MAXIMUM TWO-PHASE VESSEL BLOWDOWN FROM PIPES

F.J. Moody

ATOMIC POWER EQUIPMENT DEPARTMENT

GENERAL ELECTRIC

SAN JOSE, CALIFORNIA
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BLOWDOWN FROMPIPES

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ATOMIC POWER EQUIPMENT DEPARTMENT
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ABSTRACT

A theoretical model is proposed for maximum two-phase flow from a constant area, adiabatic pipe with friction. Graphs are given for maximum steam/water flow rate in terms of pipe geometry and entrance stagnation properties. The proposed model is used to obtain theoretical blowdown transients from saturated water systems initially at 1000, 1250, and 2000 psia. Estimated pipe flow rates and blowdown transients are compared with steam/water data. Reasonable agreement is shown.

SECTION I

INTRODUCTION

The subject of maximum two-phase flow has received increasing attention in the last 10 years. Faletti, Fauske, Moy, and Zaloudek are among investigators who have measured steam/water flows in long tubes of uniform area. Zaloudek later observed non-equilibrium effects in short tubes with L/D < 6. Recently, Fauske studied saturated water discharge through 0.25-inch-i.d. tubes up to 10.0 inch long. He concluded that steam/water flows approached phase equilibrium at discharge when the tube length exceeded 3.0 inch. Non-equilibrium effects predominated in shorter tubes.

Theoretical models by Fauske, Levy, and Moody have predicted successfully maximum steam/water flows for long tubes in terms of static pressure and steam quality at the discharge plane. These models are based on thermodynamic equilibrium and a lumped-phase or annular flow pattern with unequal phase velocities. Possible blowdown of high-pressure saturated liquid/vapor systems through pipes, valves, and orifices plays an important role in nuclear reactor design. Time-dependent core environment and containment pressure are affected directly by the nature of blowdown.

A large class of vessel blowdowns from pipes can be calculated with present maximum two-phase flow models by using pipe pressure drop correlations as proposed by Martinelli and Nelson or Levy. A general blowdown solution is sought giving: maximum flow rate in terms of vessel stagnation properties and pipe geometry; and time-dependent pressure, mass, and energy in the vessel as a function of pipe geometry.

The purpose of this work is to:

1. Develop a model for maximum steam/water pipe flow rate in terms of upstream stagnation properties and pipe geometry;
2. Obtain theoretical blowdown transients for saturated water reference systems with various pipe L/D ratios and flow areas;
3. Compare maximum flow rates and estimated blowdown transients with data.
SECTION II

THEORETICAL DEVELOPMENT

2.1 Basic Equations for Two-Phase Pipe Flow

The model shown in Figure 2-1 includes these assumptions: steady flow; annular flow without entrainment and liquid in contact with the wall; liquid and vapor in thermodynamic equilibrium at any section; uniform and linear velocities of each phase; straight pipe with constant flow area and adiabatic walls; no shaft work.

Quality, $X$, vapor void fraction, $\alpha$, and slip ratio, $K$, are defined by

$$ X = \frac{W_g}{W_f + W_g} = \frac{W_g}{W} ; \quad (1) $$

$$ \alpha = \frac{A_g}{A_f + A_g} = \frac{A_g}{A} ; \quad (2) $$

$$ K = \frac{u_g}{u_f} . \quad (3) $$

Total flow rate per unit area, $G$, (abbreviated to "flow rate" in text) may be expressed as

$$ G = \frac{W}{A} = \frac{\alpha}{X} \frac{u_g}{v_g} = \frac{1 - \alpha}{1 - X} \frac{u_f}{v_f} \quad (4) $$

Equations (1) through (4) lead to another expression for $\alpha$:

$$ \alpha = \frac{1}{1 + K \left( \frac{1 - X}{X} \right) \frac{v_f}{v_g}} . \quad (5) $$

Stagnation enthalpy, $h_0$, is constant through the pipe. Neglecting elevation terms,

$$ h_0 = X \left( h_g + \frac{u_g^2}{2g_e J} \right) + (1 - X) \left( h_f + \frac{u_f^2}{2g_e J} \right) . \quad (6) $$
Figure 2-1. Incremental Pipe Section
The two-phase momentum rate, $\Omega$, crossing any section is

$$\Omega = \left[ X u_g + (1 - X) u_f \right] GA . \tag{7}$$

The momentum equation for an incremental pipe section is therefore

$$\frac{1}{\rho C} d\Omega = -AdP - \tau_w P_w d\ell . \tag{8}$$

A formulation by Levy\(^{(10)}\) was used to express wall shear.

