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THE FLOW OF A FLASHING MIXTURE OF WATER AND STEAM AT HIGH PRESSURES

By
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May 16, 1955

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Technical Information Extension, Oak Ridge, Tennessee

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By

P. N. Haubenreich

May 16, 1955

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ABSTRACT

A series of experiments were conducted at Oak Ridge National Laboratory which involved the unsteady flow of a flashing mixture of water and steam. Saturation pressures as high as 1250 psia were reached in some cases.

Pressure drop and mass flow rate data from these experiments were analyzed to determine friction factors for pipes carrying flashing mixtures. The values which were obtained are generally higher than the friction factors reported previously for heater drain lines. The difference is believed to be due to the smaller lines and higher pressures which were involved in these experiments.

For purposes of calculating flow rates and pressure drops in flashing flow, it appears that the use of properties for constant enthalpy expansion is sufficiently accurate. Tables are presented which give the properties of water expanded isenthalpically from saturated liquid at several pressures.

CHAPTER I

INTRODUCTION

The design of valves and piping to pass a mixture of liquid and vapor is a problem which often occurs in engineering practice. In its most common form the problem is one of passing fluid which is initially liquid at or near the saturation point from one vessel through a throttling valve (or orifice) and a pipe to a receiver at a lower pressure. As the fluid passes along the line there is a decrease in pressure due to friction losses. This pressure drop is accompanied by the formation of quantities of vapor which may cause the specific volume of the mixture to increase manyfold. This type of flow is referred to as the flow of a flashing mixture, or simply as flashing flow.

Two examples of flashing flow which are important in steam power plant design are in feedwater heater drain lines and in boiler blowoff lines. In both instances the flow is controlled by a valve or orifice and the line is sized to pass the maximum anticipated flow rate. Up to the present time nearly all such lines have been sized by rule-of-thumb methods, for, despite the importance of the problem, there has been very little progress in developing suitable methods of calculation for flashing flow. As a result many valves and lines are either unnecessarily large and expensive or prove to be inadequate to pass the desired flow.

The flow of a flashing mixture of water and vapor is also encountered in a nuclear reactor of the aqueous homogeneous type. A simplified diagram of such a reactor designed for the production of useful power is shown in Figure 1 (1).* The fissionable material is carried as a suspension or in solution in water. The chain reaction raises the temperature of the fluid as it is pumped through a heat exchanger where steam is generated. In order to extract useful power, the fluid temperature, and hence the pressure, must be rather high. Because boiling may cause erratic behavior of the reactor power, the pressure is maintained several hundred psi above the saturation value corresponding to the temperature in the main circulating loop. This is accomplished by use of a pressurizer, which usually consists of a stagnant leg of liquid kept at a much higher temperature than the rest of the liquid. A volume of vapor in equilibrium with the hot leg acts as a surge chamber for the high-pressure system. The reactor system also includes low-pressure storage tanks into which the contents of the high-pressure system can be drained or dumped when necessary. The drain line connecting the high- and low-pressure systems is called the dump line. At the beginning of a dump, while there is still some overpressure, the fluid entering the line is compressed liquid. Within a short time, however, the pressure drops to the saturation value in the core. Subsequently the liquid is saturated at the line inlet and flashes as it flows through the line.

Although it is possible to insure that the contents of the high-pressure system will be discharged within a given length of time simply by putting in an oversize dump line, this is not sufficient in a two-region reactor. In this type of reactor, shown in Figure 2, there are

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* Numbers in parentheses refer to similarly numbered references in bibliography at end of paper.

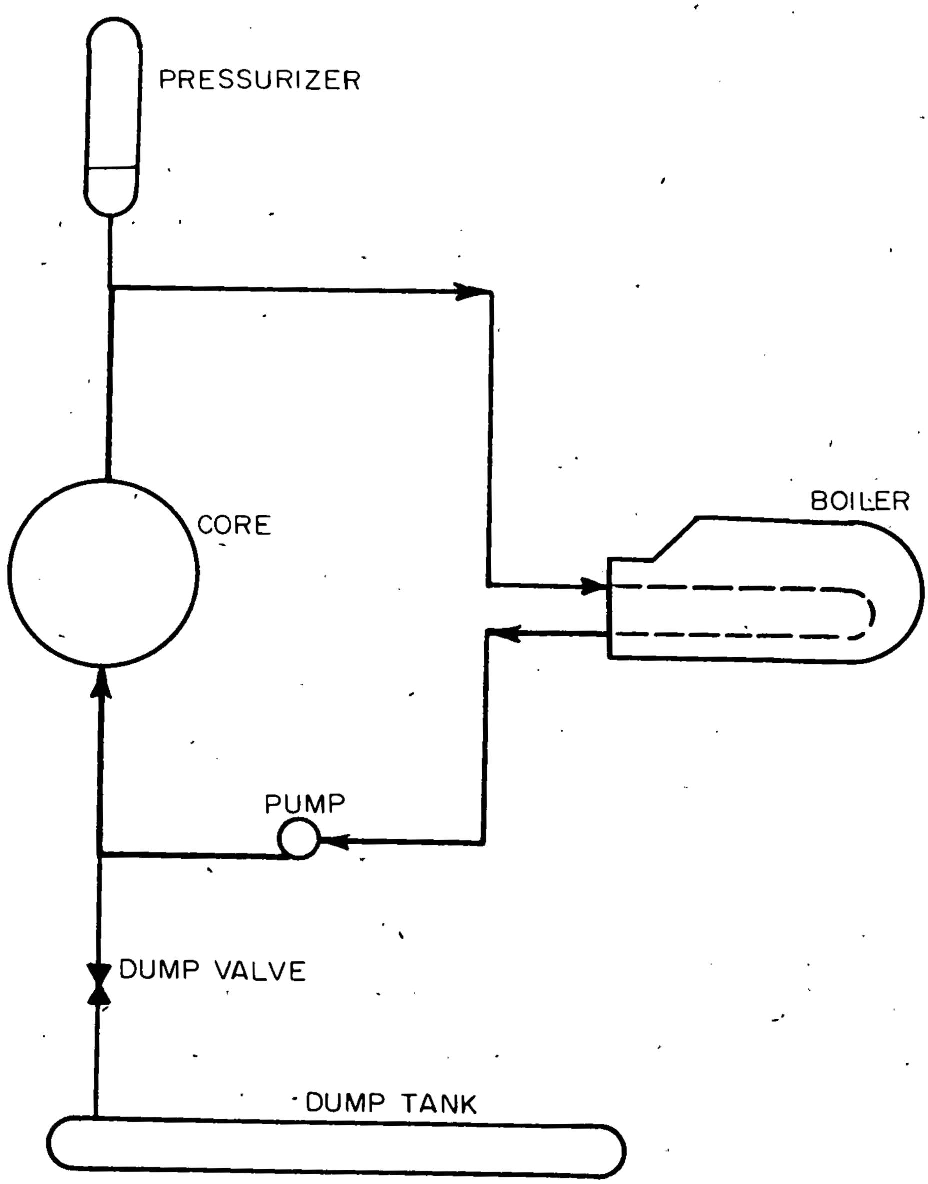


Fig. 1. Components Of An Aqueous Homogeneous Power Reactor

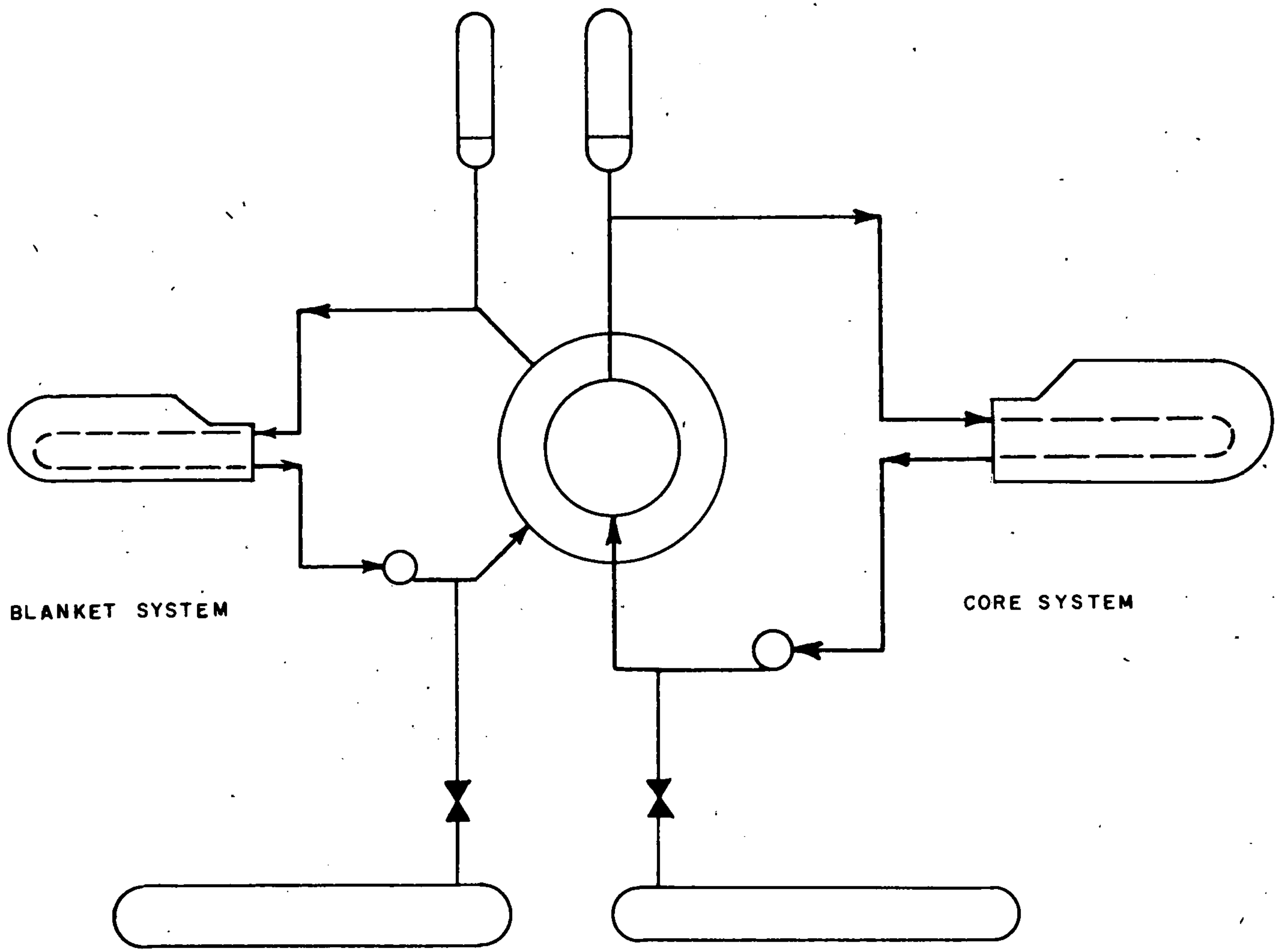


Fig. 2. Two-Region Homogeneous Reactor

really two systems. One contains the fuel solution; the other, liquid reflector or fertile material. In order to reduce the parasitic absorption of neutrons, the core tank, which separates the two high-pressure systems, is made as thin as practicable. No great thickness is required, since in normal operation of the reactor the pressures on either side can be kept nearly the same. During a dump, however, even though both systems are dumped simultaneously, the pressure in one might drop much more rapidly than in the other, thus buckling or rupturing the thin-walled tank. Regulation of the valve openings is a possible means of limiting the pressure differential. Even so, it is desirable to design the two dump lines so that the amount of valve regulation is minimized or, if possible, eliminated altogether. To accomplish this it is necessary to have a way of accurately determining the flow rate of initially saturated liquid through a line of a given length and diameter which contains a valve of a specific design.

Oak Ridge National Laboratory is presently designing and building a small two-region reactor known as the Homogeneous Reactor Test (HRT). The Laboratory is also studying large aqueous homogeneous reactors for the production of central station power. Because of the importance of the dumping problem and because of the large uncertainties in any calculations involving flashing flow, a series of experiments have been conducted to obtain information which can be used in the design of dump lines. The purpose of this paper is to determine friction factors and loss coefficients from the experimental results, and then to show how such factors can be applied in predicting flow rates for a line passing initially saturated liquid.

BACKGROUND

The amount of published data on the flow of a flashing mixture through pipes is rather meager. Bottomley reported the results of a single test made on a feedwater heater drain line[2]. Benjamin and Miller reported measurements on several drain lines[3]. Burnell published the results of a number of tests in which water was discharged from a boiler through straight pipe of various lengths and diameters[4].

Early writers predicted that in flow through a pipe a flashing mixture would exhibit critical pressure phenomena just as do vapors and gases. The tests referred to above proved this experimentally by measuring a line exit pressure well above pressures in the discharge receiver.

Benjamin and Miller analyzed their results on the basis of a finely divided mixture with equal velocity of water and steam. Starting with the dynamic equation and the continuity equation, they derived expressions, requiring numerical or graphical solution, which could be used to predict critical pressures at the pipe exit. Charts based on their equations have been widely used to estimate the capacity of boiler blowoff lines[5].

Allen analyzed the results of the above mentioned authors[6]. He also used the dynamic and the continuity equations to arrive at equations for predicting critical pressures at line exits. However, he proposed the use of approximate formulas for the thermodynamic properties which simplified the required integrations. Results were the same for either method and checked the experimental observations in the heater drain lines very well. In Burnell's tests, on the other hand, the critical pressures were much lower than predicted by the equations. According to Allen, this indicated that in the long, straight pipes of Burnell's experiments there was separation of the liquid and vapor streams, which invalidated one of the basic assumptions of the analysis.

CHAPTER III

DESCRIPTION OF EQUIPMENT AND INSTRUMENTATION

The experimental equipment, shown schematically in Figure 3, consisted essentially of a pressure vessel connected to a large receiver by a line containing a quick-opening valve. The manner of operation of the equipment is briefly as follows: Liquid was charged into the pressure vessel, the temperature and pressure brought up to the desired levels and the valve opened. As the contents of the vessel rushed through the line, pressures at several points were measured and recorded. The amount of water collected in the receiver was also measured and recorded continuously.

The pressure vessel is shown in Figure 4. The electric heaters on the flange and on the lower part of the 12-inch pipe were used to heat the main body of the liquid. Thermocouples projected upward through the blind flange, with the junctions 3-1/2 inches, 34 inches and 70 inches above the inside surface of the flange. The pressure in the vessel could be raised above the saturation value for the main body by heating the liquid in the pressurizer. Thermocouples were provided immediately above and below the pressurizer heater and near the top of the pressurizer. Vessel pressure was measured by a pressure cell located at the top of the pressurizer. The vessel was protected by a 2500-psi rupture disc in a vent line from the pressurizer.

Details of the bottom flange and the dumpline entrance are shown in Figure 5. The section of 1/2-inch sch 160 pipe had been welded in the flange for other purposes prior to this experiment. The other

