

AN EXPERIMENTAL STUDY OF TWO-PHASE ONE-COMPONENT
FLUID FLOW IN CIRCULAR PIPES

by

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INTRODUCTION

As a liquid flows inside a pipe, there is a decrease in the static pressure caused by the friction of the fluid moving past the walls. If the flow is adiabatic and if the temperature of the fluid is such that its vapor pressure is only slightly less than the static pressure of the line, then the friction may reduce the pressure until it equals the vapor pressure of the fluid. At this particular condition of equal pressures, some of the liquid will start to boil or "flash". From this point to the discharge, the pipe is said to be transporting a "flashing fluid". A flashing fluid also exhibits a resistance to flow which manifests itself in the form of friction. However, the magnitude of the frictional resistance is quite different from that of the pure liquid. The experimental evaluation of friction accompanying the adiabatic flow of a two-phase fluid through straight pipe was the subject of this investigation.

Knowledge of the pressure losses of flashing fluids frequently is of value in industrial plant designs. All steam generating facilities are confronted with such a problem of predicting the pressure loss of a flashing fluid. Hydrocarbon processing plants have a similar problem. The problem in such an industrial situation is to predict a proper pipe size required to transport the necessary volume of fluid while staying within certain pressure loss limitations. The present day design calculation procedures are both tedious and inaccurate. Because a pipe which is too small may prevent the equipment from operating

properly, the usual practice is to choose an excessively large size of pipe. This over-designing is a needless expense which could be avoided were a better knowledge of the two-phase fluid friction losses available.

THEORETICAL ANALYSIS

The term "flashing" was used in preference to "boiling" since the former denotes a mixture of vapor and liquid in which additional vapor is being formed at the expense of sensible heat of the liquid. This mechanism may be shown for water and steam flowing adiabatically a differential distance by the following steps:

1. As the pressure decreases, the saturation temperature decreases causing a corresponding decrease in the enthalpy or heat content of the water.
2. The heat liberated by the decrease in enthalpy of the water goes into latent heat of vaporization of some water to maintain the total enthalpy of the system constant.
3. The specific volume of the water and steam mixture increases rapidly since a small amount of liquid occupies a large volume when vaporized.
4. If the mass rate of flow is to be constant, the velocity of the mixture must be increased since the specific volume increased. The energy for accelerating the mixture causes a further decrease in the pressure.

The above qualitative mechanism may be expressed mathematically with a mechanical energy balance as developed in

standard texts (15,20) on fluid flow.

$$Z_1 + P_1 v_1 + \frac{u_1^2}{\alpha g} + \int P dv = Z_2 + P_2 v_2 + \frac{u_2^2}{\alpha g} + \Sigma F \quad (1)$$

This equation is valid for a single mass unit of any elastic fluid where the energy of the ultimate particles is constant. An examination of each term will be useful in later considerations.

The "Z" terms measure the difference in static head or elevation above some selected reference datum between the two points about which the energy balance is being taken. Since a horizontal pipe would have no change in such an energy term, a simplification in the analysis of any experimental data would result if level pipe were used.

When any unit mass of fluid moves into the system, it must force out a similar mass of fluid if there is to be no accumulation of material. The entering and leaving fluids are then doing work, the quantities of which are the products of the pressure and the volume.

Because the fluid is flowing, it possesses kinetic energy. This is different from the kinetic energy of the ultimate particles, for the latter is present whether or not the fluid is in motion and is dependent upon the state of the fluid. From elementary mechanics, the kinetic energy is evaluated by

$$K. E. = 1/2 m u^2. \quad (2)$$

Since the mass commonly used by engineers is actually the weight, a gravity term must be inserted as follows:

$$w = m g \quad \text{or} \quad m = w/g. \quad (3)$$

$$K. E. = w u^2/2g$$

The symbol u represents an average velocity of the fluid which is usually defined as

$$u = w v / A. \quad (4)$$

A mean velocity defined by this equation does not always yield a true mean kinetic energy. A true kinetic energy must be obtained by a summation of the instantaneous values at each point in the cross section of the pipe. For turbulent flow, the α of Equation (1) can be shown to have the value of two. Since all of the flows discussed in the paper were turbulent, α will be taken as two throughout the remainder of the paper.

If a compressible fluid is in the line, it will expand while flowing since the static pressure is continually decreasing. When any unit slug of fluid expands it will do work upon the slug of fluid preceding it. This work is represented by $\int p dv$. The $p dv$ is not a point function; and, therefore, requires a known expansion path for the evaluation. The usual procedure is to assume an approximate path and to regard the error as part of the friction term.

The ΣF term represents the total loss as a result of friction. It actually represents the mechanical energy made unavailable by the irreversibilities in the flow process.

Dittus and Hildebrand (4) and Kraft (11), after studying tubular heaters for flashing hydrocarbon mixtures, concluded that the friction term is the most significant. Each of these authors suggested that the friction term be evaluated by neglecting

any kinetic energy or expansion work. The small quantities for the latter energy terms may be calculated from the terminal conditions determined by the friction. They may then be added to the friction term as corrections. However, the authors point out that such small corrections are hardly justified since the errors in estimating the friction term for two-phase flow are possibly much larger than the kinetic energy or expansion work terms.

If the kinetic energy and expansion work are assumed to be negligible, the total energy change between two points is caused by the fluid friction. This change in energy between two points is manifested in the static pressure difference of the two points. Therefore, an investigation of the pressure drop of flashing fluids required a study of the friction factor and equations for using this friction factor to predict the pressure drop.

Dimensional analysis has shown that the frictional resistance of a moving fluid is proportional to the dimensionless ratio $D G/u$, which is usually called the "Reynolds' Number". Experimental investigation has verified this theoretical analysis and charts are readily available today for estimating a friction factor if the Reynolds' Number is known.

Once the friction factor has been evaluated, the pressure drop can be calculated. One equation in common use for determining the pressure drop is the Fanning Equation (5).

$$P = \frac{f G u L}{2 g D} \quad (5)$$

For a two-phase mixture, the viscosity term in the Reynolds' Number and the velocity term in the Fanning Equation are difficult

to evaluate. The viscosities of the two phases may differ fifty fold. There is a problem of how to average the properties of the two phases in proportion to their respective amounts. The same exists in trying to average the specific volumes of the two phases in order to calculate the velocity for the Fanning Equation. The only way to determine how to combine the properties of the two phases to form one "psuedo" property of the mixture was to obtain experimental data on the pressure drop of such mixtures and to adjust the properties to fit the observed data.

LITERATURE SURVEY

Before proceeding into the experimental evaluation of the pressure drop of flashing fluids, a review of the previous work on two-phase fluid flow was advisable.

Flow Mechanism of Two-Phase Mixtures

In estimating the friction factor for the two-phase flow, one of the first problems concerned the dispersion of the two phases. Investigations were carried out at the Massachusetts Institute of Technology (8,22) to determine the mechanism of flow with the hope that knowledge of the mechanism would permit a theoretical evaluation of the friction factor. By observing the flow of various air and water mixtures in a glass pipe, the workers concluded that four distinct types of flow were evident. They were as follows:

1. Separate flow
2. Bubbling flow
3. Slugging flow
4. Washing-out flow

"Separate flow" was defined as that condition where the two phases pass through the tube in two distinct layers. Even for separate flow, there is a noticeable wave motion on the surface of the liquid. As either the water or the gas rate increases, these waves become larger. When the waves touch the top of the tube, the surface tension tends to hold them up there and the air is entrapped as a bubble. This is called "bubbling flow". Bubbling flow is also said to occur when the air rate is so low that there is no possibility of wave motion. Given a condition of bubbling flow, an increase in air velocity will cause a change to "slugging". Here, the water is apparently carried along by the air--the latter being at a much higher velocity. When the wave motion of the water touches the top of the tube, the air is slowed considerably. This builds up the pressure of the air which pushes the air down to the bottom of the tube forcing the water ahead of it. The increased pressure behind the wall of water tends to force the water along at an increased speed. An increase in either the water or air rates causes the slugging to change because of slug breakdown, to that type of flow called "washing-out". During this period of breakdown, an increase in either component causes the slug to move faster. Since the greater friction of the slug against the bottom of the tube causes the top to move faster, the water at the front of the slug falls to the bottom. The top of the slug becomes shorter until it finally breaks down and washes out.

An important point of interest in these investigations was that the pulsating or slugging flow prevented any pressure drop

data of reliable accuracy to be taken.

The flow mechanism was also studied at the University of California by Martinelli, et. al. (12) in the investigation of pressure drop of two-phase fluids. However, the conclusions were quite different from those obtained at the Massachusetts Institute of Technology. Four types of flow were found but the criterion was based upon streamline or turbulent motion of the gas and liquid phases. Either one or both phases could be in streamline or turbulent flow. All of the types were experimentally observed except that of a turbulent liquid and a viscous gas. The most common type of flow was where both of the phases were in turbulent motion. The phases are intimately mixed to the extent that no visual separation was apparent. The liquid was carried along in a spray, the fineness of which was dependent upon the velocity of the phases. Each of the four types of flow was observed to have a characteristic pressure drop equation. However, in all four types of flow, the liquid phase was more important than the vapor phase in determining the friction factor.

The fact should be pointed out that none of these investigations used a flashing mixture. All of the studies of flow mechanism were with two-component, two-phase flow where the liquid-to-vapor ratio was constant throughout the test line. So far as could be found, no work has been done on investigating the flow mechanism of a flashing fluid.

