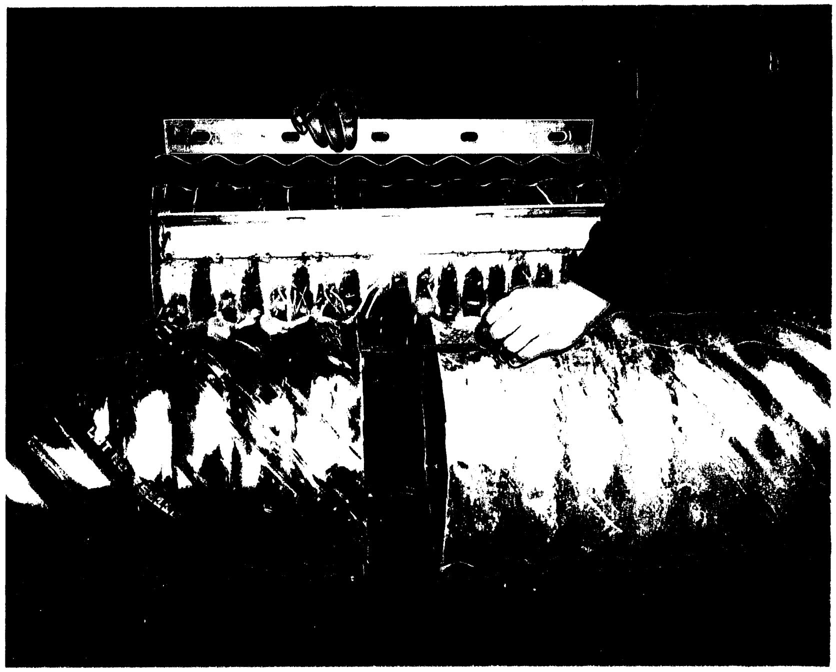


Fig. 2 - A Typical Joint with Full Length Band
(This arrangement was not used)

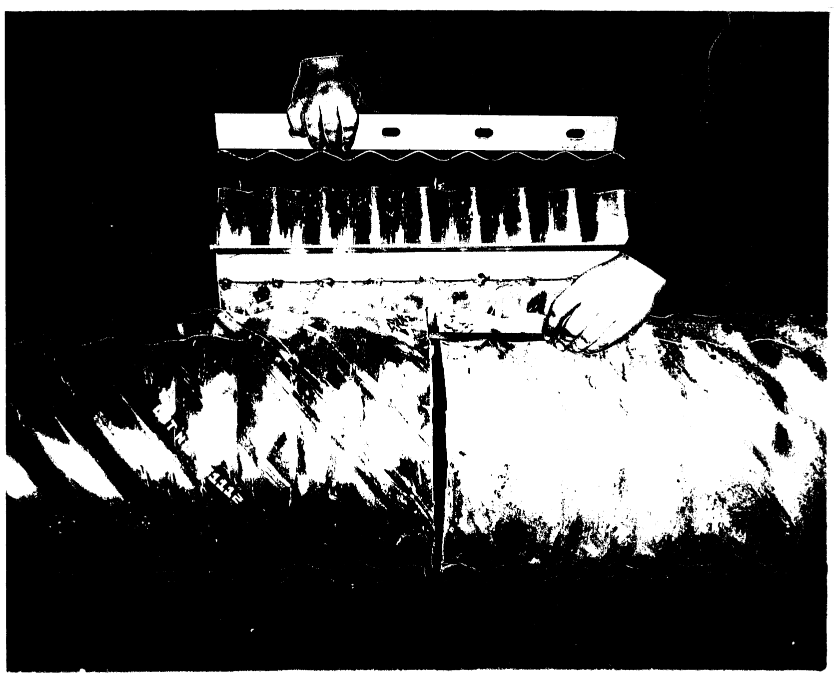
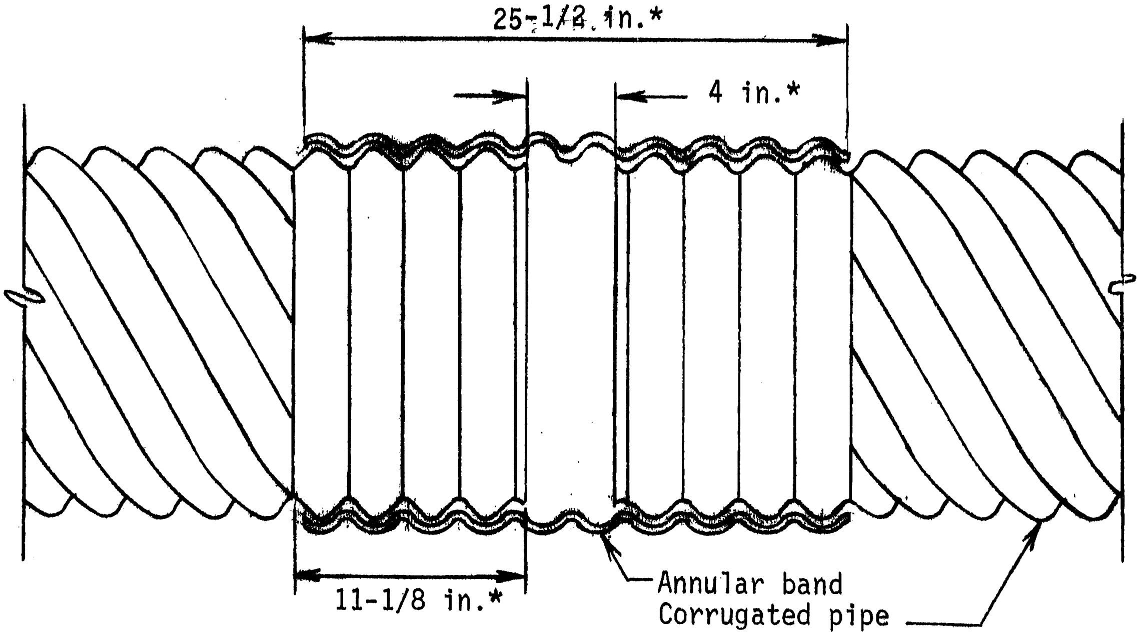
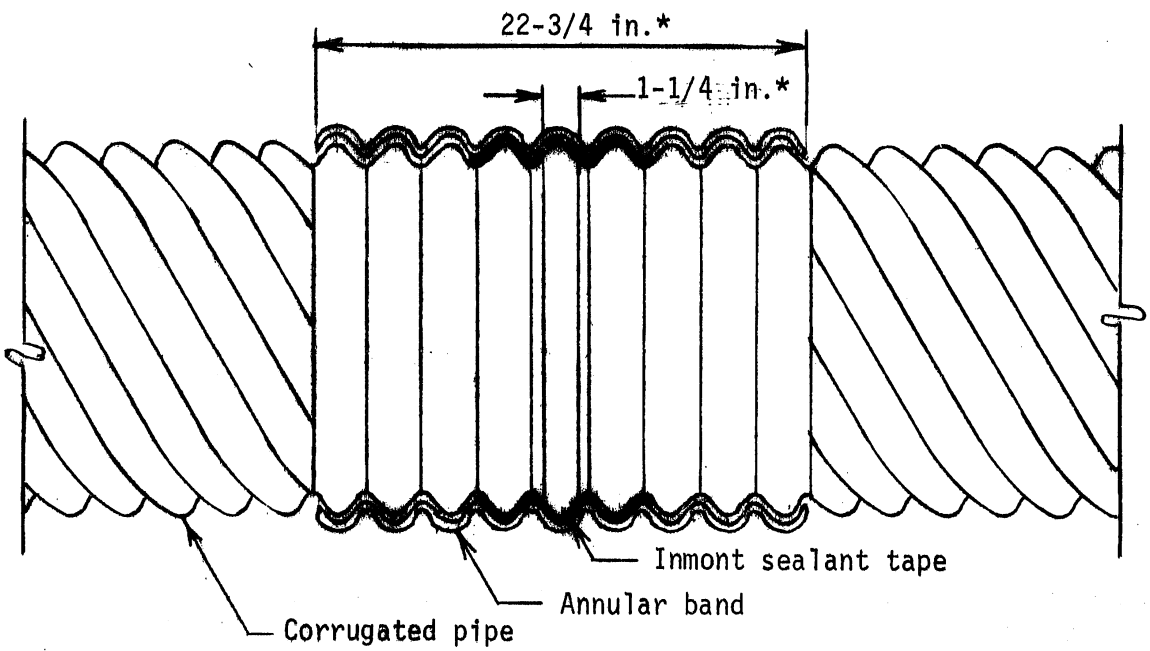


