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TECHNICAL NOTE 3889

PRESSURE LOSSES OF TITANIA AND MAGNESIUM SLURRIES IN PIPES AND PIPELINE TRANSITIONS

By Ruth N. Weltmann and Thomas A. Keller

Lewis Flight Propulsion Laboratory
Cleveland, Ohio

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SUMMARY

Comparisons of experimental and calculated pressure losses are presented for Newtonian and non-Newtonian materials. The non-Newtonian materials were slurries of titanium dioxide particles suspended in water and of magnesium particles suspended in a hydrocarbon fluid. One of the slurries showed Bingham plastic flow behavior, while the other one behaved like a pseudoplastic material. The pressure-loss data were obtained for laminar, transitional, and turbulent flow through straight pipelines and pipe transitions of 1-inch and 3/8-inch nominal pipe size. The pipeline transitions considered are 90° elbows; gate, ball, plug, and globe valves; contractions; and expansions. Transition loss coefficients, which are independent of the flow rate in the pipelines, even for laminar flow in the pipeline system, were determined for the transitions and are compared with those reported in the literature for Newtonian fluids.

The transition loss coefficients obtained for the Newtonian and non-Newtonian materials and those reported in the literature for Newtonian fluids agree within the errors of experiment. Thus, these studies indicate that, for the design of pipeline systems, such as fuel systems, where the slurries have flow characteristics similar to those treated in this paper, the transition loss coefficients for Newtonian fluids can be used in the design calculations.

INTRODUCTION

With the higher speed and larger range of aircraft, fuels that provide higher thrust or that reduce fuel weight or volume consumption or both are desired. For ram-jet engines and afterburners, the use of fuel slurries, which have a paint-like consistency such as suspensions of fine metal particles suspended in a hydrocarbon liquid, is possible, since there are no moving parts in the exhaust. Theoretical combustion studies with magnesium particles suspended in a hydrocarbon liquid have shown that greater thrust and higher combustion temperatures are obtained with these slurries than with the conventional fuels (refs. 1 and 2). Another application for paint-like slurries of metal particles suspended in a liquid is their use as fuels in homogeneous atomic reactors. Aqueous uranium and thorium slurries have been used for this purpose and were shown to exhibit non-Newtonian flow behavior (ref. 3).

In both applications these slurries are passed through pipeline systems. Thus, for the design of these pipeline systems it is important to know the pressure losses in the straight pipe sections as well as in the pipeline transitions. Pressure losses of non-Newtonian materials in straight pipeline sections are treated in references 4 to 8. Pressure losses of Newtonian materials in transitions are treated in references 4, 9, 10, and 11 for turbulent flow in the pipeline system. For laminar flow of Newtonian materials in the pipeline system, only the pressure losses in contractions (ref. 12) and in elbows (ref. 13) have been treated in the literature. 4204

The purpose of this report is to augment the experimental data for pressure losses of non-Newtonian materials in straight pipelines and to present experimental pressure-loss data for non-Newtonian materials in pipeline systems and for Newtonian materials whenever needed for purposes of comparison. This report also intends to formulate transition loss coefficients for fittings such as elbows and valves in such a way that they can be independent of the flow rate even for laminar flow of the Newtonian and non-Newtonian materials in the pipeline system.

This report presents experimental pressure-loss data for three Newtonian liquids and two non-Newtonian slurries in laminar, transitional, and turbulent flow. One of the non-Newtonian slurries had the flow behavior of a Bingham plastic material, while the other one behaved like a pseudoplastic material. The pressure losses were measured over straight pipe sections and transitions for 1-inch- and 3/8-inch-diameter pipes. Transitions such as 90° elbows; gate, ball, plug, and globe valves; contractions; and expansions are considered.

SYMBOLS

C_L transition loss coefficient for total-pressure loss

C_s transition loss coefficient for static-pressure loss

D pipeline diameter, in.

f yield value, dynes/cm^2

G rate of shear, sec^{-1}

L length of pipeline, in.

N structure number

ΔP total-pressure loss, dynes/cm^2 and lb/sq in.

Re Reynolds number

U plastic viscosity, poise

v mean velocity (from flow rate), cm/sec and ft/sec

η apparent viscosity, τ/G, poise

μ Newtonian viscosity, poise

ρ density, g/cc

τ shearing stress, dynes /cm^2 and lb/sq in.

φ friction factor

DESCRIPTION OF PIPELINE FLOW SYSTEM

A photograph of the pipeline flow system is shown in figure 1 and a schematic sketch is shown in figure 2. The pipeline system consisted of 3-foot straight and uniform pipeline sections, which were connected by flanges so that pipe fittings could be inserted or removed between any two sections. Two 12-inch-diameter tanks, one on each end of the pipeline system, were provided to hold about 20 gallons of the material being tested. Pressures of 1 to 100 pounds per square inch could be applied to either tank. This made continuous measurements possible without sample changing. The flow rate was measured with floats. The float consisted of a ball attached to a rod. The rods moved up and down in a glass tube with a change in liquid level in the tanks, so that their positions in each tank could be registered within a measured time. Each straight pipeline section and each pipe fitting was provided with four pressure taps on each end, so that pressure differentials could be measured over each section or transition and also across each flange connection. All measurements were static-pressure measurements. The differential pressures were obtained on a 10-foot-high mercury manometer, so that differential pressures up to 60 pounds per square inch could be measured.

Traps as shown in figures 1 and 2 were provided between each pipeline pressure tap and each manometer connection. This was done so that the liquid on the two sides of each differential manometer would have nearly the same level even when a pressure differential existed between these two taps, and also to prevent the liquid material from entering the airlines leading to the manometers at line pressures up to 100 pounds per square inch. The traps are especially important if non-Newtonian materials are being measured. Non-Newtonian materials frequently require a certain minimum pressure before they flow. The material, after being displaced into the trap, should flow back into the pipeline by gravity when the line pressure is released. Gravity, however, does not always supply sufficient pressure to empty the trap completely when the material is non-Newtonian.

Thus, non-Newtonian materials might slowly accumulate in the traps. This would introduce an error in pressure indication if the trap on one side of the differential manometer fills faster than on the other side. To prevent this, two measurements were made at the same applied pressure by flowing first from one tank and then from the other, so that the traps on both sides of each differential manometer were alternately subjected to the higher pressure. This procedure was carefully followed when the higher pressure measurements were made. In addition, the error due to a difference in liquid level was minimized by designing the traps so that the volume displaced by the line pressure has a large diameter-to-length ratio. The traps had to be mounted close to the pipeline to minimize the pressure-time fluctuations of the manometer indications. The pressure readings were obtained in from 10 to 100 seconds, depending upon the flow rate, by taking photographs of the manometer board at each applied pressure. Thermocouples were mounted flush with the inside pipe walls in the center of each straight pipeline section and in both tanks. The temperatures ranged around 25° C. They were recorded on a self-balancing strip-chart potentiometer. 4204

Two nominal pipe sizes were used. They were 1-inch pipe, with a measured inside diameter of 0.95 inch, and 3/8-inch pipe, with a measured inside diameter of 0.50 inch. The flanges that were provided on each end of a straight section and fitting were the same for both pipe sizes. This made it possible to alternate pipes of different sizes in the pipeline setup and thus to obtain measurements for contractions and expansions in the ratio of 1:2. Other ratios of 1:12 and $1:\infty$ (into tank) of contractions and expansions were obtained in and out of the respective tanks.

The pipelines and fittings were not especially selected nor were they machined or tapped to fit each other. They were used in random order, so that it was possible to have a better fit for one test and a poorer fit for the next test. This by necessity caused some spread in the data, since a pressure loss due to a preceding poor fit will frequently extend over some of the following length of pipeline. Because of the random selection of pipelines and fittings this spread of data should be typical for any commercial pipeline system where often no selection or special care can be exercised.

