E. Casado

The Flow of a Flashing Mixture of Water and Steam Through Pipes

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This paper presents a method of designing piping to carry a flashing mixture of water and steam. An example of such piping is the cascade drain lines between feedwater heaters. The flow formula is based on a thermodynamic analysis of the problem, and the necessary coefficients of friction have been determined from tests. Calculations indicate the possibility of a critical-pressure condition at the end of a pipe carrying a flashing mixture. Each of the drain lines included in the study exhibited this phenomenon. The paper also includes a discussion of erosion of elbows in pipe lines carrying a flashing mixture, and suggests designs for minimizing or preventing failures from this cause. A criterion for predicting the comparative life of different sizes of pipes for a given line carrying a flashing mixture is set up on the basis of the total force on the elbows due to the momentum of the flowing fluid. An example of the application of the proposed flow formula to an actual design case is given in the Appendix.

During recent years power-plant engineers have become more and more interested in the flow of a flashing mixture of water and steam through pipes in connection with the design of cascade drain lines between extraction feedwater heaters. Several such lines, designed by the rule-of-thumb methods that have been used heretofore either have caused trouble with erosion in the elbows or have been unnecessarily large and expensive.

The case of fluid flow, involving a flashing mixture of water and steam, can readily be analyzed on the basis of thermodynamic equilibrium. However, to derive a usable formula for design purposes, it was necessary to obtain test data from which coefficients of friction could be determined. Since it was thought such tests might well include the friction effect of elbows, valves, and tees, it was decided to conduct an experimental investigation using existing cascade drain lines, particularly those in which trouble with erosion was being experienced.

While the data presented in this paper are not complete, sufficient information is given to be of assistance in the design of pipe lines to carry a flashing mixture of water and steam. An analysis is made of the flow in several cascade drain lines in order to learn why some have failed because of erosion while others have not. Several other factors of particular interest in designing pipe lines to carry water and steam are also discussed.

More complete results could have been obtained by means of special equipment, but the extra cost was not considered warranted by the amount of data which could be obtained by such equipment over that which could be obtained from tests of the existing drain lines.

In this paper the term "flashing mixture of water and steam" is used in preference to "boiling water" as used by earlier writers since it is considered to be more definite and possibly more accurate. It is used to denote a mixture of saturated water and steam in which additional steam is continually being formed at the expense of the sensible heat in the water, made available as a result of the continuing reduction in pressure as the mixture flows down the pipe.

Flow of a Flashing Mixture of Water and Steam Through Pipes

When saturated water flows from a receiver at one pressure through a throttling valve (or an orifice) and pipe to a receiver at a lower pressure, the following changes take place:

(*a*) As the pressure decreases the saturation temperature also decreases, and the enthalpy of that part of the fluid that remains liquid water is reduced in proportion to the drop in temperature.

(*b*) The heat liberated by the reduction in enthalpy of the water is all absorbed as latent heat in evaporating part of the water.

(*c*) The specific volume of the mixture of water and steam increases rapidly as steam is produced.

(*d*) The energy which becomes available with the decrease in pressure is expended in accelerating the mixture of water and steam and thus increasing its kinetic energy.

For example, if saturated water flows from receiver *A*, Fig. 1, through the throttling valve and the pipe into the lower-pressure

Fig. 1 Schematic Diagram Illustrating Flow of Flashing Mixture of Steam and Water Through a Pipe Connecting High- and Low-Pressure Receivers

receiver *B*, some of the water will flash into steam immediately following the valve at point 1, and an increasing amount of water will flash into steam as the mixture flows toward receiver *B*. The amount of steam flashed at point 1 in the pipe depends upon the initial temperature in *A* and the pressure P_1. Pressures P_1 and P_2 (at points 1 and 2) are functions of the initial saturation temperature in *A*, the weight of mixture flowing through the pipe, the size and length of the pipe, and in some cases the pressure in *B*.

Equations of Flow. According to the continuity equation

$$V = \frac{wv}{A} \qquad [1]$$

where V is the velocity in ft per sec; v the specific volume in cu ft per lb; w the weight flowing in lb per sec; and A the cross-sectional area of the pipe in sq ft.

Since w/A is constant for flow through a pipe, the velocity and, therefore, the kinetic energy at any point along the pipe during expansion depend upon the specific volume of the mix-

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Contributed by the Power Division and presented at the Annual Meeting, New York, N. Y., December 1–5, 1941, of The American Society of Mechanical Engineers.

ture. The rate of increase of the velocity and kinetic energy during the expansion is a function of the rate at which the specific volume increases. Fig. 2 shows an example of the increase in specific volume of a flashing mixture of water and steam for an isentropic expansion from an initial pressure of 15 psi abs. This curve shows that, if saturated water flowing through a pipe expands from 15 psi abs to 3 psi abs, the velocity at the end of the pipe, due to the increase in specific volume of the flashing mixture, would be 175 times the velocity of the water at the entrance.

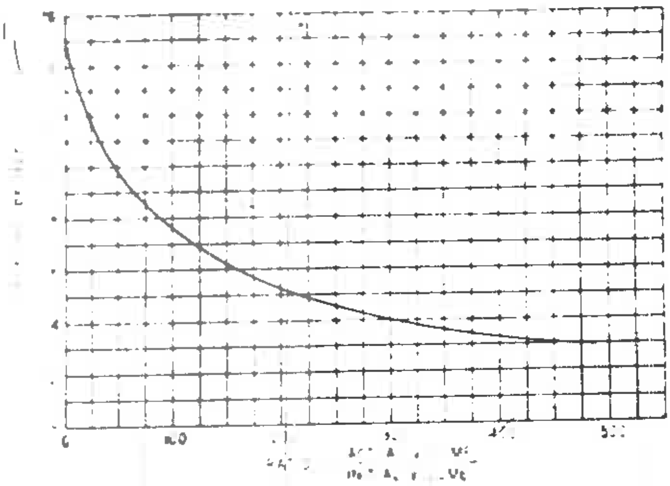

FIG. 2 INCREASE IN VOLUME OF FLASHING MIXTURE OF WATER AND STEAM DURING ISENTROPIC EXPANSION FROM SATURATED WATER AT 15 PSI ABS

A mixture of saturated water and steam in which additional steam is continually being flashed is not strictly an elastic fluid, although it is believed to behave in much the same way and will, therefore, be treated accordingly. The general energy equation for the flow of fluids can be expressed in the form

$$v\,dP + \frac{V\,dV}{g} + \frac{KV^2}{2gD}\,dx = 0 \quad \ldots\ldots\ldots [2]$$

in which v is the specific volume in cu ft per lb; P the pressure in lb per sq ft; V the velocity in ft per sec; D the diameter of the pipe in feet; K the friction factor; and x the distance in ft through which the mixture flows. This equation may be derived either from the energy relation as given by Goodenough[3] or by using Bernoulli's theorem.