Fig. 3 - A Typical Joint with Band Shortened
One Corrugation (This arrangement
was used on the test pipe)



Typical Joint Geometry
with Full Length Band



Typical Joint Geometry with
Band Shortened One Corrugation

Fig. 4 - Typical Joint Geometry

*Dimensions vary slightly for different pipes.

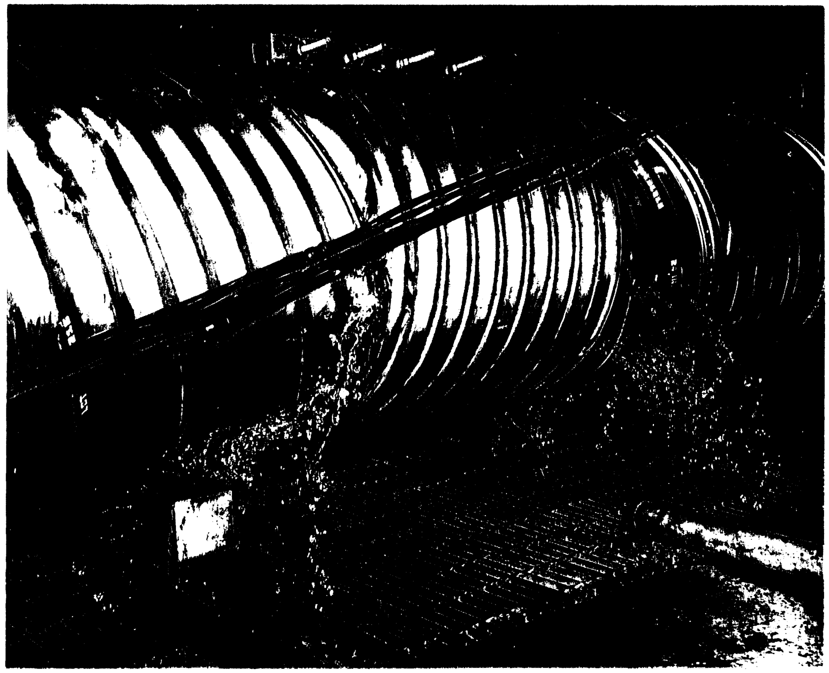


Fig. 5 - Typical Joint Leakage with a Standard Band and Sealant Tape

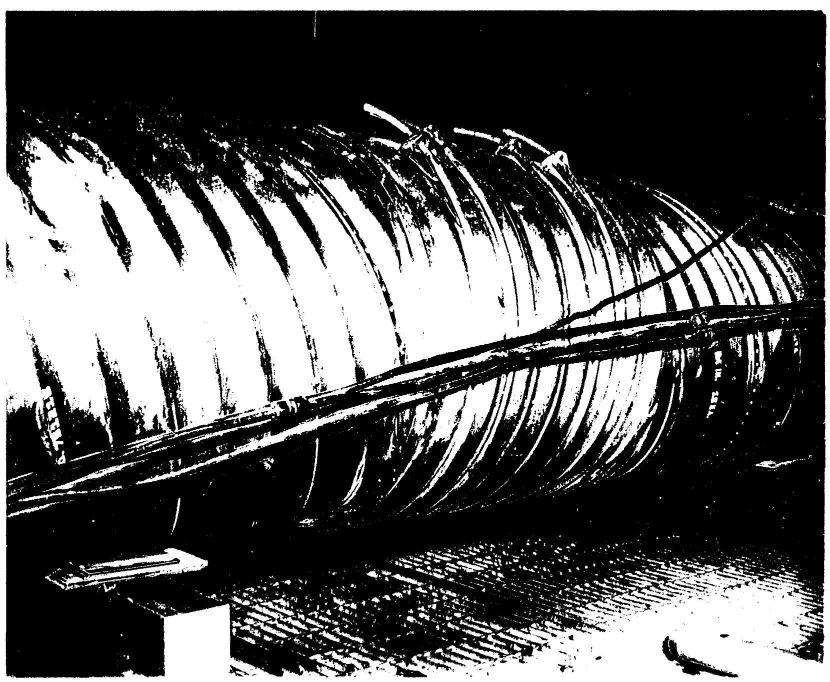


Fig. 6 - Joint Leakage Stopped with Additional Asphalt Sealant and Rod Hoops to Reinforce the Band