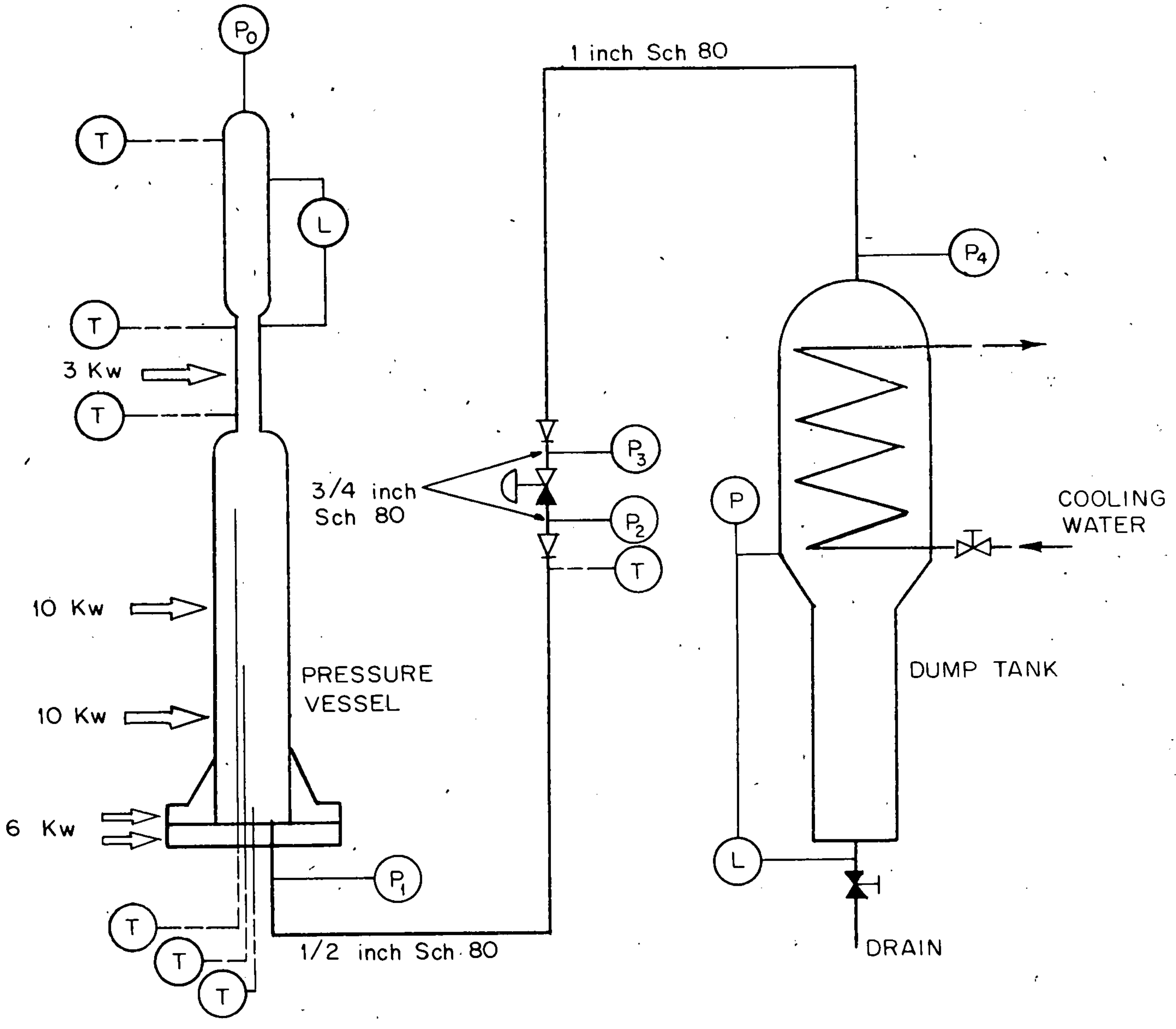


Fig. 3. Schematic Flow Diagram

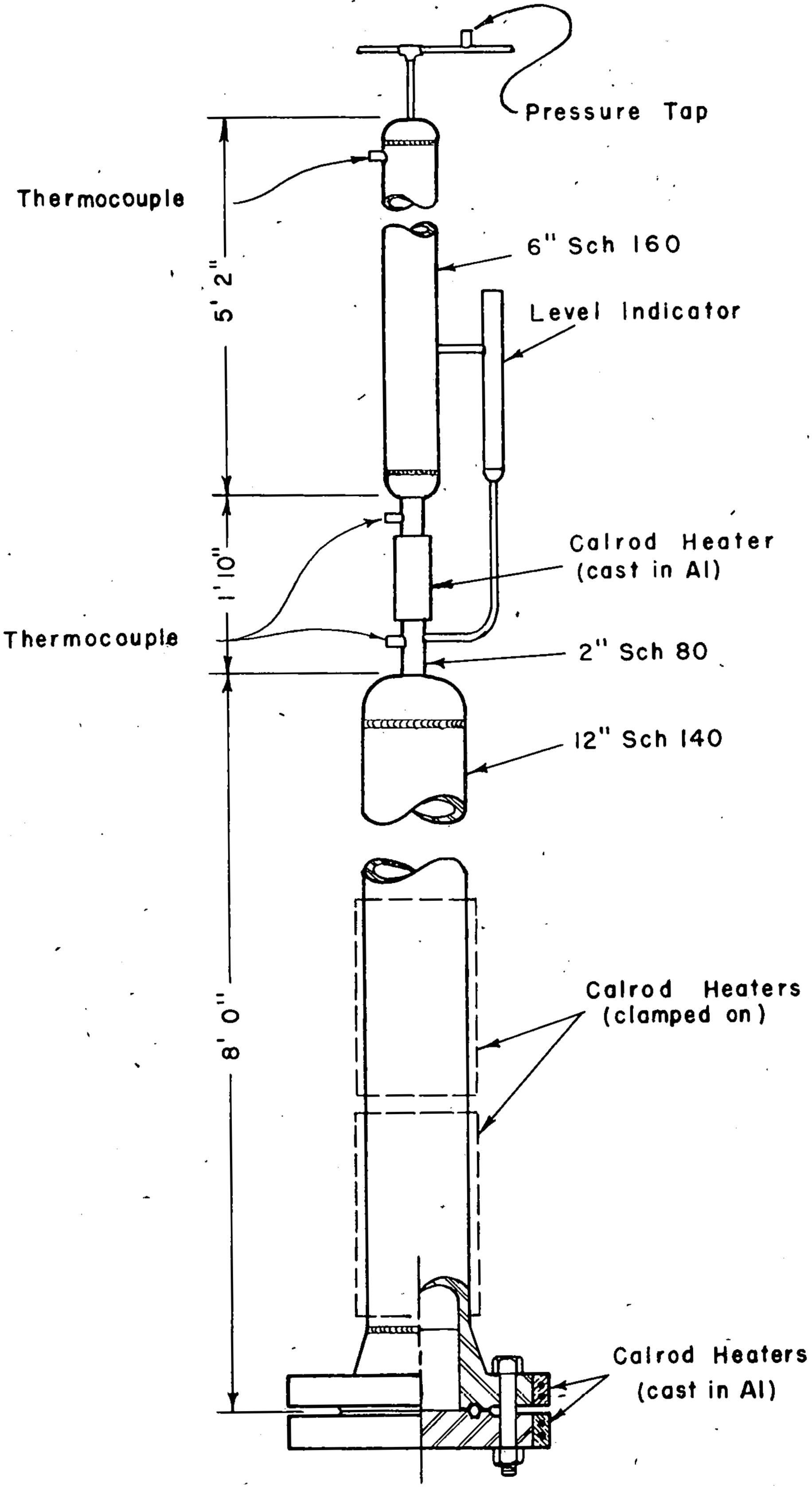
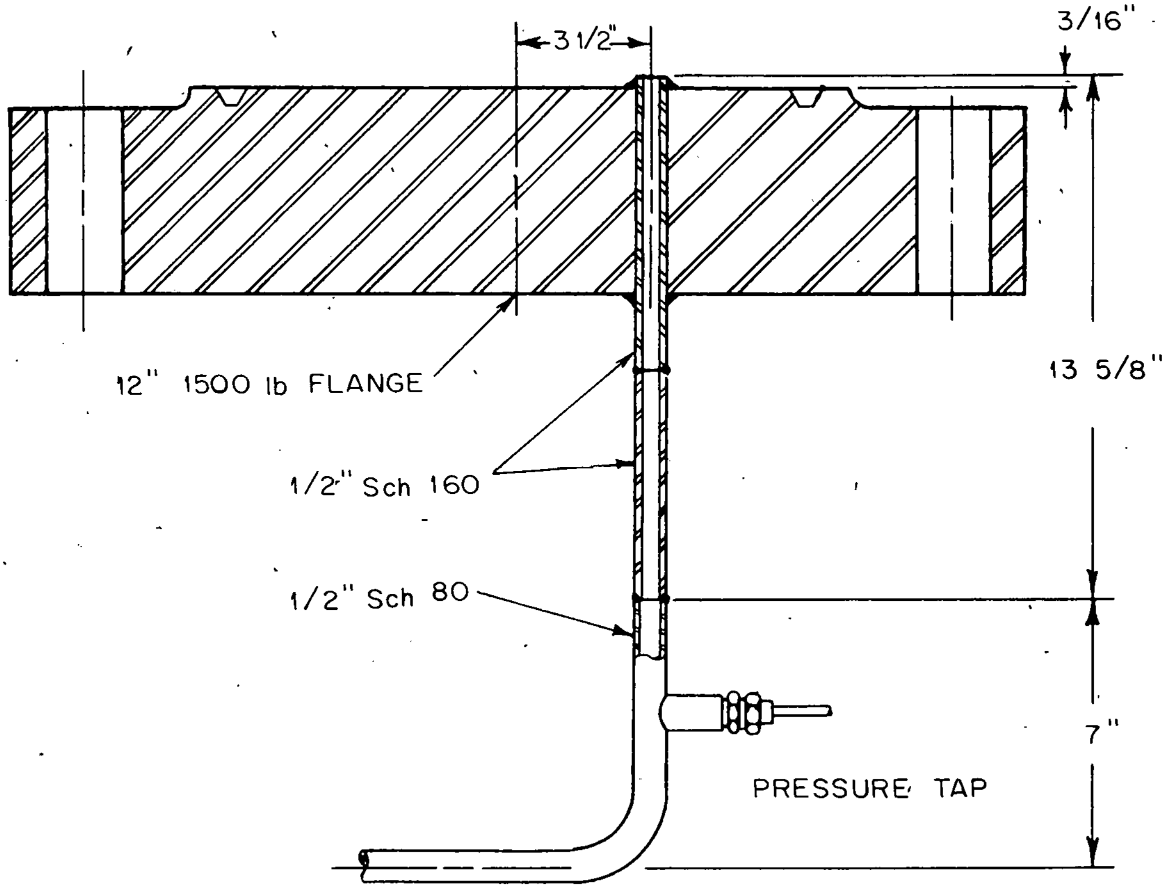
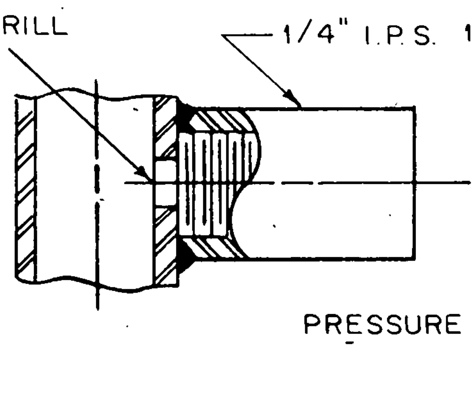


Fig. 4. Pressure Vessel





PRESSURE TAP DETAIL

Fig. 5. Bottom Flange

pressure taps in the dump line were identical with the one shown in this figure.

The layout of the dump line is shown in Figure 6. This rather unusual configuration simulated the lengths of line and the number of bends in the reactor. The 1/2-inch line was surrounded by a jacket of 1-1/4-inch sch 40 pipe for a total of 14 feet beginning about 2-1/2 ft from the line entrance. All bends in the 1-inch section and the bend in the jacketed section were on a 3-1/2 inch radius. Other bends in the 1/2-inch line were on a radius of 2 inches.

The valve was a Foxboro Type V4 3/4-inch valve with 1/2-inch trim, shown in Figure 7. The valve was placed so that flow was upward through the seat.

The dump tank is shown in Figure 8. The level in the tank was measured by a Foxboro electronic differential pressure cell arranged as shown. Because condensate would have collected in the line extending upward from the cell in any event, this line was kept full of water and a constant head maintained by a seal pot. Arranged in this way, the maximum differential pressure occurred when the tank was empty and decreased as the level rose in the tank. The range of the cell was 0-150 inches of water. The output from the cell was recorded by a Foxboro Dynalog Universal strip chart recorder. During one run a sight glass was installed on the dump tank.

Pressures in the pressurizer and dump line were measured by Baldwin SR4 pressure cells Type BS410. The cells on the pressurizer, at the line entrance, valve inlet and valve discharge were 2000 psi capacity. Output