Pressure Drops in the Flow of Two-Phase Mixtures

The majority of the work on the pressure drop of two-phase fluid flow was done at the University of California Agricultural Experimental Station by several workers (2, 13, 14). The purpose of these experiments was to arrive at a means of properly estimating the size of fuel lines to the heaters of fruit orchards. While no investigations were made on flashing fluids, much information was obtained about two-phase flow characteristics. Numerous tests were made on air-liquid mixtures. Eight different liquids were used in pipes of three different sizes at various temperatures. The air-to-liquid ratio varied from all air to all liquid. As a result of several years of testing, a method was devised for predicting the pressure drop of two-phase, two-component flow. The proposed method, however, is based upon a vague flow type modulus which is the ratio of the actual cross sectional area of the liquid in the pipe to the total pipe cross sectional area, and is accurate only to within thirty percent. Methods of estimating the modulus are dependent upon the type of flow and the composition. No correlation was made with the pressure drop equations commonly used in fluid flow problems.

Although no work was done on flashing fluids, one of the workers (14) has attempted to extend the knowledge gained in these tests to predict the pressure drop for flashing mixtures of water and steam. The method has the same limitations as that for the two-component flow described above.

Bottomley (3) was the first to publish any results on the pressure drop of flashing fluids. His test data consisted of a single run on a marine boiler. Because of the limited data, no conclusions were reached except that lines for transporting flashing fluids should be considerably larger than for a single liquid phase.

The most informative study on the pressure drop of flashing steam and water mixtures was made by Benjamin and Miller (1), although their work was primarily an investigation of erosion rates in the bends of steam boiler lines. Believing that the erosion of the elbows was being caused by the velocity increase resulting from the flashing of part of the liquid, they investigated the pressure drop of flashing fluids in order to be able to predict the extent of flashing. The tests were made at a Detroit power plant on the drain lines of steam boilers. While the technique used was excellent for the study of erosion, it has some limitations for investigating pressure drops. The plant had to continue operation at normal capacity; and as a result, the range of flows which could be obtained was narrow and limited to those occurring daily. However, sufficient data were taken to propose a method for estimating the size of lines which should be used for flashing mixtures. The calculation of the pressure drop of the flashing mixture was necessary in order to keep the fluid velocity low and to thereby minimize erosion in the bends.

In the method proposed by Benjamin and Miller (1) pressure drops were estimated by a graphical integration of a modified form of Equation (1), the mechanical energy balance equation.

For the friction factor, the authors proposed the use of an average value of all the observed friction factors. This is a logical assumption since all of the observed data showed that the friction factor varied only about twenty percent on each side of the average. The primary shortcoming of the method, however, is the failure to correlate the friction factor with the factors for all water and all steam flows in terms of the vapor-to-liquid ratio. This prohibits extending the application of the method beyond the ranges covered in the test runs or extending it to other fluids. As the authors concluded, the investigation was meant only as a guide in some boiler design problems similar to the installations on which the tests were made, and was not to be a general solution to the two-phase flow problem.

Dittus and Hildebrand (4) have published an article on the design of tubular heaters for hydrocarbons. In this article, they propose a method for calculating the pressure drop of the flashing hydrocarbon mixture. The method is composed of a series of tedious trial-and-error solutions which, in effect, amount to a step-wise integration of the energy balance equation. The method was developed mostly from experience rather than from any experimental investigations.

Kraft (11) has also proposed a method for designing tubular heaters for flashing hydrocarbons. This method is a simplification of that proposed by Dittus and Hildebrand. Kraft recommends that the pressure drop of a flashing fluid be done in a step-wise manner. The friction factor is estimated entirely from

liquid properties, as the work at the University of California indicated. However, the velocity in the Fanning Equation is determined from a weighted average of the two phases. The author believes that the pressure drop calculated in such a manner is probably high necessitating over-designing rather than under-designing. For those heaters investigated, the method appears to approximate actual conditions.

Conclusions From Previous Work

The conclusions from the previous work may be summarized as follows:

1. The microscopic mechanism of flow for a flashing fluid has not been investigated. A simpler approach appeared to be to study the macroscopic effect in hopes of determining some empirical correlation for the prediction of "psuedo" properties.
2. The friction factor is probably determined by the liquid portion of the mixture.
3. No method has yet been proposed for estimating the pressure drop of flashing fluids by the equations commonly used for single-phase fluids.

THE EXPERIMENTAL EQUIPMENT

Steam and water were chosen as the fluids with which to obtain some experimental data. This choice was determined by two factors. First, the water and steam were readily available in the laboratory; and since there was no necessity of recovering the material, no cooling or storage equipment at the exit of the

test line was needed. Second, the steam served as the medium for heating the water by condensing a portion of the steam in a mixing device. The uncondensed portion of the steam and the water at its saturation temperature were forced through the test line.

The test line was a nominal three-eighths inch diameter galvanized iron pipe. The size was a compromise between a pipe that would behave similarly to the large pipes in commercial installations, yet not so large as to require a high capacity water-steam mixing device. With the pipe size selected the pressures at which the water and steam were available in the laboratory were sufficient to obtain reasonable variations in flow rates. The test line was forty feet long with pressure taps brazed onto it at ten feet increments. The line was horizontal and straight in order to eliminate potential head and equivalent length of fittings. To make the flow approximate adiabatic conditions, the line was covered with foamglas insulation one and one-half inches thick. Figure 1 shows a diagram of the line and the arrangement of the pressure taps.

The pressure drop was measured with mercury manometers of the single leg type. Figure 2 shows the details of the manometer connections. The manometers were connected to read the pressure drop over increments of ten feet, so that the full effects of flashing were able to be studied.¹

¹ As the amount of vapor increases, the velocity increases rapidly. Referring to Equation (5), the increase in velocity should increase the pressure drop. The pressure drop, according to the formula, should be different for each increment of length. Connection in this manner permitted the measurement of the static pressure difference over each ten-foot section.

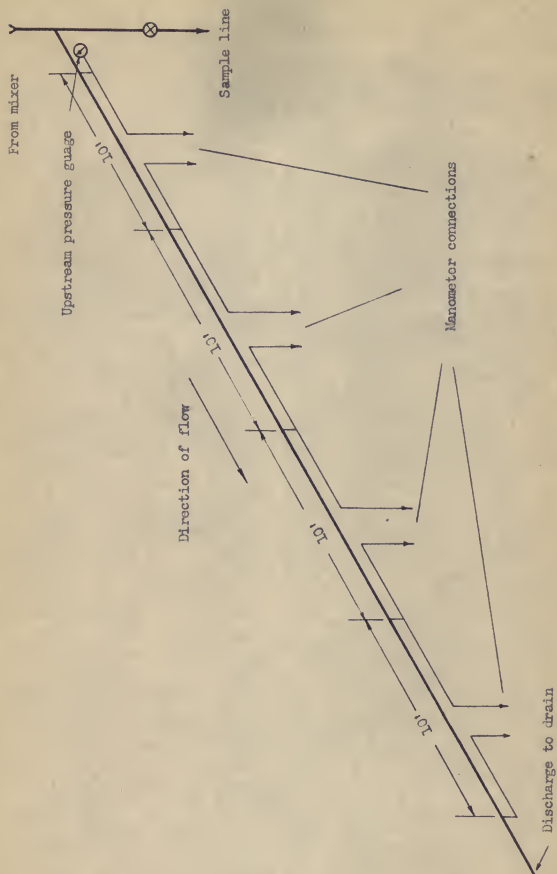


Fig. 1. Schematic diagram of the experimental test line

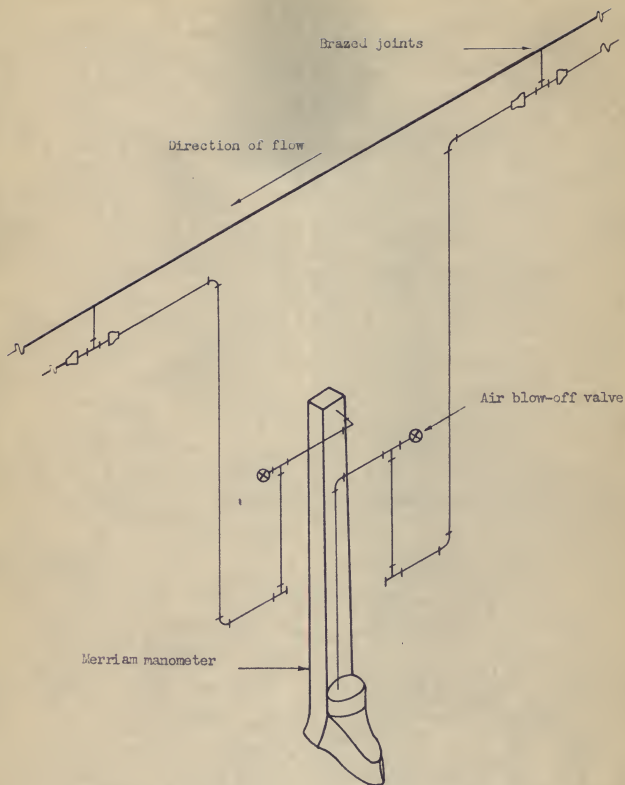


Fig. 2. Details of the manometer connections as viewed from the rear of the manometer.

The static pressure in the line was measured by a pressure gauge on the upstream end of the line. When the line was exhausting to the atmosphere, the pressure gauge reading provided a check on the total of the manometer readings.