$$\tau_w = \frac{f_{F(LSP)}}{2 \mu_c} v_f \left( \frac{1 - \alpha}{1 - \alpha} \right)^2 \tag{9}$$

Equation (9) is characterized by annular flow with liquid in contact with the wall. Levy found that this model accurately predicts pressure drop up to 0.80 steam quality. Thereafter, better results are obtained from a vapor flow model. The term $f_{F(LSP)}$ is based on the liquid Reynolds's number,

$$Re_L = \frac{DG}{\mu_L} \left( \frac{1 - X}{1 - \alpha} \right) = \frac{DG}{\mu_L} \left[ 1 + X \left( \frac{K v_f}{v_f - 1} \right) \right]. \tag{10}$$

2.2 Slip Ratio and a Consistent Pipe Flow Model

Zivi\(^{(11)}\) showed that kinetic energy in a two-phase flow is minimum if the slip ratio is given by

$$K = \left( \frac{v_f}{v_f} \right)^{1/3} . \tag{11}$$

He noted that minimum kinetic energy flow corresponds to minimum entropy production, which characterizes a steady state thermodynamic process.

The energy model proposed by Moody\(^{(8)}\) expresses $G$ from Equations (4) through (6) for constant $h_0$ and isentropic flow. Assuming $K$ and $P$ to be independent, flow rate is maximum when

$$\left( \frac{\partial G}{\partial K} \right)_P = 0 \tag{12}$$

and

$$\left( \frac{\partial G}{\partial P} \right)_K = 0 . \tag{13}$$
Equation (12) gives Zivi's slip ratio in Equation (11). The slip ratio \((v_G/v_L)^{1/3}\) tends to give slightly higher theoretical two-phase flow rates. Therefore, the local pressure-dependent slip ratio of Equation (11) is assumed valid throughout the pipe. This assumption is consistent with the energy model\(^{(6)}\) which later is employed for maximum flow rates.

2.3 Maximum Two-Phase Pipe Flow Rates

The Mach number of subsonic gas flow in a constant area, adiabatic pipe approaches unity in the direction of flow.\(^{(12)}\) Gas flow rates are limited by choking (unity Mach number) at the discharge end. Corresponding behavior for steam/water flows has been observed experimentally.\(^{(1, 2, 3, 4, 5)}\)

Figure 2-2 shows that the state of a two-phase system approaches maximum flow condition as static pressure drops. Points 1 and 4 represent entrance and exit states in a pipe. The process path between 1 and 4 was obtained from Equations (4), (5), (6), and (11) for non-maximum flow at constant \(h_0\) and \(G\). \(G_M\), the maximum flow rate corresponding to state 1 is greater than \(G\). But as pressure drops, the fluid state approaches maximum flow condition where \(G\) and \(G_M\) become identical. State 4 corresponds to the pipe exit where the flow chokes.

Figure 2-2. Two-Phase Pipe Flow Process Path
The energy and momentum equations, Equations (6) and (8), can be written in terms of $K$:

\[ G^2 \, d f_1 = - dP - f_2 \, G^2 \, d \]

\[ h_0 = f_3 + G^2 \, f_4 \]

where

\[ f_1 = \frac{J}{\lambda} \frac{1}{6_c} \left[ K (1 - X) v_f + X v_g \right] \left( X + \frac{1 - X}{K} \right) \]  
(16)

\[ f_2 = \frac{J \, P \, w}{\lambda \, A} \frac{f F (LSP)}{2 \, g_c} \, v_f \left[ 1 + \left( \frac{1}{K} \frac{v_g}{v_f} - 1 \right) X \right]^2 \]  
(17)

\[ f_3 = h_f + X h_{fg} \quad \text{and} \]

\[ f_4 = \frac{1}{2 \, g_c} \left[ K (1 - X) v_f + X v_g \right]^2 \left( X + \frac{1 - X}{K^2} \right) \]  
(19)

Since $K$ is given by Equation (11), Equations (16), (18), and (19) give $f_1$, $f_3$, and $f_4$ in terms of $P$ and $X$ only.