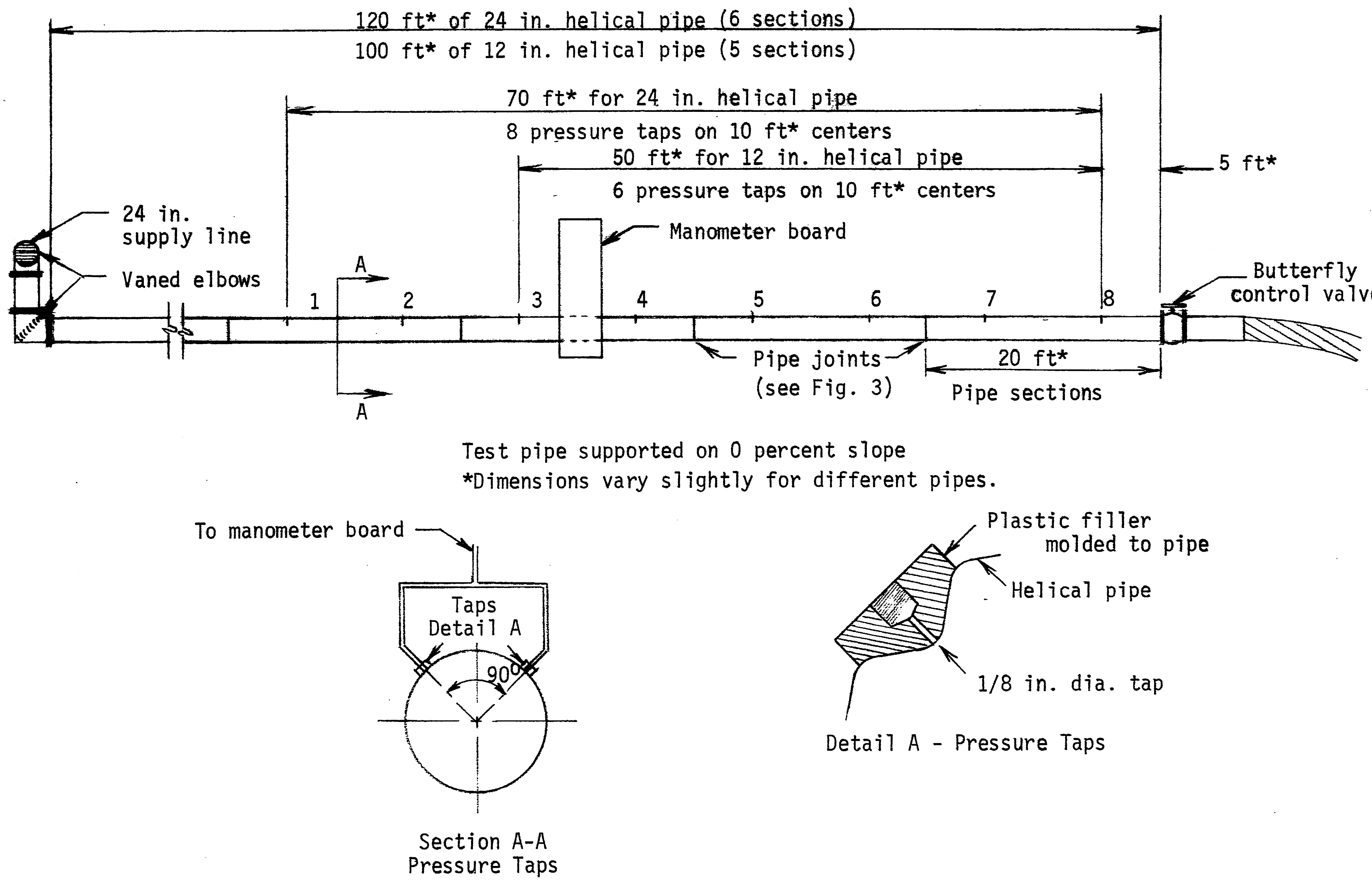


Fig. 7 - Test Arrangement for 24 in. and 12 in. Pipe

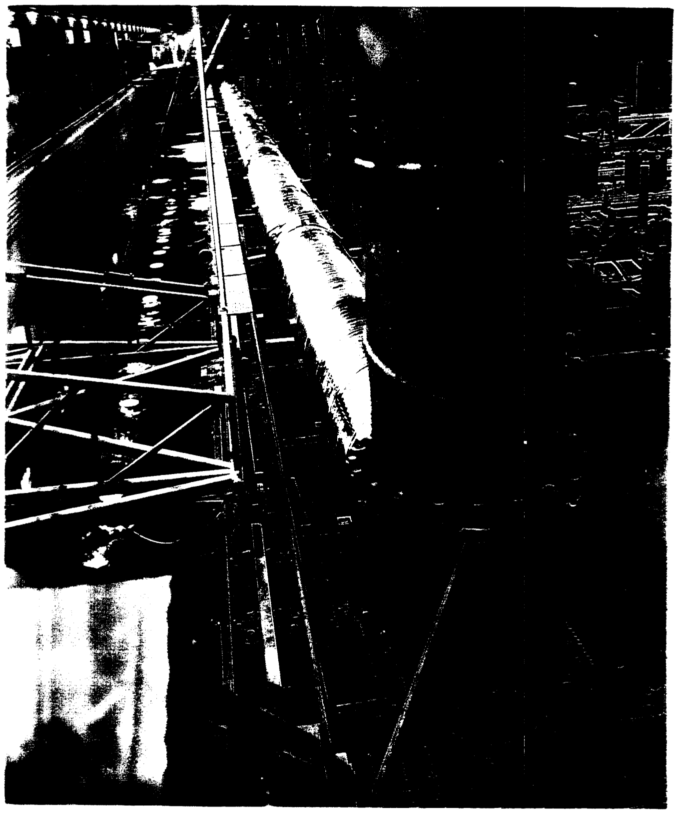


Fig. 8 - Looking Downstream at the Experimental Arrangement with 24 in. Pipe Installed

Fig. 9 - Looking Upstream at the Experimental Arrangement with 24 in. Pipe Installed