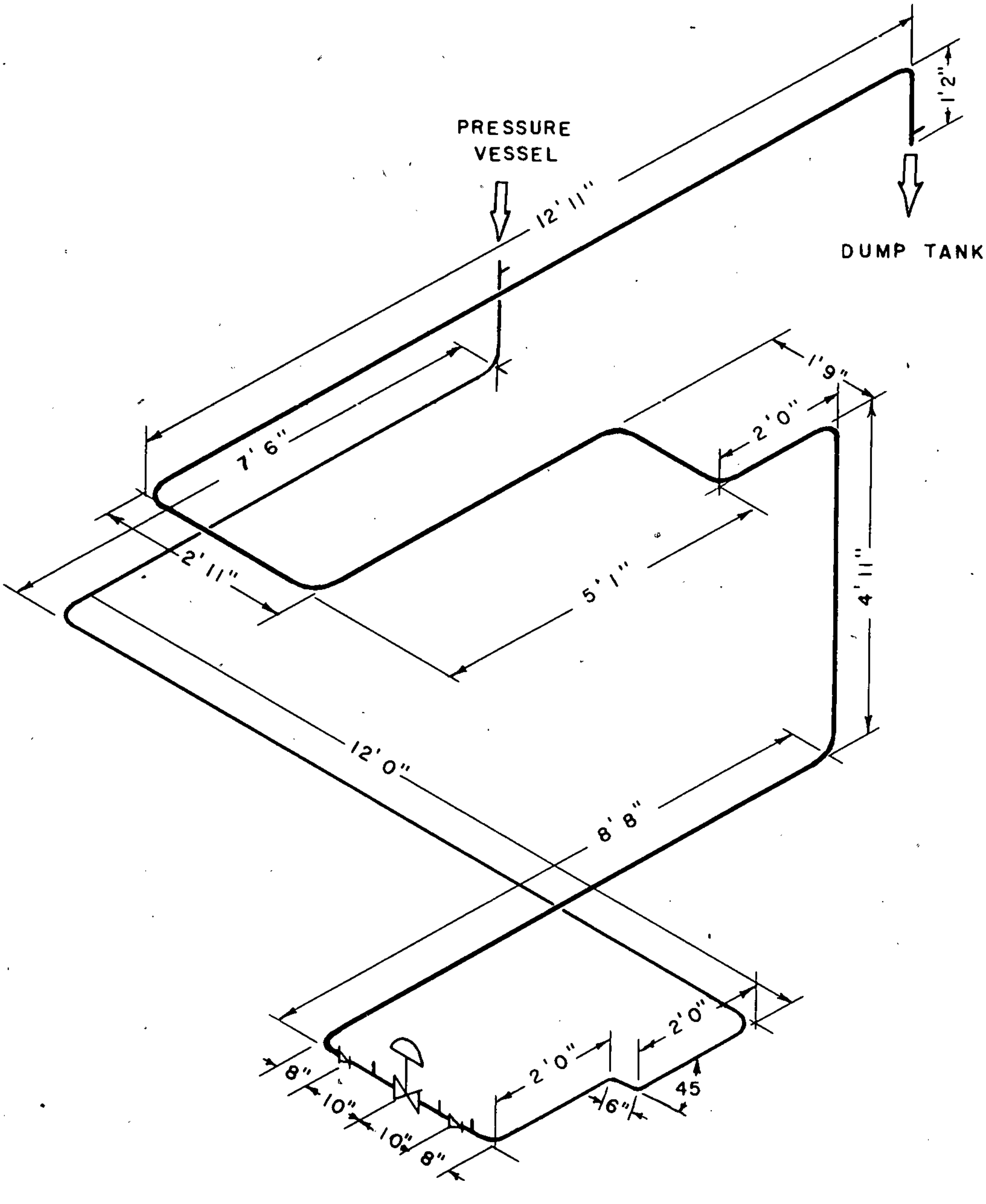


Fig. 6. Dump Line

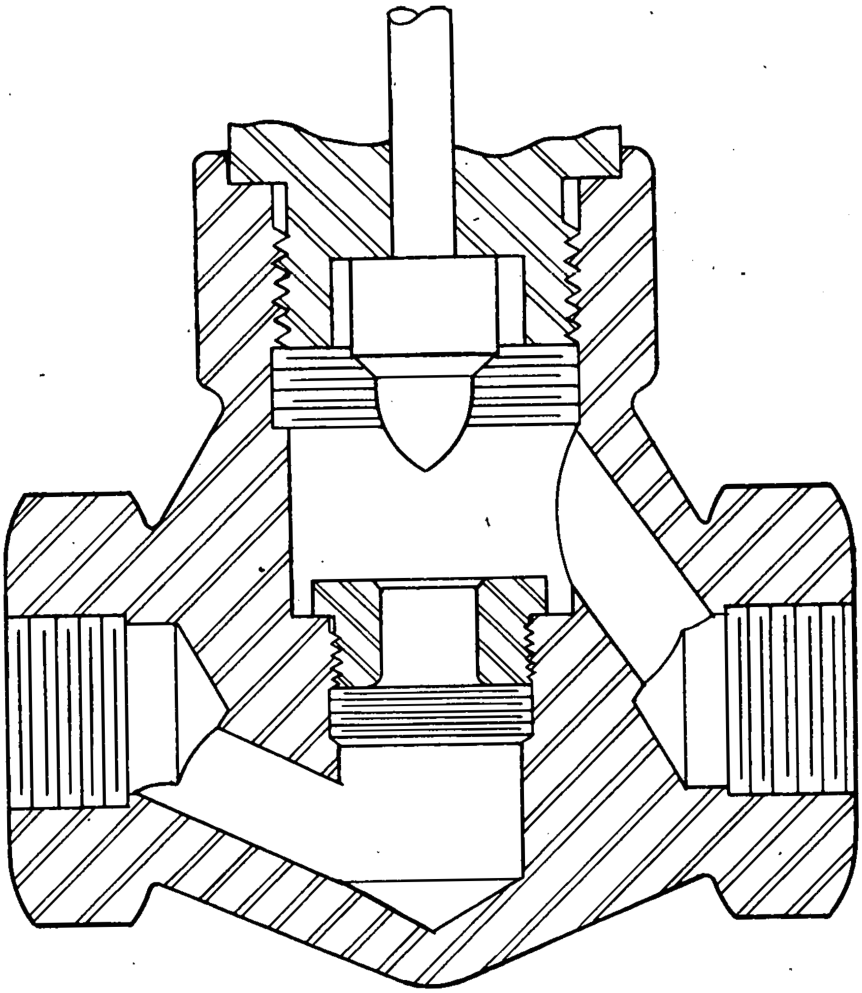


Fig. 7. Valve Body and Trim

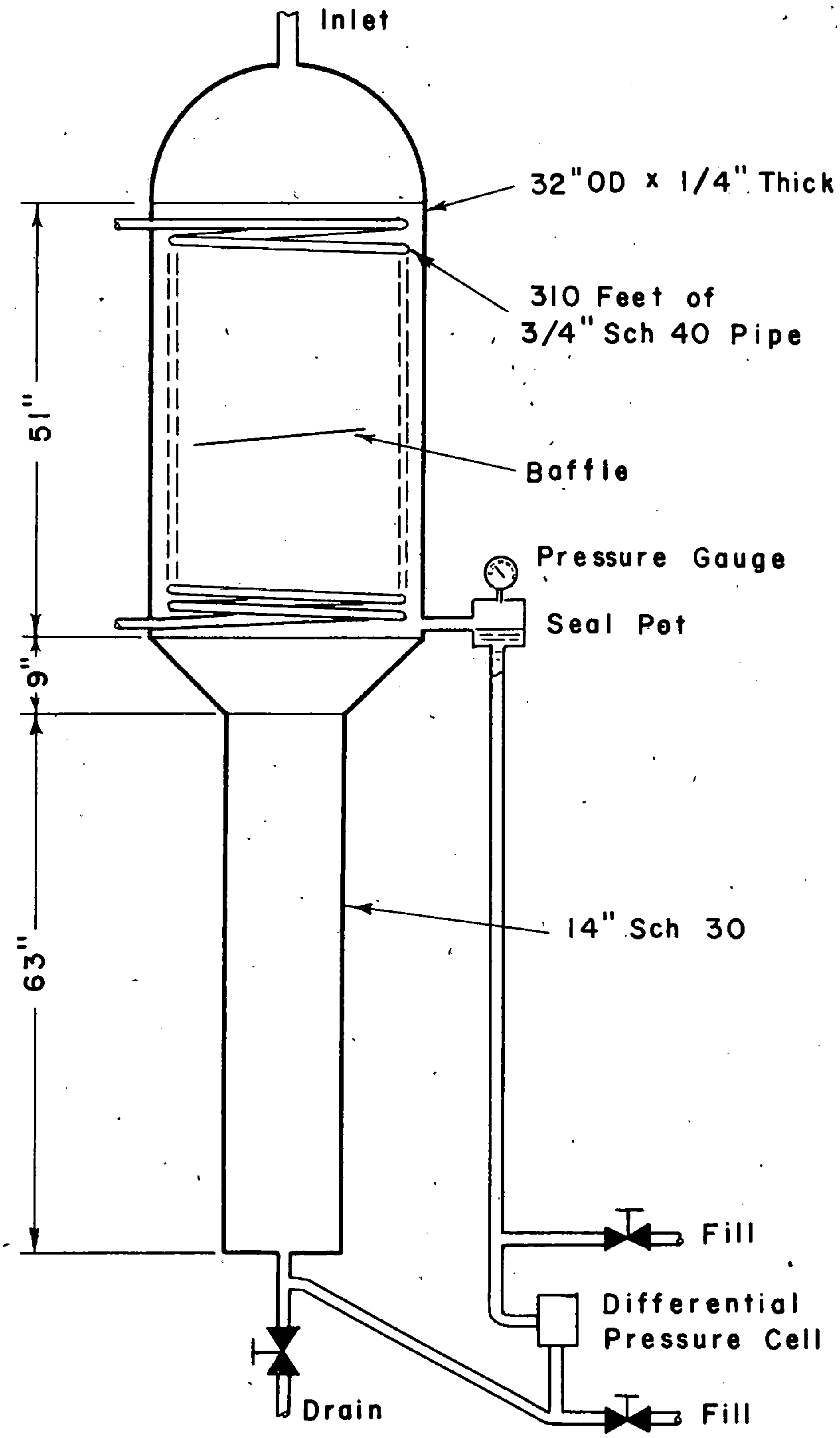


Fig. 8. Dump Tank

from these cells went to Brush Universal Amplifiers Type BL-320 which were connected to Brush Magnetic Penmotors Type BL-202. The Baldwin pressure cell at the line discharge was 1000 psi capacity. The signal from this cell went through a Swartwout A2A Strain Gage Amplifier to a Swartwout A3A Autronic Recorder.

All the thermocouples were iron-constantan. Use of Conax glands made possible the exposure of the bare junctions to the fluid. For the thermocouple at the valve inlet, a Conax gland was screwed into a threaded coupling like the one shown in Figure 5. The ceramic insulator extended into the 1/4-inch hole and ended flush with the inside of the pipe wall. The junction was made right against the end of the insulator so that it extended about one-sixteenth of an inch into the flowing stream. All temperatures except the one at the valve inlet were recorded on a Brown multipoint recorder. The valve inlet temperature was recorded on a Brown Electronik strip chart recorder.

CHAPTER IV

EXPERIMENTAL PROCEDURE AND RESULTS

To begin a run, the dump valve was closed and the pressure vessel evacuated, using a vacuum pump connected to the line at the top of the pressurizer. A Scott and Williams Hydropulse pump was then used to introduce the desired amount of water into the vessel. Following this, the heaters were turned on and the water in the vessel brought up to the predetermined temperature. If an initial overpressure was desired, the pressurizer temperature was raised above the temperature of the main body of liquid. When the temperatures were at their desired levels, the dump tank was evacuated to about 24 inches Hg vacuum. Next, a flow of cooling water was started through the dump tank coils. The system was then ready for a dump.

During the heating-up procedure, the temperatures in the pressure vessel were recorded continuously by the Brown multipoint recorder. Immediately preceding the dump, the instruments which recorded the pressures, the temperature at the valve inlet and the dump tank level were turned on. The Brush pressure recorders and the Foxboro dump tank level recorder were connected by a circuit which enabled a timing mark to be registered simultaneously on all charts. When everything was ready, a signal was given, the timing mark registered on the Brush and Foxboro charts, and marks made manually on the Swartwout chart and the Brown chart which recorded the temperature at the valve inlet. These marks served as a time datum in reading the charts. A second or two after the timing marks were made, the valve was opened, starting the dump.

The valve was left open and the recorders running until the pressures and the dump tank level became steady. At this point the pressure in the dump tank was always subatmospheric because of the cooling water flow through the condensing coils. The dump valve was then closed, the instruments turned off and the contents of the dump tank drained into a weigh tank. The measured weight of the recovered water was used later to calibrate the dump tank level record.

During most of the runs, the chart speeds for the various recorders were: the Brush recorders, 5 mm/sec; the Swartwout, 4 in/min; the Foxboro, 6 in/min; and the Brown, 8 in/min. For some of the runs in which there was an initial overpressure, the chart speed on the Brush recorders was increased to 25 mm/sec in order to follow the rapid changes in pressure during the first few seconds after the opening of the valve.

Runs made with the pressure vessel containing saturated liquid at the start of the dump covered a range of initial pressures from 150 psia to 1250 psia. In addition some dumps were made in which the liquid in the vessel was initially at 300°C and 2000 psia.

Figure 9 shows the results of a typical run in which the pressure vessel initially contained saturated liquid. The time is measured from the timing marks on the recorder charts, and the valve did not begin to open until about 1.5 seconds after the timing signal was given. After the valve began to open, it took about 1.0 second for the full travel of the stem. The rather sharp break in the curves at about 62 seconds marks

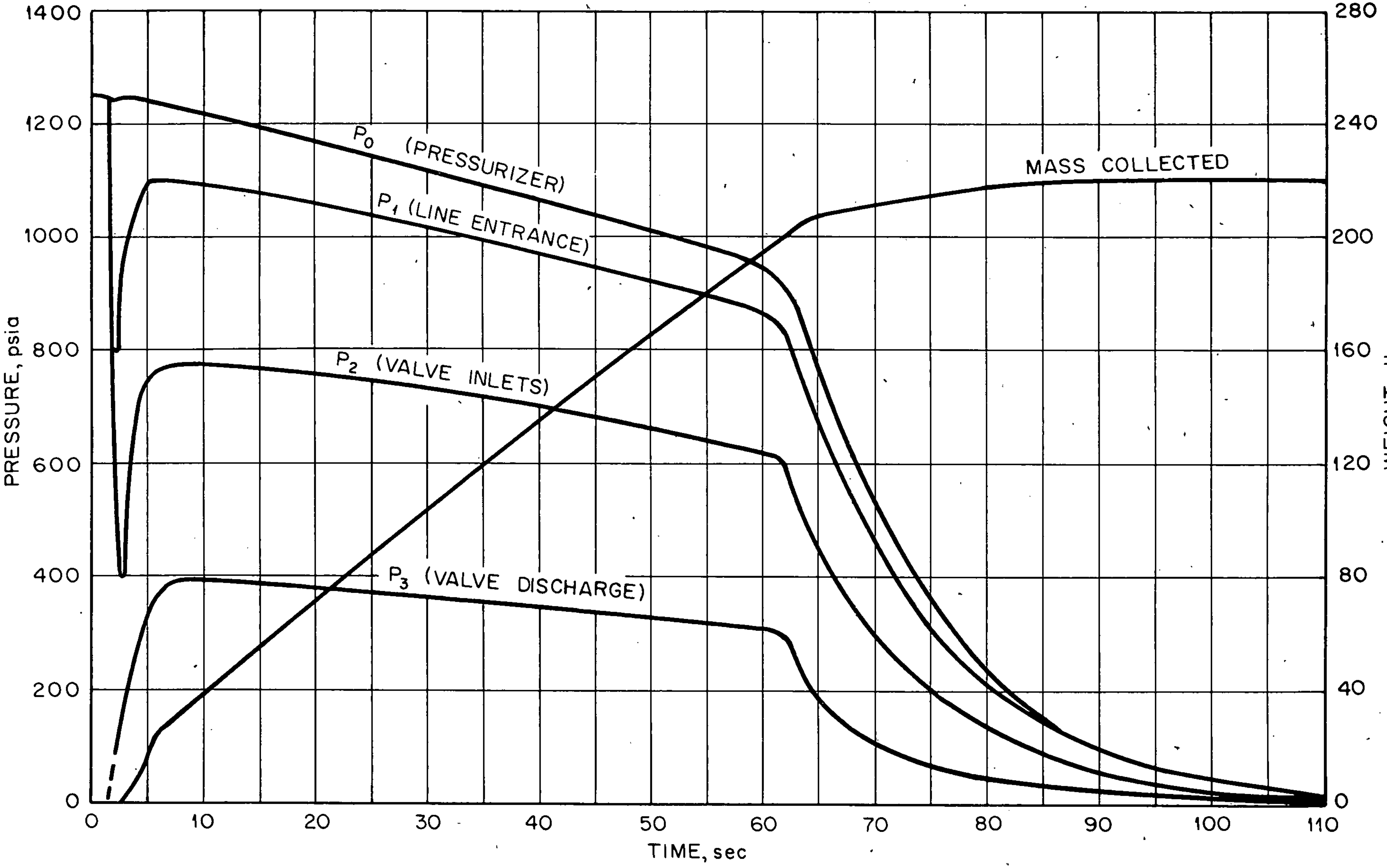


Fig. 9. Results Of Run 4

the point at which the last of the liquid left the pressure vessel and steam began to flow through the line.

Results of one of the pressurized runs are shown in Figure 10 and 11. Here the liquid in the main body of the pressure vessel was initially at 568°F and the pressure was 2005 psia. The pressurizer was at the saturation temperature for this pressure, 636°F. In this run the dump tank was provided with a sight glass. One of the recorders was equipped so that the observer watching the rise of liquid in the sight glass could press a button and register a mark on the moving chart. Marks were made for every six inches rise of the liquid between nine and 45 inches above the initial level. These readings, converted to mass of liquid collected, are indicated by the points on Figure 10.

Figure 12 illustrates the results of one of the runs at lower initial pressure. This shows the more gradual changes in pressure which were typical of the lower-pressure runs.

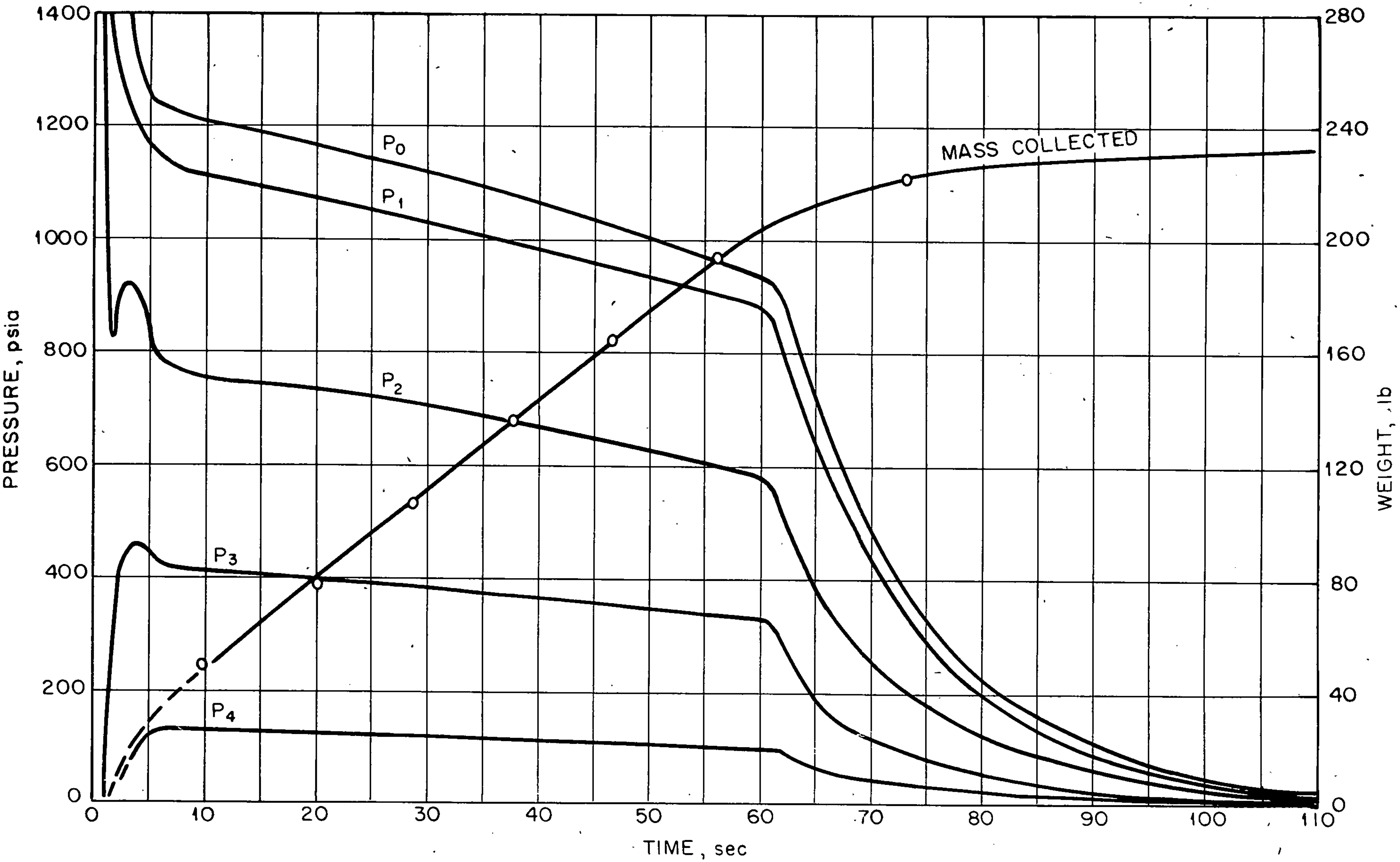


Fig. 10. Results Of Run 10

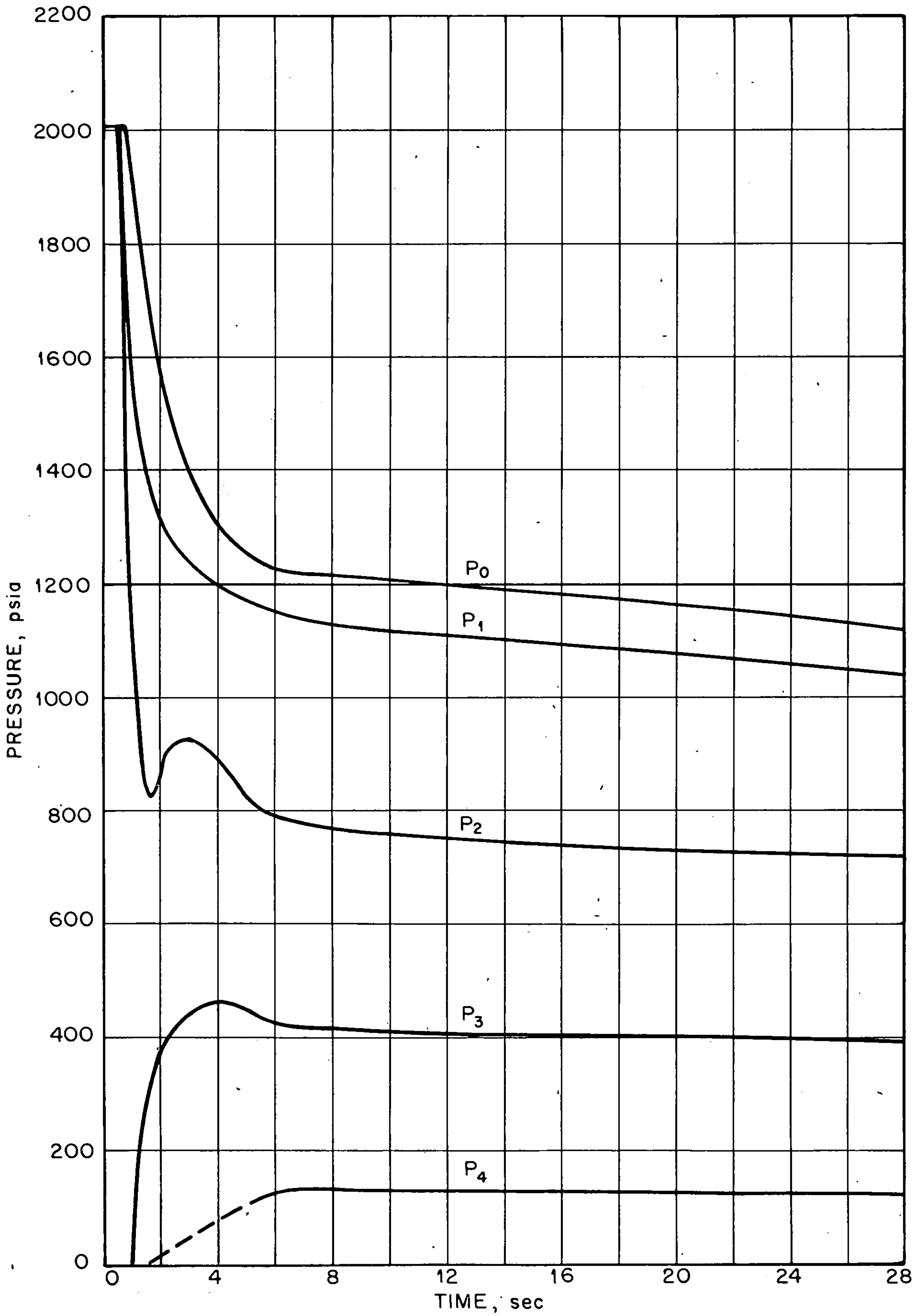


Fig. 11. Pressures During First Part Of Run 10.