Substituting for $df_1$ in Equation (14),

\[ G^2 \left[ \left( \frac{\partial f_1}{\partial P} \right)_X \, dP + \left( \frac{\partial f_1}{\partial X} \right)_P \, dX \right] = - dP - f_2 \, G^2 \, df_1 \]  
(20)

The values of $h_0$ and $G$ are constant throughout the pipe. Equation (15) provides a relationship between $P$ and $X$, typical of the process path in Figure 2-2 for which

\[ \frac{dX}{dP} = - \frac{\left( \frac{\partial f_3}{\partial P} \right)_X + G^2 \left( \frac{\partial f_4}{\partial P} \right)_X}{\left( \frac{\partial f_3}{\partial X} \right)_P + G^2 \left( \frac{\partial f_4}{\partial X} \right)_P} \]  
(21)

The term $P_w/A \, f_F (LSP)$ is replaced by $\bar{f}/D$ where $\bar{f}$ is an average Darcy friction factor in the pipe for the average liquid Reynolds' number of Equation (10), and $D$ is the hydraulic diameter.
Equation (20) can be written as

$$\Gamma \left( P; h_0, G \right) dP = \frac{\overline{F}}{D} \, dt \, .$$

(22)

where

$$\Gamma \left( P; h_0, G \right) = 2g_e \frac{J_G}{2} C^2 \left[ \left( \frac{\partial f_1}{\partial X} \right)_P \left( \frac{\partial f_3}{\partial X} \right)_P + G^2 \left( \frac{\partial f_4}{\partial P} \right)_P \left( \frac{\partial f_1}{\partial P} \right)_X \right] - \frac{2g_e}{J} \cdot$$

(23)

Referring to the exit plane (station 2 in Figure 2-3), $h_0$ and a maximum flow rate $G_M$ are sufficient to specify $P_2$ and $X_2$ from the energy model. Therefore, Equation (22) can be integrated from $P_2$ to some higher pressure $P_1$ at station 1 (Figure 2-3) giving

$$\int_{P_2}^{P_1} \Gamma \left( P; h_0, G_M \right) dP = \frac{\overline{F}}{D} \, .$$

(24)

Known values of $P_1$, $h_0$, and $G_M$ uniquely determine $X_1$ from Equation (15). Stagnation pressure at station 1 is obtained from the idealized isentropic entrance for which

$$S_0 = S_1 = S_{f_0} + \frac{S_{f_{g_0}}}{h_{f_{g_0}}} \left( h_0 - h_{f_0} \right) \, \text{ and}$$

$$S_1 = S_{f_1} + X_1 \, S_{f_{g_1}} \, .$$

(25)

(26)

Equations (23) through (26) were programmed for machine calculation using saturated steam-water properties. Values of $P_0$ and $h_0$ cover saturation states from 25.0 to 2800 psia. Results are plotted in Figure 2-4 giving $G_M$ in either of the forms

$$G_M \left( P_0, h_0, \frac{\overline{F}}{D} \right) = 0 \, \text{ and}$$

$$G_M \left( P_1, h_0, \frac{\overline{F}}{D} \right) = 0 \, \text{ for} \, \frac{\overline{F}}{D} \, \text{ between 0.0 and 100.}$$

(27)

(28)
Figure 2-3. Pipe Maximum Flow Model
Figure 2-4. Pipe Maximum Steam/Water Discharge Rate
Figure 2-4. Pipe Maximum Steam/Water Discharge Rate
Figure 2-4. Pipe Maximum Steam/Water Discharge Rate
Figure 2-4. Pipe Maximum Steam/Water Discharge Rate
Figure 2-4. Pipe Maximum Steam/Water Discharge Rate
Figure 2-5, which relates $G_m^*$, $h_o$, $P_2$, and $X_2^*$ was constructed from the energy model\(^{(8)}\) and can be used with Figure 2-4 to obtain $P_2$ and $X_2$ for maximum steam/water pipe flows.

2.4 Blowdown of Saturated Systems

Figure 2-6 shows an adiabatic, constant volume system containing an equilibrium mixture of liquid/vapor. Mass and energy escape through a single pipe at rates $W$ and $h_E W$ respectively. The term $h_E$ is stagnation enthalpy of fluid in the immediate pipe neighborhood.