The fittings over which the pressure losses were measured were 90° elbows and the most commonly used gate, ball, plug and globe valves. These fittings were chosen since the pressure losses of the more complicated fittings are frequently listed as multiples of the pressure losses of these typical fittings. The loss of a 90° elbow can be used, for instance, to represent the loss in a T fitting.

PRESSURE LOSSES IN STRAIGHT PIPELINE SECTIONS

The total-pressure loss ΔP in a pipeline system can be given as (refs. 4 and 9)

$$\Delta P = \rho \frac{v^2}{2} \left[\frac{L}{D} \varphi + C_L \right] \tag{1}$$

The transition loss coefficient C_L is zero in a straight and uniform pipeline section. The friction factor φ for Newtonian, Bingham plastic,
4204 and pseudoplastic materials can be obtained from a friction-factor diagram as shown in references 4 and 5 if the flow properties of the material and the mean velocity in the pipeline are known. The mean velocity v is obtained by dividing the measured flow rate in the pipeline by the cross-sectional area of the pipeline. The flow properties of the material and the density ρ are determined separately. The length-to-diameter ratio L/D of the pipeline is determined from length and diameter measurements.

Pressure losses of two slurries, a mineral oil, and a silicone fluid were measured in the 1-inch pipeline system. Only laminar flow was obtained. The two slurries were a suspension of titanium dioxide particles in water and a suspension of particles of magnesium in a hydrocarbon fuel. Measurements with an automatic concentric-cylinder rotational viscometer (ref. 14) indicated that the titanium dioxide slurry was a Bingham plastic material with a constant plastic viscosity and yield value at any constant temperature, while the magnesium slurry was a pseudoplastic material with a structure number and an apparent viscosity that decreased with increasing rates of shear for any constant temperature. The mineral oil and the silicone fluid were Newtonian liquids. The viscosity and density of the mineral oil were measured at the same temperature at which its pressure loss was determined over the pipeline. These two values were used in equation (1) to verify the measured dimensions of the pipeline. The calculated values of L/D agreed with the measurements within 2 percent.

The pressure-loss data for the titanium dioxide slurry in the 1-inch straight pipeline are plotted in figure 3. In this figure the squared mean velocity is plotted against the pressure loss. The solid line is calculated from equation (1). In order to do this, the friction factor φ has to be obtained. It was determined for the measured plastic viscosity of 0.26 poise and the measured yield value of 320 dynes per square centimeter by using the generalized friction diagram (refs. 4 and 5). The density ρ of this material was 1.18 grams per cubic centimeter. These values were determined at 25° C. The points were experimentally obtained for four pipeline sections, the length-to-diameter ratio of each pipeline being 34. In order to compare the pressure losses of a non-Newtonian material in a pipeline with those of a Newtonian liquid, the dashed line is calculated for a Newtonian liquid of the same viscosity and density as the Bingham plastic slurry. Turbulent flow for both materials is indicated by curve ABC. The difference of the flow behavior of these two materials in a pipeline is very striking. Turbulence sets in at much higher flow rates or flow velocities in the non-Newtonian material (namely, at about B) than in the Newtonian liquid (where it sets in at about A). The displacement between the two curves indicates that a substantial pressure difference is required before the slurry starts to flow, while at that same pressure the flow of the Newtonian liquid is already turbulent.

Figure 4 is a flow curve of rate of shear G against shearing stress τ of the magnesium slurry. This curve was measured with the concentric-cylinder rotational viscometer, which was built at the NACA Lewis laboratory (ref. 14). This flow curve was produced at a constant temperature of 25° C. Since this slurry is pseudoplastic, a flow curve taken over an extended range of rates of shear is required to calculate the structure number N and the apparent viscosity η, which is the ratio of τ/G for the respective rates of shear in the pipeline. The structure number N was obtained by plotting $\log G$ against $\log \tau$. Then N is the slope of this line. Since the plot is not quite linear, different values for N were obtained for different ranges of rate of shear (ref. 4), so that $N = 4.3$ for $G \lesssim 500 \text{ sec}^{-1}$, $N = 3.1$ for $500 \text{ sec}^{-1} \le G \le 1000 \text{ sec}^{-1}$, and $N = 2.5$ for $G \ge 1000 \text{ sec}^{-1}$. To determine both flow properties at the same rates of shear at which the pressure losses in the pipeline were measured, those rates of shear had to be determined. The rate of shear in the pipeline, which is a function of the mean velocity in the pipeline and of N, is $G = 2v(N + 3)/D$ (refs. 4 and 5). The flow curve was measured up to a rate of shear of about 3500 sec^{-1}. For higher rates of shear in the pipeline, the straight-line plot of $\log G$ against $\log \tau$ was used for extrapolation.

The squared mean velocity is plotted against the pressure loss of a magnesium slurry in a 1-inch straight pipeline in figure 5. The solid line is calculated from equation (1) by using the flow properties at the rates of shear prevailing in the pipeline for the respective flow rates to calculate φ. The density ρ was 1.10 grams per cubic centimeter. The points are the experimentally measured pressure losses, which were obtained for various test runs and different pipelines. Above a Reynolds number of 1200, transitional flow might account for the deviations between the experimental points and the calculated line. This is suggested because it was found that transitional flow starts at about $Re = 1200$ in the 3/8-inch pipeline, as will be discussed later.

The magnesium slurry and three Newtonian liquids were measured in the 3/8-inch pipeline. In this pipeline laminar, transitional, and turbulent flow could be obtained with the available pressures. The same mineral oil that was used in the 1-inch pipeline was also used to check the 3/8-inch pipeline dimensions. The pressure-loss data for a Newtonian liquid are shown in figure 6, and for a magnesium slurry in figure 7. Again, the squared mean velocity is plotted against the pressure loss. The Newtonian liquid is a silicone fluid and has a viscosity of 0.1 poise and a density of 0.98 gram per cubic centimeter at 25° C. The solid lines are calculated from equation (1), and the points are the experimental measurements obtained for various test runs and different pipelines. At Reynolds numbers of $Re \lessapprox 1200$ the flow in both the silicone fluid and the slurry is laminar, and the experimental points corroborate the calculations. That is also the case for Reynolds numbers $Re \geq 3100$, except that then the flow is turbulent. In turbulent flow the calculations were made for smooth pipelines. In the Reynolds number range between 1200 and 3100 the flow is apparently transitional; and, since these pipelines were not perfectly matched and surfaced, the experimental data deviate from the calculations in this region. However, the deviations even in this region are small when considering practical applications.

In turbulent flow the rates of shear in the pipelines are rather high, but difficult to determine. Therefore, an apparent viscosity extrapolated to infinite rate of shear was used to determine Re in the case of the pseudoplastic magnesium slurry. To obtain this viscosity the reciprocal values of the rates of shear $1/G$ were plotted against the respective apparent viscosities η (ref. 15). This plot was almost a straight line above $G = 5000\ \text{sec}^{-1}$, with an intercept at the apparent-viscosity axis. This intercept at $1/G = 0$ represents the apparent viscosity at infinite rate of shear. This apparent viscosity was equal to 0.20 poise and is used in all turbulent-flow calculations for the magnesium slurry. No structure number N is required, since in turbulent flow the pressure loss of a non-Newtonian material depends on one flow property only, the viscosity (ref. 4). To obtain turbulent flow, especially with the slurry, pressures above 60 pounds per square inch had to be applied. At those pressures minute leaks frequently developed in the airlines to the manometers. When the non-Newtonian slurries were being tested, these leaks caused the traps to fill rapidly with slurry; and, since these slurries had difficulty in flowing back into the pipelines, these leaks led to errors in the pressure-loss measurements. Thus, fewer and less reliable data were obtained for turbulent flow than for laminar and transitional flow.