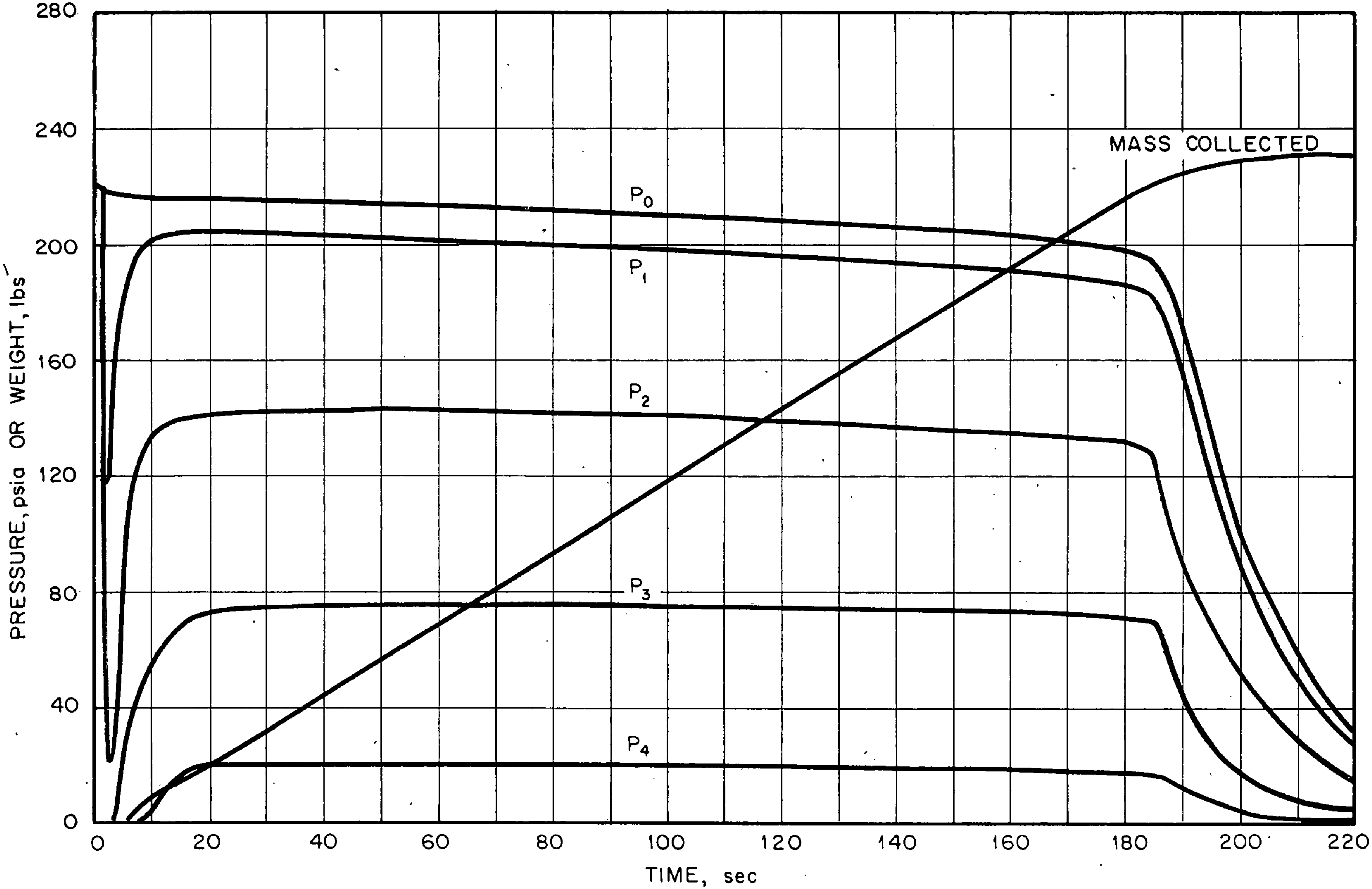


Fig. 12. Results Of Run 28.

CHAPTER V

THEORY

When a fluid which is initially saturated liquid flows through a pipe, the following changes occur:

(a) The pressure decreases because of friction.

(b) As the pressure decreases the saturation temperature also decreases and the liquid tends to become superheated.

(c) Part of the liquid becomes vapor.

(d) As evaporation occurs the temperature of the remaining fluid decreases, tending to approach the saturation temperature.

(e) As the fluid moves along the pipe, the pressure continues to decrease, more vapor is formed and the temperature continues to fall. Thus there is two-phase flow with the ratio of the two phases continuously changing.

When two-phase mixtures move through a pipe several different types of flow are possible.

1. The two phases may be finely dispersed and move with equal velocities.

2. The two phases may be mixed but traveling with unequal velocities.

3. The phases may travel separately at generally different velocities with a continuous interface between them.

4. There may be "slugging", where the flow consists of alternate slugs of liquid and vapor moving through the line.

In the case of flashing flow, vapor is being produced throughout the liquid, which tends to produce a finely dispersed mixture. Bends, valves or other irregularities which create turbulence act to maintain the dispersion. If the phases are well mixed, the difference in the average velocities of the liquid and vapor will not be great, due to the drag of one phase on the particles of the other. Thus the first type of flow described above is probably a good approximation in most situations where flashing flow occurs.

In the consideration of flashing flow some attention must be given to the possibility of the existence of metastable conditions. If there is a metastable condition, the properties of the fluid will not be those given by steam tables or equations of state which apply only to equilibrium conditions. In the case of the flow of saturated liquid through <u>sharp-edged</u> orifices, experiments have shown that the water did not flash until it was through the orifice.[(7)] However, for even very short nozzles there is evidence of flashing in the nozzle. In flow through a pipe, where the pressure drop is much more gradual than in a throttling device, it seems unlikely that any great degree of metastability could exist. Benjamin and Miller observed both temperature and pressure along lines carrying a flashing mixture.[(7)] In all cases the saturation pressure corresponding to the observed temperature checked the observed pressure within 0.5 psi. Burnell also took simultaneous pressure and temperature measurements and detected little, if any, difference in all cases. Of course, these measurements do not prove the non-existence of metastable conditions in pipeline flow, since the introduction of measuring instruments into the stream would tend to destroy the metastable condition, if such existed. However, even without measuring devices there are always disturbances in the flow. Therefore, it can be assumed that the properties of the fluid are essentially those of a mixture of the phases in equilibrium with each other.

In the analysis which follows it is assumed that liquid and vapor move with equal velocities and that the velocity is approximately constant over the cross section of the pipe. Heat transfer and changes in elevation are assumed to be negligible.

For steady flow the continuity equation is

$$\bar{G} = \frac{W}{A} = \frac{V}{v} \qquad [1]$$

For steady one-dimensional flow, where changes in elevation are negligible and where there are no internal drag forces, the differential equation of motion through a circular section is

$$v\,dp + d\left(\frac{V^2}{2g_c}\right) + \frac{4f\,V^2}{D\,2g_c}\,dL = 0 \qquad [2]$$

For a constant area channel, $\bar{G}$ is constant, so that $dV = \bar{G}\,dv$ and $d(V^2) = 2\bar{G}^2 v\,dv$. Thus for a circular channel of constant diameter, equation [2] can be written as

$$v\,dp + \bar{G}^2\,\frac{v\,dv}{g_c} + \bar{G}^2\,2f\,\frac{v^2\,dL}{g_c\,D} = 0 \qquad [3]$$

Dividing by v^2 and integrating, equation [3] becomes

$$\int_{p_1}^{p} \frac{dp}{v} + \frac{\bar{G}^2}{g_c}\ln\frac{v}{v_1} + \bar{G}^2\,\frac{2f\,L}{g_c\,D} = 0 \qquad [4]$$

This can be rearranged to give

$$G^2 = \frac{g_c \int_p^{p_1} \frac{dp}{v}}{\ln\left(\frac{v}{v_1}\right) + \frac{2f\ L}{D}} \qquad [5]$$

In order to carry out the integration indicated above, it is necessary to trace the changes in the state of the fluid as it expands from p_1 to p. The differential form of the energy equation for steady flow with no heat transfer, no significant change in elevation and no external work is

$$dh + \frac{1}{J}\ d\left(\frac{V}{2g_c}\right) = 0 \qquad [6]$$

By integration

$$h + \frac{V^2}{2g_c J} = h^o = \text{constant} \qquad [7]$$

Substituting V = Gv, equation [7] becomes

$$h + \frac{G^2}{2g_c J}\ v^2 = h^o \qquad [8]$$

For constant-area flow, G is constant. When G and h^o are fixed, equation [8] is sufficient to determine the states through which the fluid passes. Thus this equation represents a curve on the h-s diagram whose position and shape are determined by the parameters G and h^o. Such a curve is called a Fanno line.

Return now to equation [5]. If the conditions at the line entrance are held constant, and if the length of the drain line is fixed so that fL/D cannot change, the mass velocity, as shown by this equation, will depend only on the downstream pressure, p. If values of p not much less than p_1 are chosen, and the mass velocity calculated by equation [5], it is found that G increases with decreasing p, as would be expected. However, if this process is repeated at successively lower values of p, it is found that the calculated value of G reaches a maximum and then begins to decrease.

If an experiment were attempted, keeping the inlet conditions and line length fixed and gradually reducing the pressure of the discharge receiver, the flow rate would increase and reach a maximum as predicted by the equation. If the receiver pressure were reduced still further, however, there would be no more change in either the mass flow rate or the pressures inside the line. The reason for this behavior is that at the maximum flow rate the velocity at the end of the line has become so high that pressure impulses can no longer travel upstream. Therefore the conditions in the pipe are unaffected by changes in the receiver pressure. The pressure in the end of the line when such a condition exists is called a critical pressure. It is equal to the pressure at which the mass flow rate, calculated by equation [5] is at its maximum, provided the conditions of the experiment match those assumed in the derivation of equation [5] and in the evaluation of the integral and volume ratio in this equation.

The Fanno line also indicates the existence of a critical pressure. If the Fanno line is extended to sufficiently low pressures, the entropy will reach a maximum and then begin to decrease with decreasing pressure.

(See Figures 15 and 16.) The pressure at which the entropy is at its maximum value is the critical pressure.

It must be remembered that the Fanno line only represents conditions along a pipe. Thus it can be used to predict the critical pressure when there are no heat losses and when the mass flow rate is known. It cannot be used to predict the flow through a given line. Equation [5], on the other hand, can be used to predict mass flow rates and critical pressures, if the resistance of the line (fL/D) is known and the integral and the volume ratio can be evaluated.

CHAPTER VI

ANALYSIS OF EXPERIMENTAL RESULTS

In the development of the equations describing the flow through the dump line, it was assumed that a steady flow existed. In the experiment the flow was never actually steady. However, at times during each dump conditions were changing slowly enough so that the assumption of steady flow is a good approximation. As can be seen from the experimental data, after the loss of the initial overpressure, all the line pressures changed rather slowly until steam began to enter the line. During this period the conditions at the line inlet did not change appreciably in the time required for fluid to pass through the pipe. Only the data taken during this period were used in the analysis of flashing flow.

The equation used in calculating line friction factors is equation [5]. In this equation there appears the term L/D, which must be taken as the value for a length of straight pipe having the same resistance as the actual line. Calculations of equivalent lengths for the 1/2-inch and 1-inch lines are given in Appendix A. For the 1/2-inch line the equivalent L/D was 570; for the 1-inch line it was 549.

The procedure which was first used for calculating friction factors is as follows: Taking the results of one of the runs, plotted as in Figure 9, a time was chosen when the pressures were not changing rapidly, i.e., when saturated liquid could be assumed to be entering the line. The pressures in the pressurizer, in the line entrance, at the valve inlet, at the valve discharge and at the line exit were read from the curves. The mass discharge rate at this time was obtained by measuring the rate of increase of the mass collected in the dump tank. The pressurizer pressure fixed the stagnation enthalpy, since the water in the pressure vessel was saturated liquid at this pressure. The pressure at the beginning and end of a section of line were used to evaluate the integral and the volume ratio in equation [5]. In order to simplify the calculation of specific volume as a function of pressure, the expansion was assumed to be isenthalpic. (As will be shown later, the error introduced by this is small, especially in the line preceding the valve.) The resistance of the line was then found from*

$$\frac{2fL}{D} = \frac{g_c \int_{p_2}^{p_1} \frac{dp}{v}}{G^2} - \ln \frac{v_2}{V_1}$$

Since the equivalent L/D of the line was known, the friction factor could be readily calculated.

Preliminary calculations using the above procedure gave friction factors which varied considerably from run to run. This was attributable to relatively small differences in the indicated pressure drops which caused fairly large variations in the calculated friction factors. Rather than average the calculated factors, it was decided to average the data before analysis. Results of all the runs were compared and the best

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* Subscripts shown are for the 1/2-inch line. For the 1-inch line the pressures would be p_3 and p_4.

average values of pressure drops and flow rates determined. These values were then used to calculate friction factors.

The graphs which were used to compare the results of the various runs are shown in Figures 13 and 14. All of the points shown were taken from the curves (such as Figure 9) in regions where saturated liquid was entering the line. The pressurizer pressure was chosen as the ordinate since the condition of the water entering the line is fixed by this quantity alone. Fixing the conditions at the inlet fixes the pressures along the line and the mass discharge rate. This is true because (a) the valve setting remained the same in all runs, and (b) the flow was unaffected by dump tank pressure because of the existence of a critical pressure at the line exit.

Friction factors were again calculated using specific volumes based on isenthalpic expansion, but this time with the pressures and discharge rates taken from the curves through the experimental points. The results are shown in Table I. For the 1/2-inch line, f was calculated directly from W, p_1 and p_2. Because of the critical pressure at the end of the line, it was not possible to solve directly for the f in the 1-inch section. Therefore values of f were determined which would give the observed W when p_3 was as observed and the critical pressure existed at the end of the line. As shown in Table I, the calculated line discharge pressures were fairly close to the observed values of p_4. Sample calculations of the friction factors are given in Appendix B.

In the process of the calculations described above, v, $\ln\left(\frac{v}{v_o}\right)$ and $g_c \int_p^{p_o} \frac{dp}{v}$ were computed as functions of p for several values of p_o.