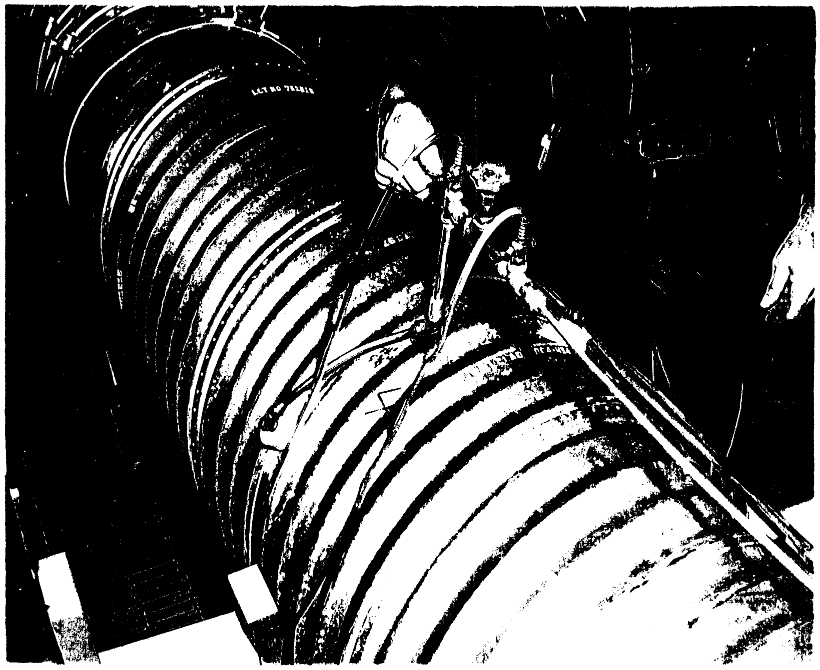


Fig. 10 - Piezometer Tap Arrangement

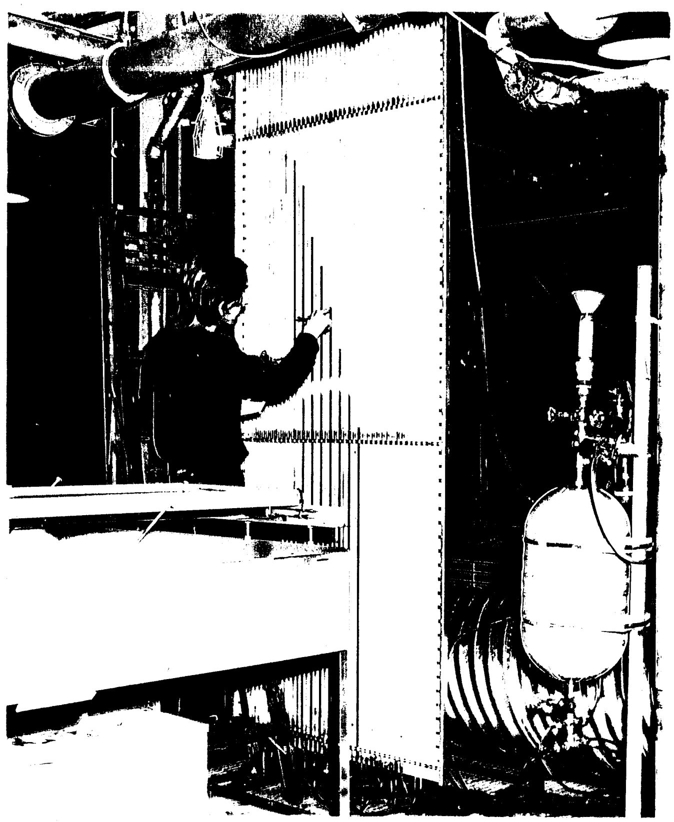


Fig. 11 - Reading the Piezometric Pressures on the Manometer Board

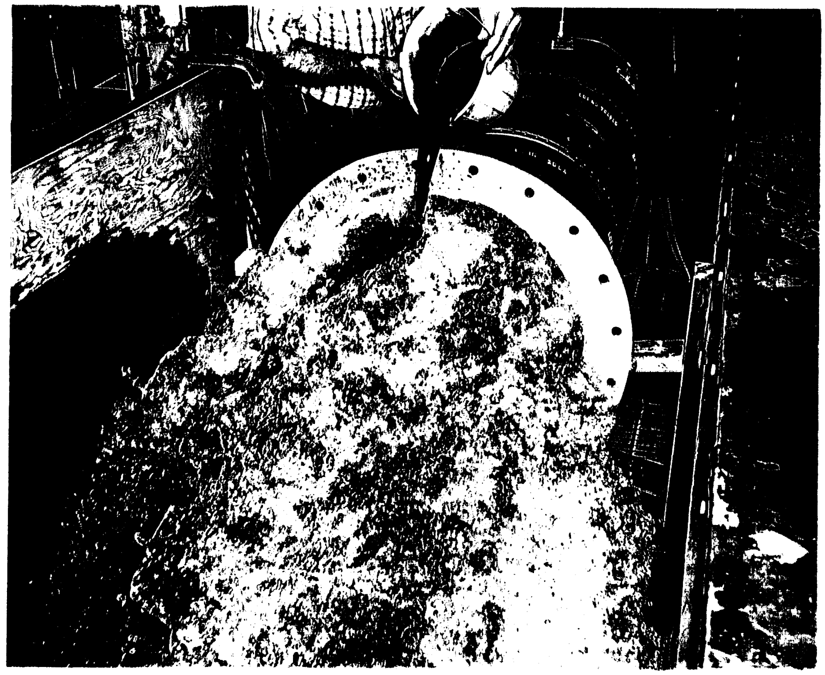


Fig. 12 - Dye Shows the Helical Flow at the 24 in. Pipe Exit with Valve Removed

Fig. 13 - A Cloth Streamer Shows the Helical Flow at the 12 in. Pipe Exit with Valve Removed

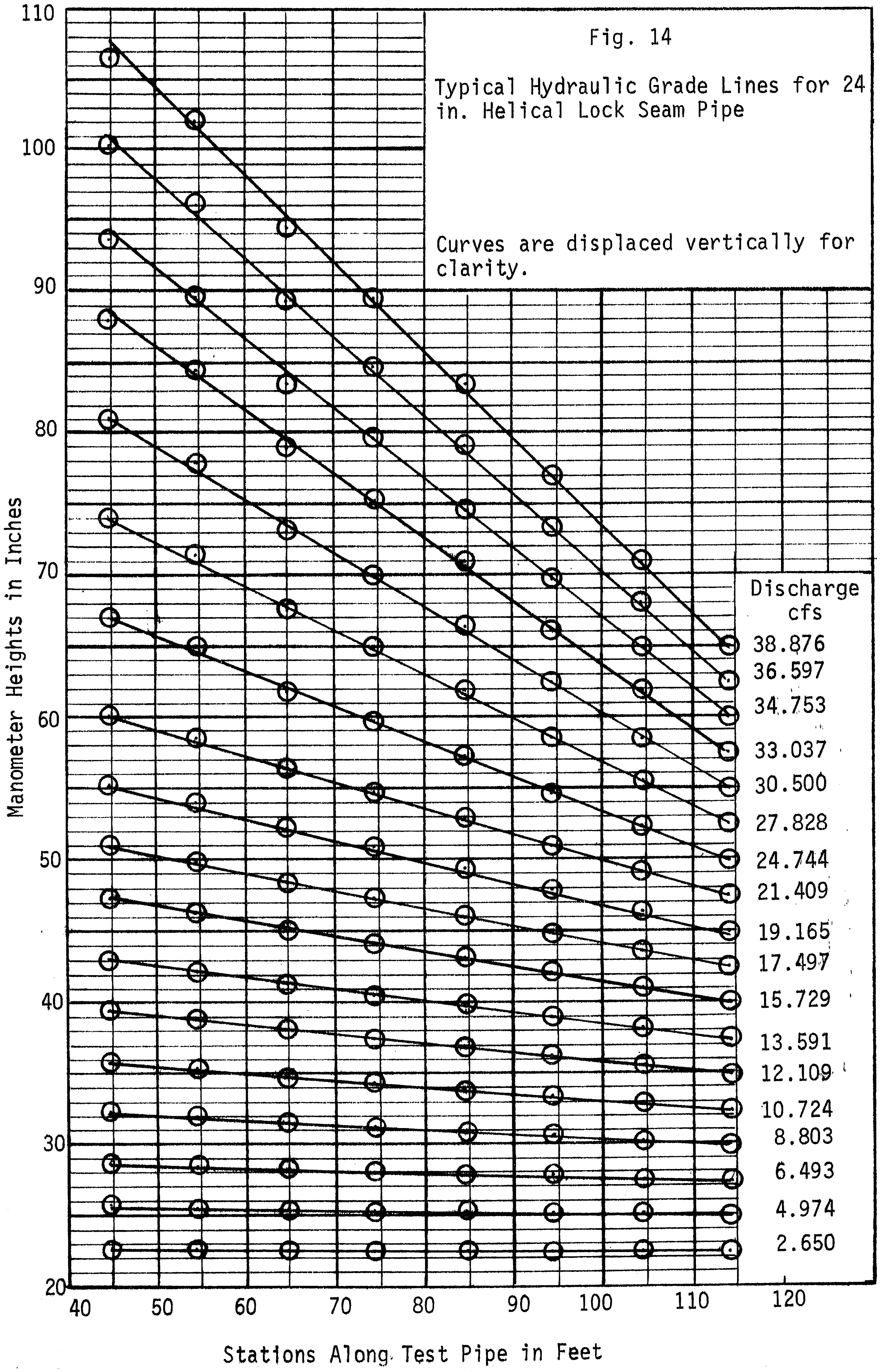


Fig. 14
Typical Hydraulic Grade Lines for 24 in. Helical Lock Seam Pipe
Curves are displaced vertically for clarity.
Discharge cfs
38.876
36.597
34.753
33.037
30.500
27.828
24.744
21.409
19.165
17.497
15.729
13.591
12.109
10.724
8.803
6.493
4.974
2.650
Manometer Heights in Inches
Stations Along Test Pipe in Feet
110
100
90
80
70
60
50
40
30
20
40
50
60
70
80
90
100
110
120