PRESSURE LOSSES IN PIPELINE TRANSITIONS

The transition loss coefficient C_L for a pipeline transition as given in equation (1) is the coefficient for the total-pressure loss. In this investigation static-pressure losses were measured, from which C_L was determined. In most fittings such as elbows and valves the transition loss coefficient C_L equals the transition loss coefficient C_s, which is obtained from static-pressure measurements. However, in transitions such as expansions and contractions that is not the case. The following relations (ref. 10) exist:

For expansions:

$$C_L = C_s + 1.0 - v_0^2/v^2 \tag{2}$$

For contractions:

$$C_L = C_s - 1.0 + v_0^2/v^2 \tag{3}$$

where v_0 is the mean velocity in the larger pipe and v is the mean velocity in the smaller pipeline.

In the literature, the pressure losses in pipe transitions are frequently given in numbers of "velocity heads." For transitions such as contractions and expansions, which are considered to have "zero" length, the number of velocity heads is identical to the transition loss coefficient C_L. This, however, is not necessarily so when considering transitions such as elbows, valves, and other fittings. Fittings in analogy to a pipeline have a definite length, which is the length that separates the two pipeline sections between which the fitting is inserted. Therefore, the pressure losses that are measured over these pipeline transitions are considered as composed of a "transition loss" and of an "equivalent pipe loss." The latter is equivalent to the pressure loss in a straight pipeline section with a length equal to that of the transition and with a diameter equal to that of the connecting pipelines. If the equivalent pipe loss is small compared with the transition loss, the number of velocity heads and the transition loss coefficient C_L will be approximately equal. This is usually the case in turbulent flow. However, in laminar flow the equivalent pipe loss can become large compared with the transition loss; and, since in laminar flow the equivalent pipe loss increases with increasing Reynolds number at a rate that is less than proportional to v^2, the number of velocity heads will then not be a constant. Thus, to determine transition loss coefficients, which are independent of the velocity in the pipeline system, the equivalent pipe losses were deducted from the measured pressure losses. This was also done to obtain the transition loss coefficients for contractions and expansions, since the pressure losses of these transitions also had to be measured over a finite length. The equivalent pipe loss is calculated from equation (1) by treating the transition as if it were a straight and uniform pipeline for which $C_L = 0$ with a length equal to that over which the pressure loss was measured and with a diameter equal to that of the connecting pipelines. In contractions and expansions the equivalent pipe loss in the larger diameter pipeline can usually be neglected. In laminar and turbulent flow the equivalent pipe losses were calculated by using equation (1) and references 4 and 5; however, in transitional flow the experimental data were used to calculate the equivalent pipe losses.

The transition losses that are available in the literature for Newtonian liquids are given either as numbers of velocity heads or as transition loss coefficients. These values from the literature are listed in table I together with the transition loss coefficients obtained in these studies.

For turbulent flow, experimental data are available in the literature for some valves, elbows, contractions, and expansions (refs. 9 and 11). In laminar flow the transition loss coefficients for contractions, also called "entrance loss coefficients," have been calculated (ref. 12) and experimentally verified. Some experimental data are given for elbows for laminar and turbulent flow in reference 13. The experimental data for elbows for laminar flow (ref. 13) were recalculated to represent numbers of velocity heads. These values were found to decrease with increasing Reynolds number. This is not surprising, since the equivalent pipe loss would increase with increasing Reynolds numbers at a rate that is less than proportional to v^2.

The transition loss coefficients obtained in these studies for elbows in laminar flow seem independent of the velocity in the pipeline. An example of this is shown in figure 8. Figures 8 and 9 are examples of the experimental pressure losses obtained over the different transitions, from which the transition loss coefficients were calculated. Figure 8 gives the pressure-loss measurements across a 3/8-inch 90° elbow in laminar and transitional flow. Figure 9 represents the pressure losses over 1-inch gate and globe valves in laminar flow. The points are the experimental pressure-loss measurements obtained for two test runs. The solid lines represent the computed pressure losses over the transition when the calculated constant value of C_L is used in equation (1), and the dashed lines represent the equivalent pipe losses obtained from equation (1) for $C_L = 0$.

Table I indicates a moderate spread in the data for the transition loss coefficients that were obtained for the same transition and pipe size for the different Newtonian liquids. This spread, as previously mentioned, is due to the fact that the pipelines and pipe transitions were used in random order and no attempt was made to fit them to each other. The transition loss coefficients from any test without change in setup were within 10 percent or ±0.2, whichever was greater. But the coefficients tabulated in the table are averages obtained from many tests, where some measurements were made after a complete reassembly of the pipeline system. Therefore, the errors in the tabulated data could be greater than ±10 percent or ±0.2 because of changes in alinement for the different tests. Two distinct average values were obtained with the magnesium slurry for the transition loss coefficients in the 3/8-inch globe valve and the 3/8-inch elbow. The two values for the globe valve are within the stated uncertainty. The two values for the elbow differ appreciably. These measurements, in contrast to all others, included the length of two flanges. Since the transition loss coefficient over a flange could range from 0 to about 0.4 depending upon the alinement, a difference in alinement of the elbow in regard to its neighboring pipelines could account for the fact that two distinct values were repeatedly obtained for this transition loss coefficient. The measurements that gave one of these two values are shown in figure 8.

A study of table I shows that the transition loss coefficients obtained for the different Newtonian liquids vary over the same range as those obtained for the non-Newtonian materials. This indicates that the transition loss coefficients obtained from Newtonian liquids are valid at least for such non-Newtonian slurries as were treated in these studies.

The transition loss coefficients for the fittings, which were obtained for laminar flow in the pipelines, do not seem to differ much from those obtained for turbulent flow. This is not surprising, since the pressure losses in fittings are due to a combination of contractions and expansions and the transition loss coefficients C_s for static-pressure losses in contractions are not much higher for laminar flow than for turbulent flow, namely in the ratio of 2.2 to 1.5. In fact, a careful study of table I indicates that the transition loss coefficients for the fittings might be somewhat smaller in turbulent flow than in laminar flow.

Two pipe sizes were used to determine whether the transition loss coefficient changes with pipe size. Even though the data are not sufficient to give a definite answer, they indicate that the transition loss coefficients are not much affected by the pipe size. Again, this would be expected, since the transition loss coefficients for the contractions and expansions also are hardly affected by the pipe size. The transition loss coefficient for the globe valve is somewhat higher for the 3/8-inch valve than for the 1-inch valve. The ratio of the orifice diameters of these two valves is 2.3, while the ratio of the pipe diameters is only 1.9. Therefore, the difference in transition loss coefficients for these two valves might be explained by differences in the construction of the two valves rather than in the pipe size.

CONCLUDING REMARKS

Experimental pressure-loss data have been presented for two non-Newtonian slurries that flow like a Bingham plastic and a pseudoplastic material. These data show that pressure losses in straight and uniform pipelines can be calculated if the density, the flow properties for the pipeline flow condition, and the mean velocity in the pipeline are known. For the pseudoplastic slurry these data are determined for laminar, transitional, and turbulent flow in the pipeline. The calculated and experimental pressure losses agree closely in the laminar- and turbulent-flow regions but deviate for transitional flow. However, these deviations are negligible in most cases where practical applications are considered.

Pressure losses over pipeline transitions such as valves, 90° elbows, contractions, and expansions were measured for these same slurries and some Newtonian liquids for laminar, transitional, and turbulent flow in the pipeline. Transition loss coefficients, which are independent of the flow rate in the pipelines, even for laminar flow in the pipeline system, were determined for these transitions. The data indicate that for non-Newtonian slurries such as those considered in these studies the transition loss coefficients are equal to those obtained for Newtonian liquids within the spread of the experimental data. Thus, in designing a pipeline system for similar non-Newtonian slurries, such as a fuel system for a ram jet or an afterburner or for a homogeneous reactor, the sizing of the pipeline system can be done by using the transition loss coefficients established for Newtonian fluids.