From Equation [1] $(w/A)v$ can be substituted for V in Equation [2] to give

$$v\,dP + \left(\frac{w}{A}\right)^2 \frac{v\,dv}{g} + \frac{K}{D}\left(\frac{w}{A}\right)^2 \frac{v^2}{2g}\,dx = 0$$

or

$$\frac{1}{v}\,dP + \left(\frac{w}{A}\right)^2 \frac{1}{g}\frac{dv}{v} + \frac{K}{D}\left(\frac{w}{A}\right)^2 \frac{1}{2g}\,dx = 0$$

substituting ρ for $1/v$ and integrating

$$\int_{P_1}^{P_2} \rho\,dP - \left(\frac{w}{A}\right)^2 \frac{1}{g}\int_{\rho_1}^{\rho_2}\frac{d\rho}{\rho} + \left(\frac{w}{A}\right)^2 \frac{1}{2g}\frac{K}{D}\int_0^L dx = 0$$

[3] "Principles of Thermodynamics," by G. A. Goodenough, second and third editions, Henry Holt and Company, New York, N. Y., 1912, 1920, respectively.

or

$$\int_{P_1}^{P_2} \rho\,dP + \left(\frac{w}{A}\right)^2 \frac{1}{2g}\log_e\left(\frac{\rho_1}{\rho_2}\right)^2 + \left(\frac{w}{A}\right)^2 \frac{1}{2g}\frac{KL}{D} = 0 \quad [3]$$

The density (or specific volume) used in Equation [3] is determined by assuming isentropic expansion in the pipe. In the actual expansion, the fluid friction and pipe friction impose a throttling action which causes the expansion to depart to some extent from the isentropic. However, in the usual case, the density of the mixture will be very nearly the same whether an isentropic expansion or a constant-enthalpy expansion is assumed.

No attempt has been made in developing Equation [3] to in-

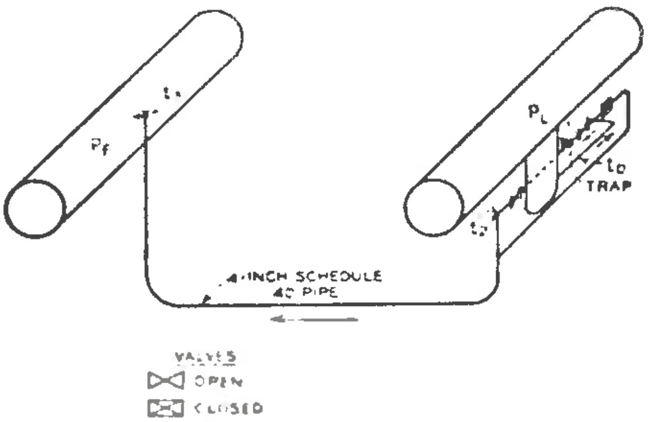

FIG. 3 DIAGRAM OF FIRST HEATER DRAIN LINE TESTED
(Refer to Table 1 for results of test.)

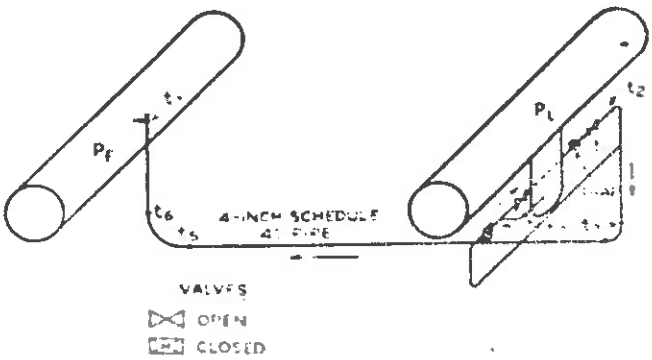

FIG. 4 DIAGRAM OF SECOND HEATER DRAIN LINE TESTED
(Refer to Table 2 for results of test.)

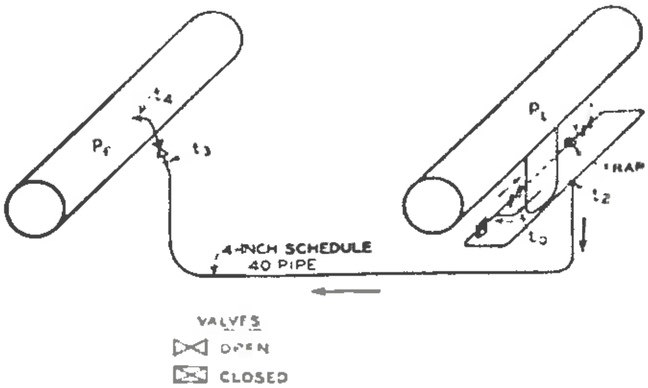

FIG. 5 DIAGRAM OF THIRD HEATER DRAIN LINE TESTED
(Refer to Table 3 for results of test.)

TABLE 1 RESULTS OF TEST OF DRAIN LINE SHOWN IN FIG. 3

Run number	1	2	[illegible]	4
Duration of test, hr	[illegible]	[illegible]	[illegible]	[illegible]
Flow through pipe, lb per sec	[illegible]	[illegible]	[illegible]	[illegible]
Initial pressure, p_1, psi abs	[illegible]	[illegible]	[illegible]	18.2
Final pressure, p_2, psi abs	[illegible]	[illegible]	[illegible]	[illegible]
Friction factor, K	[illegible]	[illegible]	[illegible]	[illegible]

Temperature-measuring point	Distance from trap (equivalent length), ft	Run No. 1 Temp, F	Run No. 1 Saturation pressure, psi abs	Run No. 2 Temp, F	Run No. 2 Saturation pressure, psi abs	Run No. 3 Temp, F	Run No. 3 Saturation pressure, psi abs	Run No. 4 Temp, F	Run No. 4 Saturation pressure, psi abs
t_s Saturation temp		264	37.0	250	29.8	248	[illegible]	223	18.2
t_1	0		37.0		30.0		[illegible]		[illegible]
t_2	[illegible]	[illegible]	[illegible]	225	18.8	[illegible]	14.2	[illegible]	[illegible]
t_3 End of pipe	[illegible]	[illegible]	18.2	207	14.2	[illegible]	[illegible]	[illegible]	[illegible]

* This test was of a preliminary nature to determine if a critical-pressure condition existed at end of pipe.
[illegible] previous test data.
Extrapolated.