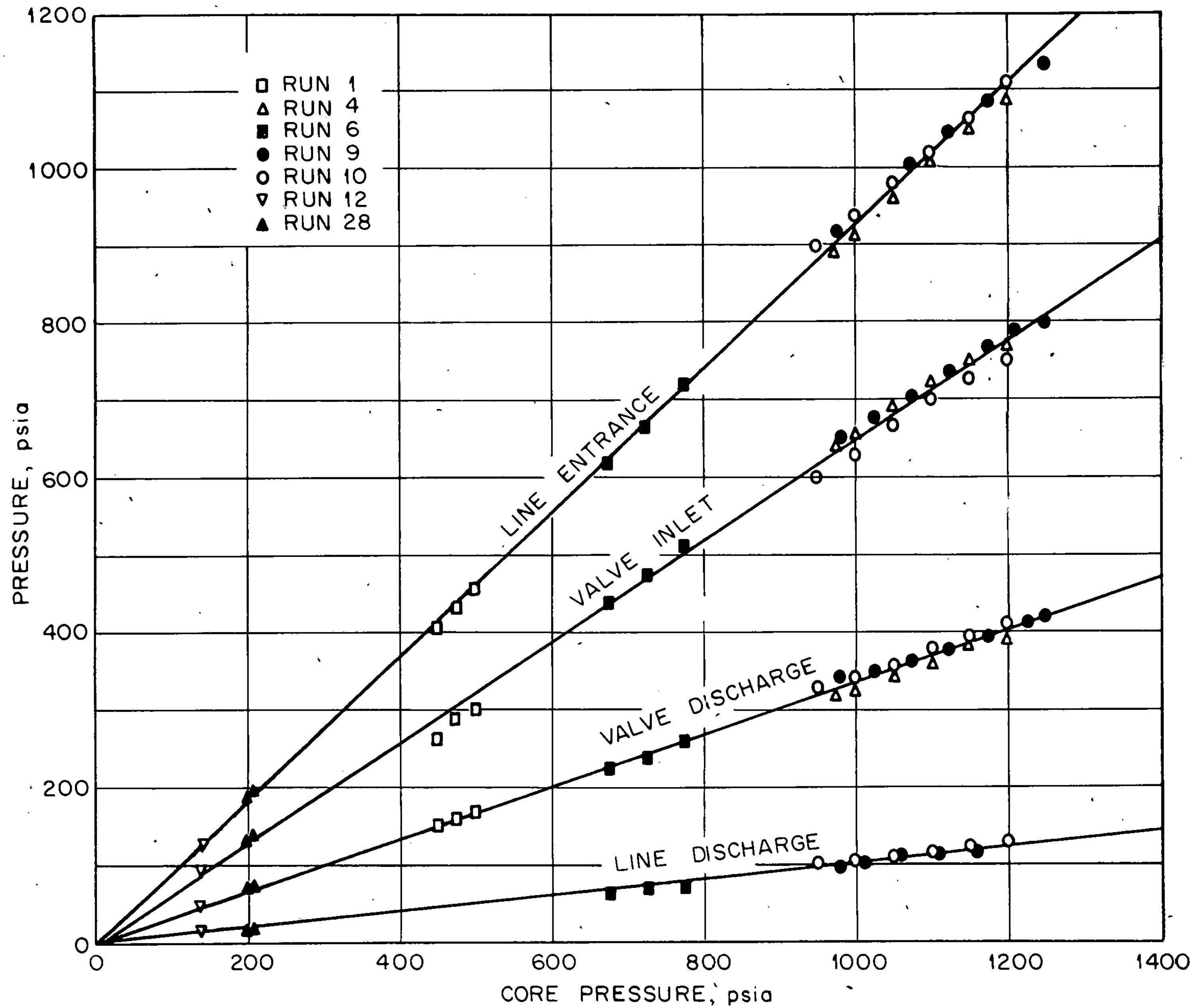

Fig. 13. LINE PRESSURES

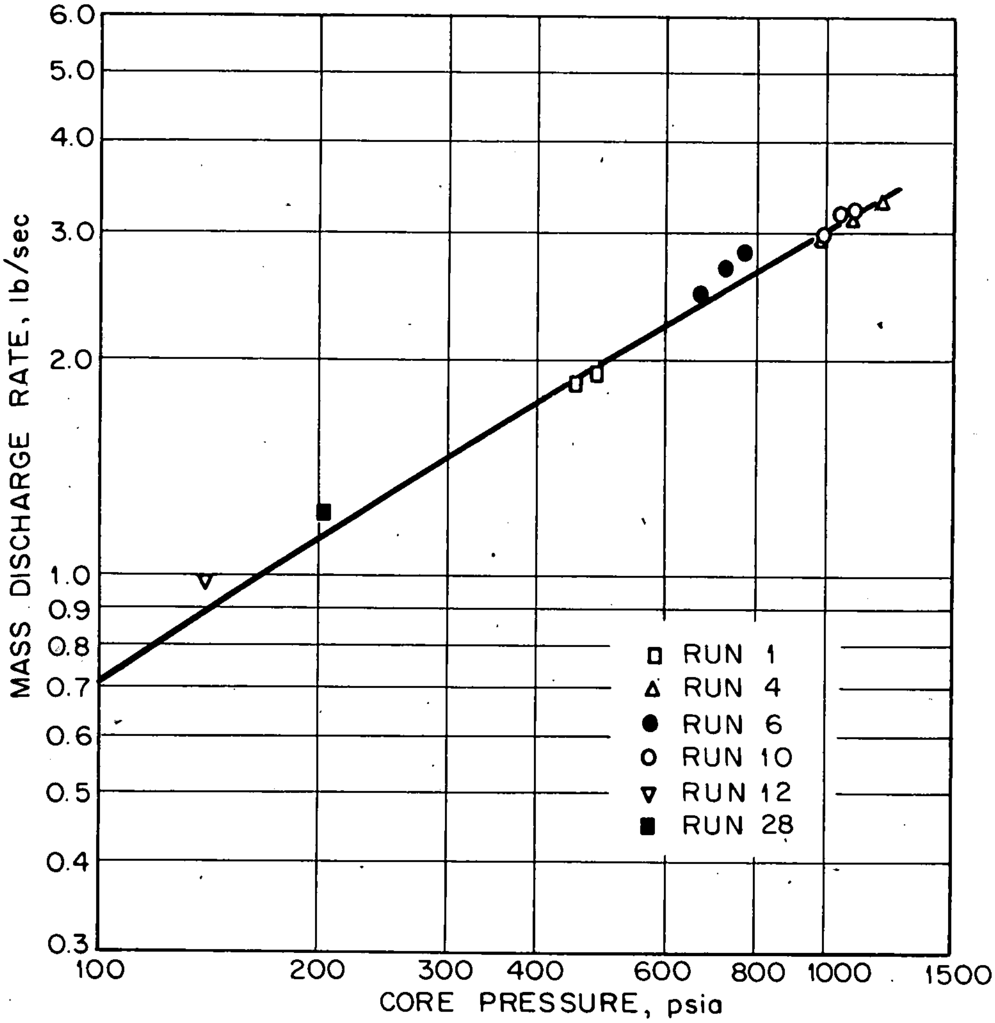

Fig. 14. Mass Discharge Rates

TABLE I

LINE FRICTION FACTORS BASED ON ISENTHALPIC EXPANSION

Observed Quantities				
p_0, psia	1100	450	200	140
p_1, psia	1020	416	188	133
p_2, psia	710	295	133	95
p_3, psia	366	153	71	50
p_4, psia	115	47	22	16
W, lb/sec	3.20	1.91	1.20	0.98
Calculated Quantities				
4f (1/2-inch)	0.0248	0.0209	0.0179	0.0168
4f (1-inch)	0.0186	0.0120	0.0082	0.0063
p_4, psia	118	57	30	24

These results are given in Appendix C.

As explained in a previous section, even if the flow were adiabatic, the expansion of the fluid would not be at constant enthalpy because of the decrease in enthalpy due to the increase in kinetic energy. Instead, if there were no heat losses, the expansion through the line would be at constant stagnation enthalpy. Consideration of this type of expansion results in more difficult calculations. In the case of isenthalpic expansion from saturated liquid at p_o, the properties of the fluid depend only upon the pressure to which the expansion has gone. For constant stagnation enthalpy expansion, on the other hand, the properties depend also upon the value of G. As a result, the properties must be calculated for each value of G considered. Furthermore, calculation of the properties becomes an iterative process, hence is much more laborious. A sample calculation for constant stagnation enthalpy is given in Appendix D.

The first calculations for constant stagnation enthalpy were for p_o = 1100 psia. W was taken as 3.20 lb/sec, which is equivalent to G = 1970 lb/sec-ft^2 through the 1/2-inch sch 80 section of the dump line and G = 641 lb/sec-ft^2 through the 1-inch sch 80 section. The Fanno lines for this expansion are shown in Figure 15.

Because the pressure was still 720 psia as the fluid left the 1/2-inch line, the specific volume and the velocity were not high enough in this section to cause very much deviation from constant enthalpy. At the end of the 1/2-inch line the calculations showed that $h^o - h$ = 0.4 Btu/lb and that v = 0.0744 ft^3/lb compared to 0.0748 ft^3/lb assuming

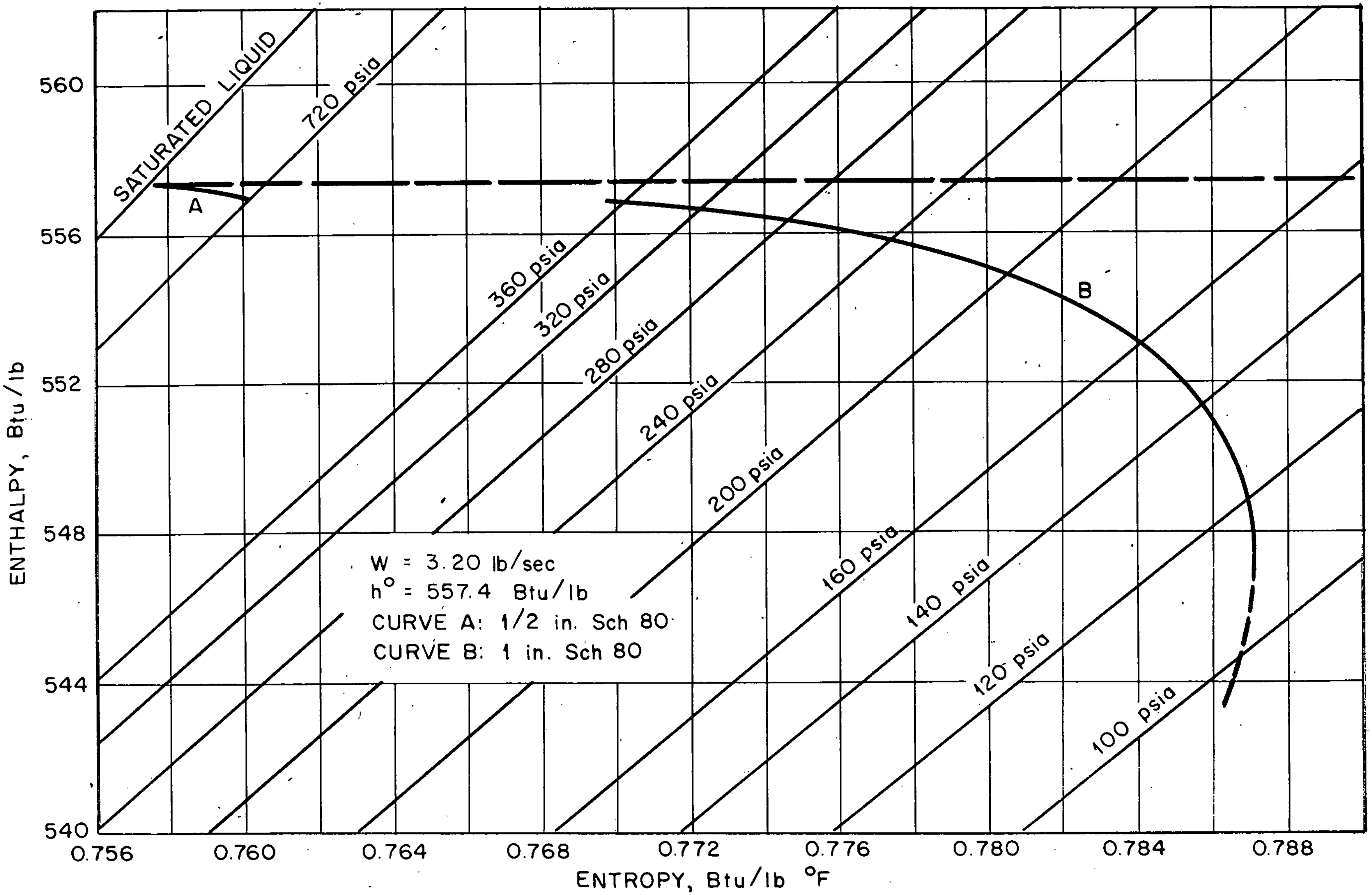

Fig. 15. Fanno Lines For Expansion From Saturated Liquid At 1100 psia

constant enthalpy. This was not enough difference to make any significant change in ln $\frac{v_2}{v_1}$ or $g_c \int_{p_2}^{p_1} \frac{dp}{v}$. Thus there was no change in the calculated value of f for the 1/2-inch line.

In the 1-inch section the expansion to low pressure produced large specific volumes and high velocities so that a greater difference appeared between the constant stagnation enthalpy properties and those assuming constant enthalpy. The effect on the calculated friction factor was not great however, as 4f only changed to 0.0190 from 0.0186.

Results obtained at a lower initial pressure were also analyzed on the basis of constant stagnation enthalpy flow. For p_o = 140 psia the flow rate, as taken from the slope of the dump tank level record, was 0.98 lb/sec. This is equivalent to G = 196 lb/sec-ft^2 in the 1-inch section of the dump line. The Fanno line for this mass velocity and h^o equal to the enthalpy of saturated liquid at 140 psia is curve A in Figure 16. On this curve the pressure at the point of maximum entropy, that is, the pressure which would be expected to exist at the end of the line, is about 24 psia. This is somewhat higher than the 16 psia observed, but agrees with the 24 psia calculated from equation [5] on the basis of isenthalpic expansion (Table I). Thus, there appears to be some question as to the accuracy of either the observed values of pressure or mass flow rate or the assumption of constant stagnation enthalpy. However, if it is assumed that the observed values of p_o, p_1, p_2, p_3 and W are correct and that the expansion is at a constant stagnation enthalpy, the friction factors calculated for the 1/2-inch and

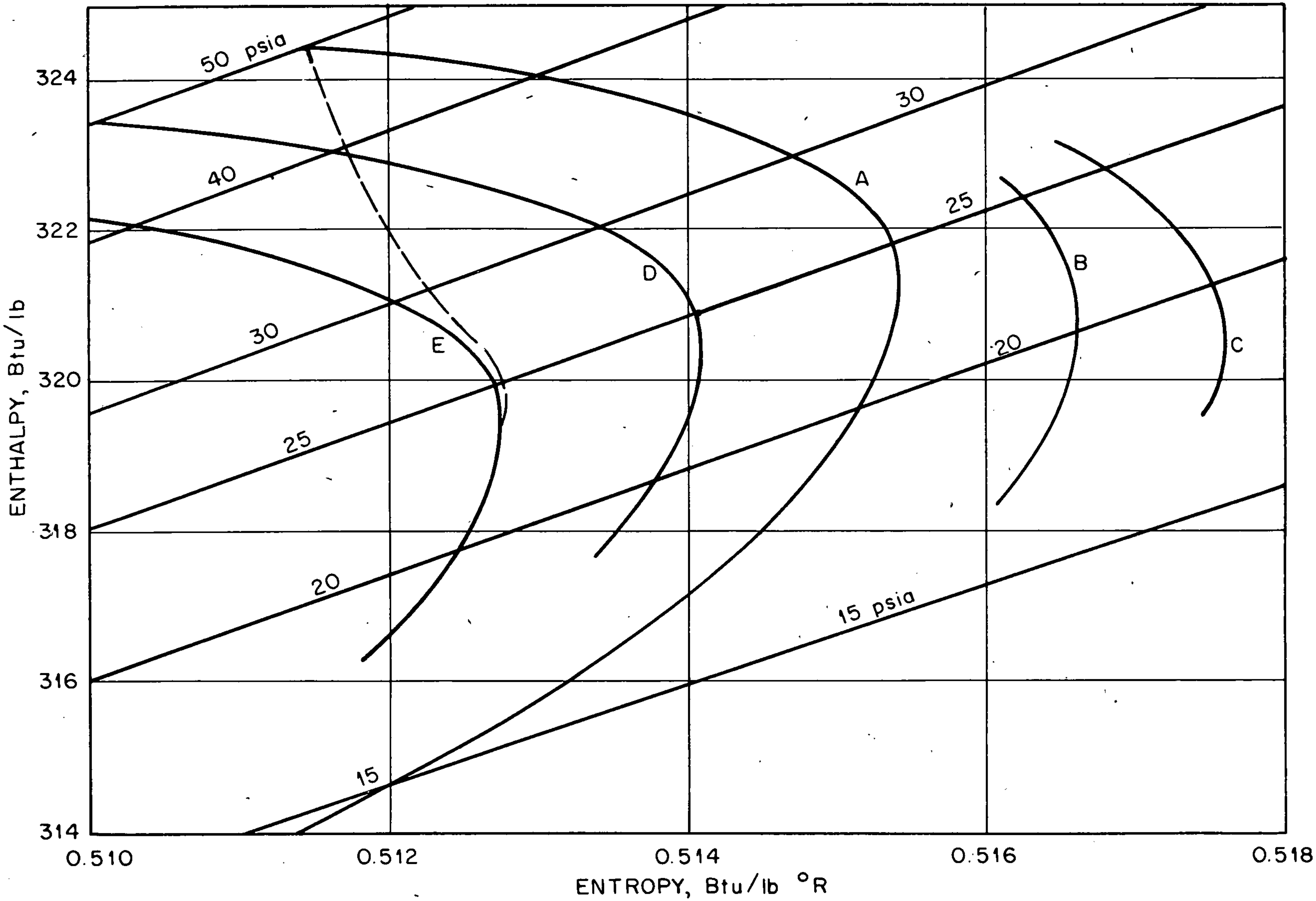

Fig. 16. Fanno Lines For Low-Pressure Expansions

1-inch sections are 0.0168 and 0.00645 respectively.

In an effort to resolve the discrepancy between the observed and calculated values of p_4, the Fanno line was calculated having the same stagnation enthalpy as before, but with $\dot{G}$ = 174 lb/sec-ft^2 or W = 0.87 lb/sec. This is shown as curve B in Figure 16. From this curve p_4 would be predicted as about 20 psia. The results of further reducing $\dot{G}$ to 160 lb/sec-ft^2 (W = 0.8 lb/sec) with h^o the same as before are shown by curve C in Figure 16. The predicted value of p_4 is reduced only to around 18 or 19 psia.

Although the Fanno-type expansion describes the flow through the dump line more accurately than the constant-enthalpy assumption, it is not exactly correct, either. Because the dump line was not insulated, there were heat losses which would cause the stagnation enthalpy of the fluid to decrease along the length of the line.

The rate of heat loss from the 1-inch section has been estimated for a time during a dump when the water entering the dump line was saturated liquid at 140 psia. For this condition, observed line pressures were p_3 = 50 psia and p_4 = 16 psia. For these pressures, the temperatures of the flashing mixture would be 281°F entering the 1-inch line and 216°F leaving. The principal resistance to heat transfer is the outside coefficient, so that the pipe should be at nearly the same temperature as the fluid. Because in flashing flow the pressure and temperature decrease slowly at first and more rapidly toward the end of the line, the average temperature difference between the pipe and the air was approximately 200°F. An equation for the heat transfer coefficient for transfer from a horizontal cylinder to air by natural convection is [8]

$$h_c = 0.27\left(\frac{\Delta T}{D_o}\right)^{0.25}$$

The surface area of the section of 1-inch line was 13.5 ft^2, giving a calculated rate of heat loss of 1.3 Btu/sec. Under the assumed conditions the mass flow rate was 0.98 lb/sec, so the decrease in stagnation enthalpy of the fluid while passing through the line was one or two Btu per pound.

Curve D in Figure 16 is the Fanno line for G = 196 lb/sec-ft^2 and h^o = 323.82 Btu/lb, one Btu/lb less than h_f at 140 psia. Curve E is the Fanno line for the same G, but for h^o = 322.82 Btu/lb. From the relative positions of curves A, D and E, the path on the h-s diagram of an expansion with continuous heat loss can be inferred, at least qualitatively. If the fluid entered the 1-inch line at 50 psia, with h^o = 324.82 Btu/lb and lost 2.0 Btu/lb while in the line, the path of the expansion would be somewhat as indicated by the dashed line. This curve represents expansion with both friction and heat losses.

In a constant stagnation enthalpy expansion the entropy changes are entirely internal. Therefore such an expansion can only move in the direction of increasing entropy. Hence the point of maximum entropy is the limiting point which corresponds to the condition at the end of the line. This is not necessarily the case, however, if there is a flow of heat from the system. If there is sufficient cooling, the entropy of the fluid could actually begin to decrease before the end of the line.