The construction of a mixing device which would provide a smooth flow over wide ranges of vapor-to-liquid ratio was the major obstacle in the design of the equipment. Pounding and "bumping" occurring in an open mixer is a typical occurrence when steam strikes cold water. In order to obtain reliable pressure data, bumping had to be eliminated or smoothed out before injection into the test line.

The first device tried was a mixing chamber into which the steam and water were injected. The chamber consisted of a nominal one and one-quarter inch pipe six inches in length. The steam and water were mixed by forcing the water into the pipe perpendicular to the steam in an attempt to obtain intimate mixing. This arrangement was completely unsuccessful.

Believing that small steam bubbles would smooth out the flow, an injection nozzle was made. A piece of nominal one-inch pipe eighteen inches in length was plugged at one end. Fifty holes, each five thirty-seconds-inch in diameter, were drilled in the pipe. The nozzle was inserted concentrically into a two-inch pipe as shown in Fig. 3. Steam was admitted inside the nozzle, and as the bubbles of steam emerged from the holes, they were to be swept along by the water. This device was a considerable improvement over the simple chamber but still failed to produce a flow smooth enough to obtain pressure data.

A jet evacuator was tried but with absolutely no success. Using either water or steam as the motive fluid failed to produce even flow.

The device finally used was a centrifugal pump arranged as shown in Fig. 4. The impellers of the pump were to provide sufficient agitation to break up the steam bubbles and mix them intimately with the water. The steam is injected into the water by means of a nozzle placed as closely as possible to the impellers of the pump. The nozzle consisted of a regular one-half inch pipe plugged with a cap. Four holes, each three thirty-seconds of an inch in diameter, were drilled into the cap through which the steam was admitted. Provision for recycling of the mixed material was made; however, this was not used. The fact should be pointed out that the sole purpose of the pump was to serve as an agitator to mix the steam and water and not in any way to act as a pressure booster. Actually, there was a loss of pressure through the pump. The mixing chamber of the pump served to bring the water to the saturation temperature. Additional steam was admitted downstream of the mixture.

Certain auxiliary equipment indicated in Fig. 4 was also needed. The water pressure of the main was found to cycle with variations of as much as five pounds over periods ranging from two to ten seconds. Therefore, to stabilize the pressure of the water, a surge tank with an air pocket was placed between the water main and the mixer. Another surge tank was placed between the steam main and the mixer to remove small pulsations in the steam pressure and to act as a moisture separator. A rotameter

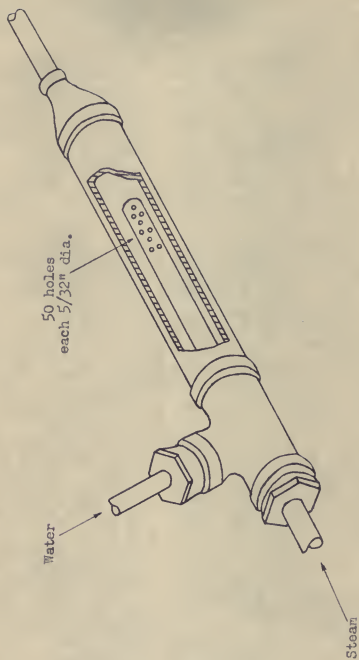


Fig. 3. Sketch of a mixing nozzle.

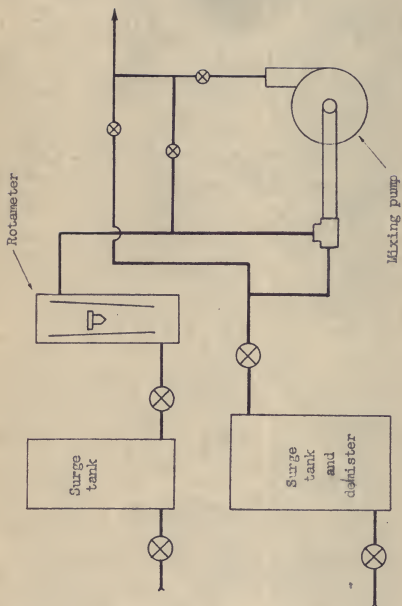


Fig. 4. Sketch of the mixing section and auxiliary apparatus.

was used to measure the water flow.

PROCEDURE

The procedure for obtaining data for a typical run was as follows:

1. The rate of water flow through the rotameter was adjusted to slightly above that which was desired for the run.
2. The steam valve was opened to admit sufficient steam to obtain approximately the desired vapor-to-liquid ratio as indicated by opening the sample line and making a visual observation to the mixture.
3. A final adjustment on the water rate was made so that the rotameter was exactly as desired. The increase in pressure of the system after the admission of the steam tended to slow the water rate to about the desired value since it was over-ranged initially.
4. The pressure differentials were read from the manometers.
5. The calorimetric data were obtained.
 - a. An empty thirty gallon steel drum open at one end was weighed.
 - b. Approximately 125 pounds of water were added and the weight of the drum and water were obtained.
 - c. The temperature of the water was taken with a thermometer calibrated in tenths of a degree Centigrade.
 - d. The mixture of steam and water from the sample line was admitted to the drum. The water was stirred while the mixture was being added in order to keep the water temperature approximately uniform throughout.
 - e. The final weight of the drum, water, and sample was obtained.

- f. The final temperature of the contents of the drum was taken. The tests were so conducted that the water was heated from a few degrees below room temperature to a few degrees above room temperature.

6. All calorimeter measurements were repeated as a check.

A minimum of time elapsed between the pressure reading and the calorimeter readings since the apparatus had a tendency to change spontaneously. This effect was especially noticeable for runs made using mixtures of high quality.

The best operating scheme was to close the recycle valve in the mixing section and not run the pump. The pump was actually serving only as a chamber in which the water and the steam were mixed as the impellers were stationary. The two valves for admitting steam downstream of the mixing section were always fully opened, permitting the maximum amount of steam to enter at those points.

THE DATA

The experimental data taken in this investigation are reported in Tables 1, 2 and 3 of the Appendix. Table 1 is the calibration of the rotameter which was used to measure the water rate. These calibration data have been shown graphically in Fig. 5.

The particular friction factor for the test line was determined from the data shown in Table 2. The experimental friction factor has been shown as a function of the Reynolds' Number in Fig. 6. The solid line is the calculated value of the factor and the dashed line is the value reported by Pigott (21) for clean, new

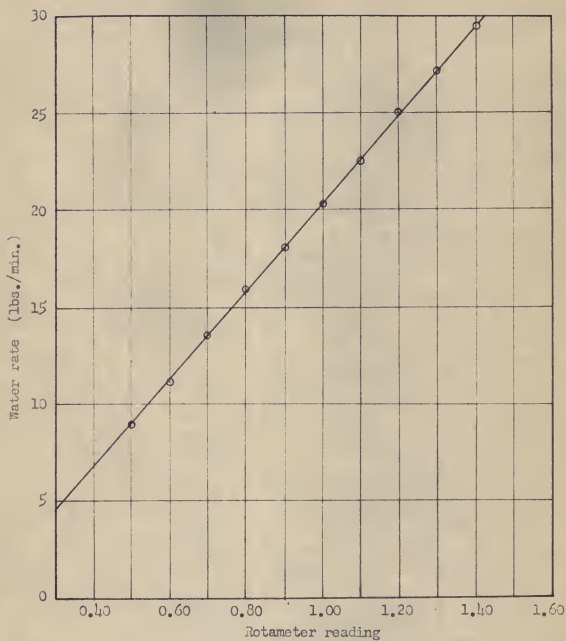


Fig.5. Calibration of the rotameter.

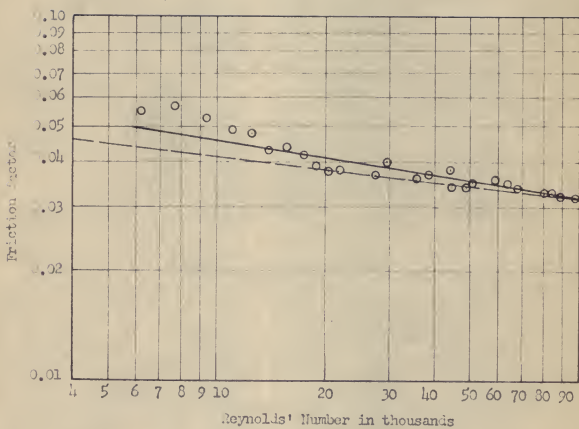


Fig. 6. Friction factor for the experimental test line.

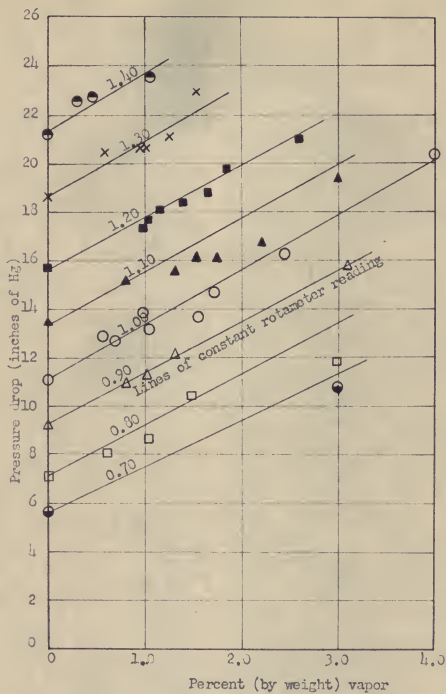


Fig. 7. Pressure drop over forty feet of pipe for various qualities and water rates.

three-eighths inch diameter pipe as used in the test line.