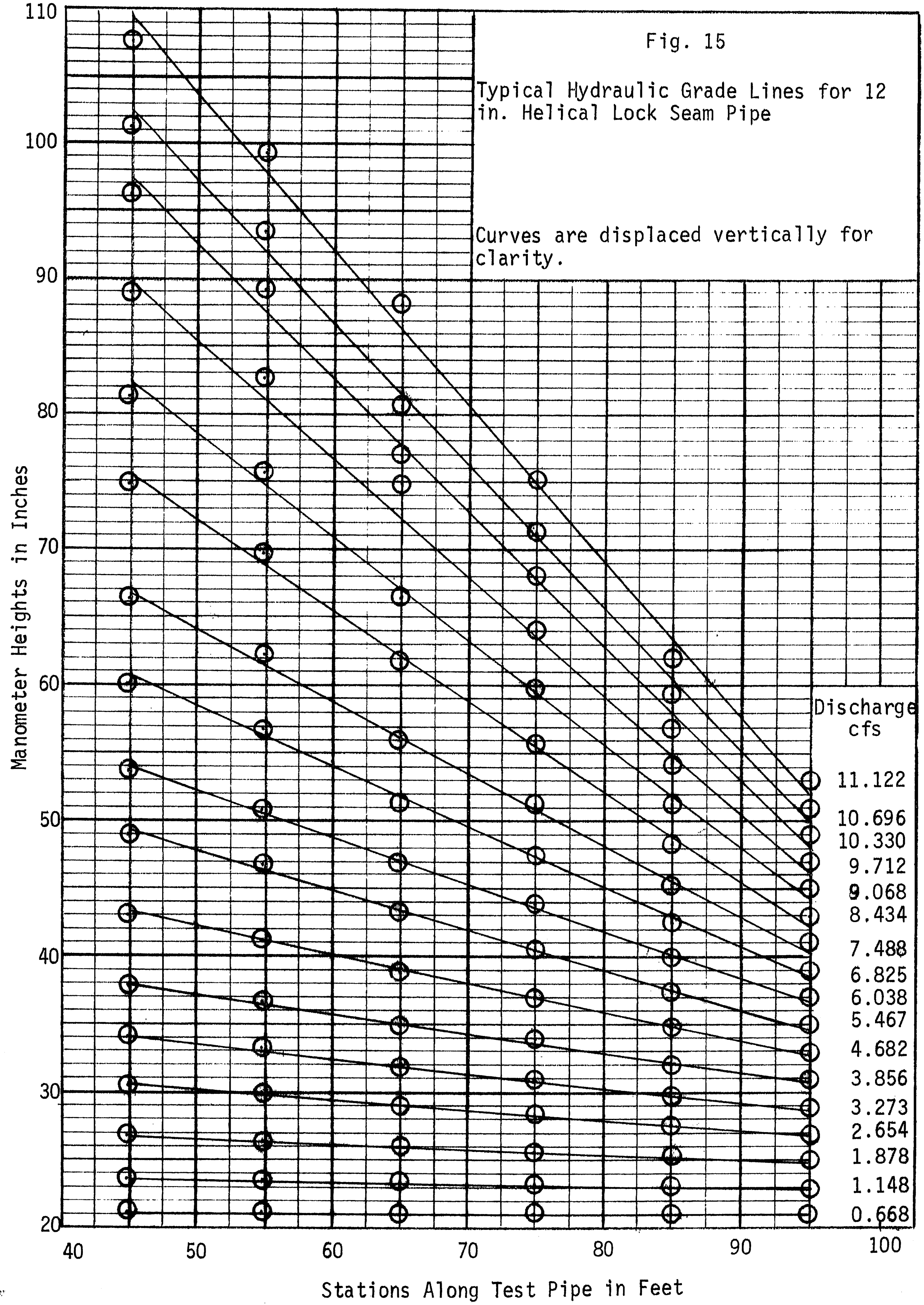


Fig. 15
Typical Hydraulic Grade Lines for 12 in. Helical Lock Seam Pipe
Curves are displaced vertically for clarity.
Discharge cfs
11.122
10.696
10.330
9.712
9.068
8.434
7.488
6.825
6.038
5.467
4.682
3.856
3.273
2.654
1.878
1.148
0.668
Manometer Heights in Inches
Stations Along Test Pipe in Feet
110
100
90
80
70
60
50
40
30
20
40
50
60
70
80
90
100