A state equation for saturated liquid/vapor is

$$
\frac{E}{M} = \frac{e_f^*}{v_{fg}^*} \left( \frac{V}{M} - v_f \right).
$$

No mass or energy sources are considered; therefore, the following conservation equations apply.

$$
W + \frac{dM}{dt} = 0
$$

$$
W h_E + \frac{dE}{dt} = 0
$$

Differentiating Equation (29) and combining with Equations (30) and (31) to obtain system pressure rate, it follows that

$$
\frac{dP_0}{dt} = - \frac{\left( h_E + \frac{e_{fg}^*}{v_{fg}^*} v_f - e_f \right)}{\frac{V}{M} \left( \frac{e_{fg}^*}{v_{fg}^*} \right) - \left( \frac{e_{fg}^*}{v_{fg}^*} v_f \right) + e_f^*} \frac{W}{M}.
$$

After substituting

$$
W = GA
$$

$$
M^* = \frac{M}{M_i}
$$

$$
V = v_i M_i
$$

$$
E^* = \frac{e_i^* M}{e_i M_i}
$$

2-13
Figure 2-5. Pipe Exit Properties for Maximum Steam/Water Discharge
Figure 2-6. Saturated System Blowdown Model
Equations (30), (31), and (32) become

\[
\frac{dM^*}{dt^*} = -G 
\]

\[
\frac{dE^*}{dt^*} = - \frac{h_E}{c_j} G 
\]

\[
\frac{dP_0}{dt^*} = - \left( \frac{h_E + \frac{d_{fg}}{v_{fg}} v_f - e_f}{v_{fg}} \right) G 
\]

\[
\left[ \frac{v_f}{M^*} \left( \frac{d_{fg}}{v_{fg}} \right)' - \left( \frac{d_{fg}}{v_{fg}} v_f \right)' + e_f' \right] M^* 
\]

Equation (27) is the form used to express \( G \), where \( h_E = h_0 \).

The value of \( h_E \) depends on \( P_0 \) and liquid/vapor action in the system. Three characteristic blowdowns are considered:

1. Saturated liquid blowdown characterized by

\[
h_E = h_f (P_0) 
\]

2. Homogenized mixture blowdown characterized by

\[
h_E = h_f (P_0) + \frac{h_{fg} (P_0)}{v_{fg} (P_0)} \left[ \frac{v_f}{M^*} - v_f (P_0) \right] 
\]

3. Saturated vapor blowdown characterized by

\[
h_E = h_g (P_0) 
\]

Equations (38), (39), and (40) were integrated numerically to express \( P_0, M^*, \) and \( E^* \) in terms of \( t^* \). Results are shown in Figure 2-7 for three steam/water reference systems initially filled with saturated water at 1000, 1250, and 2000 psia. Saturated liquid, mixture, and vapor blowdowns are included for each reference system. Curves for \( T/L/D \) values from 0.0 to 100 are included. Liquid blowdown curves were obtained by taking \( h_E = h_f (P_0) \) until all liquid was gone, leaving only saturated vapor in the system. Thereafter, blowdown was completed with \( h_E = h_g (P_0) \). Liquid disappeared from the 1000, 1250, and 2000 psia reference systems at \( P_0 \) values of 660, 770, and 1150 psia respectively.
Figure 2-7. Blowdown from 1000 psia Steam/Water Reference System
Figure 2-7. Blowdown from 1250 psia Steam/Water Reference System
Figure 2-7. Blowdown from 2000 psia Steam/Water Reference System
SECTION III

RESULTS AND DISCUSSION

3.1 Use of Illustrations

Maximum discharge rates of saturated steam/water from constant area pipes can be estimated from Figure 2-4 when values for \( P_0 \), \( h_0 \), and \( \frac{L}{D} \) are known. Friction factor changes slowly with Reynold's number. An approximate \( R_{eL} \) can be computed from Equation (10) based on estimated \( G_M \) and pipe inlet conditions. An average \( f \) can be obtained from standard charts.\(^{(14)}\) Using \( G_M \) obtained from Figure 2-4, and \( h_0 \), Figure 2-5 can be used to estimate pressure at the discharge plane, \( P_2 \). If \( P_2 \) exceeds receiver pressure, maximum flow occurs. However, if receiver pressure exceeds \( P_2 \), maximum flow does not occur, and Figure 2-4 is not valid. An improved solution for \( G_M \) can be obtained by evaluating \( R_{eL} \) at an average pressure in the pipe.

Steam/water discharge is often related to pipes with local turns, valves, area changes, etc. Unfortunately, the two-phase pressure loss at a local restriction is difficult to represent accurately in terms of equivalent pipe length as is often done in single-phase flow. This technique may be employed in the present model for first approximations. However, it may be necessary to examine all upstream area restrictions to estimate the actual location where flow choking occurs.