The data for the test runs with various liquid-to-vapor ratios at the outlet are included in Table 3. These data, which are shown graphically in Fig. 7, represent the maximum variations obtainable with the apparatus. The water rate ranged from one and one-half to four gallons per minute. The maximum vapor content at the outlet was four percent on a weight basis or ninety-eight percent on a volume basis. Data from only one calorimetric determination has been included for each run. Because the apparatus was somewhat unstable, it did not always continue to operate at one condition for more than a few minutes; therefore, first determination of quality was believed to be the more accurate. The second determination of quality served mainly as an order-of-magnitude check on the first trial. Sample calculations are shown in the Appendix to illustrate the method for finding the quality from the experimental data. Three measurements of quality were made on the downstream end of the test line to study heat loss. No difference in the upstream and downstream qualities could be determined within the accuracy of the calorimetric measurements.

The data are believed to have a maximum probable error of about ten percent. The mercury in the manometers was not steady causing a possible error in the readings of two-tenths of an inch on each manometer, but since most of the manometer readings were larger than two inches, the probable error is well within the ten percent mentioned above. The calorimetric measurements were the limiting factors in precision of the experimental data.

In many instances, unstable operation of the apparatus for periods over a few minutes prevented checks on the quality determination. Furthermore, calculations based upon a weighing accuracy of one-quarter of a pound and a temperature accuracy of two-tenths of a degree Fahrenheit showed that the possible error in the quality was about ten percent. For those runs where the apparatus was stable, calorimetric check determinations were slightly better than the ten percent figure.

ANALYSIS OF THE DATA

For the analysis of a flashing fluid, the data plotted in Fig. 7 must be modified. This graph is a plot of the outlet quality when exhausting to the atmosphere versus the static pressure decrease over the entire test line. However, in most of the runs a flashing mixture existed only in part of the forty feet of pipe. The other portion was a single phase liquid. These data must be used to calculate the length of line in which flashing occurred and the pressure drop over only this length of pipe. A sample calculation of the type used for determining the pressure drop of the flashing mixture is included in the Appendix. The results of the calculations are shown graphically in Fig. 8, where pressure drop per foot of pipe has been plotted for varying percent vapors and for different water rates. The data of Fig. 8 rather than Fig. 7 were used for correlating purposes since they show the pressure drop of the flashing mixture only.

An attempt was made to combine the fluid properties of the two phases in a manner such that the single "psuedo" properties

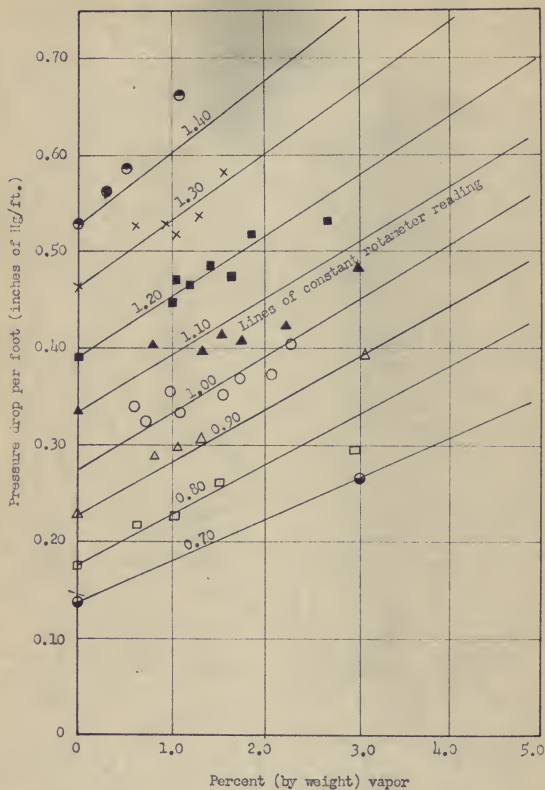


Fig. 8. Pressure drop per foot for various water and steam mixtures.

of the mixture might be used for the viscosity term in the Reynolds' Number and for the velocity term in the Fanning Equation. The following methods of averaging were tried:

1. Average the properties in proportion to the volume fraction of each phase.
2. Average the properties in proportion to the weight fraction of each phase.
3. Average the properties arithmetically irrespective of the amount of each phase present.

None of these averaging methods nor any combination of them (as using the weight average for the viscosity and the volume average for the velocity) were suitable for predicting a pressure drop corresponding to the experimental data of Fig. 8. Included in this method of averaging was the procedure recommended by Kraft (11) for flashing hydrocarbons. His estimation procedure gave results from fifty percent to five hundred percent above the experimentally measured pressure drop.

Since the viscosity and/or velocity of a flashing fluid were not simple functions of the individual phase properties, some other procedure had to be used for correlating the data.

On the basis of the conclusions reached at the University of California by Boelter and Kepner (2), the assumption was made that the friction factor of the mixture was determined by the liquid phase only. The Reynolds' Number for estimating the friction factor was calculated by neglecting the mass of vapor and using only the mass of the liquid portion of the mixture. The friction factor for the mixture was then assumed to be the same

as when calculated for the single liquid phase.

Using the assumed friction factor, each term in the Fanning Equation for pressure drop was known except the velocity. The experimental pressure drops were used to calculate a velocity. Sample calculations for this procedure are included in the Appendix. The problem of correlation of the data was reduced to one of predicting this back-calculated velocity.

A logical assumption at this point was that for a constant water rate, an increase in the quality, or percent vapor, should produce a proportional increase in the velocity. Hence, if the fractional increase in velocity were plotted against quality, a straight line would result for each constant water rate. Three such lines are shown in Fig. 9 which is a plot of the data of Table 5. The family of lines for the varying water rates was evident.

Since this method of predicting the velocity for the Fanning Equation was both reasonable and convenient, a chart was constructed based upon smoothed experimental data. This chart, shown as Fig. 10, is a plot of average quality versus increase in velocity caused by flashing divided by the velocity for the single liquid phase.

The smoothing of the data was done as follows;

1. The pressure drop per foot of pipe for the liquid at the boiling point was plotted against water rate.
2. By the method of least squares, the equation of the best straight line through the points was determined.
3. At convenient water rate intervals, the pressure drop

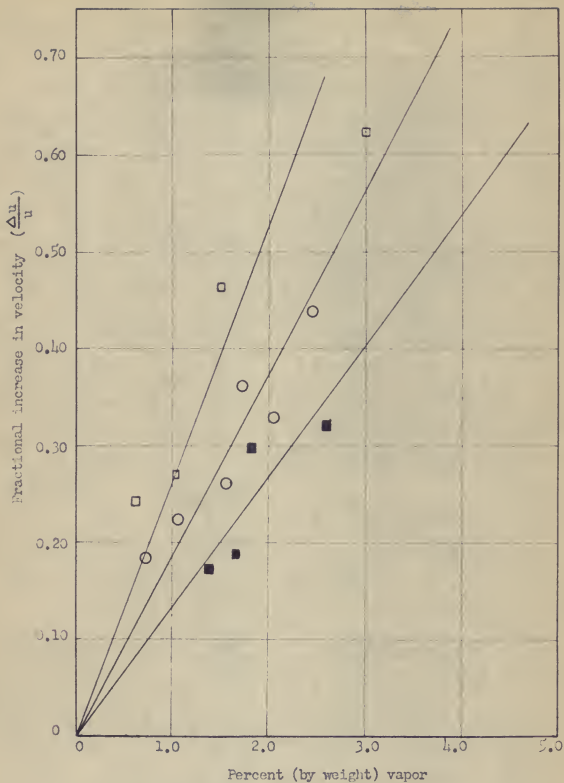


Fig. 9. Fractional increase in velocity caused by flashing at three different water rates.

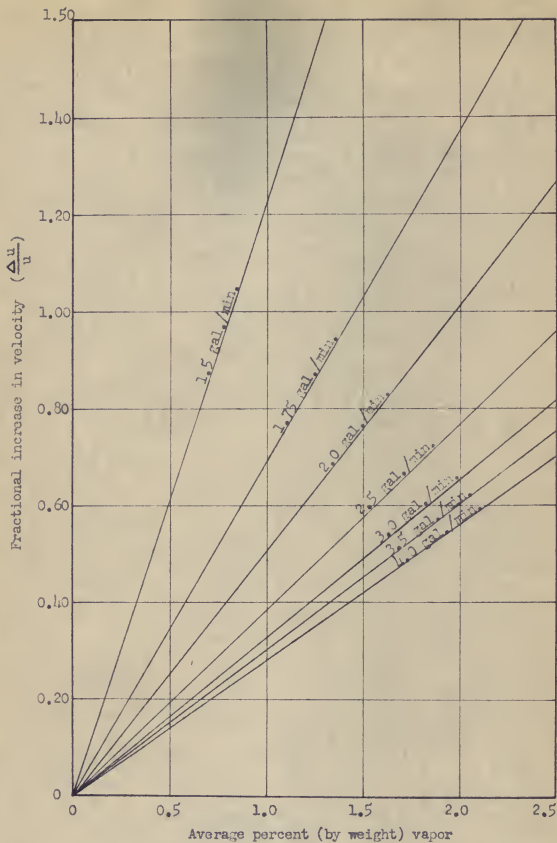


Fig. 10. Chart for estimating the fractional increase in the velocity caused by part of the liquid flashing.

was calculated from the equation determined in step 2. These are shown in Table 6 of the Appendix.