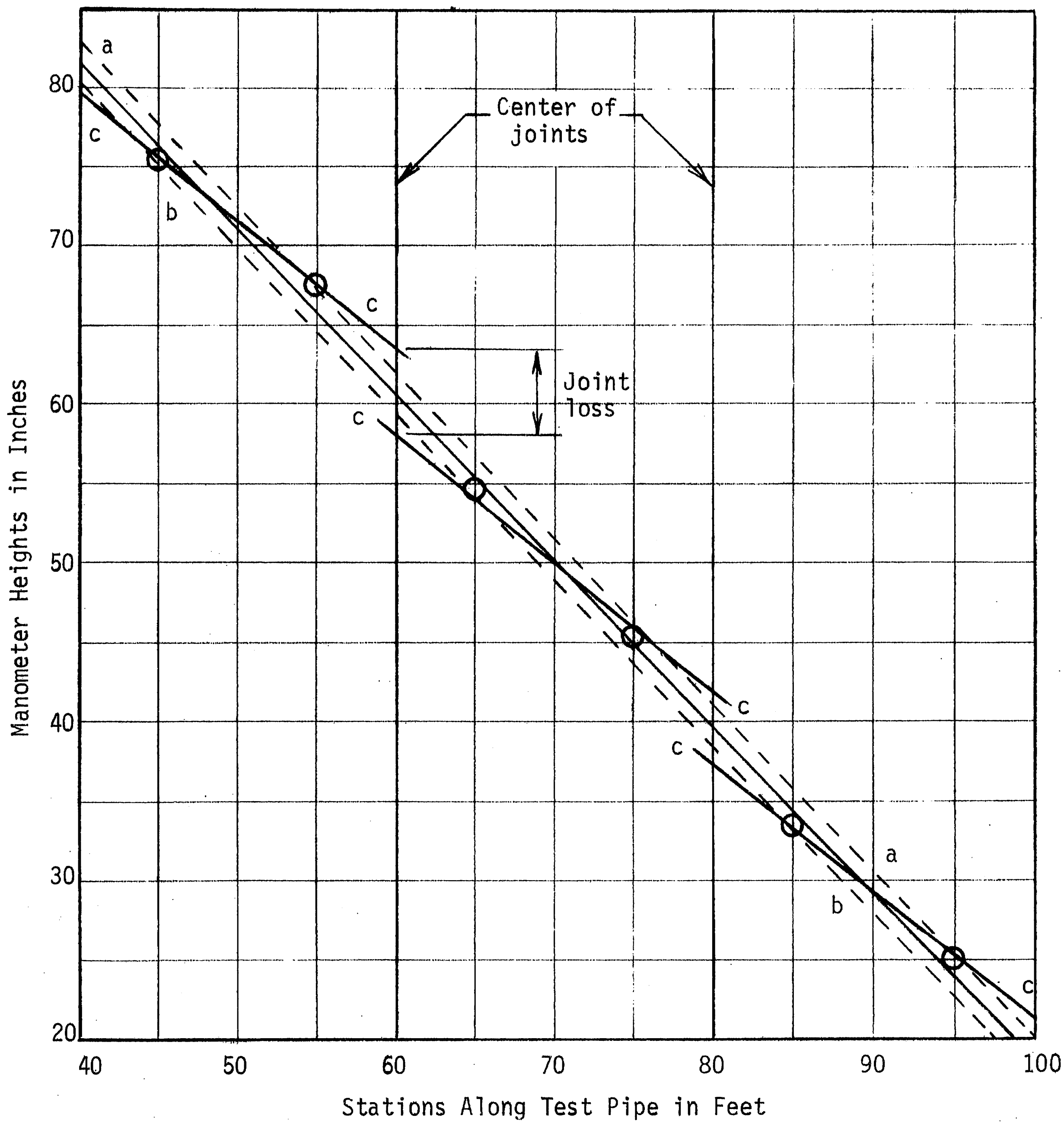


Fig. 16 - A Typical Hydraulic Grade Line for the 12 in. Helical Lock Seam Pipe Showing the Joint Loss

Discharge = 10.696 cfs

Darcy $f_{aa} = f_{bb} = f_{av.} = 0.0252$, $f_{cc} = 0.0195$

Manning $n_{aa} = n_{bb} = n_{av.} = 0.0116$, $n_{cc} = 0.0102$

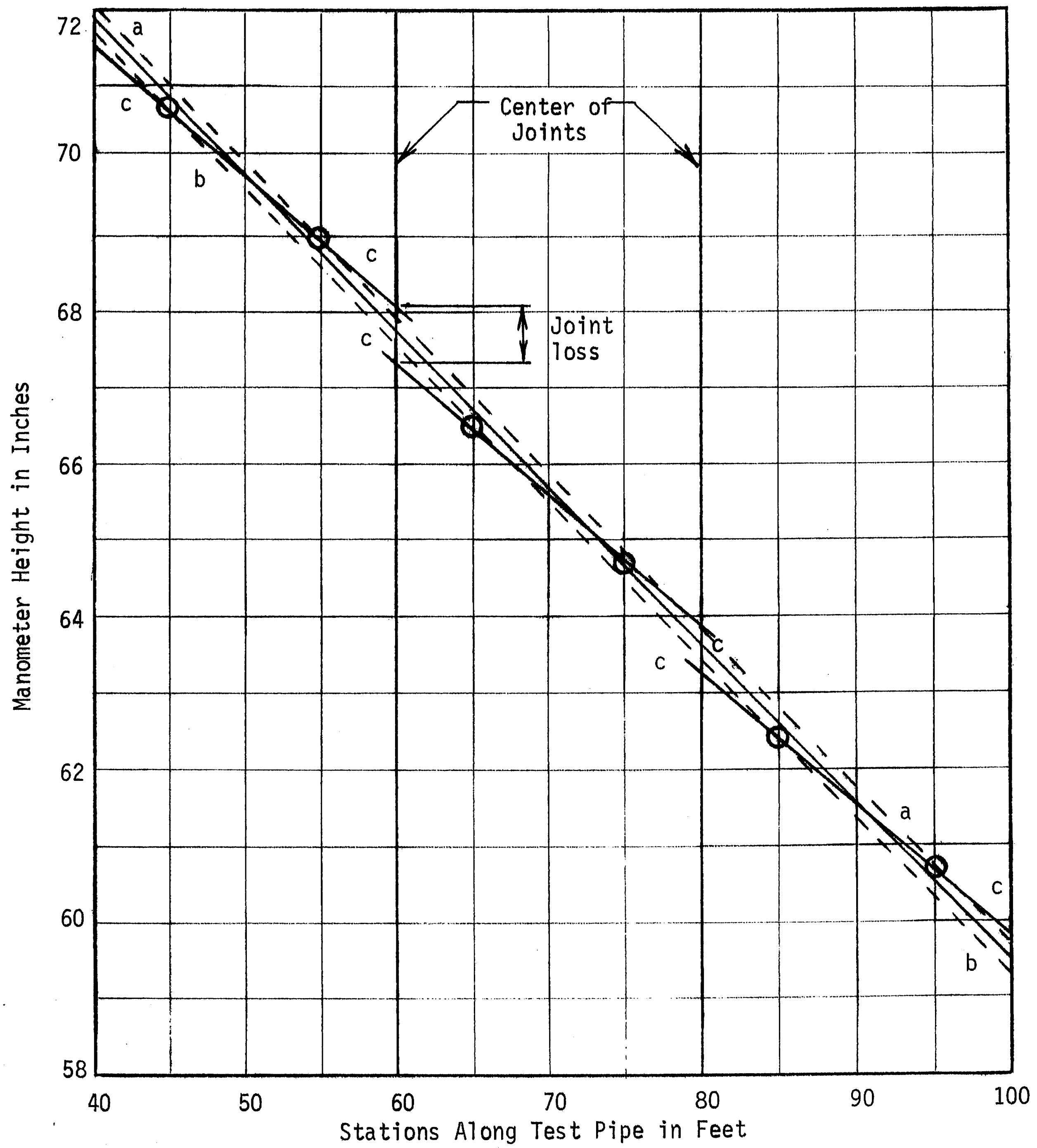


Fig. 17 - A Typical Hydraulic Grade Line for the 12 in. Helical Lock Seam Pipe Showing the Joint Loss