These considerations suggested the possibility that taking into account a decrease in stagnation enthalpy might result in a lower calculated critical pressure. To check on this, a calculation was made using specific volumes calculated for constant entropy. It was assumed that the fluid entered the 1-inch line at 50 psia and that the entropy throughout the line was 0.5114 Btu/lb ^{o}R. As can be seen from Figure 16, this would require the removal of heat at a rather high rate, since 2 Btu/lb would have to be removed by the time the fluid reached a pressure of about 36 psia. For a mass flow rate of 0.98 lb/sec, these constant-entropy calculations indicated a critical pressure of about 22 psia and a friction factor (4f) of 0.0067. These figures are quite close to those calculated on the basis of either constant enthalpy or constant stagnation enthalpy. This is true because at a given pressure there is not much difference between the specific volumes calculated by any of the assumptions.

As a result of the foregoing work, it appeared that the values of the friction factors were fairly well established. Efforts were next directed toward establishing a correlation between the friction factor and Reynolds number and/or the relative roughness of the pipe.

Reynolds numbers were calculated at each end of each section of pipe for two different flow rates. The results are shown in Table II. The Reynolds number is defined by

$$Re = \left(\frac{D\,G}{\mu}\right) = \left(\frac{D\,W}{A\,\mu}\right)$$

The viscosity used in the numbers in Table II is that of saturated

TABLE II

COMPARISON OF REYNOLDS NUMBERS
AND FRICTION FACTORS

Nominal Pipe Size, in.	1/2	1/2	1	1
Mass Flow Rate, lb/sec.	3.20	0.98	3.20	0.98
Reynolds Numbers x 10^{-6}				
Entering	6.4	2.7	4.4	1.8
Leaving	6.9	2.9	5.2	2.0
Average	6.6	2.8	4.8	1.9
Friction Factor (4f)	0.025	0.017	0.019	0.006

steam at the observed pressure.

The relative roughnesses of the 1-inch and 1/2-inch pipe were 0.0019 and 0.0033 respectively. This is based on an absolute roughness of 0.0018 inches, which is appropriate for commercial steel.

For comparison purposes, the Reynolds number and the relative roughness of the pipe were calculated for a typical run reported by Benjamin and Miller.[3] In this run, the mass flow rate was 25.3 lb/sec through a 4-inch sch 40 pipe, corresponding to G = 1.99 lb/sec-in^2. The saturation pressure was 42.8 psia, the pressure entering the line from the trap was 35.8 psia and the pressure at the end of the line was 27.4 psia. The friction factor was calculated by the authors to be 0.0119. The Reynolds number, based on the viscosity of steam, was approximately 1.1 x 10^7 throughout the pipe. The relative roughness of 4-inch sch 40 pipe is 0.00045.

The discussion up to this point has dealt entirely with the pressure drops and friction factors in the pipelines. However, in order to predict the total pressure drop through a drain line, entrance losses and pressure drop through the valve or throttling device must obviously be included in the calculation.

The line friction factors which have been obtained from this experiment can readily be applied to other drain lines having different dimensions. The pressure drop data for the line entrance and the valve do not have such general applicability, however. This is so because the pressure drops due to such resistances are strongly dependent on the geometry, which can vary widely in different designs.

The entrance losses observed in the experiment are rather high. As shown in Figure 5, the 1/2-inch pipe projects above the surface of the flange. This type of entrance results in considerably higher pressure drops than if the entrance were rounded or even if the opening were square, but flush with the wall of the container. Other resistances to flow, located between the vessel and the first pressure tap, result from the welds connecting the sections of pipe. It is quite possible that some weld metal projected inside the pipe, reducing the flow area and greatly increasing the head losses. It has not be possible to ascertain whether this is the case, since the equipment has not been dismantled at the time of writing.

Because it was desired to calculate flow rates through the existing dump line under conditions other than those which had already been observed, an empirical relation was derived which would give the pressure drop associated with the entrance section. The usual equation for the losses caused by a pipe entrance is

$$p_L = K_L \frac{\rho V^2}{2g_c}$$

where p_L is the head loss, expressed in terms of pressure, K_L is a coefficient which depends on the shape of the entrance, ρ is the density and V is the velocity in the pipe. The value of K_L for an entrance where a pipe projects into a reservoir is about 0.8 for non-flashing flow.[9] By fitting an equation of the type indicated above to the experimental data, a coefficient of 4.5 was determined for the entrance section. This was based on the density of saturated liquid and the velocity appropriate for this density. While the above equation fitted the data quite well, an empirical relation was derived which had the advantage of being simpler to apply and which fitted just as well. This relation is

$$_0\Delta p_1 = 7.8\ W^2$$

where $_0\Delta p_1$ is the pressure drop from the vessel to the first pressure tap in psi and W is the mass flow rate in lb/sec.

The problem of the flow of a flashing mixture through valves is a wide and involved field in itself. There appears to be no simple way of relating the capacity of valves for flashing fluids to the characteristics observed with liquids. For example, although angle style bodies have higher flow coefficients for liquids, their capacities are less than those for globe valves when there are large pressure drops and considerable flashing in the valve.[10] Because of the complexity of the problem, no attempt was made to draw any general conclusions from the observed pressure drops through the valve in the experimental dump line. A simple empirical relation for the pressure drop in terms of mass flow rate was obtained from the experimental data. This relation is

$$_2\Delta p_3 = 45\ W^{1.75}$$

where $_2\Delta p_3$ is the pressure drop through the valve in psi, and W is the mass flow rate in lb/sec.

Having determined friction factors for the lines and empirical relations for the pressure drops in the line entrance and through the valve, a procedure was developed whereby this information could be used to predict the effect of changes in the dimensions of the dump line.

The procedure consists essentially of solving the simultaneous equations for the pressure drops through the various components of the dump line by graphical means. A description of the method and a sample calculation are given in Appendix E.

In an application of this calculational procedure, the mass flow rate through the existing dump line was calculated as a function of p_o, using the line friction factors obtained at a p_o of 1100 psia. The results are indicated by the curve in Figure 14. Almost identical results were obtained by considering the entire resistance of the dump line to be equal to a 1-inch sch 80 pipe with $2\ fL/D = 145$. In both cases the mass flow rates predicted for low initial pressures are less than those actually observed.

CHAPTER VII

SUMMARY

The dump experiments reported here consisted of raising a quantity of water in a pressure vessel to high temperature and pressure and then opening a valve, allowing the water to rush through the drain line into a receiver which was held at low pressure. Instrumentation was provided to record pressures at various points along the drain line during the dump. The level of the water in the dump tank was also recorded so that mass flow rates could be established. Although the experiment was one in which the flow was unsteady, there were times during each dump when pressures and the flow rate were changing very slowly. This permitted analysis on the basis of steady flow.

By choosing times during the dumps when saturated liquid at a known pressure was entering the line, the specific volume of the flashing mixture could be calculated as a function of the pressure to which the fluid had expanded. In most of the calculations, the properties of fluid which had expanded at constant enthalpy were used. Calculations in which the stagnation enthalpy was assumed to remain constant showed that the specific volumes were very close to those for constant enthalpy expansion. Heat losses from the line were also shown to have little effect on the specific volume of the flashing mixture.

The range of initial pressures was such that measurements were taken with saturated liquid entering the line at pressures from 140 to 1250 psia. The observed mass flow rates during periods when the water in the pressure vessel was saturated liquid can be represented closely by

$$W = 0.055\ p_o^{0.58}$$

where W is the mass flow rate, in lb/sec and p_o is the pressure in the vessel in psia.

The resistance of the drain line to flow was divided into four parts for analysis: the line entrance, the 1/2-inch line preceding the valve, the valve, and the 1-inch line downstream from the valve. Each part was analyzed and flow coefficients or friction factors calculated.

The equation used to analyze the pressure drops through the sections of pipe is

$$G^2 = \frac{g_c \int_{p_2}^{p_1} \frac{dp}{v}}{\ln \frac{v_2}{v_1} + \frac{2fL}{D}}$$

This equation was developed from the dynamic and continuity relations for flow through a circular pipe of uniform diameter. The subscripts 1 and 2 as used here refer to the conditions at the entrance and discharge ends of the section respectively.

The friction factors calculated by the above equation were found to vary with pressure. Friction factors based on properties evaluated for constant enthalpy expansion are shown in Figure 17. There is little difference in these values and those calculated assuming other types of expansion. In the case of the 1/2-inch line, the maximum difference caused by the different assumptions as to the type of expansion was less than one percent. For the 1-inch line, consideration of the effect on specific volume of heat losses and changes in kinetic energy resulted

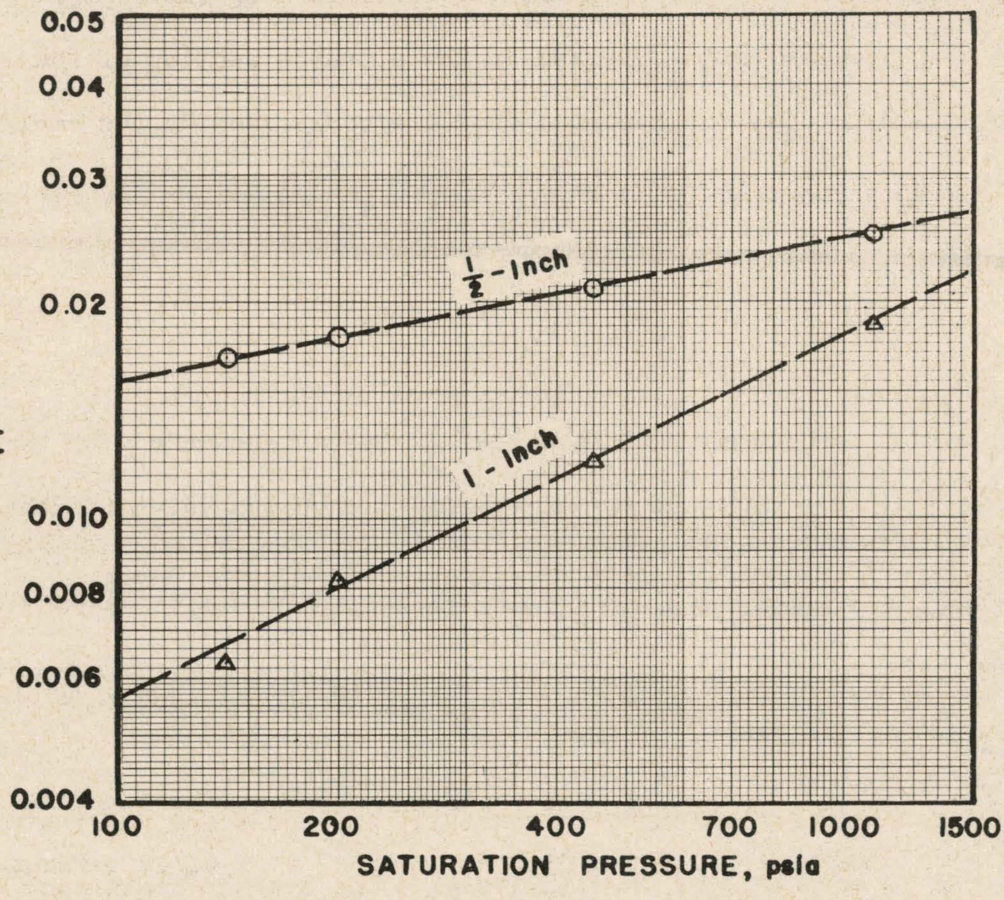

FIG. 17. Friction Factors

in calculated factors no more than six percent higher than those shown.

During a dump, the pressure in the dump tank was kept quite low by the use of condensing coils. The pressure in the tank was always low enough so that a critical pressure existed at the line discharge. The observed line discharge pressures compared fairly well with the critical pressures which were calculated for the observed mass flow rates.

The pressure drops observed in the entrance section of the line were quite high. The experimental data are correlated by the empirical relation

$$\Delta p = 7.8 W^2$$

where Δp is the pressure drop in the entrance section in psi, and W is the mass flow rate in lb/sec.

All of the dumps which have been analyzed were made with the valve fully opened. The observed pressure drop through the valve for these conditions can be represented by a simple empirical equation

$$\Delta p = 45 W^{1.75}$$

where Δp is the pressure drop through the valve in psi, and W is the mass flow rate in lb/sec.

A method was developed for combining the equations for the pressure drops in each part of the line to give the overall resistance. By the use of this calculational procedure, the effect of changes in the dimensions of the line can be predicted.

CHAPTER VIII

CONCLUSIONS

From the experimental results it appears that the friction factor for a pipe carrying a flashing mixture is a function of the pressure. The diameter of the pipe also seems to have an effect. The factor for the 1-inch section of the experimental line was always less than that for the 1/2-inch section. Both decreased with decreasing pressure.

No correlation was observed between the friction factor and the Reynolds number.

It would have been desirable to have run tests with lines of various dimensions, different type valves and different shape inlets. The primary purpose of the experiment was to test a particular drain line, however. Because all the data were taken with one arrangement, it was impossible to separate the effects of pressure, pipe size, quality of the mixture, etc. For example, the difference in the friction factors for the 1-inch and 1/2-inch sections may be due to: (a) the difference in diameters, (b) the lower pressures in the 1-inch line, or (c) the relative location of the two lines, the 1/2-inch line being above the valve where there was only a moderate amount of vapor, and the 1-inch line downstream where the specific volumes and velocities were much higher.

In general, the data obtained in the experiment appear to be reliable.

Several times during the experimental program, the pressure cells were dead-weight calibrated to within 0.25 per cent of the full scale rating, i.e., 5 psi for the 2000 psi cells and 2.5 psi for the 1000 psi cell at the line discharge. The calibration was not carried out before each run, however, so that the accuracy of the pressure cells was not always within these limits. The estimated maximum error due to the cells during any run is one percent. A larger error was involved in reading the Brush charts which recorded pressures. In order to give quick response, recorders were used with small charts, so that full scale was 40 mm or less. These charts could be read to within about 1.5 percent of full scale. The Swartwout chart recording the line discharge pressure was larger and could be read to within 2 psi. The cumulative effect of the errors in the pressure measurements is sufficient to explain the small discrepancies between the calculated and observed critical pressures.

It is difficult to assess the probable error in the dump tank level measurement. The differential pressure cell was accurate to within one percent. There was some error, however, due to the delay caused by the water condensing in the upper part of the dump tank and then running down the walls. Because the level was increasing rather uniformly, the mass flow rate, which was derived from the rate at which the level rose, was probably accurate to within about two percent.

The probable error in the experimental measurements is not enough to invalidate any of the conclusions.

Although the flow through the drain line was never truly steady, the data which were used in calculating friction factors was always taken at times when the pressures and flow rates were changing slowly enough so that the assumption of steady flow introduced no appreciable error.

In calculating the specific volume of a flashing mixture for use in determining flow rates and pressure drops, the assumption of constant enthalpy is usually sufficiently accurate. Heat losses and increases in kinetic energy cause the enthalpy of the fluid to decrease as it moves through the line, but in the experimental dump line, which was not insulated, the reduction in the enthalpy was not great enough to cause very much change in the calculated friction factors. The maximum difference between the friction factor calculated for constant enthalpy and that based on the estimated actual properties of the fluid was less than six percent. In most cases the difference was insignificant. Of course, if a cooler were used to remove large quantities of heat, the effects could not be ignored.

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APPENDIX A

EQUIVALENT LENGTH OF EXPERIMENTAL DUMP LINE

The friction losses suffered by a fluid passing through a line containing bends or other irregularities are greater than the losses for a straight, uniform pipe of the same length. In pressure drop calculations, a simple method of taking into account the added losses due to fittings, bends or valves is to represent the irregularity as an equivalent increase in the length of straight line. In the case of bends, experiments have shown the increase in equivalent length to be a function of the ratio of the radius of the bend to the pipe diameter.

In the section of 1/2-inch line between the pressure taps at the line entrance and at the valve entrance, there were two 45° bends and four 90° bends. The length of line, exclusive of the pipe in the 90° bends, was 23 feet. One of the 90° bends was on a radius of 3.5 inches; the other 90° bends and the 45° bends were on 2-inch radii. The equivalent length of this section of line was calculated as follows:

1/2-inch sch 80 pipe, D = 0.546 in.

<u>23 ft. of pipe</u>, $L/D = \frac{23 \times 12}{0.546} = \underline{\underline{506}}$

<u>one 90° bend</u>

$R = 3.5$ in., $R/D = 6.41$

equivalent $L/D = \underline{\underline{19}}$

<u>three 90° bends</u>

$R = 2$ in., $R/D = 3.66$

equivalent $L/D = 3 \times 13 = \underline{\underline{39}}$

two 45^{o} bends

assume equivalent to one 90^{o} bend

R = 2 in., R/D = 3.66

equivalent L/D = 6 (bend resistance only)

Total equivalent length

equivalent L/D = 506 + 19 + 39 + 6 = 570

The equivalent L/D for the bends were taken from Figure 20, Crane Technical Paper No. 409.[10] Because the length of pipe in the 90^{o} bends was not included in the 23 feet, the equivalent L/D for these bends is for the total resistance, which includes the resistance due to length and the bend resistance. The length of pipe in the 45^{o} bends was included in the straight pipe, so the equivalent L/D for these bends is only for the bend resistance.

The equivalent length of 1-inch pipe between pressure taps was calculated as follows:

1-inch sch 80 pipe, D = 0.957 in.

straight pipe

35 ft.-6 in., L/D $= \frac{35.5 \times 12}{0.957} =$ 445

eight 90^{o} bends

R = 3.5 in., R/D = 3.66

equivalent L/D = 8 x 13 = 104

Total equivalent length

equivalent L/D = 445 + 104 = 549

APPENDIX B

CALCULATION OF LINE FRICTION FACTORS

The following are sample calculations for determining the friction factors in the 1/2-inch line preceding the dump valve and in the 1-inch line after the valve.

Friction Factors for p_o = 1100 psia

Calculate f for 1/2-inch and 1-inch lines for the following conditions:

p_o =	1100 psia	p_3 =	366 psia
p_1 =	1020 psia	p_4 =	120 psia
p_2 =	710 psia	$\dot{W}$ =	3.20 lb/sec

Assume constant-enthalpy expansion with $h = h_{fo} = 557.4$ Btu/lb.