4. A plot was then made of pressure drop per foot versus quality for the convenient water rates. The lines were drawn to pass through the calculated liquid pressure drop per foot and drawn to have the same slope as an approximate water rate on Fig. 8. This chart is included for reference as Fig. 13 in the Appendix.

5. Points were then read from Fig. 14 to calculate the chart of Fig. 10. Table 7 in the Appendix is a summary of the calculations.

Figure 10 enabled a prediction of "pseudo" velocity that was to be used in the Fanning Equation. A method for calculating the pressure drop for adiabatic flow was as follows:

1. The pressure drop up to the point of initial flashing is found by examination of the pressure-enthalpy diagram. A pressure-enthalpy diagram for water and steam is included in the Appendix as Fig. 13. The length of pipe for this single phase fluid may then be found from the Fanning Equation.

2. The static pressure at the point where there is about one percent of the material vaporized may be read from the pressure-enthalpy diagram. A pressure drop may be calculated as the difference between the saturation pressure and the pressure at one percent vaporization. There would be an average quality of one-half of one percent in this interval.

3. Estimate the increase in velocity caused by the vapor above that velocity of all liquid flow by reading on Fig. 10

at an average quality of one-half of one percent for the particular water rate.

4. Calculate the velocity of the liquid from the usual w v/A formula.

5. Calculate the "psuedo" velocity by

$$u^* = u + u \left(\frac{\Delta u}{u} \right) \quad (6)$$

6. Estimate the friction factor from the Reynolds' Number of the liquid phase.

7. Calculate the length of pipe needed to have the calculated pressure change found in step 1.

$$L = \frac{2}{f} \frac{p_g D}{G u^*} \quad (7)$$

8. From the pressure-enthalpy diagram, obtain the pressure where two percent of the material is vaporized. Repeat steps 2 through 7 to find the length of pipe needed to vaporize the second percent.

9. Continue taking increments until the length of pipe corresponds to that needed in the physical requirements of the design problem.

A sample problem is included in the Appendix illustrating the above procedure.

CONCLUSIONS

The proposed method for calculation is based upon a relatively narrow range of water velocities and liquid-to-vapor ratios. The straight lines on Fig. 8 would be expected to curve upward at higher vapor fractions until they became asymptotic

to the all vapor phase flow lines. This would cause the lines of the graph for estimating the velocity increase to curve upward also. A semilogarithmic plot would probably be necessary since the velocity increase would be multiples of the liquid velocity rather than fractions larger as observed in this experiment. Because these lines on Fig. 8 might curve upward, care was taken not to extend the lines beyond the range of experimental data.

A calculation method of the type proposed has certain advantages over any previously suggested.

1. No trial and error calculations are involved for determining the pressure drop over a specified length of line.
2. The same friction factor chart is used as for a single phase fluid.
3. The calculations are algebraically simple and permit rapid solutions of each incremental step.

The main disadvantage of this method is that a chart for estimating the increase in velocity must be experimentally found for each fluid. However, if the data for steam and water were extended and similar data were taken for at least one other fluid, a general method might be found for approximating the "psuedo" velocity.

The proposed method still requires a step-by-step solution, but overcoming this tediousness would be difficult since conditions all along the line carrying a flashing fluid are changing.

The chart used for calculating the "psuedo" velocity is limited to water and steam flowing through a three-eighths inch

pipe. However, the chart could probably be used for other size lines if the parameter were mass velocity or velocity rather than gallons per minute. This would require experimental verification.

RECOMMENDATIONS FOR FUTURE WORK

Future experimental work on this problem should consist of extending the data for water and steam mixtures, then of obtaining similar data for at least one other fluid. Complete experimental data of this nature for at least two, and preferably several more fluids, would probably permit a general solution to the problem of pressure drop of a flashing fluid. In order to obtain the necessary data, certain changes in the apparatus as outlined below would be needed.

The data for water and steam mixtures could probably be doubled if the water pressure on the present apparatus were increased to equal the pressure of the steam. At present, when the steam valve is opened to increase the quality of the mixture to above the range investigated here, the water, because it is at a lower pressure, is prevented from entering the mixing section. Because the increase in range would still not permit a liquid-to-vapor range from all water to all steam, this suggestion is regarded to be of secondary importance to those for completely revising the mixing section.

As mentioned in the discussion of the operational technique, the best results were obtained with the mixing pump not running. This fact suggested that a mixing chamber be installed to replace

the pump. A possible arrangement would be to have a nozzle, similar to the present one, exhausting into a pipe acting as a chamber. A large pipe would replace the pump as the chamber. Downstream from the mixing chamber, several (four or five) injection nozzles such as the two now used should be located to permit the addition of steam to the hot water from the mixing chamber. In effect, this is a modification of the unsuccessful mixer shown in Fig. 3. The failure of the nozzle shown in Fig. 3, however, is now believed to have resulted from trying to admit all of the steam into the mixing chamber. In other words, there were too many holes in the first nozzle. A new nozzle should be designed to admit only sufficient steam to heat the water to the saturation temperature, and the steam to provide excess vapor should be admitted by the jets downstream of the mixer.

The type of mixer suggested should have one advantage over the present pump arrangement. A pressure drop of fifteen to twenty pounds per square inch existed across the mixing pump and the connecting pipe to the test line. A mixer of the simple chamber type would be expected to reduce this pressure loss in two ways. First, the simple chamber would be expected to offer less frictional resistance than the pump chamber; and second, the more compact simple chamber might eliminate the span of pipe between the present mixer and the test line.

A more versatile apparatus would be one making use of electrical heaters. If electrical heaters were used, a surface heat exchanger should be used to bring the liquid close to its saturation temperature. The hot liquid should then be passed over the

electric heater which would supply sufficient heat to vaporize the portion desired. This arrangement is believed to have the advantage of extreme accuracy in measuring the vapor-to-liquid ratio. The surface heat exchanger is suggested so that the size of the heater might be reduced to that just sufficient to supply the latent heat of vaporization.

The recommendation to study another fluid is necessary before any general calculation procedure can be proposed since most fluids will exhibit a greater flashing effect than water because of their lower latent heats of vaporization. For example, the latent heat of vaporization for a hydrocarbon is in the order of one-fifth that for water. Since the specific heats of the liquid are about the same, an equal decrease in the enthalpy of the liquid portion of the mixture will vaporize four to five times more hydrocarbon than water. The flashing effect is, thus, more prominent in substances whose latent heats of vaporization are relatively low. The substance chosen must be one having thermal properties and viscosities for both the liquid and the vapor phases available in the literature. Acetone might be a suitable fluid with which to work.

For handling more costly fluids (as acetone) storage and condensing facilities would be necessary. The condenser would also permit direct measurement of the quantity of vapor at the exit. To do this, heat extracted by the condenser would be measured by the inlet and outlet water temperatures; these data, with the enthalpy data of the two phases, would be sufficient to calculate the quality of the mixture entering the condenser. With this

arrangement, a simple surface heat exchanger or an electric heater could be employed at the upstream end to generate the vapor.

The existing test line would be satisfactory for extending the data. However, an improvement suggested is a method to check pressures. Thermocouples could be installed at the pressure taps. Measurement of the temperature would then be related to the pressure if the liquid and vapor were in equilibrium. If the vapor actually flows faster than the liquid, the two phases may not be in equilibrium and the pressure as determined by the temperature measurement and that measured by the manometer would not agree. This might lead to some revealing mechanism of the flow patterns and finally permit a general solution to the problem of predicting the pressure drop of flashing fluids.

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NOMENCLATURE

A	Cross sectional area of the pipe (sq. ft.)
α	Proportionality factor in the kinetic energy term depending upon the type of flow.
D	Diameter of pipe (ft.)
F	Frictional resistance
f	Fanning friction factor
G	Mass velocity (lb./hr.-sq. ft.)
g	Gravitational constant (32.2 ft./sec.-sec.)
L	Length of pipe (ft.)
m	Mass (slugs)
P	Static pressure (lb./sq. in. or lb./sq. ft.)
u	Velocity of fluid (ft./sec.)
u*	"Pseudo" velocity (ft./sec.)
v	Specific volume (cu. ft./lb.)
w	Weight (lb.)
Z	Height above a reference datum plane (ft.)
ν	Viscosity of fluid (lb./sec.-ft.)

KEY TO SYMBOLS ON FIGURES

Symbol	Rotameter reading
●	0.70
□	0.80
△	0.90
○	1.00
▲	1.10
■	1.20
×	1.30
⊙	1.40

These symbols have been used on Figs. 7, 8, and 9.

APPENDIX

Sample Calculations

A. Calculate the quality from the bomb calorimeter data.

Illustrated on Run No. 2

Weight of barrel and water at start of sampling	144 1/2 lbs.
Weight of empty barrel	25 lbs.
Weight of water in barrel	119 1/2 lbs.
Initial temperature of water	19.8 C.
Weight of barrel, water, and sample mixture	160 1/4 lbs.
Weight of sample mixture	15 3/4 lbs.
Final temperature of water and sample mixture	30.8°C.
Enthalpy of water @ 19.8° C.	35.85 B.T.U./lb.
Enthalpy of water @ 30.8° C.	55.60 B.T.U./lb.
Enthalpy increase of water	19.75 B.T.U./lb.