Discharge = 4.682 cfs

Darcy $f_{aa} = f_{bb} = f_{av.} = 0.0259$, $f_{cc} = 0.0217$

Manning $n_{aa} = n_{bb} = n_{av.} = 0.0118$, $n_{cc} = 0.0108$

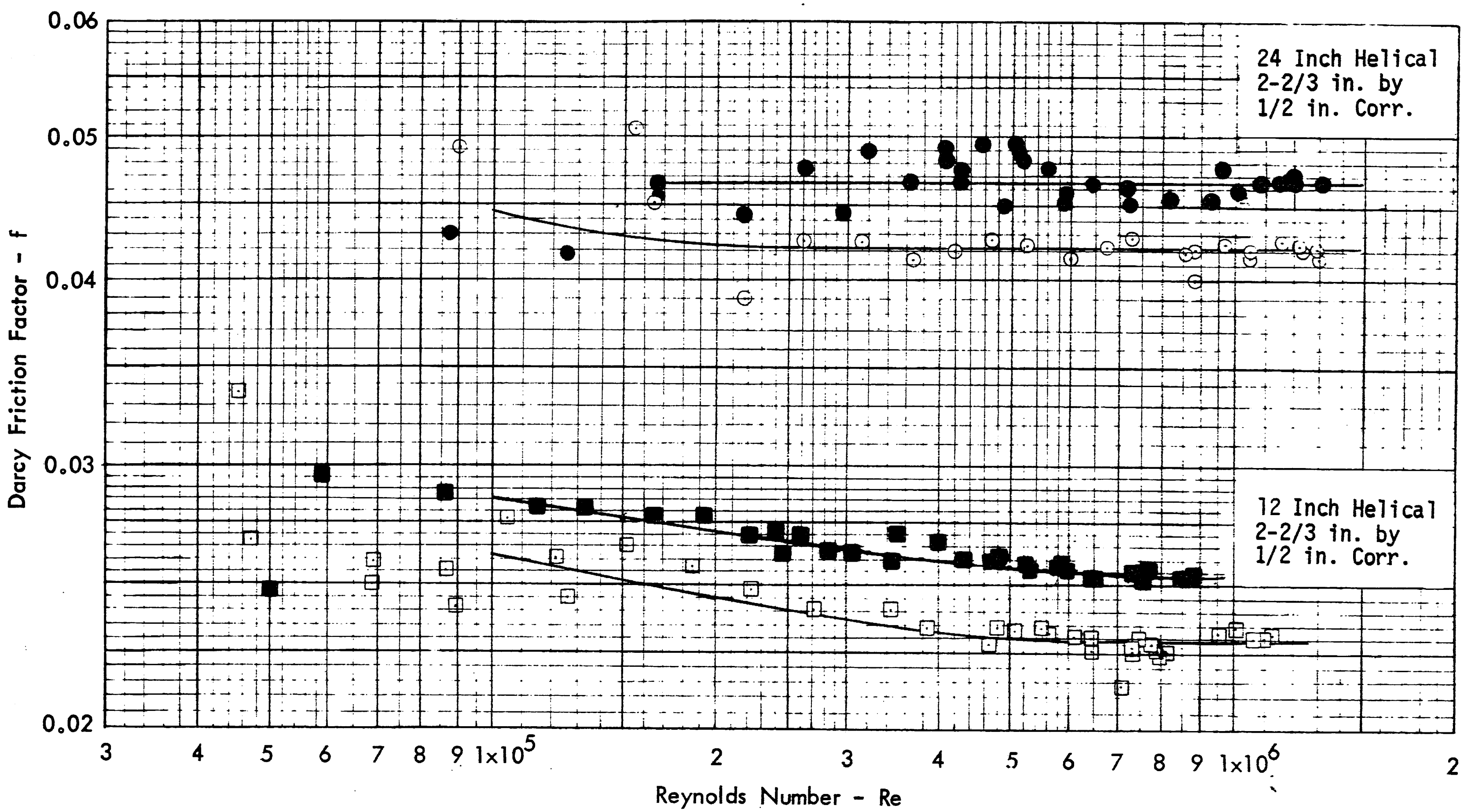


Fig. 18 - Variation of Darcy Friction Factor f with Reynolds Number - 12 Inch and 24 Inch Corrugated Pipe

○ ⊡ Tests in 1971 on helical corrugated pipe with joints at 40 ft spacing and using helical joint connectors.

● ■ Tests in 1976 on helical corrugated pipe with joints at 20 ft spacing using 4 annular corrugations on each side of each joint.

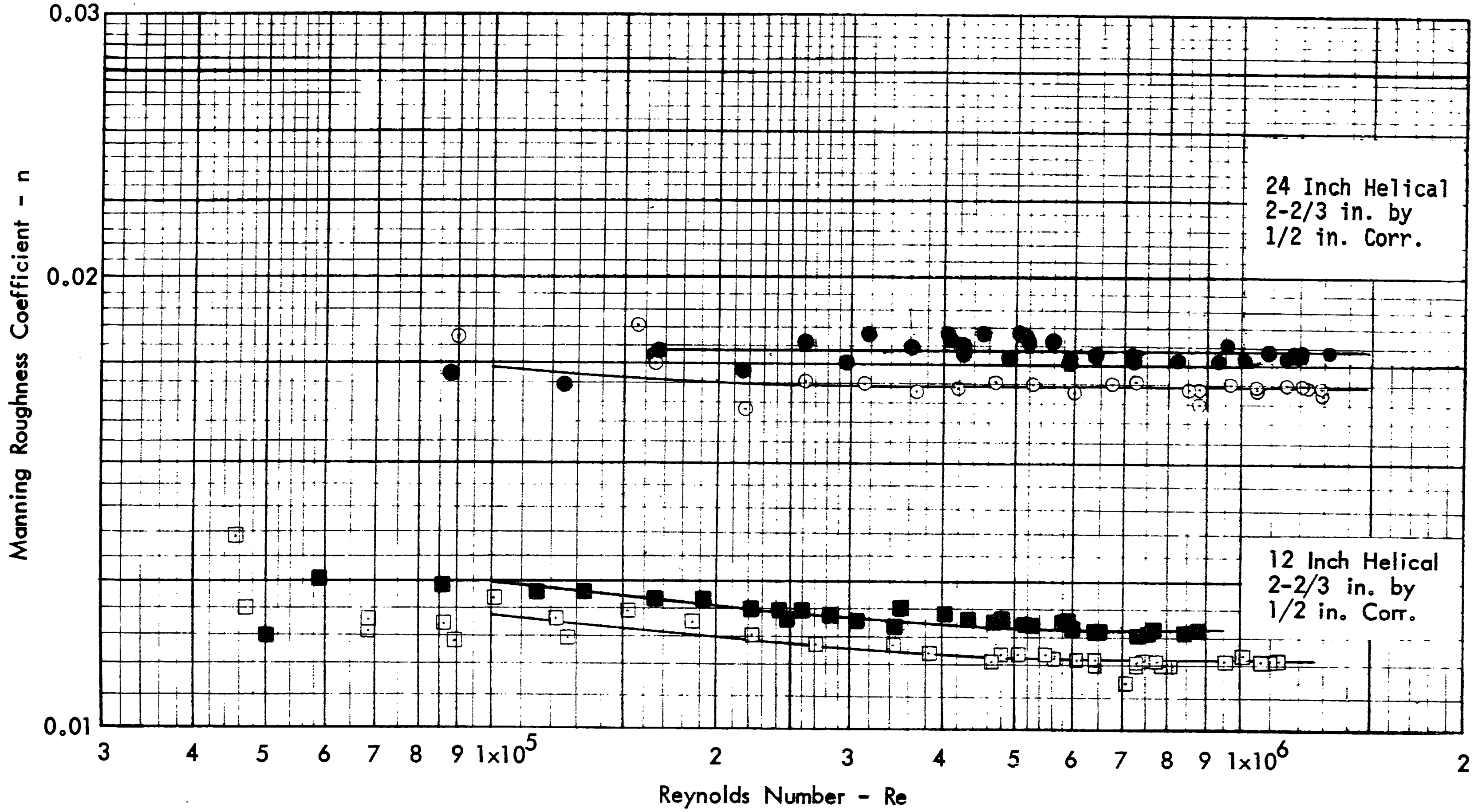


Fig. 19 - Variation of ~~Manning~~ n with Reynolds Number - 12 Inch and 24 Inch Corrugated Pipe

⊙ ⊡ Tests in 1971 on helical corrugated pipe with joints at 40 ft spacing and using helical joint connectors.