TABLE 2 RESULTS OF TEST OF DRAIN LINE SHOWN IN FIG. 4

Run number	1	2	3	4
Duration of test, hr	1	1	1	1
Flow through pipe, lb per sec	25.1	21.1	17.5	[illegible]
Initial pressure, p_1, psi abs	[illegible]	41.9	[illegible]	[illegible]
Final pressure, p_2, psi abs	10.16	9.02	7.64	6.47
Friction factor, K	[illegible]	0.0118	[illegible]	0.0119

Temperature-measuring point	Distance from trap (equivalent length), ft	Run No. 1 Temp, F	Run No. 1 Saturation pressure, psi abs	Run No. 2 Temp, F	Run No. 2 Saturation pressure, psi abs	Run No. 3 Temp, F	Run No. 3 Saturation pressure, psi abs	Run No. 4 Temp, F	Run No. 4 Saturation pressure, psi abs
t_s Saturation temp		271	42.8	265	38.2	255	[illegible]	245	27.4
t_1*	0		37.8		30.5		[illegible]		[illegible]
t_2	5.0	261	[illegible]	251	30.5	241	[illegible]	[illegible]	[illegible]
t_3	10.2	260	[illegible]	250	29.6	[illegible]	[illegible]	[illegible]	[illegible]
t_4	22.0	257	34.5	249	29.2	[illegible]	[illegible]	[illegible]	[illegible]
t_5	[illegible]	251	30.4	243	26.3	241	[illegible]	[illegible]	[illegible]
t_6	42.6	251	30.4	243	26.2	241	[illegible]	219	[illegible]
t_7 End of pipe	48.9	245	27.4	236	23.0	225	18.8	211	[illegible]

* Extrapolated.

TABLE 3 RESULTS OF TEST OF DRAIN LINE SHOWN IN FIG. 5*

Flow through pipe, lb per sec	22.2
Initial pressure, p_1, psi abs	41.4
Final pressure, p_2, psi abs	8.4
Friction factor, K	0.0116

Temperature-measuring point	Distance from trap (equivalent length), ft	Temp, F	Saturation pressure, psi abs
t_s Saturation temp	...	269	41.4
t_1[b]	0		35.0
t_2	40	256	33.0
t_3	82.5	244	26.8
t_4 (End of pipe)[b]	90.3	...	22.0

* This line was tested because of trouble with trap.
[b] Extrapolated.

clude the effect of a static head due to a vertical section of pipe. In most of the heater drain lines encountered by power-plant engineers, a major part of the pipe is horizontal; therefore the effect of the static head is of secondary importance compared to the thermodynamic head.

Before Equation [3] can be used, values for the friction factor K must be determined experimentally. All of the other quantities in the equation will be given for each design case under consideration.

Experimental Investigation of Flow of a Flashing Mixture of Saturated Water and Steam Through Pipes

The experimental work was carried out on some of the heater drain lines on 30,000-kw and 60,000-kw steam turbines at the Connors Creek Power Plant. Schematic sketches of the drain lines tested are shown in Figs. 3, 4, and 5.

During each test, temperature measurements were taken at various points along the drain-line piping, as shown in the sketches and, since the flashing mixture in the pipe was a mixture of saturated water and steam, the pressures at the various points were obtained from steam tables.[4] The quantity of drains flow-

[4] "Thermodynamic Properties of Steam," by J. H. Keenan and F. G. Keyes, John Wiley & Sons, Inc., New York, N. Y., 1936.

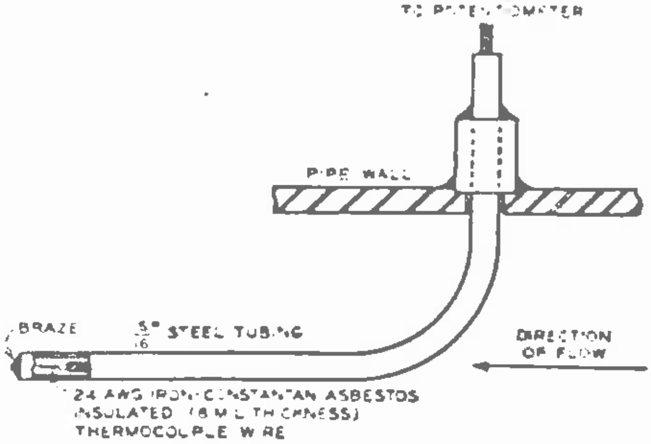

Fig. 6 Design of Thermocouple Well Used in Tests

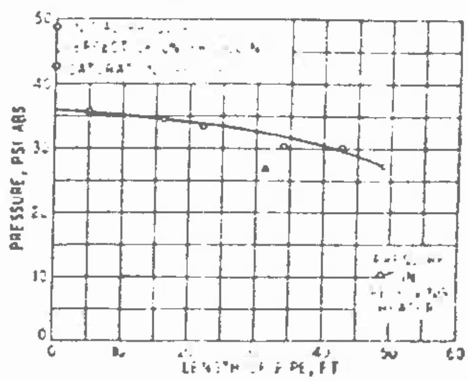

Fig. 7 Measured Pressures Along Drain Line Shown in Fig. 4, as Found in Run No. 1 Given in Table 2

ing through the line was determined by heat-balance calculation.

All temperatures were determined with iron-constantan thermocouples and a temperature indicator of the potentiometer type. It is believed that the measured temperatures are in error by no more than ±1 F. The design of the thermocouple well is shown in Fig. 6.

Test Results

Included in the results of the tests are the pressures at various points along the drain lines, the weight flowing through the pipe, and the values for the friction coefficient K from Equation [3]. In determining the values for K, the length of pipe used in Equation [3] was the equivalent length determined by use of data from various sources.[4]

Table 1 gives the results of a test of the drain line illustrated in Fig. 3. This test was conducted primarily to verify the existence of a critical-pressure condition in the end of the pipe or at the entrance to the low-pressure receiver. The weight of mixture flowing through the pipe was estimated from previous test data, and the values determined for K have been given here mainly for the record.

Table 2 gives the results of a test of the drain line illustrated in Fig. 4. This test was conducted to determine the friction factor K. Fig. 7 shows the pressures at the various points along the pipe for run No. 1 plotted against the equivalent length.