Figure 2-7 can be used to estimate system pressure, mass, and energy at various times during a vessel blowdown. The value of \( P_0 \) is shown as a function of \( t^* \) in Graphs I, II, and III. Graph IV gives \( M^* \) and \( E^* \) in terms of \( P_0 \) for liquid, mixture, and vapor blowdowns. Values of \( t^* \) may be found from Graph I, II, or III for corresponding \( M^* \), \( E^* \), and \( P_0 \) in Graph IV.

Liquid blowdown corresponds to mass loss from a low point on the system if vapor entrainment is minor. Mixture blowdown applies to rapid mass loss from the system during which vapor forms faster than it can separate from the remaining liquid. Vapor blowdown implies flow from a high point on the system, slow enough for internal vapor separation without liquid entrainment.

Systems in practice usually are not initially filled with saturated water. However, Figure 2-7 may be used to estimate time-dependent \( P_0 \), \( M^* \), and \( E^* \) for saturated systems containing some vapor by excluding the initial vapor volume.

3.2 Comparison with Fauske’s Data

Figure 3-1 (a) through (e) compares calculations with the present model with Fauske’s data for maximum steam/water discharge rates from straight pipes.\(^{(2)}\) Static pressure taps were at one diameter, 6, 12, 24, 36, and 48 inches from the discharge end. Fauske obtained discharge plane pressure by extrapolating static pressure readings. His exit quality was calculated from the energy equation with a homogeneous kinetic energy term. This calculation was reversed to obtain actual test values of stagnation enthalpy for plotting data in Figure 3-1 (a) through (e). Static pressure at the first tap 48 inches upstream from discharge corresponds to \( P_1 \) in the present model. An average \( R_{eL} \) for the range of pipe flow data is about \( 10^5 \). Therefore, \( f \approx 0.01 \) for smooth pipes was used for all comparisons with Fauske’s data.
Figure 3-1. Comparison with Fauske's Maximum Steam/Water Pipe Flow Data
Figure 3-1. Comparison with Fauske's Maximum Steam/Water Pipe Flow Data
Figure 3-1. Comparison with Fauske's Maximum Steam/Water Pipe Flow Data
The present theory shows reasonable agreement with Fauske's data for test section TS IV short, Figure 3-1 (a) and (b), having an inside diameter of 0.125 inch. Theoretical flow rates tend to over-predict the data at lower stagnation enthalpies. The rounded value of \( \frac{L}{D} = 4.0 \) corresponds to \( f = 0.01 \) and \( L/D = 384 \). Test values of discharge quality ranged from 2.0 to 43.0 percent.

Figure 3-1 (c), (d), and (e) shows Fauske's data for lower pressure in test section TS II long, having an inside diameter of 0.269 inch. Using \( f = 0.01 \) with \( L/D = 180 \), a rounded value \( \frac{L}{D} = 2.0 \) was used for the theoretical comparison. Test discharge qualities are between 14.0 and 62.0 percent. Again, theoretical flow rates over-predict the data at lower stagnation enthalpy. Best predictions are shown for \( h_0 \) around 500 Btu/lb. Under-predicted flow rates are noted at \( h_0 \) above 500 Btu/lb.

Figure 3-1 (f) shows Fauske's most recent data for saturated water discharge from 0.25-inch inside diameter tubes. The stagnation reservoir was a 35.0 ft\(^2\) tank which contained saturated water up to 2000 psia. Blowdowns were relatively slow so that stagnation pressure and enthalpy were nearly constant for each run. All tubes had sharp entrance sections and ranged in length from zero (orifice) to 10 inches. Fauske concluded that equilibrium was approached at discharge in those tests when the tube length exceeded 3.0 inches. Shorter tubes exhibited non-equilibrium effects. Therefore, the present model was compared for tube lengths greater than 3.0 inches. The sharp-edged entrance pressure loss factor was assumed to be 0.5 single-phase velocity heads. Using \( f = 0.01 \), values of \( \frac{L}{D} \) for the various pipe lengths range between 0.1 and 0.4. Therefore, \( \frac{L}{D} \) was estimated between 0.5 and 1.0 for these tests. Calculations are shown for the present model at \( \frac{L}{D} \) values of 0.0 and 1.0. A calculation for \( \frac{L}{D} = 2.0 \) is shown for comparison and shows better agreement with data.

Summarizing the comparison with Fauske's data, Figure 3-1 shows that the present model predicts maximum two-phase pipe flow rates to within \( \pm 30.0 \) percent. This is not surprising since Levy\(^{10}\) noted similar deviations in two-phase pressure drop predictions with his momentum exchange model.