The friction factor in the 1/2-inch line is found from

$$G^2 = \frac{g_c \int_{p_2}^{p_1} \frac{dp}{v}}{\ln\left(\frac{V_2}{V_1}\right) + \frac{2f\,L}{D}}$$

From Appendix A, L/D = 570. From Appendix C, for isenthalpic expansion,

$$g_c \int_{710}^{1020} \frac{dp}{v} \doteq 1503 \ (\text{lb/sec-in}^2)^2$$

$$\ln\left(\frac{v_{710}}{v_{1020}}\right) = 0.9674$$

For 3.20 lb/sec through 1/2-inch sch 80 pipe ($A = 0.234$ in^2).

$$\bar{G}^2 = \left(\frac{3.20}{0.234}\right)^2 = 187 \text{ (lb/sec-in}^2)^2$$

Thus

$$\frac{2fL}{D} = \frac{1503}{187.0} - 0.9674 = 7.0700$$

$$4f = \frac{2 \times 7.0700}{570} = \underline{\underline{0.0248}}$$

The friction factor in the 1-inch line must be determined so that the maximum value of G is that observed. For a 1-inch sch 80 pipe passing 3.20 lb/sec,

$$\bar{G}^2 = \left(\frac{3.20}{0.719}\right)^2 = 19.81 \text{ (lb/sec-in}^2)^2$$

A friction factor must be chosen and $\bar{G}^2$ calculated for several values of p_4 so that a plot can be made to determine the maximum value of $\bar{G}^2$. As a first approximation to 2fL/D, calculate as described above using the observed values of G, p_3 and p_4.

$$g_c\int_{120}^{366} \frac{dp}{v} = 130.1 \text{ (lb/sec-in}^2)^2$$

$$\ln\left(\frac{v_{120}}{v_{366}}\right) = 1.4567$$

$$\frac{2fL}{D} = \frac{130.1}{19.81} - 1.4567 = 5.1107$$

Using this value, calculate G^2 for several values of p_4.

p_4	$g_c \int_{p_4}^{366} \frac{dp}{v}$	$\ln \frac{v_4}{v_{366}}$	G^2
140	125.4	1.2722	19.65
120	130.1	1.4567	19.81
115	131.1	1.5067	19.81
100	133.9	1.6702	19.75

When G^2 is plotted against p_4, the p_4 for the maximum G^2 is about 118 psia. In this instance it is not necessary to repeat the calculation, since the maximum calculated G^2 is near enough to the observed value. (If the maximum had been much above the observed value, a larger value of 2fL/D would have been chosen and the calculation repeated.) The equivalent L/D for this section of line is 549, so that

$$4f = \frac{2 \times 5.1107}{549} = \underline{\underline{0.0186}}$$

Friction Factors for p_o = 140 psia

Calculate 4f for the 1-inch line under the following conditions:

p_o	=	140 psia	p_4	=	16 psia
p_3	=	50 psia	W	=	0.98 lb/sec

For 0.98 lb/sec through a 1-inch sch 80 pipe

$$G^2 = 1.858 \ (\text{lb/sec-in}^2)^2$$

Constant Enthalpy

Assume constant enthalpy with

$$h = h_{fo} = 324.82 \text{ Btu/lb}$$

As a first approximation, calculate 2fL/D assuming W, p_3 and p_4 are all correct. For constant enthalpy

$$g_c \int_{16}^{50} \frac{dp}{v} = 5.92 \ (\text{lb/sec-in}^2)^2$$

$$\ln\left(\frac{v_{16}}{v_{50}}\right) = 1.633$$

So that

$$\frac{2fL}{D} = \frac{5.92}{1.858} - 1.633 = 1.553$$

Using this value, calculate G^2 for several values of p_4.

p_4	$g_c \int_{p_4}^{50} \frac{dp}{v}$	$\ln\left(\frac{v_4}{v_{366}}\right)$	G^2
40	2.68	0.3640	1.398
30	4.50	0.7926	1.918
25	5.14	1.0473	1.977
20	5.63	1.3459	1.942
15	5.97	1.7139	1.827

This is plotted as Curve A in Figure 18. According to this curve, the pressure at the end of the line would be about 25 psia and the mass flow rate would be 1.01 lb/sec (G^2 = 1.977 $(\text{lb/sec-in}^2)^2$).

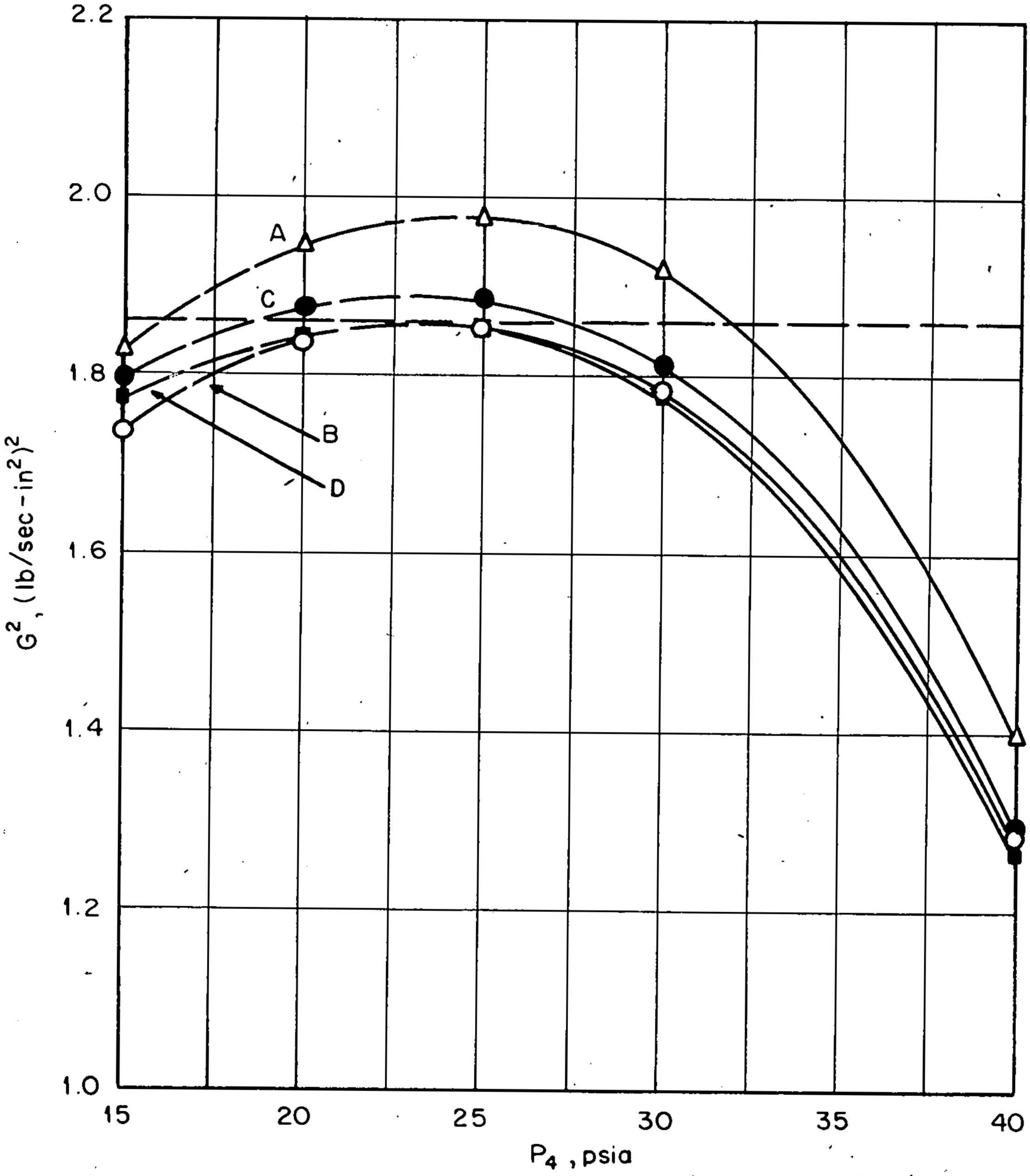

Fig. 18. Calculated Mass Velocities In 1-inch Line

The line resistance which would give the observed mass flow rate at the maximum point on the curve of $\bar{G}^2$ vs. p_4 is found by trial and error to be $2fL/D = 1.728$. This corresponds to

$$4f = \frac{2 \times 1.728}{549} = 0.00630$$

$\bar{G}^2$ vs. p_4 for this resistance is shown as curve B in Figure 18.

Constant Stagnation Enthalpy

Assume that the stagnation enthalpy in the 1-inch section is constant at

$$h^o = h_{fo} = 324.82 \text{ Btu/lb}$$

The properties of the fluid for this h^o and $W = 0.98$ lb/sec are given in Appendix D. Use these properties to calculate the friction factor for the 1-inch line which would give $W = 0.98$ when there is a critical pressure at the end of the line.

First try $2fL/D = 1.728$, the value determined on the basis of constant enthalpy. Results of calculations using this resistance are indicated by curve C, Figure 18. The maximum flow rate is too high, so a greater resistance must be tried. The value which gives the desired flow rate is $2fL/D = 1.780$ or $4f = 0.00648$. For this value, $\bar{G}^2$ as a function of p_4 is shown by curve D, Figure 18.

APPENDIX C

PROPERTIES OF WATER EXPANDED ISENTHALPICALLY FROM SATURATED LIQUID

The tables on the following pages give some of the properties of water which has been expanded at constant enthalpy from saturated liquid at various pressures. The equations used in computing the specific volume at a given pressure are

$$h = h_{fo}$$

$$v = v_f + \frac{h - h_f}{h_{fg}} v_{fg}$$

The properties which were used in these equations were taken from the 1936 edition of the steam tables by Keenan and Keyes.[12]

TABLE III

PROPERTIES OF WATER EXPANDED ISENTHALPICALLY
FROM SATURATED LIQUID AT 1200 PSIA

p psia	$10^3\ v$ ft^3/lb	$\ln \frac{v}{v_o}$	$g_c \int_p^{p_o} \frac{dp}{v}$ (lb/sec in^2)^2
1200	22.3	0.0000	0
1150	26.19	0.1608	463
1100	30.58	0.3157	859
1050	35.36	0.4609	1200
1000	40.73	0.6023	1495
960	45.44	0.7117	1703
920	50.75	0.8222	1889
880	56.58	0.9310	2056
840	62.98	1.0381	2205
800	70.20	1.1467	2340
760	78.19	1.2545	2461
720	87.27	1.3644	2569
680	97.50	1.4752	2666
640	109.3	1.5890	2753
600	122.7	1.7050	2830
560	138.3	1.8247	2899
520	156.5	1.9483	2959

TABLE III

PROPERTIES OF WATER EXPANDED ISENTHALPICALLY
FROM SATURATED LIQUID AT 1200 PSIA
(CONTINUED)

p psia	10^3 v ft^3/lb	$\ln \frac{v}{v_o}$	$g_c \int_p^{p_o} \frac{dp}{v}$ (lb/sec in^2)^2
480	178.1	2.0777	3013
440	204.0	2.2134	3060
400	235.4	2.3566	3101
360	274.4	2.5099	3136
320	324.0	2.6761	3166
280	388.6	2.8579	3191
260	428.9	2.9566	3202
240	476.2	3.0612	3212
220	532.6	3.1731	3221
200	600.9	3.2957	3229
180	685.1	3.4249	3236
160	790.9	3.5686	3242
140	928.5	3.7289	3247
120	1114	3.9106	3252
100	1375	4.1216	3255
80	1771	4.3747	3258

TABLE IV

PROPERTIES OF WATER EXPANDED ISENTHALPICALLY
FROM SATURATED LIQUID AT 1100 PSIA

p psia	10^3 v ft^3/lb	$\ln \frac{v}{v_o}$	$g_c \int_p^{p_o} \frac{dp}{v}$ (lb/sec-in^2)^2
1100	22.0	0	0
1050	26.43	0.1835	465
1000	31.39	0.3554	854
960	35.75	0.4855	1121
920	40.70	0.6152	1355
880	46.13	0.7404	1562
840	52.08	0.8618	1745
800	58.83	0.9836	1906
760	66.29	1.1030	2050
720	74.79	1.2236	2177
680	84.37	1.3442	2289
640	95.41	1.4671	2389
600	108.0	1.5911	2478
560	122.8	1.7195	2555
520	139.8	1.8492	2623
480	160.3	1.9860	2683
440	184.7	2.1277	2736
400	214.5	2.2773	2780

TABLE IV

PROPERTIES OF WATER EXPANDED ISENTHALPICALLY
FROM SATURATED LIQUID AT 1100 PSIA
(CONTINUED)

p psia	10^3 v ft^3/lb	$\ln \frac{v}{v_o}$	$g_c \int_p^{p_o} \frac{dp}{v}$ $(lb/sec\text{-}in^2)^2$
360	251.5	2.4364	2819
320	298.5	2.6078	2851
280	360.0	2.7951	2879
240	443.4	3.0035	2901
200	562.4	3.2412	2919
180	642.9	3.3750	2926
160	744.0	3.5210	2933
140	875.8	3.6841	2938
120	1053.1	3.8685	2943
100	1304.1	4.0822	2947
80	1684.5	4.3381	2950

TABLE V

PROPERTIES OF WATER EXPANDED ISENTHALPICALLY
FROM SATURATED LIQUID AT 1000 PSIA

p	10^3 v	$\ln \frac{v}{v_o}$	$g_c \int_p^{p_o} \frac{dp}{v}$
psia	ft^3/lb		$(lb/sec\ in^2)^2$
1000	21.6	0	0
980	23.56	0.0868	198
960	25.60	0.1699	380
940	27.88	0.2552	548
920	30.16	0.3338	702
900	32.53	0.4095	844
880	35.16	0.4872	977
840	40.65	0.6323	1213
800	46.9	0.7752	1418
760	53.8	0.9127	1596
720	61.7	1.0495	1751
680	70.6	1.1845	1887
640	80.9	1.3204	2005
600	92.6	1.4556	2108
560	106.4	1.5945	2199
520	122.4	1.7347	2277

TABLE V

PROPERTIES OF WATER EXPANDED ISENTHALPICALLY
FROM SATURATED LIQUID AT 1000 PSIA
(CONTINUED)

p psia	10^3 v ft^3/lb	$\ln \frac{v}{v_o}$	$g_c \int_p^{p_o} \frac{dp}{v}$ $(lb/sec\ in^2)^2$
480	141.6	1.8804	2345
440	164.6	2.0309	2404
400	192.5	2.1875	2454
360	227.4	2.3541	2497
320	271.8	2.5325	2533
280	330.0	2.7265	2563
240	409.0	2.9411	2587
200	522.0	3.1851	2607
180	598.5	3.3067	2615
160	694.9	3.4712	2622
140	820.5	3.6373	2627
120	989.7	3.8248	2632
100	1230	4.0418	2637
80	1594	4.3012	2640

TABLE VI

PROPERTIES OF WATER EXPANDED ISENTHALPICALLY
FROM SATURATED LIQUID AT 700 PSIA

p psia	10^3 v ft^3/lb	$\ln \frac{v}{v_o}$	$g_c \int_p^{p_o} \frac{dp}{v}$ $(lb/sec\text{-}in^2)^2$
700	20.5	0	0
680	23.89	0.1531	202
660	27.65	0.2992	377
640	31.62	0.4333	528
600	40.49	0.6806	778
560	51.03	0.9120	975
520	63.26	1.1268	1133
480	78.07	1.3372	1261
440	96.06	1.5445	1365
400	118.07	1.7509	1449
360	145.71	1.9612	1517
320	181.18	2.1791	1571
280	228.06	2.4092	1616
240	292.33	2.6825	1650

TABLE VI

PROPERTIES OF WATER EXPANDED ISENTHALPICALLY
FROM SATURATED LIQUID AT 700 PSIA
(CONTINUED)

p psia	10^3 v ft^3/lb	$\ln \frac{v}{v_o}$	$g_c \int_p^{p_o} \frac{dp}{v}$ $(lb/sec\text{-}in^2)^2$
200	384.97	2.9328	1677
180	448.12	3.0846	1688
160	528.03	3.2487	1697
140	632.75	3.4295	1705
120	774.6	3.6319	1711
100	976.8	3.8639	1717
80	1285.6	4.1386	1721
60	1811.0	4.4812	1724

TABLE VII

PROPERTIES OF WATER EXPANDED ISENTHALPICALLY
FROM SATURATED LIQUID AT 450 PSIA

p psia	$10^3 v$ ft^3/lb	$\ln \frac{v}{v_o}$	$g_c \int_p^{p_o} \frac{dp}{v}$ $(lb/sec\text{-}in^2)^2$
450	19.5	0	0
440	23.00	0.1651	106
420	30.33	0.4417	276
400	38.61	0.6830	407
380	47.95	0.8997	511
360	58.55	1.0994	595
340	70.63	1.2870	665
320	84.52	1.4666	723
300	100.60	1.6407	772
280	119.34	1.8115	812
260	141.44	1.9814	847
240	167.82	2.1525	876
220	199.66	2.3262	900
200	238.76	2.505	921
180	287.6	2.691	938
160	350.1	2.888	952
140	432.5	3.099	963.7

TABLE VII

PROPERTIES OF WATER EXPANDED ISENTHALPICALLY FROM SATURATED LIQUID AT 450 PSIA (CONTINUED)

p psia	10^3 v ft^3/lb	$\ln \frac{v}{v_o}$	$g_c \int_p^{p_o} \frac{dp}{v}$ (lb/sec-in^2)^2
120	545.1	3.330	972.9
100	707.1	3.590	980.2
80	956.9	3.893	985.7
70	1139.4	4.068	987.8
60	1385.3	4.263	989.6
50	1738.1	4.490	991.1
40	2275.3	4.760	992.2
30	3188.5	5.097	993.0

TABLE VIII

PROPERTIES OF WATER EXPANDED ISENTHALPICALLY
FROM SATURATED LIQUID AT 200 PSIA

p	10^3 v	$\ln \frac{v}{v_o}$	$g_c \int_p^{p_o} \frac{dp}{v}$
psia	ft^3/lb		$(lb/sec\text{-}in^2)^2$
200	18.39	0	0
196	23.05	0.2256	43.2
192	28.50	0.4383	78.3
188	33.96	0.6130	107.2
180	45.84	0.9131	152.5
170	62.57	1.2244	194.7
160	81.83	1.4927	226.2
150	104.33	1.7356	250.6
140	130.65	1.9605	269.8
130	161.87	2.1749	285.2
120	199.27	2.3827	297.7
110	244.66	2.5879	307.9
100	300.63	2.7939	316.2
90	370.98	3.0043	322.9
80	461.50	3.2226	328.3
70	581.53	3.4537	332.65

TABLE VIII

PROPERTIES OF WATER EXPANDED ISENTHALPICALLY
FROM SATURATED LIQUID AT 200 PSIA
(CONTINUED)

p psia	10^3 v ft^3/lb	$\ln \frac{v}{v_o}$	$g_c \int_p^{p_o} \frac{dp}{v}$ (lb/sec-in^2)^2
60	746.58	3.7036	336.07
50	985.41	3.9811	338.70
40	1356.6	4.3008	340.66
30	2000.0	4.6890	342.04
25	2530.3	4.9242	342.54
20	3345.1	5.2033	342.93

TABLE IX

PROPERTIES OF WATER EXPANDED ISENTHALPICALLY
FROM SATURATED LIQUID AT 140 PSIA

p psia	$10^3 v$ ft^3/lb	$\ln \frac{v}{v_o}$	$g_c \int_p^{p_o} \frac{dp}{v}$ (lb/sec-in^2)^2
140	18.02	0	0
135	29.31	0.4864	50.0
130	41.62	0.8370	84.2
120	70.21	1.3600	127.38
110	105.27	1.7650	153.90
100	148.95	2.1121	172.02
90	204.46	2.4288	184.98
80	276.64	2.7312	184.47
70	373.36	3.0310	201.48
60	507.8	3.3386	206.67
50	704.5	3.6660	210.44
40	1013.8	4.0300	213.12
30	1556.4	4.4586	214.94
25	2007.9	4.7133	215.58
20	2706.6	5.0119	216.07
15	3910.4	5.3799	216.41
10	6415.8	5.8750	216.64

APPENDIX D

CALCULATION OF PROPERTIES FOR CONSTANT STAGNATION ENTHALPY

In fluid flow with no heat transfer and no external work, the stagnation enthalpy remains constant. For cases where changes in elevation are negligible, this condition can be written as

$$h + \frac{G^2}{2g_c J} v^2 = h^o$$

Described below are calculations for finding h and v for a mixture of liquid and vapor when p, h^o and G are known quantities.