Table 3 shows the results of a test of the drain line illustrated in Fig. 5. This drain line, which operates with about the same initial and final pressures and quantity of mixture flowing as the one illustrated in Fig. 4, but which is somewhat different in piping arrangement, was investigated because of trouble with the float-operated drainer trap.

Analysis of Results

As far as the authors know only two other writers[6,7] have discussed the subject of the flow of a flashing mixture of water and steam through pipes and only one of them offered any test results.[7] His investigation consisted of a single test, and it was felt that additional test data were needed to establish a basis for design. As previously stated, the results of the tests given by the authors in this paper are not complete; however, they do include data that apparently are not available elsewhere, and it is for this reason that this paper has been written. These data offer assistance in designing piping to carry a flashing mixture of water and steam, and it is hoped that other investigators will publish such information as they may have on the subject to provide a still better basis for design work.

Critical Pressure at End of Pipe. It has been known for some time that when an elastic fluid flows through a pipe a critical pressure will occur at the end of the pipe when the ratio of the velocity to the specific volume is a maximum. This critical pressure with an elastic fluid, such as steam, occurs when its velocity equals the velocity of sound in the fluid.

It is interesting to note that a critical-pressure condition can also occur when a flashing mixture of water and steam flows through a pipe, as shown by the test results in Tables 1, 2, and 3. Since the velocity of sound in a mixture of steam and water is not known, it is impossible to state whether or not the velocity attained by the mixture when the critical-pressure condition occurs is that of sound. It may be stated, however, that the critical pressure in the end of the pipe is reached when the increase in energy made available by an increment drop in pressure balances the resulting increase in kinetic energy and the increase in friction. From Equation [3], the relation can be expressed

$$\left(\frac{w}{A}\right)^2 \frac{1}{2g} \log_e \left(\frac{v_2}{v_1}\right)^2 + \left(\frac{w}{A}\right)^2 \frac{1}{2g}\frac{KL}{D} = 144 \int_{p_2}^{p_1} \frac{1}{v}\, dp$$

or

$$\left(\frac{w}{A}\right)^2 \frac{1}{2g} \times \frac{1}{144} = \frac{\int_{p_2}^{p_1} \frac{1}{v}\, dp}{\log_e \left(\frac{v_2}{v_1}\right)^2 + \frac{KL}{D}} \qquad [4]$$

in which ($p \times 144$) has been substituted for P in Equation [3].

Fig. 8 is a graph of solutions of Equation [4] for various downstream-end pressures, assuming a fixed initial pressure of 35 psi abs. As shown by the curve, the maximum value of $\left(\frac{w}{A}\right)^2 \frac{1}{2g} \times \frac{1}{144}$ and therefore the maximum capacity of the pipe is reached when the downstream-end pressure is about 22 psi abs. From

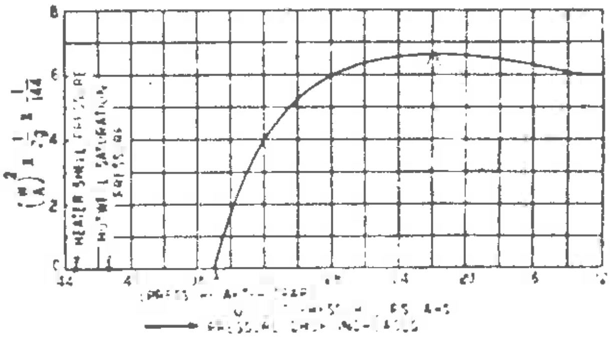

Fig. 8 Illustration of Graphical Method Used in Connection With Equation [4] to Determine Pressure After Trap and End or Critical Pressure in a Pipe Carrying Flashing Mixture of Steam and Water

(The pipe size in this case is 4 in., and the equivalent length following the trap is 90.3 ft. The value of K is considered as 0.0120 and w equal to 22.2 lb per sec.)

Table 3 and Fig. 5, it is seen that the 22 psi-abs downstream end pressure is the critical pressure found on test at the end of the line, and the 35-psi-abs upstream pressure is that found on test near the outlet of the trap, while the flow of 22.2 lb per sec through a 4-in. pipe corresponds to the maximum value of 6.65 shown in Fig. 8, for $\left(\frac{w}{A}\right)^2 \frac{1}{2g} \times \frac{1}{144}$. In this calculation, which is given in detail in the Appendix, the equivalent length was 90.3 ft and the value for K was taken as 0.0120. From this example, it is seen that, when the saturation temperature, quantity of the water, and length and diameter of the pipe are known, it is possible by use of Equation [4] to determine the critical pressure at the end of the pipe, and the initial pressure following the throttling valve or orifice. Also Equation [4] can be used to determine the best pipe size when the saturation temperature, quantity of saturated water, and the length of the line are known.

Friction Factor, K. The friction factor for the flow of a mixture of water and steam through a pipe in which a critical pressure condition exists at the end of the pipe was found to vary from 0.0116 to 0.0131, see Tables 2 and 3. In his test Bottomley[7] found the friction factor to be 0.0120 which compares favorably with the values found by the authors.

[4] "Engineering Data on Flow of Fluids in Pipes, and Heat Transmission," Crane Company, Chicago, Ill., 1935. "Piping Handbook," by J. H. Walker and S. Crocker, third edition, McGraw-Hill Book Company, Inc., New York, N. Y., 1939.

[6] "Discharge Capacity of Traps," by A. E. Kittredge and E. S. Dougherty, *Combustion*, vol. 6, Sept., 1934, pp. 14–19.

[7] "Flow of Boiling Water Through Orifices and Pipes," by W. T. Bottomley, Trans. North East Coast Institution of Engineers and Shipbuilders, vol. 53, 1936–1937, pp. 65–100.