The wall shear expression given by Equation (9) is more applicable to lower quality flows. Therefore, flow rate over-predictions from the present model at lower \( h_0 \) are probably due to inherent over-predictions of maximum flows from the energy model at lower values of \( X \). However, use of Equation (9) for wall shear would introduce pipe pressure drop errors at higher qualities because of liquid entrainment and departure from an assumed annular flow pattern. Higher liquid entrainment would decrease effective viscosity near the wall, and consequently reduce wall shear for a given flow rate. Therefore, a somewhat higher flow rate would be expected than the present model estimates at higher qualities and stagnation enthalpies.

### 3.3 Correction for High Quality Flows

Using Equation (5), \( \tau_w \) given by Equation (9) can be written as follows for high quality flows:

\[
\tau_w (HQ) \approx \left( \frac{f (LSP)}{2 \kappa_c} \right) \tau_1 \left( \frac{v_g}{v_f} \right)^{4/3} G^2 .
\]  

\[ (44) \]
Equation (44), which contains liquid properties, cannot be justified for a case where liquid is almost totally absent. A better expression for wall shear in high quality mixture flow would be

$$\tau_W(HQ) = \frac{f_F(MHQ)}{2g_c} \left(\frac{v_{(MHQ)}}{v_f}\right)^3 \left(\frac{v_f}{v_g}\right)^4 G^2.$$  \hspace{1cm} (45)

The formulation of Equation (9) was used in the present model at all qualities, whereas Equation (45) would be preferred at higher values of X or HQ. For high quality flows, the friction factor in Equation (44) can be adjusted to a new value $f_F(HQ)$ to give approximately the same wall shear $\tau_W(HQ)$ as Equation (45) for a given flow rate G. Equating the two expressions for $\tau_W(HQ)$,

$$f_F(HQ) = f_F(MHQ) \left(\frac{v_{(MHQ)}}{v_f}\right)^3 \left(\frac{v_f}{v_g}\right)^4.$$  \hspace{1cm} (46)

The present model therefore should give better predictions at higher stagnation enthalpy if $\bar{T}$ is approximated by

$$\bar{T}(HQ) = \bar{T}(MHQ) \left(\frac{v_{(MHQ)}}{v_f}\right)^3 \left(\frac{v_f}{v_g}\right)^4.$$  \hspace{1cm} (47)

E.g., a suitable friction factor for estimating saturated vapor flow from the present model would be

$$\bar{T}_g = \bar{T}_{(CSP)} \left(\frac{v_f}{v_g}\right)^{1/3}.$$  \hspace{1cm} (48)

where $\left(\frac{v_f}{v_g}\right)$ should be evaluated at an average static pressure in the pipe.

3.4 Comparison with Blowdown Tests

The present blowdown model was compared with pressure suppression tests for: a full-scale, 1/112 segment of the Bodega Bay atomic power plant \(15\) and a full-scale, 1/48 segment of the Humboldt Bay plant. \(16\) Figure 3-2 is a typical primary system schematic for either series of tests. The model reactor vessel initially contained saturated steam/water at 1250 psia. A double rupture disc assembly was used for the purpose of initiating blowdown. Various nozzles and orifices were used to simulate specific blowdown flow areas. Transient vessel pressure traces were obtained for each test. Table 3-1 gives important data for all tests considered here.
Figure 3-2. Blowdown System for Bodega and Humboldt Pressure Suppression Tests
### TABLE 3-1