● ■ Tests in 1976 on helical corrugated pipe with joints at 20 ft spacing using 4 annular corrugations on each side of each joint.

TABLE I. SUMMARY OF FRICTION MEASUREMENTS FOR 24 INCH HELICAL LOCK SEAM PIPE

Re-corrugated ends--4 annular corrugations at each end of each pipe.

Note that the test length was made up of 20 ft pipe lengths.

Average measured diameter = 2.03083 ft

Q cfs	$\bar{V}$ fps	S per cent	Water Temp. Deg. F	ν ft^2/sec $\times 10^5$	Re $\times 10^{-6}$	Darcy f	Manning n
38.876*	12.002	5.1469	34.0	1.859	1.3111	0.0467	0.0179
36.597	11.298	4.5950	33.0	1.895	1.2108	0.0470	0.0179
36.435	11.248	4.4751	33.0	1.895	1.2054	0.0462	0.0178
35.363*	10.917	4.2351	34.0	1.859	1.1926	0.0464	0.0178
34.753	10.729	4.1031	33.0	1.895	1.1498	0.0466	0.0179
33.037	10.199	3.7192	33.0	1.895	1.0930	0.0467	0.0179
30.500	9.416	3.1313	33.0	1.895	1.0091	0.0462	0.0178
28.452*	8.784	2.8194	34.0	1.859	0.9596	0.0478	0.0181
27.828	8.591	2.5555	34.0	1.859	0.9385	0.0452	0.0176
24.744	7.639	2.0276	33.0	1.895	0.8186	0.0454	0.0176
21.409	6.609	1.5117	34.0	1.859	0.7220	0.0452	0.0176
21.308*	6.578	1.5297	34.0	1.859	0.7186	0.0462	0.0178
19.165	5.916	1.2357	34.0	1.859	0.6463	0.0461	0.0178
17.604	5.435	1.0366	34.0	1.859	0.5937	0.0459	0.0177
17.497	5.402	1.0078	34.0	1.859	0.5901	0.0451	0.0176
16.625	5.132	0.9610	34.0	1.859	0.5607	0.0477	0.0181
15.729	4.856	0.8686	33.0	1.895	0.5204	0.0481	0.0182
15.311	4.727	0.8374	34.0	1.859	0.5164	0.0490	0.0183
15.017	4.636	0.8146	34.0	1.859	0.5065	0.0495	0.0184
14.527	4.485	0.6935	34.0	1.859	0.4899	0.0451	0.0176
13.591	4.196	0.6683	34.0	1.859	0.4584	0.0496	0.0184
12.720	3.927	0.5495	34.0	1.859	0.4290	0.0466	0.0179
12.674	3.913	0.5555	34.0	1.859	0.4274	0.0474	0.0180
12.109	3.738	0.5147	34.0	1.859	0.4084	0.0481	0.0181
12.041	3.717	0.5207	34.0	1.859	0.4061	0.0492	0.0184
10.724	3.311	0.3923	34.0	1.859	0.3617	0.0468	0.0179

*Downstream valve removed

TABLE II. SUMMARY OF FRICTION MEASUREMENTS FOR 12 INCH HELICAL LOCK SEAM PIPE

Re-corrugated ends--4 annular corrugations at each end of each pipe.

Note that the test length was made up of 20 ft pipe lengths.

Average measured diameter = 0.96467 ft

Q cfs	$\bar{V}$ fps	S per cent	Water Temp. Deg. F	ν ft^2/sec x 10^5	Re x 10^{-6}	Darcy f	Manning n
11.122*	15.218	9.5104	40.0	1.664	0.8822	0.0255	0.0117
10.696	14.635	8.6971	34.0	1.859	0.7594	0.0252	0.0116
10.684*	14.618	8.6971	40.0	1.664	0.8475	0.0253	0.0116
10.591	14.491	8.4813	34.0	1.859	0.7520	0.0251	0.0116
10.330	14.134	8.1826	34.0	1.859	0.7334	0.0254	0.0117
9.712*	13.288	7.2531	40.0	1.664	0.7703	0.0255	0.0117
9.068	12.407	6.2739	34.0	1.859	0.6438	0.0253	0.0116
8.799	12.038	5.9253	36.0	1.791	0.6484	0.0254	0.0116
8.434	11.540	5.4772	34.0	1.859	0.5988	0.0255	0.0117
7.961	10.893	4.9461	36.0	1.791	0.5867	0.0259	0.0118
7.488*	10.245	4.3652	40.0	1.664	0.5940	0.0258	0.0117
7.353	10.060	4.2158	34.0	1.859	0.5220	0.0259	0.0118
7.198	9.848	4.0083	36.0	1.791	0.5305	0.0257	0.0117
6.825	9.338	3.6681	34.0	1.859	0.4846	0.0261	0.0118
6.372	8.719	3.1701	36.0	1.791	0.4696	0.0259	0.0118
6.038	8.261	2.8631	35.0	1.823	0.4372	0.0260	0.0118
5.467	7.479	2.4066	36.0	1.791	0.4029	0.0267	0.0119
4.994	6.833	2.0465	34.0	1.859	0.3546	0.0272	0.0121
4.682	6.406	1.7095	36.0	1.791	0.3450	0.0259	0.0118
4.323	5.915	1.4805	34.0	1.859	0.3069	0.0263	0.0118
3.856	5.276	1.1834	36.0	1.791	0.2842	0.0264	0.0119
3.488	4.773	0.9610	34.0	1.859	0.2477	0.0262	0.0118
3.412	4.669	0.9510	34.0	1.859	0.2423	0.0271	0.0120
3.273*	4.478	0.8697	40.0	1.664	0.2596	0.0269	0.0120
3.012	4.121	0.7386	36.0	1.791	0.2220	0.0270	0.0120
2.655	3.632	0.5909	35.0	1.823	0.1922	0.0278	0.0122
2.238	3.063	0.4200	36.0	1.791	0.1650	0.0278	0.0122

*Downstream valve removed

TABLE II. (cont.)

Q cfs	$\bar{V}$ fps	S per cent	Water Temp. Deg. F	ν ft^2/sec x 10^5	Re x 10^{-6}	Darcy f	Manning n
1.878	2.570	0.3004	34.0	1.859	0.1335	0.0282	0.0123
1.528	2.090	0.1992	37.0	1.759	0.1146	0.0283	0.0123
1.148	1.571	0.1145	37.0	1.759	0.0862	0.0288	0.0124
0.835	1.142	0.0622	34.0	1.859	0.0593	0.0296	0.0126
0.668	0.914	0.0332	37.0	1.759	0.0501	0.0247	0.0115

LIST OF REFERENCES

(1) Silberman, E. and Dahlin, W.Q., Friction Factors for Helical Corrugated Aluminum Pipe, Project Report No. 112, University of Minnesota, St. Anthony Falls Hydraulic Laboratory, December 1969.

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