TABLE 4 RESULTS OF STUDY OF FLOW CHARACTERISTICS THROUGH SEVERAL EXISTING CASCADE DRAIN LINES

No. of line	Nominal diam of pipe, in	Pressure — Receiver — High (sat), psi abs	Pressure — Receiver — Low, psi abs	Pressure — End of line, psi abs	Specific volume at end of line (v), cu ft per lb	Density at end of line (ρ), lb per cu ft	Flow (w), lb per sec	Velocity of mixture (V), fps	Momentum (1.414 MV), lb	Kinetic energy $\left(w\frac{V^2}{2g}\right)$ ft-lb	Density × velocity head $\rho\frac{V^2}{2g}$, psf	Lines having elbows replaced
1*	4	152.0	45.8	47.2	0.800	1.25	14.00	127	78	3500	310	X
2*	4	42.8	10.5	27.3	0.426	2.35	25.30	122	136	5870	545	X
3	3	208.8	93.8	93.8	0.340	2.94	5.98	40	11	150	73	
4	4	93.8	36.2	36.2	0.800	1.25	11.30	102	51	1840	200	
5*	4	37.0	6.5	18.2	0.900	1.11	18.20	185	148	9700	590	X
6	4	108.8	37.9	37.9	0.821	1.22	10.72	171	81	4870	555	X
7	4	47.9	13.0	19.3	0.833	1.20	19.13	180	151	9630	605	X
8	6	13.0	2.8	10.4	0.427	2.34	29.90	64	84	1870	145	X
9	4	216.0	108.0	108.0	0.312	1.20	13.64	48	29	4910	115	
10	6	108.0	37.7	37.7	0.834	1.20	20.00	110	128	5000	225	
11	6	11.3	2.5	4.1	3.900	0.256	7.70	150	51	2700	90	
12	4	25.1	7.2	7.4	3.030	0.330	6.94	238	73	6100	290	
13	6	7.2	2.2	4.6	1.500	0.667	13.08	98	56	1950	100	
14*	4	41.4	8.4	22.0	0.704	1.42	22.2	177	172	10,800	690	

* Test results; all others are calculated.

In studying these results it is important to keep in mind that all the tests were conducted with existing installations of 4-in. pipe and for a limited number of pressure combinations. To obtain complete information concerning the friction factor for the flow of a flashing mixture of water and steam through pipes, it would be necessary to investigate the flow through several sizes of pipes, with several pressures in the low-pressure receiver for each of a large number of initial pressures in the high-pressure receiver, and with various quantities for each pipe size and each pressure combination. To conduct such a test, it would have been necessary to build special test equipment involving the expenditure of more time and money than was believed warranted by the importance of the authors' particular problem. A value of 0.0120 for the friction factor will give results that are sufficiently accurate for many design purposes.

Design of Pipe Lines to Carry a Flashing Mixture of Water and Steam

Past practice in the design of pipe lines to carry a flashing mixture of water and steam has depended more upon the judgment of the designer than upon any actual information. For instance, some designers have chosen pipe sizes to give a certain water velocity, neglecting the effect of steam forming in the line, while others have gone to the opposite extreme and chosen excessively large pipes. A line designed by the latter would usually be unnecessarily expensive, while one designed by the former would be liable to troubles with erosion in the elbows and in some cases operating difficulties with float-operated drainers.

Erosion in Drain-Line Piping. The results of a study of the flow characteristics through several existing cascade drain lines are given in Table 4. It should be noted that not all of these lines have a critical pressure at the end of the line, and neither have all the lines had an elbow replaced. The cause of erosion in some drain lines has not definitely been determined; however, there is some indication that it is the result of cavitation which occurs at the elbows due to the momentum of the flowing mixture. For instance, when a saturated mixture of water and steam flows around a bend, the increase in pressure at the outer wall, which results from the change in direction of flow, will cause steam bubbles to collapse, constituting the cavitation condition.

For design purposes it is important to be able to determine in advance whether a line carrying a mixture of water and steam will have a satisfactory life or whether it will wear out in a short time due to erosion. Three methods for correlating experience with erosion have been investigated, namely, (*a*) force on elbows due to the momentum of the flowing mixture $\left(1.414\,\frac{wV}{g}\right)$, (*b*) kinetic energy $\left(\frac{wV^2}{2g}\right)$, (*c*) the product of the velocity head and

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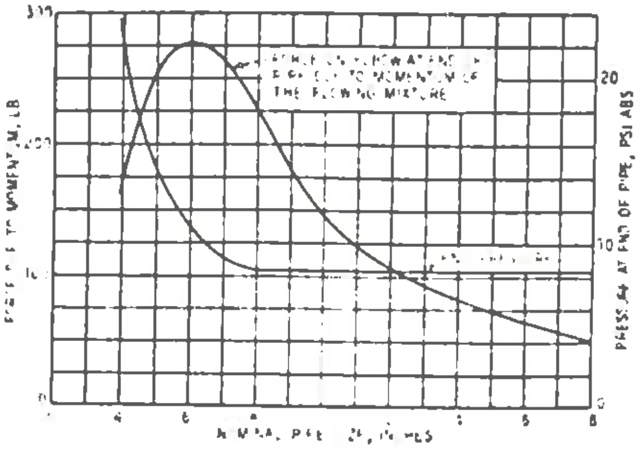

Fig. 9 Curves Showing Relation Between Downstream-End or Critical Pressure, Force Due to Momentum, and Pipe Size for Particular Design

(Case in which the weight of flashing mixture flowing is 22.2 lb per sec, and the initial saturation pressure of the water is 41.4 psi abs.)

the density $\left(\rho\frac{V^2}{2g}\right)$. Of these (*a*) seems to give the most consistent results based on experience with existing lines, as shown by the data given in Table 4. For example, all the lines except three, having a force on the elbow at the end of the line of 75 lb or more, have had at least one elbow replaced. Two of the exceptions have been in service only a short time, while the other is subject to corrosion. All of the lines with a force below 75 lb have not as yet failed by erosion. The methods (*b*) and (*c*) do not give results that are consistent in all cases with experience, as the reader can observe by comparing the values for lines Nos. 4 and 8 in Table 4.

It is important to note that in some cases increasing the pipe size will not always result in a lower velocity at the end of the line. As an example, take line No. 14 in Table 4, for which test data are given in Table 3. If the pipe in this line is increased from a 4-in. to a 6-in. size, the pressure at the end of the line will be about 11 psi abs or 2.5 to 3 psi above the pressure in the low-pressure receiver. The specific volume then is 2.56 cu ft per lb, the velocity is 285 fps, and the force due to the momentum of the flowing mixture at the end of the line is 277 lb.

Fig. 9 shows how the force due to momentum varies for different pipe sizes, as well as the pressures at the end of the line. From the standpoint of this force, it is seen that the 4-in. pipe is better than either the 6-in. or 8-in. pipe. Also in order to get the value of the force due to momentum down to about 75 lb it would be necessary to use a 4-in. pipe, as shown in Fig. 9. It is probable, however, that the larger sizes of pipes, because of the greater projected area at the elbows, will stand a considerably greater total force than pipes ranging up to 8 in. diam. For this particular line, it would probably be necessary to use at least a 10-in. pipe to reduce the erosion to the point where it would not be objectionable.