**SYSTEM DATA FOR BLOWDOWN TESTS**

<table>
<thead>
<tr>
<th></th>
<th>Bodega - 40</th>
<th>Bodega - 21</th>
<th>Bodega - 16</th>
<th>Humboldt - 22</th>
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<tr>
<td><strong>Vessel Initial Pressure (psia)</strong></td>
<td>1250</td>
<td>1250</td>
<td>1250</td>
<td>1250</td>
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<tr>
<td><strong>Saturation Temperature (°F)</strong></td>
<td>572.4</td>
<td>572.4</td>
<td>572.4</td>
<td>572.4</td>
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<tr>
<td><strong>Saturation Enthalpy (Btu/lbm)</strong></td>
<td>578.6</td>
<td>578.6</td>
<td>578.6</td>
<td>578.6</td>
</tr>
<tr>
<td><strong>Initial Water Temperature (°F)</strong></td>
<td>537.4</td>
<td>572.4</td>
<td>572.4</td>
<td>572.4</td>
</tr>
<tr>
<td><strong>Initial Water Enthalpy (Btu/lbm)</strong></td>
<td>532.5</td>
<td>578.6</td>
<td>578.6</td>
<td>578.6</td>
</tr>
<tr>
<td><strong>Initial Water Subcooling (Btu/lbm)</strong></td>
<td>48.1</td>
<td>0.0</td>
<td>0.0</td>
<td>0.0</td>
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<tr>
<td><strong>Flashing Pressure (psia)</strong></td>
<td>940</td>
<td>1250</td>
<td>1250</td>
<td>1250</td>
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<tr>
<td><strong>Nozzle Throat Area (sq in.)</strong></td>
<td>8.25</td>
<td>8.25</td>
<td>----</td>
<td>----</td>
</tr>
<tr>
<td><strong>Orifice Throat Area (sq in.)</strong></td>
<td>----</td>
<td>----</td>
<td>20.6</td>
<td>0.95</td>
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<tr>
<td><strong>Back Pressure (atm)</strong></td>
<td>1.0</td>
<td>1.0</td>
<td>1.0</td>
<td>1.0</td>
</tr>
<tr>
<td><strong>Upstream Pipe Area (sq in.)</strong></td>
<td>115</td>
<td>115</td>
<td>115</td>
<td>26</td>
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</table>
3.5 Estimation of Equivalent \( f_{L/D} \) for Bodega Bay Test 40

An estimation of \( f_{L/D} \) is required in the present theory for predicting blowdown tests. Such an estimation can be made by either of two methods: a summation of standard single-phase geometric loss coefficients plus actual \( f_{L/D} \) components associated with the system tested; or calculation of an equivalent \( f_{L/D} \) from measured irreversible pressure drop at a known cold water flow rate. The latter method was chosen. Although blowdown rate was not measured in any of the tests, a cold water flow rate can be estimated in Bodega Test 40 from the initial rate of vessel pressure drop. Test 40 initially contained 579°F water which was pressurized to 1250 psia by introducing steam at the top just before blowdown. The sub-cooled water did not flash in the vessel until pressure dropped from 1250 to 940 psia. Therefore, initial pressure drop was due to expansion of vapor in the vessel. Approximating the vapor by an ideal gas, adiabatic pressure and volume changes are related by

\[
dV_g = -\frac{V_g}{P_0} dP.
\] (49)

Vessel total volume is constant. Therefore,

\[
\frac{dV_g}{dt} = -\frac{dV_L}{dt}.
\] (50)

The initial blowdown rate is then

\[
W_{Li} = -\rho_L \frac{dV_L}{dt} = \rho_L \frac{V_{gi}}{P_{0i}} \left( \frac{dP_0}{dt} \right).
\] (51)

Figure 3-3 (a) shows vessel pressure reduction for cold water blowdown before water saturation pressure was reached in the vessel. Initially, \( dP_0/dt = -1350 \) psi/sec, which led to an initial water blowdown rate of \( W_{Li} = 990 \) lb/sec, or \( Q_{Li} = 14,300 \) lb/sec-ft² in the nozzle throat. Estimated nozzle throat static pressure dropped below water saturation pressure. However, vaporization probably did not occur; or if it did, the effect on flow rate was negligible. This conclusion was reached by comparing the cold water flow rate with calculations for an orifice and 1250 psi pressure differential.

An equivalent \( f_{L/D} = 1.6 \) based on nozzle throat area was obtained by assuming all pressure drop occurred in an equivalent pipe with the same area, using the equation

\[
\frac{f_{L}}{D} = 2 \frac{g_c (P_0 - P_R) \rho_L}{G_{Li}^2}.
\]

(52)

However, expansion losses downstream from the nozzle are included in this calculation. Such expansion losses do not affect flows which choke in the nozzle throat. Subtracting an expansion loss of 1.0 velocity head gave a rounded value of \( f_{L/D} \approx 1.0 \). This value was applied directly to Bodega Test 21, which had a saturated blowdown with the same nozzle used in Test 40.
Figure 3-3. Comparison with Blowdown Tests
Figure 3-3. Comparison with Blowdown Tests
3.6 Comparison with Bodega Test 21

The present model using $\tilde{f}_{L/D} = 6.0$ is compared with Test 21 in Figure 3-3 (b). There is a sharp initial dip in test pressure which essentially recovers in 1.0 second. This characteristic appeared in all saturated blowdown tests and is not shown by the present model. It may be explained by a combination of two effects: initial discharge of slightly sub-cooled liquid, not restricted by a two-phase mechanism; and the delay time for vapor bubbles to form and expand in liquid.