The transition loss coefficients for the fittings seem to be nearly the same for laminar and turbulent flow in the pipeline system, at least within the spread of the experimental data, except that those for contractions are higher in laminar than in turbulent flow. The data obtained for the two pipeline sizes seem to indicate that the transition loss coefficients are almost independent of the pipeline size.

Lewis Flight Propulsion Laboratory
National Advisory Committee for Aeronautics
Cleveland, Ohio, September 20, 1956

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TABLE I. - TRANSITION LOSS COEFFICIENTS FOR PIPELINE TRANSITIONS

Flow	Transition	Mineral oil (μ = 1.6, ρ = 0.88)		Silicone fluid (μ = 0.1, ρ = 0.98)		Silicone fluid (μ = 0.2, ρ = 0.98)	Mg slurry (pseudoplastic) N and η		TiO_2 slurry (Bingham U = 0.26, f = 320)	Literature, Newtonian fluids	
		Transition loss coefficients C_s for pipe diameters of -									
		0.50"	0.95"	0.50"	0.95"	0.50"	0.50"	0.95"	0.95"	C_s	C_L
Laminar	Expansion:										
	1:∞					0					
	1:12		0					0	0		
	1:2	0		0		0.1	0				
	Contraction:										
	∞:1					2.2				2.16	1.16
	12:1		2.2					2.2	2.2	2.16	1.16
	2:1	2.0		2.1		2.1	2.0			2.16	1.16
	Valve:										
	Gate	1.0	0.4	0.8		0.7	0.7	0.8			
	Ball		1.1					.7	1.2		
	Plug	1.2		1.3			1.1				
	Globe	7.6	4.2			6.5	5.8,7.1	4.6	4.4		
	90° Elbow	0.6	1.5	1.5		1.3	0.8,1.5	1.4	1.5	0.7-2.5	[a,b]0.7-2.5
Transi-tional	Expansion:										
	1:∞					0					
	1:12										
	1:2			0.1			0				
	Contraction:										
	∞:1					1.5					
	12:1										
	2:1			1.4			1.4				
	Valve:										
	Gate			0.7		0.6	0.6				
	Ball										
	Plug			1.3			1.0				
	Globe			7.7	4.3	4.5	5.0				
	90° Elbow			1.0		1.3	0.8,1.5				
Turbu-lent	Expansion:										
	1:∞					0				0	1.0
	1:12									0	1.0
	1:2			0.1			0			-0.4	.6
	Contraction:										
	∞:1					1.5				1.5	0.5
	12:1									1.5	.5
	2:1			1.4			1.3			1.3	.3
	Valve:										
	Gate			0.7						0.1-0.2	[a]0.1-0.2
	Ball										
	Plug			1.3			1.0				
	Globe			7.7						6-10	[a]6-10
	90° Elbow			0.7			0.9			0.7-1.3	[a]0.7-1.3

[a]Number of velocity heads.

[b]Calculated from experimental data of ref. 13.

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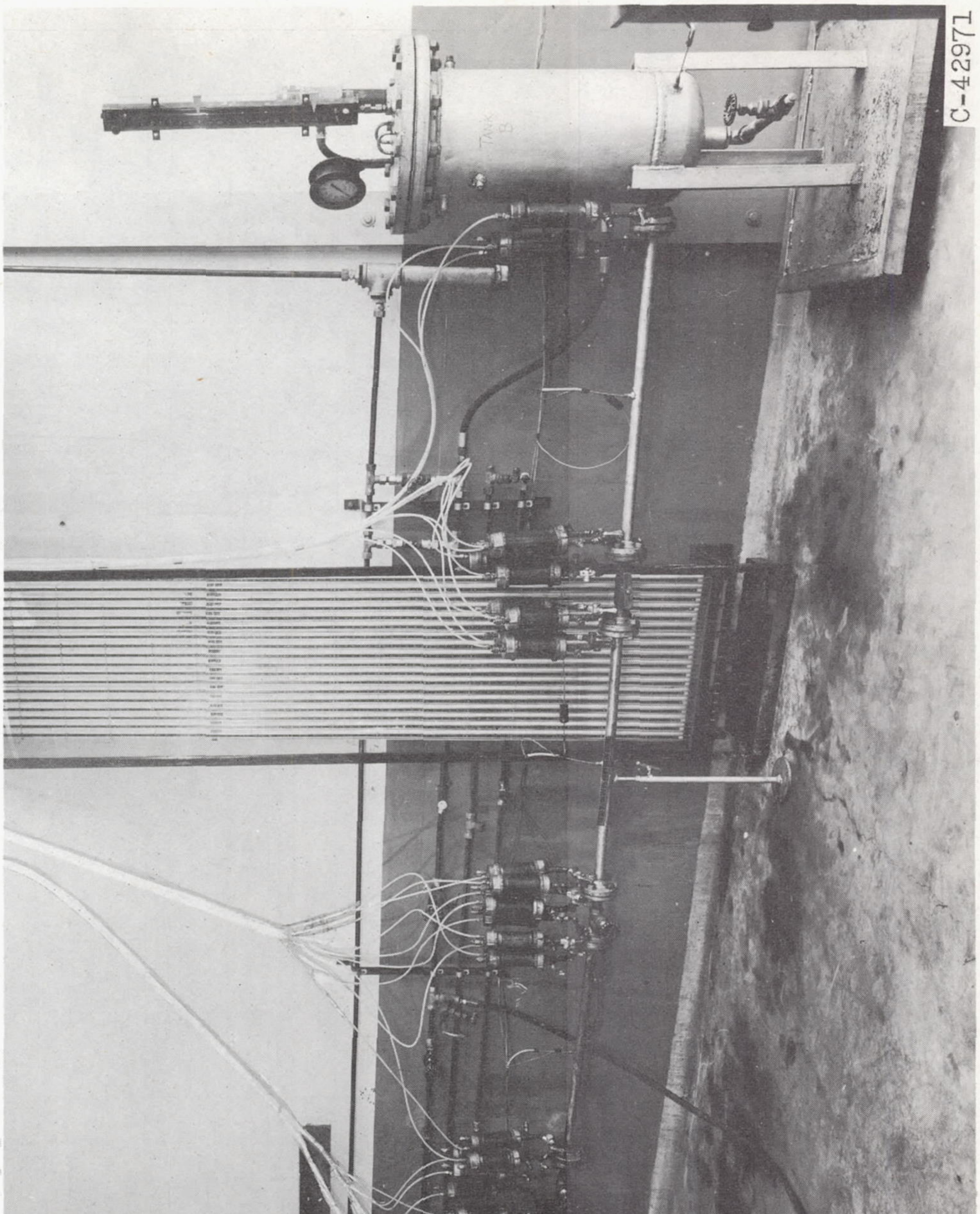

Figure 1. - Photograph of pipeline flow system.