Heat added to water per pound of sample:

$$\frac{19.75 \text{ (B.T.U./lb.)} \times 119 \frac{1}{2} \text{ (lbs.)}}{15 \frac{3}{4} \text{ (lb. of sample)}} = 149.85 \text{ B.T.U./lb.}$$

Heat added to raise temperature of barrel:

$$\frac{25 \text{ (lb.)} \times 0.012 \text{ (B.T.U./lb.}^{\circ}\text{F.)} \times 19.4 \text{ (}^{\circ}\text{F.)}}{15 \frac{3}{4} \text{ (lb. of sample)}} = 3.79 \text{ B.T.U./lb.}$$

Final enthalpy of water is same as enthalpy of sample mixture, or 55.60 B.T.U./lb.

$$\text{Initial enthalpy of mixture:} \\ 55.60 + 149.85 + 3.79 = 209.24 \text{ B.T.U./lb.}$$

Enthalpy of saturated liquid @ 1 atm.	180.07 B.T.U./lb.
Enthalpy of saturated vapor @ 1 atm.	1150.40 B.T.U./lb.

X = weight fraction of vapor in the mixture

$$180.07 (1 - X) + 1150.40 X = 209.24$$

$$X = 0.0301$$

B. Calculate the pressure drop per foot for the flashing mixture.

Run No. 5.

Manometer readings:

#1	1.8 inches of Hg
#2	2.0 " " "
#3	2.3 " " "
#4	2.5 " " "

The enthalpy of the mixture as calculated from calorimetric measurements was 190.2 B.T.U./lb. From the pressure-enthalpy diagram for water (Fig. 12), the saturation pressure corresponding to this enthalpy was 17.9 psia. This is 3.2 psi. (6.5 in. of Hg) above atmospheric pressure. Since the line was exhausting to the atmosphere, the length of pipe carrying the flashing fluid was that distance measured from the discharge end over which there was a pressure drop of 3.2 psi.

Total pressure drop desired	6.5 in. of Hg
Pressure drop over last ten feet of pipe (Manometer #4)	2.5 " " "
Difference	4.0 " " "
Pressure drop over the third ten feet of pipe (Manometer #3)	2.3 " " "
Difference	1.7 " " "

The pressure drop over the next ten feet of pipe was 2.0 inches of Hg, which is larger than the remaining 1.7 inches of Hg. A linear interpolation was made.

$$10 \times (1.7/2.0) = 8.5 \text{ ft.}$$

The total length of pipe was:

$$10 + 10 + 8.5 = 28.5 \text{ ft.}$$

The pressure drop per foot was:

$$\frac{6.5}{28.5} = 0.228 \text{ in. of Hg/ft.}$$

Sample Problem

Water at a temperature of 250°F. and a pressure of 35 psia. is being pumped from a waste heat boiler. The water is presently being sent to a steam boiler. A modification to another section of the plant resulted in a possible use for this water as a heat source and diluent. A 3/8 inch pipe, now idle, runs between the two units and possibly could be adapted to transport the water with a minimum of plant changes. Three gallons per minute of the water will be needed. The water must exhaust into a reactor whose pressure is 16 psia. If the equivalent length (includes bends and fittings) is 150 feet, determine if the pressure drop caused by friction would prohibit using this line. Assume that the line is well insulated.

1. From the steam tables (9), the enthalpy of water at 250°F. is 218.5 B.T.U./lb. The saturation pressure at this enthalpy is 29.8 psia. (Fig. 12). The pressure drop before the liquid begins to flash is 35.0 - 29.8 or 5.2 psi. Calculate the length of pipe for a pressure drop of 5.2 psi. for all water flow.

$$Re = \frac{D G}{\mu} = \frac{0.0411 \text{ (ft.) } 3 \text{ (gal./min.) } 7.99 \text{ (lb./gal.)}}{0.20 \text{ (centp.) } 0.000672 \frac{\text{(lb./ft.}^2\text{sec.)}}{\text{(Centp.)}} 60 \frac{\text{(sec.)}}{\text{min.}} 0.00133 \text{ (ft.}^2\text{)}}$$

$$Re = 91,800$$

$$f = 0.032 \text{ (from Fig. 6)}$$

$$u = w \sqrt{A} = \frac{3 \text{ (gal./min.) } 7.99 \text{ (lb./gal.) } 0.0170 \text{ (ft.}^3\text{/lb.)}}{60 \text{ (sec./min.) } 0.00133 \text{ (ft.}^2\text{)}}$$

$$u = 5.11 \text{ ft./sec.}$$

$$G = u \rho = 5.11 \text{ (ft./sec.) } 59.6 \text{ (lb./ft.}^3\text{)} = 303 \text{ lb./sec. ft.}^2$$

$$L = \frac{2}{f} \frac{P G D}{G u}$$

$$L = \frac{(2) 5.2 \frac{(\text{lb.})}{(\text{in.}^2)} 144 \frac{(\text{in.}^2)}{(\text{ft.}^2)} 32.2 \frac{(\text{ft.})}{(\text{sec.}^2)} 0.0411 (\text{ft.})}{0.032 \quad 5.11 (\text{ft./sec.}) \quad 303 (\text{lb./sec. ft.}^2)}$$

$$L = 39.7 \text{ ft.}$$

2. The pressure where there is 1% vapor is 25.3 psia. The pressure drop over the interval where the average vapor is 0.5% (by weight) is $29.8 - 25.3 = 4.5$ psi.

3. Estimate the fractional increase in velocity. From Fig. 10,

$$\left(\frac{\Delta u}{u}\right) = 0.17$$

4. The liquid velocity will be 99.5% of the velocity for the all water flow; or within the accuracy of the data, the same as for all liquid flow which was 5.11 ft./sec.

5. Calculate the "psuedo" velocity.

$$u^* = u + u \left(-\frac{\Delta u}{u} \right) = 5.11 + 5.11 (0.17) = 5.98 \text{ ft./sec.}$$

6. The friction factor is the same as for all liquid flow.

7. Calculate the length for the 1% vaporized.

$$L = \frac{(2)(4.5)(144)(32.2)(0.0411)}{(0.032)(303)(5.98)} = 29.6 \text{ ft.}$$

8. Summary of all steps down to the final pressure of 16 psia.

Outlet vapor %	1.0	2.0	3.0	3.5
Outlet pressure psia.	25.3	21.0	17.6	16.0
ΔP over increment psi.	4.5	4.3	3.4	1.6
$\Delta u/u$	0.17	0.33	0.49	0.53
u^* ft./sec.	5.98	6.80	7.62	7.82
L ft.	29.6	24.8	17.5	8.0

9. The total length of pipe is the sum of the increments above plus that prior to flashing.

$$\text{total length} = 39.7 + 29.6 + 24.8 + 17.4 + 8.0 = 119.5 \text{ ft.}$$

Therefore the 150 feet of line would be too long.

For comparison, had no flashing occurred, the pressure drop would have been about 17 psi. for the 150 feet.

Table 1. Calibration of the rotameter.

Rotameter reading	Time in minutes	Pounds of water	Pounds of water per minute
0.50	5.0	45.0	9.0
0.60	5.0	56.0	11.2
0.70	5.0	66.0	13.6
0.80	5.0	80.0	16.0
0.90	4.0	72.5	18.1
1.00	4.0	81.0	20.3
1.10	4.0	90.5	22.6
1.20	3.0	75.0	25.0
1.30	3.0	81.5	27.2
1.40	3.0	88.5	29.5

Table 2. Calculation of the friction factor for the experimental circular pipe.

Rotameter reading	Water temp. °F.	Pressure drop in. H ₂ O	Water rate lb./min.	Velocity ft./sec.	Density lb./cu. ft.	Viscosity centipoises	Reynolds' No. in thousands	Friction factor
0.55	61	2.4	9.0	1.81	62.3	1.11	6.2	0.055
0.60	61	4.0	11.2	2.25	62.3	1.11	7.7	0.057
0.70	61	6.2	13.6	2.73	62.3	1.11	9.4	0.053
0.70	209	6.6	15.9	3.34	59.9	0.28	43.3	0.038
0.80	61	8.8	16.0	3.22	62.3	1.11	11.1	0.049
0.80	152	9.4	15.0	3.28	61.2	0.42	29.3	0.040
0.80	212	7.1	18.7	3.74	59.9	0.27	51.8	0.035
0.90	61	8.4	18.1	3.64	62.3	1.11	12.5	0.048
0.90	209	8.2	21.2	4.44	59.9	0.28	58.2	0.036
1.00	61	9.7	20.3	4.08	62.3	1.11	14.0	0.043
1.00	205	11.0	23.7	4.96	59.9	0.29	62.6	0.035
1.10	61	12.0	22.6	4.54	62.3	1.11	15.5	0.044
1.10	108	10.4	22.6	4.58	61.9	0.63	27.5	0.037
1.10	146	10.4	22.6	4.52	61.3	0.45	38.5	0.037
1.10	203	13.4	26.4	5.51	59.9	0.30	67.2	0.034
1.20	61	14.0	25.0	5.23	62.3	1.11	17.3	0.042
1.20	210	15.7	29.2	6.10	59.9	0.28	79.7	0.033
1.30	61	15.5	27.2	5.47	62.3	1.11	18.8	0.039
1.30	205	10.4	31.8	6.66	59.9	0.29	84.0	0.033
1.40	61	17.6	29.5	5.93	62.3	1.11	20.4	0.038
1.40	106	15.7	29.5	5.97	61.9	0.64	35.3	0.036
1.40	130	15.2	29.5	6.01	61.5	0.51	44.4	0.034
1.40	206	21.1	34.5	7.20	59.9	0.29	97.3	0.032
1.53	121	19.0	32.5	6.61	61.5	0.52	47.8	0.034
1.60	61	24.2	34.2	6.87	62.3	1.11	23.6	0.038

Table 3. Experimental pressure data.