In many lines it is desirable to use the smaller size pipe and to resort to various schemes for retarding erosion. One scheme, which has been tried with only partial success, is to install tees in place of elbows in such a manner that the momentum of the flowing mixture will be partially dissipated against a blind flange. Another scheme is the installation of elbows with extra-heavy walls. In line with this it should be pointed out that cast-iron elbows, because of thicker walls, last longer than steel elbows. To avoid a forced shutdown of a turbine, telltales can be installed on the elbows to indicate by a leak when the wall is getting thin. Each telltale should be equipped with a valve which can be closed to permit the elbow or tee to be replaced at a convenient time.

Another scheme for minimizing erosion which has been considered is to separate the trap from its usual direct connection to the float operator, and move the trap proper to the downstream end of the drain line. The trap would then be actuated by the remote float through a mechanical or hydraulic linkage. This remote-control arrangement was not favored, however, because it would complicate the heater layout with additional control lines.

Whenever feasible the use of orifices[4] in place of float-operated traps for draining feedwater heaters offers another solution to the erosion problem. By installing the orifice near the end of the drain line so it will discharge into the low-pressure heater through a tee, as shown in Fig. 10, the erosion can be concentrated on the blind flange and in the tee, which can be examined and replaced if necessary whenever the turbine is down for inspection. This scheme has the additional advantage that smaller size pipe can be used than would be possible if the orifice (or trap) were located at the beginning of the line.

[4] "The Flow of Saturated Water Through Throttling Orifices," by M. W. Benjamin and J. G. Miller, Trans. A.S.M.E., vol. 63, July, 1941, pp. 419-426.

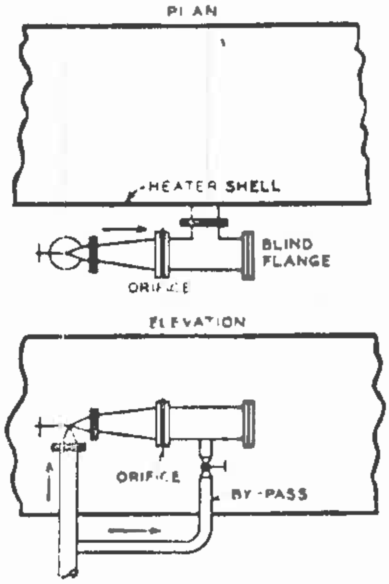

Fig. 10 Sketch Showing Installation of an Orifice Near End of Drain Line Discharging Into Low-Pressure-Heater Shell Through Tee

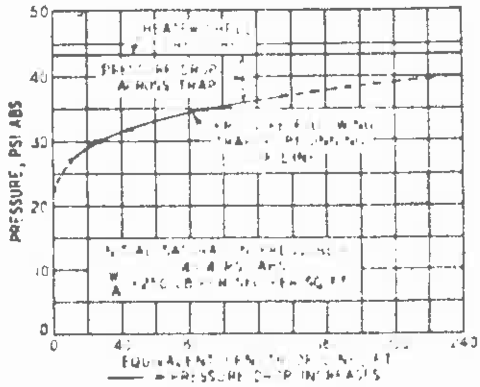

Fig. 12 Curves Showing Relation Between Length of Drain Line and Pressure Drop Across Trap for Actual Design Case

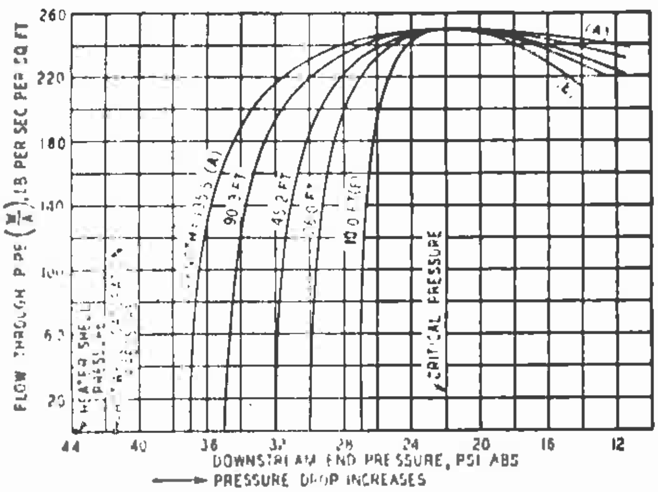

Fig. 11 Curves Showing Relation Between Pressure Following Trap, Downstream-End or Critical Pressure, and Length of 4-In. Drain Line for Constant Flow of 22.2 Lb Per Sec and Saturation Pressure of 41.4 Psi Abs in Hot Well of Upstream Heater

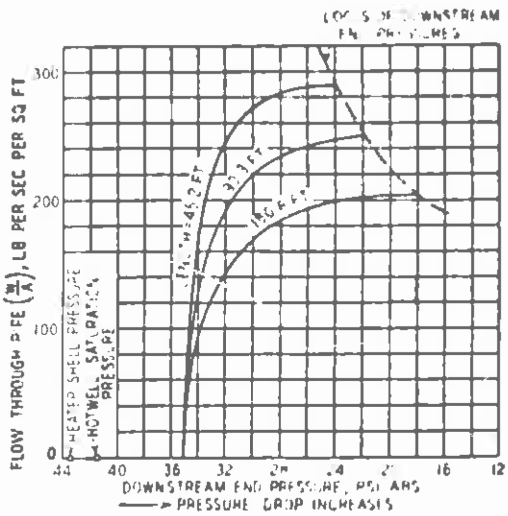

Fig. 13 Curves Showing Relation Between Capacity and Length of Pipe Following Trap, With Constant Initial Pressure After Trap and Constant Hot-Well Saturation Pressure in Upstream Heater