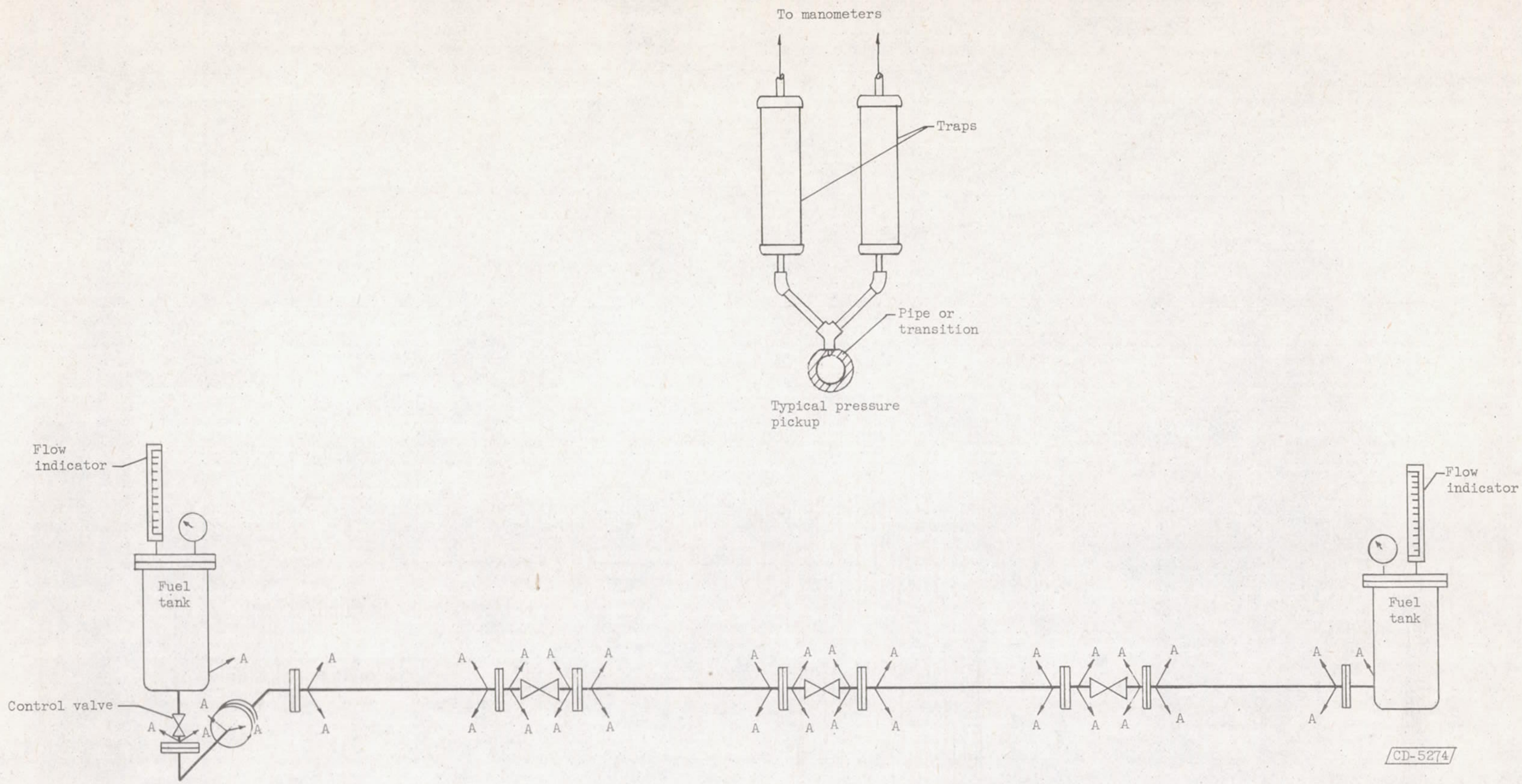

Figure 2. - Schematic sketch of pipeline flow system ("A" indicates connection to manometer traps).

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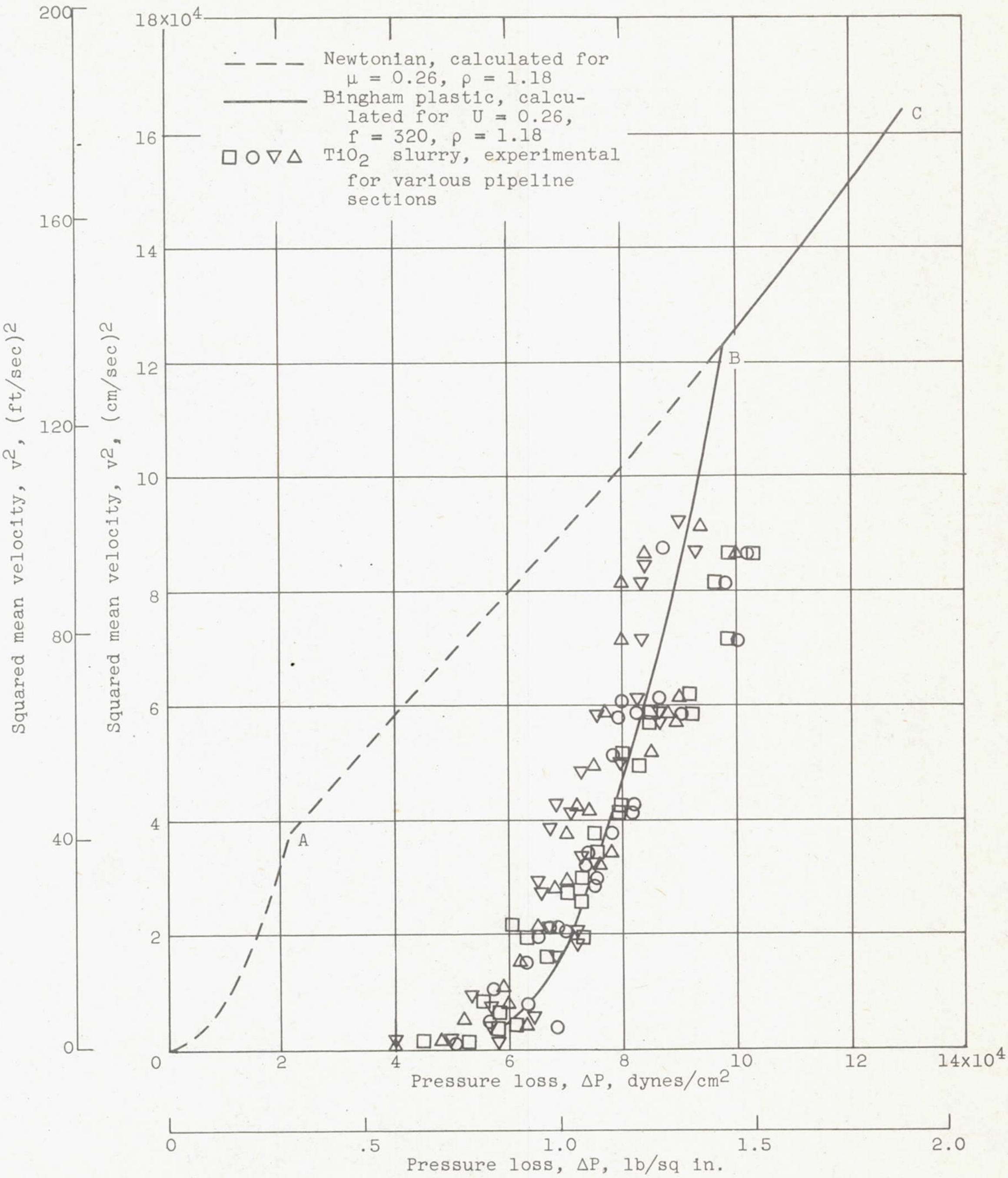


Figure 3. - Pressure losses of titanium dioxide slurry in 1-inch straight pipelines. Length-diameter ratio, 34.