Run. no.	Rotameter reading	Manometer #1	Manometer #2	Manometer #3	Manometer #4	Total pressure loss in. H ₂ O	Weight of barrel & water	Initial temp. °C.	Weight after adding sample	Final temp. °C.
1	0.70	1.4	1.4	1.4	1.4	17.5	146.1	19.7	169	31.8
2	0.70	2.0	2.4	2.3	3.4	17.7	144.1	19.8	169	32.0
3	0.80	1.7	1.8	1.9	1.8	7.1	139.7	19.7	161.5	32.3
4	0.80	1.7	1.9	2.1	2.3	8.0	143.5	22.5	150.5	32.5
5	0.80	1.6	2.0	2.3	2.5	8.4	144.7	19.1	164.3	31.1
6	0.80	2.1	2.4	2.6	3.1	10.4	137.7	19.3	150	33.8
7	0.80	2.4	2.7	3.1	3.3	11.8	137	19.3	158	34.2
8	0.90	2.3	2.3	2.3	2.3	9.2	89.7	21.7	107	34.4
9	0.90	2.5	2.7	2.8	2.9	10.9	111.7	18.6	131	33.6
10	0.90	2.5	2.8	2.9	3.1	11.3	108.3	19.7	126.3	34.4
11	0.90	2.9	2.9	3.1	3.2	12.1	117	19.4	132.3	31.6
12	0.20	2.0	3.4	4.3	5.1	15.7	107.7	19.7	122.3	34.1
13	1.00	2.7	2.8	2.7	2.9	11.0	98.7	21.4	119.3	37.8
14	1.00	3.0	3.2	3.3	3.3	12.8	144.3	23.1	160.3	32.5
15	1.00	3.0	3.1	3.2	3.3	12.0	136	22.0	151	31.6
16	1.00	3.3	3.4	3.5	3.6	13.8	143.7	18.9	160.3	29.4
17	1.00	3.0	3.2	3.3	3.4	13.1	142.3	19.2	161.3	31.0
18	1.00	3.1	3.3	3.5	3.7	13.5	97	19.1	111.3	33.1
19	1.00	3.1	3.4	3.7	4.1	14.5	86.3	19.2	103	30.5
20	1.00	3.4	3.5	3.8	4.0	14.8	137.7	18.7	151.3	28.4
21	1.00	3.4	3.6	4.3	4.7	16.2	139.3	18.5	152.3	33.9
22	1.00	4.7	4.8	5.1	5.5	20.3	96.3	18.5	110	30.3
23	1.10	3.3	3.3	3.4	3.4	13.4	104	23.0	119.3	34.1
24	1.10	3.6	3.7	3.8	4.0	15.1	103.3	20.0	118	32.8
25	1.10	3.7	3.8	3.9	4.1	15.5	105	19.7	121	33.9
26	1.10	3.0	3.0	4.1	4.3	15.1	105.3	22.8	120.3	35.0
27	1.10	3.8	3.9	4.1	4.2	16.0	104.3	24.4	114.3	33.8
28	1.10	3.8	4.0	4.3	4.6	16.7	104	19.7	117.3	32.8
29	1.10	3.9	4.4	5.0	5.0	18.3	120	18.5	134.3	31.4
30	1.20	3.6	3.9	4.1	4.1	15.7	91.3	24.1	109	30.2
31	1.20	4.1	4.3	4.4	4.5	17.3	141	27.8	158	47.5
32	1.20	4.0	4.3	4.5	4.6	17.6	90.3	20.6	107.3	30.1
33	1.20	4.3	4.5	4.6	4.7	18.1	139.3	18.1	158	30.3
34	1.20	4.2	4.5	4.7	5.0	18.4	93.3	18.6	114.3	35.2
35	1.20	4.5	4.7	4.7	4.8	18.5	137	18.5	156.3	31.8
36	1.20	4.5	4.8	5.1	5.4	18.8	92.3	19.1	109.3	37.2
37	1.20	4.9	5.1	5.3	5.7	21.0	97.3	18.1	115	36.4
38	1.30	4.6	4.7	4.7	4.8	18.5	138.3	19.1	167.3	34.0
39	1.30	4.9	5.0	5.2	5.3	20.4	145.3	22.5	166.3	34.1
40	1.30	5.0	5.1	5.2	5.3	20.5	145	17.7	159	32.1
41	1.30	5.0	5.1	5.2	5.2	20.5	133.3	18.0	159.3	34.5
42	1.30	5.1	5.2	5.3	5.4	21.0	144	18.0	166.3	32.0
43	1.30	5.5	5.7	5.8	5.9	22.9	144.3	18.0	166.3	31.9
44	1.40	5.2	5.2	5.3	5.4	21.1	138.3	19.1	167.3	34.6
45	1.40	5.5	5.5	5.7	5.7	23.5	135.3	19.2	164.3	35.9
46	1.40	5.5	5.6	5.7	5.9	24.7	99	18.0	113.3	31.6
47	1.40	5.5	5.6	5.9	6.2	25.5	94.3	19.2	109.3	33.6

Table 4. Determination of the pressure drop per foot of pipe for the flashing mixture.

Run no.	Rotameter readings	Percent vapor	Enthalpy Btu./lb.	Sat'n press. psia.	Sat'n press. less 14.7	Conversion to in. Hg	Length of pipe carrying the flashing fluid	Press. drop per foot in. Hg/ft.
1	0.70	0.00	177.0	14.7	0.0	0.0	40.0	0.140
2	0.70	3.01	209.2	25.4	10.7	10.7	40.0	0.268
3	0.80	0.00	180.1	14.7	0.0	0.0	40.0	0.177
4	0.80	0.63	186.2	16.6	2.2	4.5	20.5	0.220
5	0.80	1.04	190.2	17.9	3.2	6.5	28.5	0.228
6	0.80	1.49	194.6	19.4	4.7	9.6	36.2	0.262
7	0.80	2.97	208.9	25.3	10.6	11.8	40.0	0.295
8	0.90	0.00	180.0	14.7	0.0	0.0	40.0	0.230
9	0.90	0.81	187.9	17.2	2.5	5.1	17.9	0.285
10	0.90	1.02	190.5	18.0	3.3	6.7	22.3	0.300
11	0.90	1.31	192.8	18.8	4.1	8.3	26.9	0.308
12	0.90	3.10	209.8	25.7	10.0	15.7	40.0	0.392
13	1.00	0.00	180.0	14.7	0.0	0.0	40.0	0.275
14	1.00	0.57	185.7	16.5	1.8	3.7	10.9	0.340
15	1.00	0.70	186.9	16.9	2.2	4.5	13.8	0.326
16	1.00	0.99	189.7	17.8	3.1	6.3	17.7	0.356
17	1.00	1.06	190.4	18.0	3.3	6.7	20.0	0.335
18	1.00	1.56	195.2	19.7	5.0	10.2	29.1	0.351
19	1.00	1.73	196.8	20.3	5.6	11.4	30.1	0.379
20	1.00	2.05	200.0	21.5	6.8	13.8	37.1	0.372
21	1.00	2.46	203.9	23.2	8.5	16.2	40.0	0.405
22	1.00	4.01	218.9	30.0	15.3	20.3	40.0	0.507
23	1.10	0.00	180.1	14.7	0.0	0.0	40.0	0.335
24	1.10	0.82	188.0	17.2	2.5	5.1	12.1	0.405
25	1.10	1.32	192.9	18.9	4.2	8.6	21.6	0.398
26	1.10	1.54	195.0	19.6	4.9	10.0	24.1	0.415
27	1.10	1.75	197.0	20.4	5.7	11.6	28.5	0.407
28	1.10	2.23	201.7	21.8	7.1	14.5	34.2	0.423
29	1.10	3.01	209.3	25.5	10.8	19.3	40.0	0.482
30	1.20	0.00	180.1	14.7	0.0	0.0	40.0	0.392
31	1.20	0.99	189.7	17.8	3.1	6.3	14.1	0.447
32	1.20	1.03	190.1	17.9	3.2	6.5	13.8	0.471
33	1.20	1.14	191.2	18.3	3.6	7.3	15.7	0.465
34	1.20	1.41	193.8	19.2	4.5	9.2	19.0	0.485
35	1.20	1.64	196.0	20.0	5.3	10.8	24.9	0.474
36	1.20	1.83	197.9	20.8	6.1	12.4	23.9	0.519
37	1.20	2.58	205.1	23.6	8.9	18.1	34.1	0.531
38	1.30	0.00	180.1	14.7	0.0	0.0	40.0	0.465
39	1.30	0.60	185.9	16.6	1.9	3.9	7.4	0.527
40	1.30	0.94	189.2	17.6	2.9	5.9	11.2	0.527
41	1.30	1.00	189.8	17.8	3.1	6.3	12.1	0.511
42	1.30	1.27	192.4	18.7	4.0	8.1	15.1	0.537
43	1.30	1.53	195.0	19.6	4.9	10.0	17.1	0.585
44	1.40	0.00	174.1	14.7	0.0	0.0	40.0	0.527
45	1.40	0.51	182.8	15.6	0.9	1.8	3.2	0.562
46	1.40	0.46	184.8	16.2	1.5	3.1	5.3	0.505
47	1.40	1.07	190.5	18.0	3.3	6.7	10.1	0.662

Table 5. Correlation of experimental data.