During run 12, at a time when the pressure in the pressure vessel was 140 psia, the rate of increase of the mass in the dump tank was indicated as 0.98 lb/sec. Calculate the properties of the fluid as it passed through the 1-inch sch 80 section of the dump line, assuming that the stagnation enthalpy remains constant throughout the entire line. The pressure at the beginning of the 1-inch section was 50 psia.

The flow area of 1-inch sch 80 pipe is 0.719 in^2. Thus

$$G = \frac{0.98 \times 144}{0.719} = 196.3 \text{ lb/sec-ft}^2$$

$$V = Gv = 196.3v \text{ ft/sec}$$

$$2g_cJ = 2 \times 32.174 \frac{lb_m\text{-}ft}{lb_f\text{-}sec^2} \times 778.16 \frac{ft\text{-}lb_f}{Btu}$$

$$= 5.007 \times 10^4 \frac{ft^2/sec^2}{Btu/lb_m}$$

$$\frac{V^2}{2g_cJ} = \frac{(196.3v)^2}{5.007 \times 10^4} = \frac{v^2}{1.2994} \quad \frac{\text{Btu}}{\text{lb}}$$

Also

$$h^o = h_f(140 \text{ psia}) = 324.82 \text{ Btu/lb}$$

Therefore

$$h + \frac{v^2}{1.2994} = 324.82$$

This is the working equation.

Calculate properties at 50 psia. As a first approximation to v, take the value at p = 50 psia and h = 324.82. For these conditions

$$x = (h - h_f)/h_{fg} = 0.0809$$

$$v = v_f + xv_{fg} = 0.7045 \text{ ft}^3/\text{lb}$$

$$\frac{V^2}{2g_cJ} = \frac{v^2}{1.2994} = 0.38 \text{ Btu/lb}$$

Recalculate properties for $h = h^o - 0.38 = 324.44$ Btu/lb

$$x = 0.0805$$

$$v = 0.7010 \text{ ft}^3/\text{lb}$$

$$V^2/(2g_cJ) = 0.38 \text{ Btu/lb}$$

Thus no further approximations are required. Other quantities which are of interest can now be calculated.

$$V = 196.3v = 138 \text{ ft/sec}$$

$$s = s_f + xs_{fg} = 0.5114 \text{ Btu/lb-}^o\text{R}$$

$$1/v = 1.426 \text{ lb/ft}^3$$

The same procedure must be repeated at each pressure. At lower pressures where the kinetic energy is greater, several approximations are usually required. Results for several pressures are given in Table X. An h-s diagram of the expansion is curve A, Figure 16.

TABLE X

PROPERTIES OF WATER EXPANDED AT CONSTANT STAGNATION ENTHALPY

$h^o = 324.82$ Btu/lb

$G = 196.3$ lb/sec-ft^2

p psia	h Btu/lb	v ft^3/lb	V ft/sec	x	s Btu/lb °R	$g_c\int_p^{50} \frac{dp}{v}$ (lb/sec in^2)^2	$\ln\left(\frac{v}{v_{50}}\right)$
50	324.44	0.7010	138	0.0805	0.5114	0	0
40	324.04	1.0051	197	0.0943	0.5130	2.705	0.3600
30	323.02	1.5303	301	0.1102	0.5147	4.546	0.7803
25	321.87	1.9575	384	0.1192	0.5154	5.196	1.0265
23	321.16	2.1808	428	0.1229	0.5154	5.417	1.1350
20	319.62	2.5979	510	0.1285	0.5150	5.696	1.3096
18	318.13	2.9496	580	0.1324	0.5145	5.854	1.4369
15	314.65	3.6349	714	0.1377	0.5120	6.065	1.6455

APPENDIX E

PROCEDURE FOR CALCULATING FLOW RATES

The expressions relating flow rate to the pressure drops through the various sections of the experimental dump line can be combined to predict the flow rate when the initial pressure is known. The effect of changes in the dimensions of the lines preceding and following the valve can be determined. The procedure followed in making such calculations is illustrated by the example below.

As shown in Figure 6, the dump line pursues a tortuous path from the valve to the dump tank. It is desired to calculate the effect of shortening this section of the dump line by taking a more direct route from the valve to the tank. As installed, the equivalent L/D of the 1-inch line was 549. This can be reduced to 334 by eliminating three 90° bends and 14 feet of straight pipe. Calculate W through the shortened line when saturated liquid is entering the dump line at 1100 psia. Assume that the dump tank pressure is kept low enough so that a critical pressure exists at the end of the line. Also assume that the specific volume at any pressure is adequately represented by the value for constant enthalpy expansion.

The pressure drop in the line entrance is given by

$$_{o}\Delta p_1 = 7.8\ W^2$$

Thus for p_o = 1100 psia,

$$p_1 = 1100 - 7.8\ W^2$$

This relation is plotted in Figure 19.

The equation for the flow through the 1/2-inch line is

$$G^2 = \frac{g_c \int_{p_2}^{p_1} \frac{dp}{v}}{\ln \frac{v_2}{v_1} + \frac{2fL}{D}}$$

For this section of the line L/D = 570 and 4f = 0.0248 so 2fL/D = 7.068. With the information in Table IV, the above equation is used to compute G^2 for several values of p_2 when p_1 is 1010, 1015 and 1020 psia. W is given by $W = A\sqrt{G^2}$. The results of these computations are indicated by the three curves in the lower part of Figure 19. These three curves show how W varies with p_2 for a fixed p_1. But p_1 and W are related by the expression for the entrance pressure drop, indicated by the curve at the top of the figure. The values of W corresponding to the chosen values of p_1 are indicated by the vertical dashed lines. Thus a curve through the intersection of these dashed lines with the lower curves gives p_2 as a function of W.

The pressure drop through the valve is given by

$$_2\Delta p_3 = 45\ W^{1.75}$$

A curve of p_3 as a function of W is obtained by subtracting the pressure drop, calculated by this relation, from the value of p_2 taken from the curve of p_2 vs. W. The result is the steep curve in the right half of Figure 20.

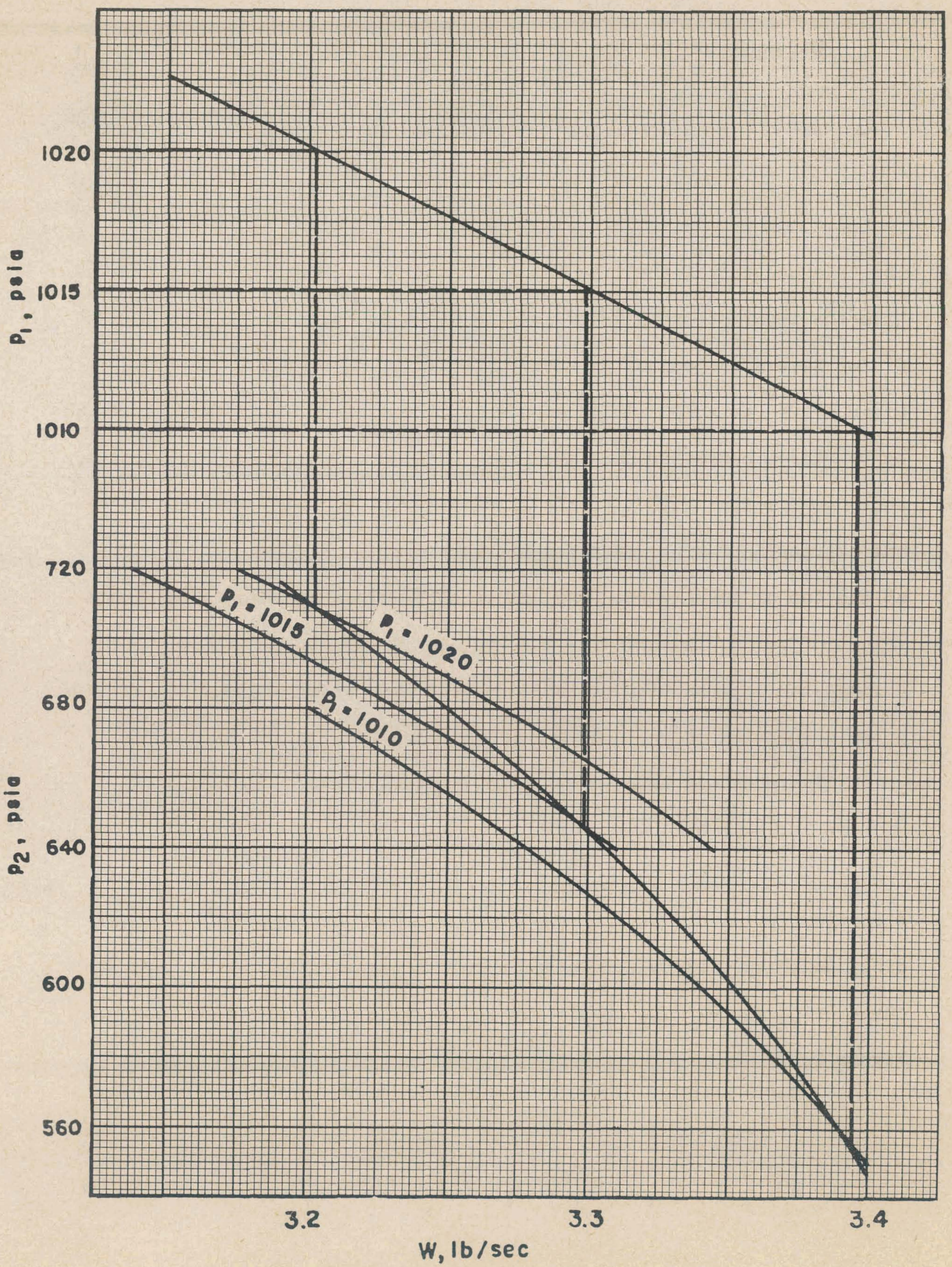

Fig. 19. Graphical Solution of Equations

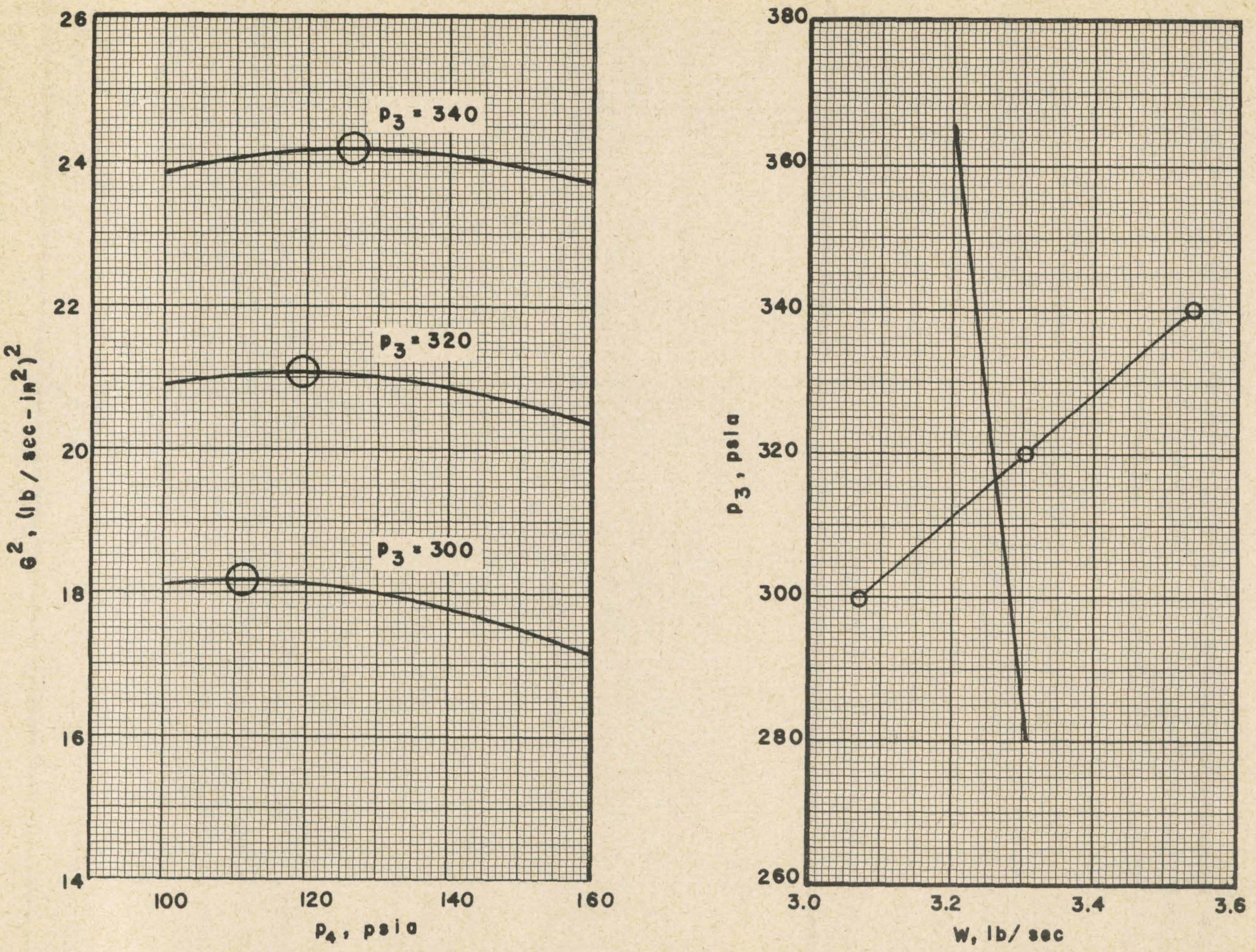

Fig. 20. Graphical Solution of Equations

The equation for the 1-inch line is

$$G^2 = \frac{\int_{p_4}^{p_3} \frac{dp}{v}}{\ln \frac{v_4}{v_3} + \frac{2fL}{D}}$$

Three reasonable values for p_3 are chosen and G^2 calculated for several values of p_4. The results are indicated by the three curves on the left side of Figure 20. It was postulated that a critical pressure existed at the end of the line. Therefore, for a given p_3, G^2 is the maximum given by the curve. From these maxima, W is computed for each value of p_3. Results are indicated by the points and line in the right half of Figure 20. The intersection of the two curves gives the desired value of W. This value can be used with the curves already drawn to determine values of the pressures. The results for the shortened line are compared below with the original dump line.

	Original Line	Shortened Line
p_o, psia	1100	1100
p_1, psia	1020	1017
p_2, psia	710	666
p_3, psia	366	316
p_4, psia	115	118
W, lb/sec	3.20	3.26

NOMENCLATURE

A	flow area
D	inside diameter
D_o	outside diameter
f	friction factor
G	mass velocity
g_c	gravitational conversion factor, $32.174 \frac{lb_m}{lb_f} \frac{ft}{sec^2}$
h	enthalpy
h^o	stagnation enthalpy
h_c	natural convection heat transfer coefficient
J	conversion factor, 778.16 ft-lb_f/Btu
K_L	loss coefficient
L	length
p	pressure
p_L	head loss
R	radius of bend
s	entropy
V	velocity
v	specific volume
W	mass flow rate
ΔT	temperature difference between cylinder and air
ρ	density
μ	dynamic viscosity

NOMENCLATURE
(CONTINUED)

Subscripts

o	pressure vessel
1	line entrance
2	valve entrance
3	valve discharge
4	line discharge
f	saturated liquid
fg	vaporization