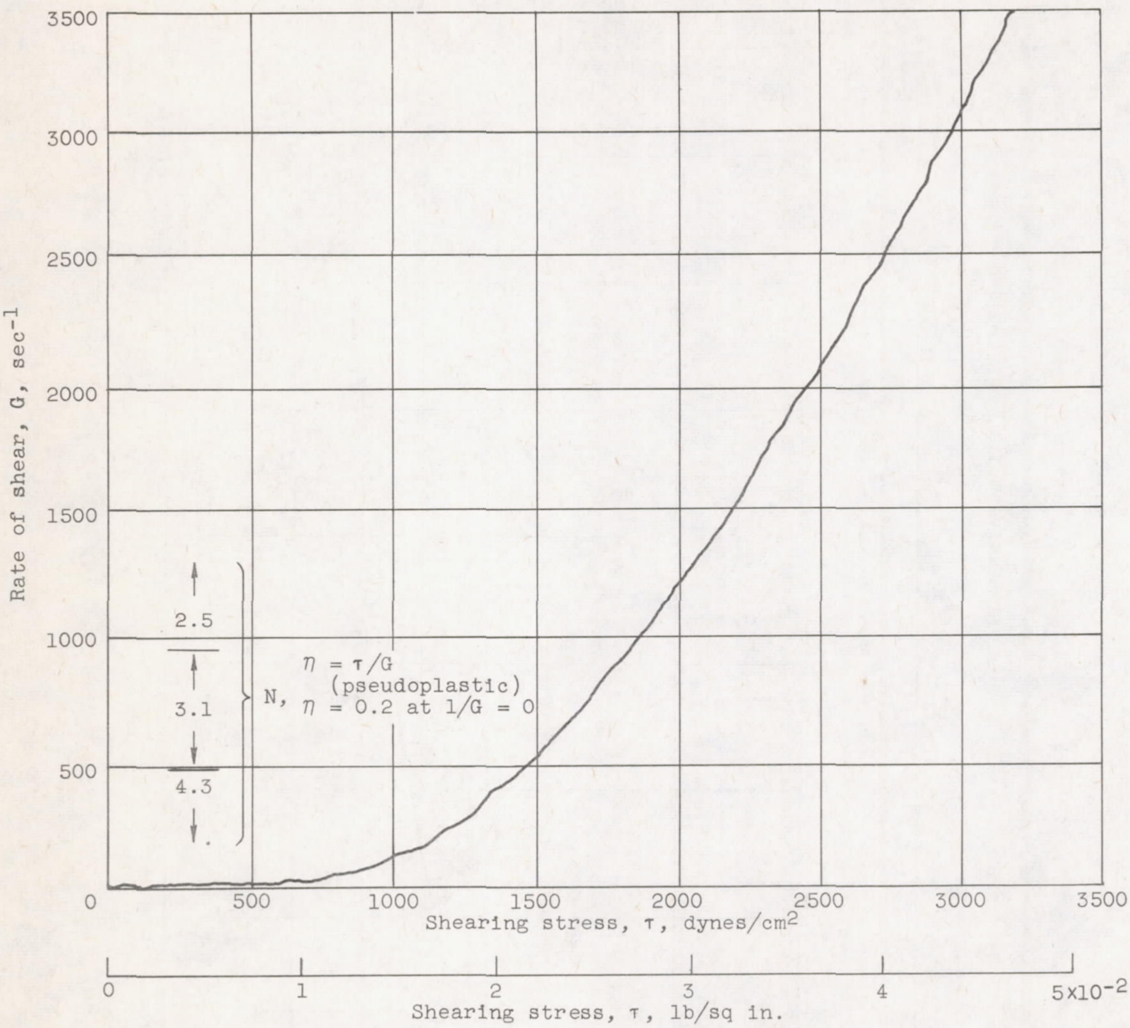


Figure 4. - Flow curve of a magnesium slurry.

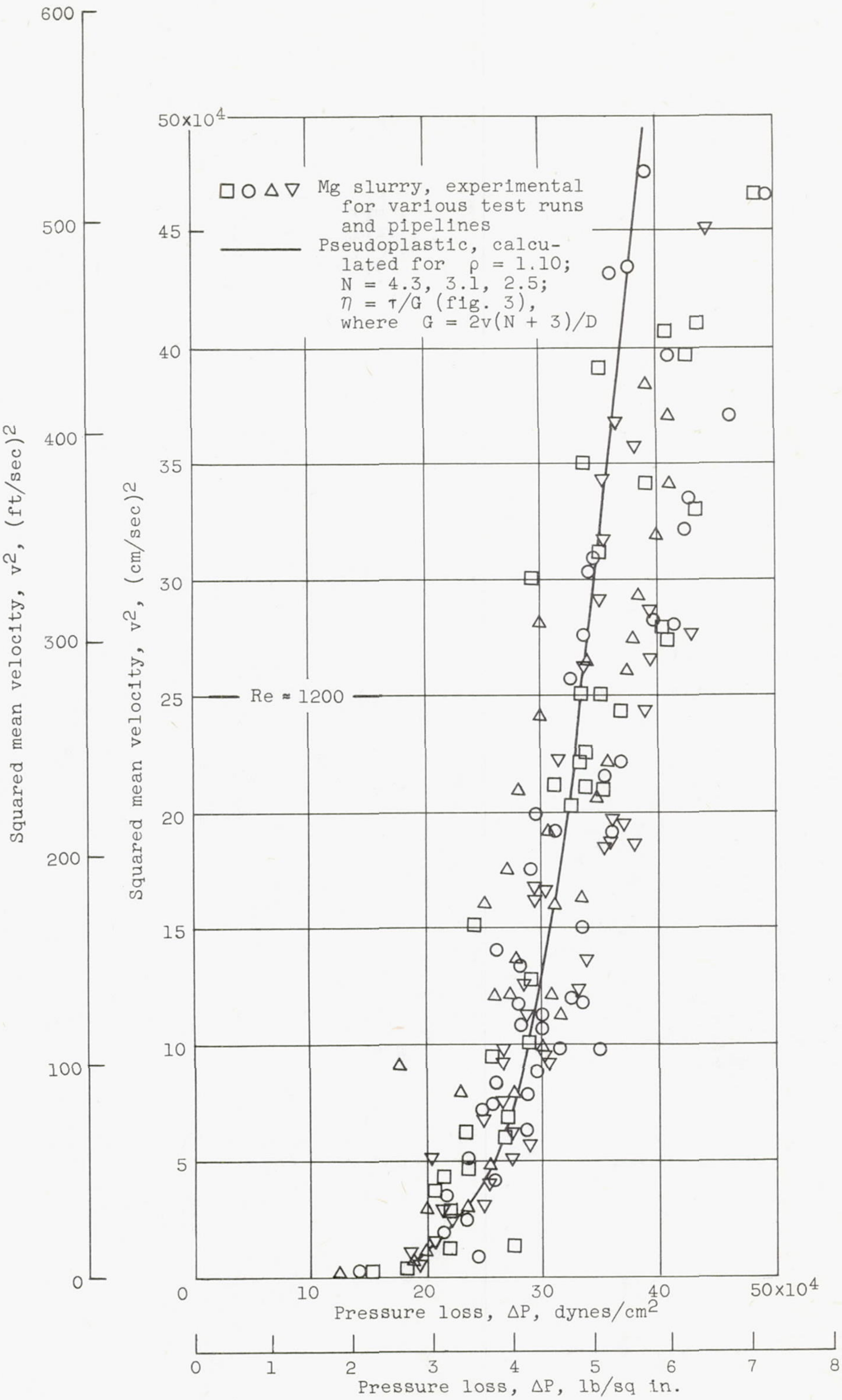


Figure 5. - Pressure losses of magnesium slurry in 1-inch straight pipelines. Length-diameter ratio, 34.