Effect of a Variation in Length on Flow Through Pipes. The effect on the flow characteristics of varying the length of pipe is mainly of interest in lines where the combination of flow and initial temperature is such that a critical-pressure condition will exist at the end of the line. If for example several lengths are assumed for the drain line, shown in Fig. 5 (No. 14, Table 4) while maintaining a constant flow of 22.2 lb per sec it is seen in Fig. 11 that the critical pressure is constant, but the pressure at the beginning of the line (immediately following the trap) varies from 27 psi abs for a length of 10 ft to 37 psi abs for a length of 135.5 ft. This variation of initial pressure with length is also shown in Fig. 12 along with the pressure drop available for forcing the water through the trap. For instance, for a length of 90 ft, which is the installed equivalent length of this line, the available drop is 8.4 psi. The trap as originally installed did not function properly at high loads on the turbine. An investigation showed that, because of the head required to push the water from the heater hot well to the trap, the float chamber on the trap was only half full, or the trap valve was only half-opened, when the heater flooded, and the available pressure differential of 8.4 psi was not great enough to overcome the friction in the half-opened valve. This trouble was overcome by connecting the float chamber directly to the hot well of the heater, thus making the float respond directly to the water level in the hot well. There were two other possible solutions to this problem, however. One was the readjustment of the linkage between the float and the valve so that when the float chamber was half full of water the valve would be wide open. This solution was discarded mainly because it necessitated the installation of a stop to prevent the valve from lifting out of the seat. The other solution was to install 6-in. pipe or larger in place of the 4-in. pipe. With a 6-in. pipe the pressure following the trap valve would be about 19 psi abs, giving a pressure drop across the valve of approximately 25 psi. However, this scheme has the disadvantage of being subject to even more severe erosion than exists in the 4-in. pipe, as shown in Fig. 9.

If the initial temperature of the saturated water and the pressure of the mixture immediately following the trap are held constant, an increase in length of pipe results in a decrease in capacity and critical pressure. This relationship is illustrated in Fig. 13, for the drain line shown in Fig. 5 (No. 14, Table 4). It is interesting to note that the curves in Fig. 13, are similar to those for steam, showing the relationship between pressure drop, length of pipe, and weight of steam flow.[9]

Conclusions

For practical purposes a flashing mixture of water and steam flowing through a pipe can be treated as an elastic fluid. The results of tests show that a critical-pressure condition can exist in the end of a pipe carrying a mixture of water and steam, which is similar to the critical-pressure condition that will exist in a line carrying steam (or any other elastic fluid), in which the pressure drop in the pipe is sufficient to produce an acoustic velocity. Whether or not the velocity of a mixture flowing through a pipe having a critical-pressure condition is the acoustic velocity is a question that cannot be answered until more is known about the velocity of sound through a mixture of a liquid and vapor. The existence of a critical-pressure condition in a pipe carrying a mixture of water and steam depends upon the combination of the following factors: (*a*) The initial saturation temperature of the water leaving the high-pressure receiver, (*b*) the quantity flowing; (*c*) the size and length of the pipe, (*d*) the pressure of the receiver into which the pipe discharges.

It is important to keep in mind that the data presented in this paper are not complete, and the authors hope that other investigators who have the proper testing facilities will be encouraged to gather more information concerning this subject. The data obtained in the authors' tests, however, are useful in making calculations from which pipe lines carrying a flashing mixture of water and steam can be designed to minimize erosion in the elbows, while avoiding the use of unnecessarily large pipe. Similarly, the data are helpful in determining the loss of pressure in the piping adjacent to an orifice or float-operated trap, so that a proper size of orifice can be provided, or so that sufficient pressure differential across the trap is assured.

While the data presented in this paper concern only the flow of a flashing mixture of water and steam, it seems probable that the analysis given here would be applicable to the flow of any flashing mixture of a liquid and its vapor. For the latter, however, it might be necessary to determine new values for the friction coefficient.

Acknowledgment

The authors gratefully acknowledge the encouragement and help of Messrs. P. W. Thompson, Sabin Crocker, and W. A. Carter in preparing this paper; and of Mr. A. C. Pasini and the technical staff at the Connors Creek Power Plant in collecting the test data.

[9] "How to Design Steam Piping for Maximum Capacity and High-Pressure Drop," by M. W. Benjamin, *Heating-Piping and Air Conditioning*, vol. 8, 1936, pp. 475–478. [10] "Steam and Gas Turbines," by A. Stodola, vol. 1, 1927 ed., McGraw-Hill Book Company, Inc., New York, N. Y., p. 63.

Appendix

Nomenclature

A = cross-sectional area of pipe, sq ft
D = diameter of pipe, ft
g = acceleration due to gravity, 32.2 fps per sec
K = friction coefficient
L = over-all length, ft
P = pressure, psf abs
p = pressure, psi abs
q = quality of steam
t = temperature, F
v = specific volume, cu ft per lb
v_m = specific volume of mixture of water and steam, cu ft per lb
V = velocity, fps
w = flow of mixture of water and steam, lb per sec
x = distance to any point along pipe, ft
ρ = density, lb per cu ft
Keenan and Keyes Steam Table nomenclature for Table 5.

Example to Illustrate the Solution of Equation [5]

In order to solve the equation

$$\left(\frac{w}{A}\right)^2 \times \frac{1}{2g} \times \frac{1}{144} = \frac{\int_{p_1}^{p_2} \rho\, dp}{\log_e \left(\frac{\rho_1}{\rho_2}\right)^2 + \frac{KL}{D}}$$

the densities of the mixture for various pressures must first be determined. Table 5 shows the method of calculating the densities for constant-entropy expansion for an initial temperature of 269.3 F, or saturation pressure of 41.1 psi abs (see Table 5). Fig. 14 shows how these densities vary as the pressure decreases

The integral $\int_{p_1}^{p_2} v\,dp$ can best be evaluated arithmetically, since, to integrate directly, it would be necessary to find the equation of the curve in Fig. 14. The arithmetical integration is shown in Table 6, together with the solution of Equation [4]. This shows that, for an initial pressure, following the trap, of 35 psi abs and a downstream-end pressure of 22 psi abs, the value of $\left(\frac{w}{A}\right)^2 \times \frac{1}{2g} \times \frac{1}{144}$ is 6.64 (see also Fig. 8), which corresponds to a flow of 22.2 lb per sec through a 4-in. pipe.

It should be pointed out that it usually requires at least three trials to find a solution. For instance if the initial pressure p_1 had been chosen as 36 psi abs the maximum value of $\left(\frac{w}{A}\right)^2 \times \frac{1}{2g} \times \frac{1}{144}$ would have been greater than the required value of 6.64.