Run no.	Percent vapor	Pressure drop per foot in. Hg./ft.	Reynolds' No. in thousands	Friction factor x 100	Mass velocity lb./hr.-sq. ft.	Mass velocity x velocity	Velocity ft./sec.	Increase in velocity ft./sec.	Fractional increase in velocity
3	0.00	0.177	51.6	3.5	234	950	4.04	-----	-----
4	0.63	0.220			235	1180	5.02	0.98	0.243
5	1.04	0.228			236	1220	5.17	1.13	0.270
6	1.49	0.262			237	1400	5.81	1.17	0.463
7	2.87	0.295			241	1580	6.56	2.52	0.624
13	0.20	0.275	62.6	3.4	297	1510	6.38	-----	-----
15	0.70	0.326			299	1800	6.02	0.94	0.165
17	1.26	0.335			299	1840	6.22	1.14	0.224
18	1.56	0.351			302	1930	6.40	1.32	0.200
19	1.73	0.379			302	2090	6.82	1.64	0.302
20	2.05	0.472			304	2050	6.75	1.57	0.329
21	2.46	0.405			305	2230	7.51	2.23	0.438
22	4.01	0.507			308	2790	8.25	5.97	0.741
30	0.00	0.392	79.7	3.3	356	2230	6.09	-----	-----
31	0.99	0.447			370	2540	6.57	0.75	0.125
34	1.41	0.485			371	2760	7.44	1.35	0.222
35	1.54	0.474			372	2850	7.23	1.14	0.167
36	1.83	0.519			373	2950	7.80	1.61	0.235
37	2.58	0.531			376	3020	8.03	1.44	0.218

Table 6. Pressure drop per foot of pipe for water near the boiling point after the data were corrected.

Run no.	Rotameter reading	Pressure drop/foot (exp.)	Pressure drop/foot* (calculated)
1	0.70	0.140	0.121
3	0.80	0.177	0.177
8	0.90	0.230	0.233
13	1.00	0.275	0.289
23	1.10	0.335	0.345
30	1.20	0.392	0.401
38	1.30	0.465	0.457
44	1.40	0.527	0.513

$$* \frac{\Delta p}{L} = 0.361 R - 0.272$$

$$\frac{\Delta p}{L} = \text{pressure drop per foot.}$$

$$R = \text{rotameter reading.}$$

Table 7. Data for chart to estimate the increase in velocity caused by flashing.

Water rate gal./min.	Percent vapor	Pressure drop per foot in. H ₂ O/ft.	Reynolds' No. in thousands	Friction factor x 100	Mass velocity lb./hr.-sq. ft.	Mass velocity x velocity	Velocity ft./sec.	Increase in velocity ft./sec.	Fractional increase in velocity
1.50	2.00	0.083	32.3	3.7	150	420	2.75	----	----
1.50	2.50	0.114			154	932	6.07	3.26	1.254
1.50	4.00	0.286			156	1445	7.25	4.47	0.520
1.75	0.00	0.135	42.9	3.5	175	777	4.83	----	----
1.75	2.00	0.235			179	1222	6.35	1.52	0.599
1.75	4.00	0.335			182	1741	9.55	3.22	1.371
2.00	0.00	0.164	47.1	3.5	201	905	4.80	----	----
2.00	2.00	0.284			205	1520	7.42	2.62	0.514
2.00	4.00	0.385			209	2060	9.87	4.97	1.514
2.50	0.00	0.270	58.9	3.4	251	1511	5.02	----	----
2.50	2.00	0.395			256	2145	6.37	1.35	0.351
2.50	4.00	0.512			261	2779	10.33	4.91	0.703
3.00	0.00	0.375	70.5	3.3	301	2130	7.07	----	----
3.00	2.00	0.508			307	2883	9.40	2.33	0.330
3.00	4.00	0.642			312	3645	11.07	4.60	0.592
3.50	0.00	0.474	82.5	3.3	351	2591	7.67	----	----
3.50	2.00	0.631			358	3582	10.00	2.33	0.304
3.50	4.00	0.790			365	4450	12.00	4.51	0.501
4.00	0.00	0.574	94.2	3.2	401	3360	6.37	----	----
4.00	2.00	0.753			410	4411	10.77	3.40	0.286
4.00	4.00	0.953			418	5475	13.09	4.72	0.554

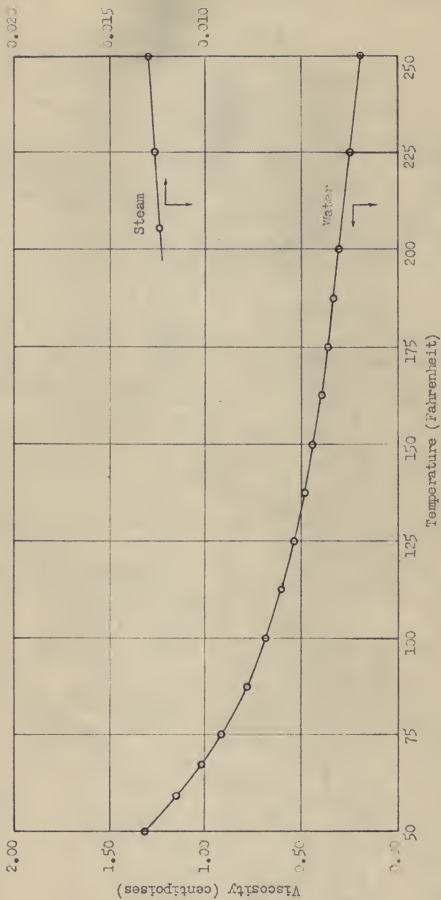


Fig. 11. Viscosity of steam and water (20).

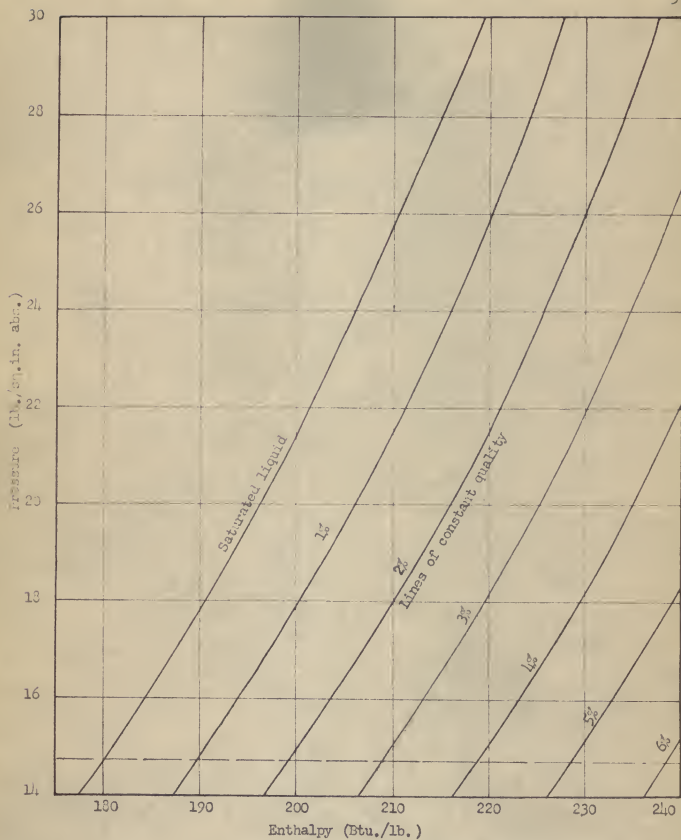


Fig. 12. Pressure-enthalpy diagram for water (9).

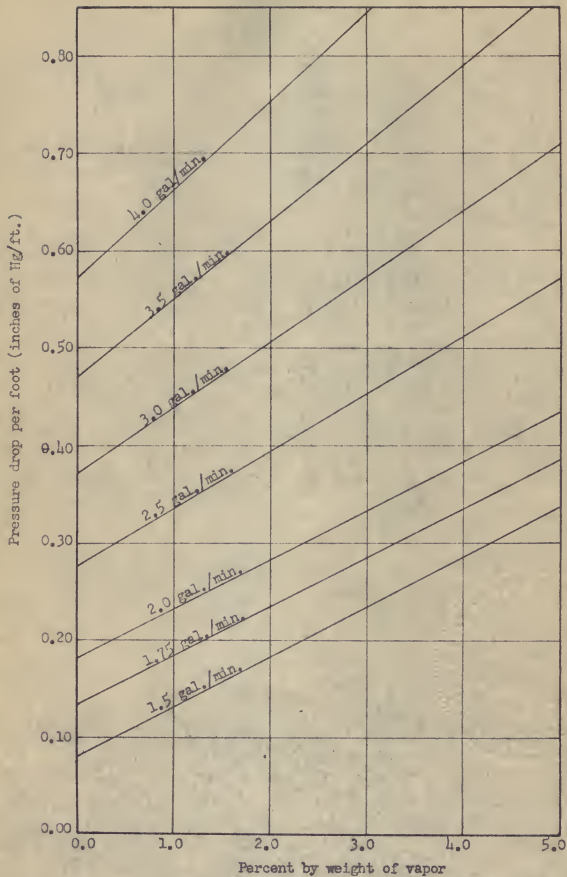


Fig. 13. Pressure drop per foot vs. quality after the data were smoothed.