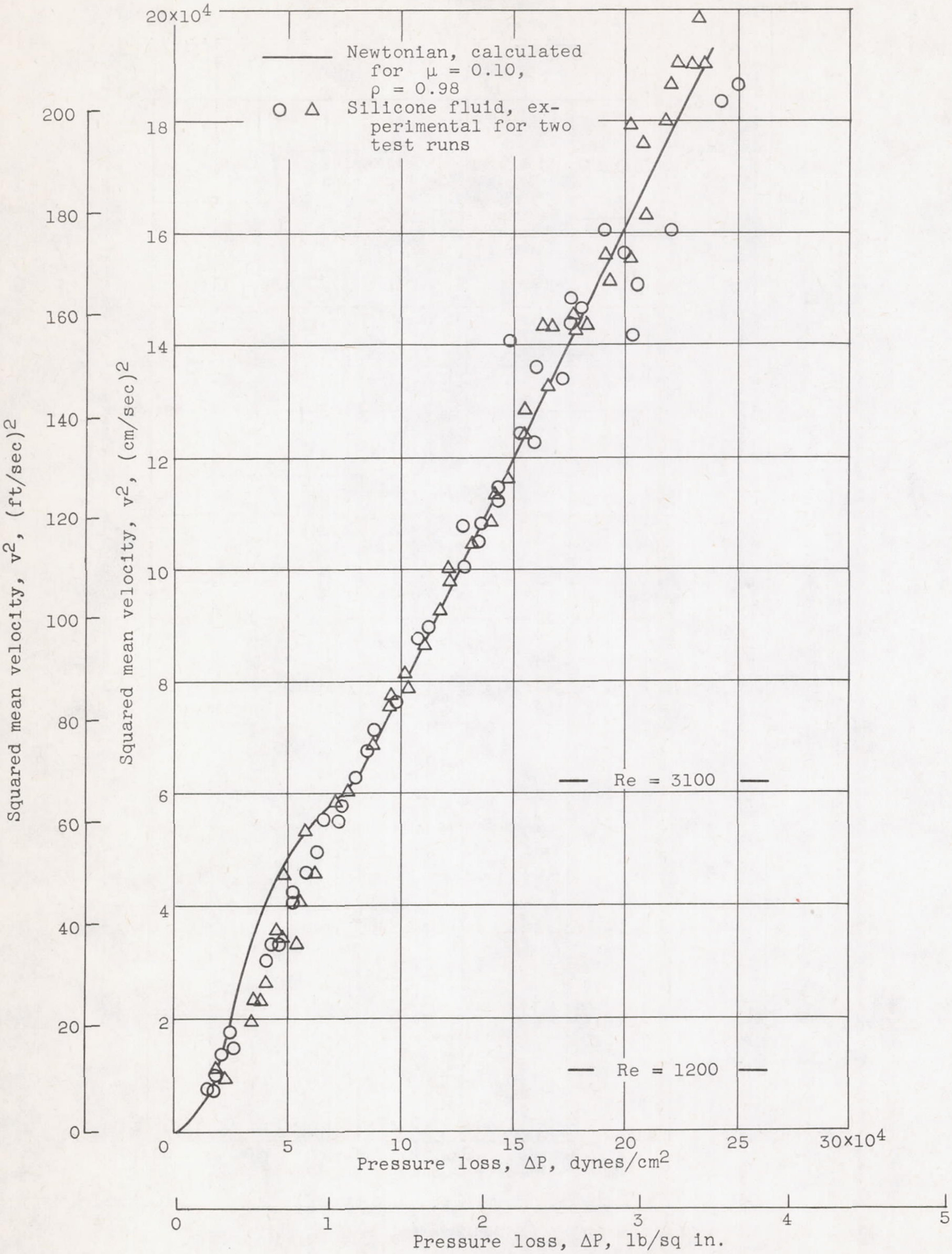


Figure 6. - Pressure losses of silicone fluid in 3/8-inch straight pipelines. Length-diameter ratio, 66.

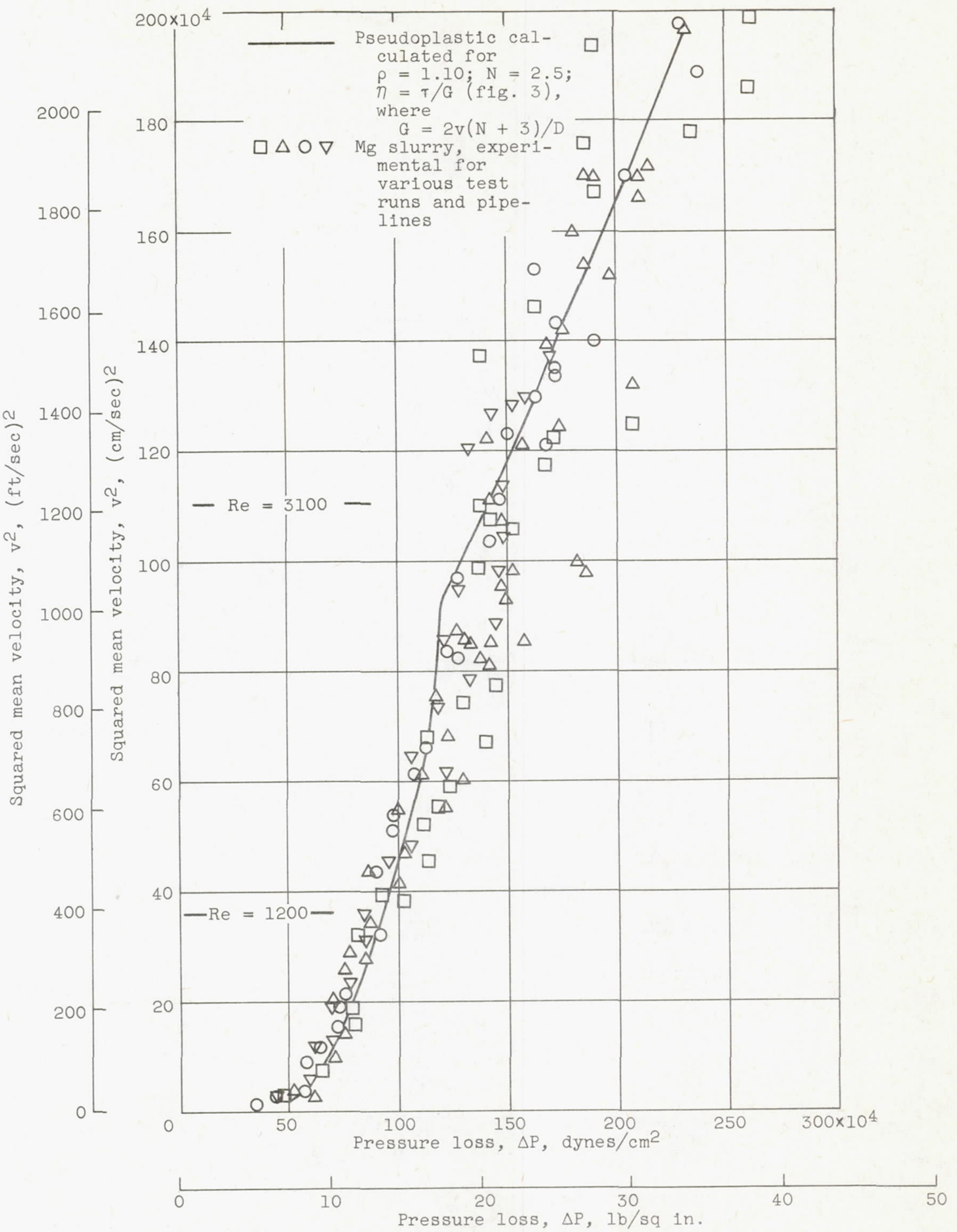


Figure 7. - Pressure losses of magnesium slurry in 3/8-inch straight pipelines. Length-diameter ratio, 66.