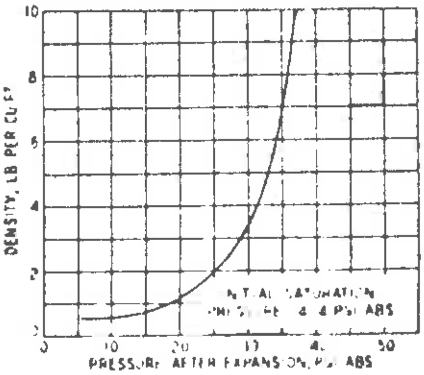

Fig. 14 Curve Giving Densities of Mixture of Water and Steam When Saturated Water Is Expanded Isentropically From Initial Saturation Pressure of 41.4 Psi Abs

TABLE 5 CALCULATION OF DENSITIES OF MIXTURE OF STEAM AND WATER FOR CONSTANT-ENTROPY EXPANSION

p	41.4	36	32	28	24	20	16	8.4
S_1	0.3948							
S_f		0.3831	0.3733	0.3623	0.3500	0.3356	0.3184	0.2708
S_{fg}		1.3017	1.3209	1.3425	1.3672	1.3962	1.4413	1.5312
$(S_1 - S_f)$		0.0117	0.0215	0.0325	0.0448	0.0592	0.0764	0.1240
$g = \frac{S_1 - S_f}{S_{fg}}$		0.0090	0.0163	0.0242	0.0328	0.0424	0.0534	0.0810
$(1 - g)$		0.9910	0.9837	0.9758	0.9672	0.9576	0.9466	0.9190
v_f	0.01716	0.01709	0.01704	0.01698	0.01691	0.01683	0.01674	0.01634
v_g		11.588	12.940	14.663	16.938	20.089	24.75	45.25
$(1 - g)v_f$		0.0169	0.0168	0.0166	0.0164	0.0161	0.0158	0.0152
$(g)v_g$		0.1043	0.2109	0.3548	0.5556	0.8518	1.3217	3.6653
v_m		0.1212	0.2277	0.3714	0.5720	0.8679	1.3375	3.6805
ρ	58.3	8.25	4.39	2.69	1.75	1.15	0.75	0.27

TABLE 6 ILLUSTRATION OF METHOD OF SOLVING EQUATION [4]

p_1	35								
p_2	34	33	32	31	30	29	28	27	26
dp	1	1	1	1	1	1	1	1	1
v_1	6.83								
v_2	5.85	5.05	4.39	3.87	3.41	3.01	2.69	2.40	2.16
v_a	6.35	5.45	4.72	4.13	3.64	3.21	2.85	2.55	2.28
$v_a dp$									
$\Sigma v_a dp$	6.35	11.80	16.52	20.65	24.29	27.50	30.35	32.90	35.18
v_1/v_2	1.17	1.35	1.56	1.77	2.01	2.28	2.55	2.86	3.17
$(v_1/v_2)^2$	1.37	1.84	2.43	3.13	4.04	5.20	6.55	8.19	10.05
$\log_e (v_1/v_2)^2$	0.32	0.60	0.89	1.14	1.40	1.65	1.88	2.10	2.30
$\frac{KL}{D} = \frac{0.012 \times 90.3}{0.3355}$	3.23	3.23	3.23	3.23	3.23	3.23	3.23	3.23	3.23
$\log_e (v_1/v_2)^2 + KL/D$	3.55	3.83	4.12	4.37	4.63	4.88	5.11	5.33	5.51
$\frac{\Sigma v_a dp}{\log_e (v_1/v_2)^2 + KL/D}$	1.79	3.08	4.01	4.72	5.25	5.64	5.93	6.17	6.40

p_2	25	24	23	22	21	20	19	18
dp	1	1	1	1	1	1	1	1
v_2	1.94	1.78	1.57	1.42	1.27	1.15	1.03	0.93
v_a	2.05	1.85	1.66	1.49	1.34	1.21	1.09	0.98
$\Sigma v_a dp$	37.23	39.08	40.74	42.23	43.57	44.78	45.87	46.85
v_1/v_2	3.53	3.89	4.36	4.78	5.39	5.95	6.65	7.17
$(v_1/v_2)^2$	12.45	15.10	19.00	22.85	29.00	35.40	44.20	51.40
$\log_e (v_1/v_2)^2$	2.52	2.71	2.92	3.14	3.37	3.57	3.79	3.99
$\frac{KL}{D}$	3.23	3.23	3.23	3.23	3.23	3.23	3.23	3.23
$\log_e (v_1/v_2)^2 + KL/D$	5.75	5.94	6.15	6.36	6.60	6.80	7.02	7.22
$\frac{\Sigma v_a dp}{\log_e (v_1/v_2)^2 + KL/D}$	6.47	6.58	6.63	6.64	6.61	6.59	6.54	6.49

Discussion

W. T. Bottomley.[10] The Society has granted the writer permission to discuss, in conjunction with this current paper, a paper[11] which the authors previously presented.

Dealing first with their previous paper, the authors do not appear to have fully understood the data which the writer gave in his paper,[7] or they would have realized that their inference, that there is no evidence of a critical pressure in sharp-edged orifices when passing saturated water, is probably not correct.

The orifice the writer used was not sharp-edged but was a converging nozzle, having a cold-water discharge coefficient of nearly unity and was formed by drilling a hole in a 1/4-in. plate, forming a well-rounded entry.

The pressure at the discharge side of the orifice, as measured by the temperature of the water, was not atmospheric pressure but was about 45 to 50 per cent of the initial saturation pressure.[12] Although the final discharge pressure at the other side of the heat exchanger was atmospheric pressure, the resistance through the heater, because of the large volume of flashing steam and water, accounted for the high back pressure at the nozzle discharge.

The writer did not assume that there was a critical pressure at the throat but deduced the fact from the results of the experiments. The last line in Table 1 of the writer's paper[7] gives the estimated throat pressure calculated from the discharge rate, assuming no vaporization before the throat and shows that the throat pressure was well above the pressure on the discharge side. The initial pressures were 48 to 61 psi abs, and the calculated critical pressure at the throat was about 67 per cent of the initial pressure.

Although these conditions were different from those of the authors', yet Fig. 2, of their paper, shows that the maximum discharge rates are in agreement but for the sharp-edged orifice, it was necessary to drop the back pressure to zero to obtain the maximum discharge; whereas, with a nozzle the maximum discharge was obtained with a much higher back pressure.

At first sight, it may appear that the agreement of the maximum flow of saturated water through converging nozzles, with the maximum flow through sharp-edged orifices, is a coincidence, but the writer proposes to show that it is an indication there is a

[10] Merz and McLellan, Carliol House, Newcastle-on-Tyne, England.

[11] "The Flow of Saturated Water Through Throttling Orifices," by M. W. Benjamin and J. G. Miller, Trans. A.S.M.E., vol. 63, 1941, pp. 419–426.

[12] Second, third, and last lines of Table 1, reference 7.