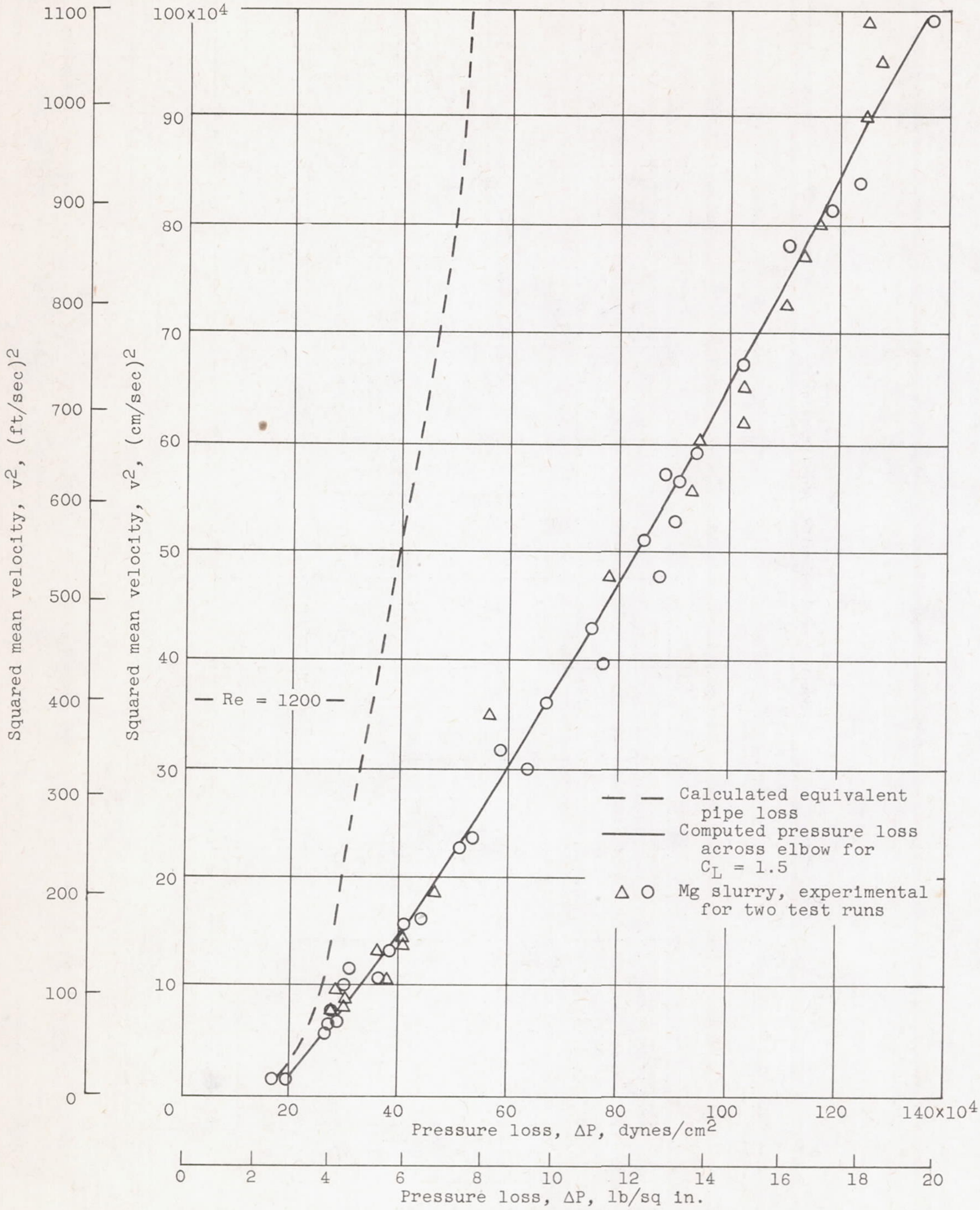
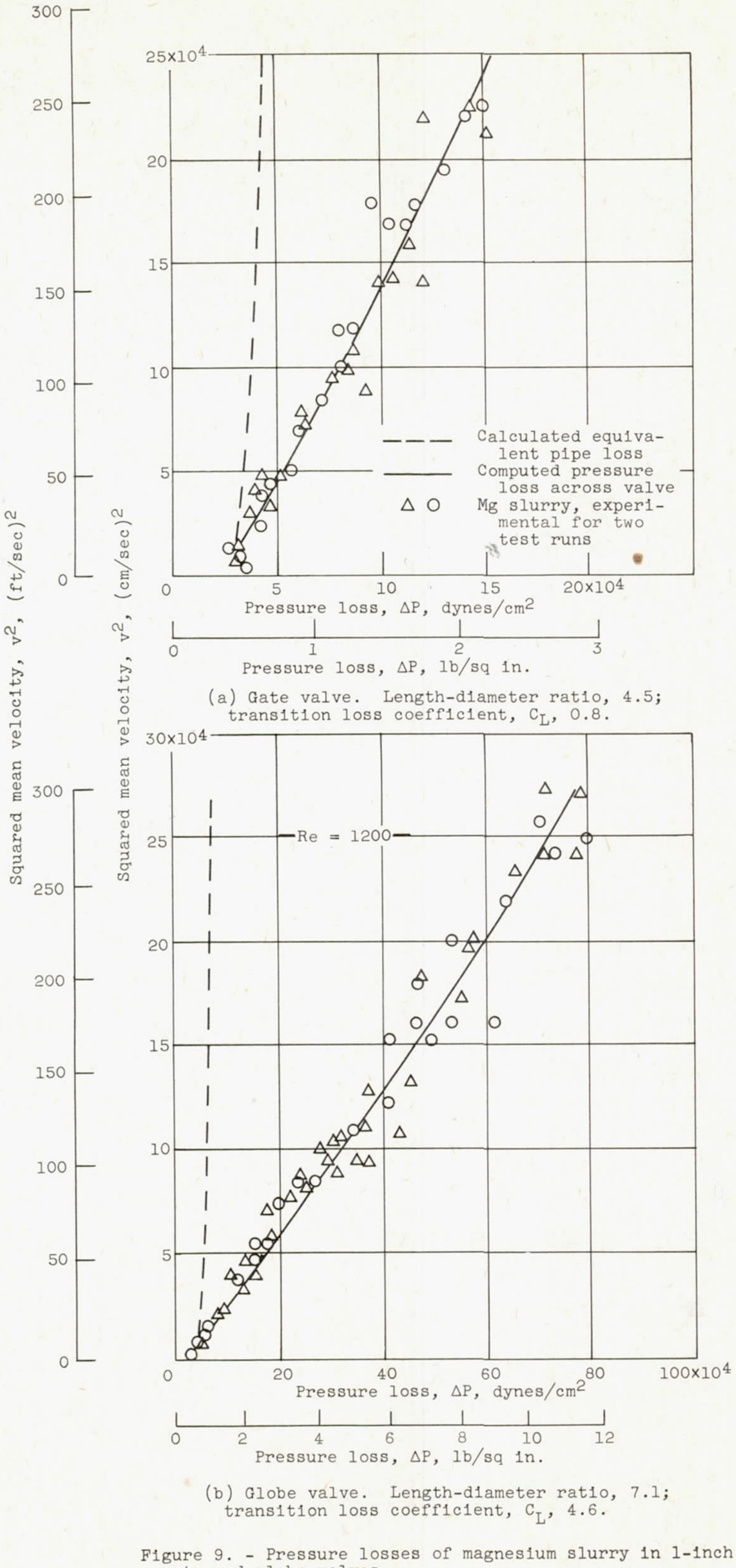


Figure 8. - Pressure losses of magnesium slurry in 3/8-inch 90° elbow. Length-diameter ratio, 24; transition loss coefficient, C_L, 1.5.



(a) Gate valve. Length-diameter ratio, 4.5; transition loss coefficient, C_L, 0.8.

(b) Globe valve. Length-diameter ratio, 7.1; transition loss coefficient, C_L, 4.6.

Figure 9. - Pressure losses of magnesium slurry in 1-inch gate and globe valves.