Also, the experimental traces show a "knee," or sudden increase in pressure drop rate. The theoretical liquid blowdowns in Figure 2-7 also show a knee when saturated water blowdown is followed by saturated vapor blowdown. This is important evidence regarding the nature of steam/water behavior in the vessel during blowdown. The present model was applied to Bodega Test 21 by assuming complete water expulsion followed by vapor expulsion. This is reasonable since blowdown proceeded from a low point on the vessel. The theoretical knee in Figure 3-3 (b) is seen to occur at a pressure of 850 psia where the trace indicates the knee at 675 psia. Therefore, vapor must have formed faster than it could separate from the liquid during blowdown, prolonging appearance of the actual knee. Vapor entrainment in liquid gives higher stagnation enthalpy near the break, the effect of which is twofold at a given stagnation pressure: it decreases maximum flow rate (Figure 2-4), and it increases initial pressure drop rate (Figure 2-7). This points out the importance of liquid/vapor action in a vessel during blowdown, which is another problem.

3.7 Comparison with Bodega Test 16

One would expect the knee in Figure 3-3 (b) to occur earlier in time and at lower pressures as the nozzle throat area increased.

Bodega Test 16, Figure 3-3 (c), is similar to Bodega Test 21 except for an increased blowdown area, and orifice replacing the nozzle. Although the knee is still evident, the blowdown area increase of almost 3 times that in Test 21 nearly caused its disappearance. A large nozzle might cause such rapid vapor formation that a nearly homogeneous liquid/vapor mixture would fill the vessel, and the knee would disappear altogether.

A remark is in order concerning the estimation of $\tilde{f}_{L/D} = 6$, used for comparison with Bodega Test 16. It was found that Bodega Test 17 (not shown) was nearly identical to Bodega Test 21 except that an orifice was used instead of a nozzle. Blowdown areas were the same in each case, and the pressure-time characteristics were nearly identical. It was concluded that exchanging an orifice for a nozzle caused negligible effects on flow rate for these blowdown tests, upstream irreversibilities being unaffected. Blowdown area evidently is the important variable. Therefore, $\tilde{f}_{L/D} = 1$ in Test 21 was simply increased by the square of the throat area ratio in Tests 16 and 21.

It is unlikely that a high concentration of vapor would occupy the vessel lower region during blowdown while liquid was still present. Therefore, steam/water action in the vessel for all Bodega and Humboldt tests should lie somewhere between: (1) a homogeneous mixture filling the vessel; and (2) completely separated phases with water occupying the vessel lower region until fully expelled. The present model was therefore used to estimate both liquid and homogeneous blowdowns for Bodega Test 16. Both theoretical blowdowns are seen to be too slow in Figure 3-3 (c). This leads to the conclusion that the estimated $\tilde{f}_{L/D} = 6$ is too high. A theoretical curve for $\tilde{f}_{L/D} = 6$ and a homogeneous system are shown for comparison. It clearly estimates too fast a blowdown.
Admittedly, the present model may encounter difficulty if applied to cases with predominant or complex geometric pressure loss effects.

3.8 Comparison with Humboldt Test 22

Humboldt Test 22 had a very small blowdown area equal to 0.95 sq in. Upstream velocities in the 26-sq-in. pipe could not exceed 10 fps, based on cold water orifice flow calculations from 1250 psia. Therefore, geometric and friction pressure losses between the vessel and orifice could be neglected. The value of $\tau L/D = 0$ was used in the present theory to compute a liquid blowdown followed by vapor. The result is shown in Figure 3-3 (d). Agreement is very good.

Fauske confirmed by his tests that saturated water flow through an orifice did not flash until it cleared the aperture. However, the water flow rate in Humboldt Test 22 apparently was restricted by a two-phase mechanism. This implies that a small fraction of vapor was entrained in liquid near the orifice due to homogeneous boiling.
## NOMENCLATURE

### Major Symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Area, (\text{ft}^2)</td>
</tr>
<tr>
<td>D</td>
<td>Hydraulic diameter (4A/P_w), (\text{ft})</td>
</tr>
<tr>
<td>E</td>
<td>Internal energy, (\text{Btu})</td>
</tr>
<tr>
<td>E*</td>
<td>Internal energy fraction defined by Equation (36), dimensionless</td>
</tr>
<tr>
<td>e</td>
<td>Specific internal energy, (\text{Btu/lb}_{\text{m}})</td>
</tr>
<tr>
<td>(f'_F)</td>
<td>Local Fanning friction factor, dimensionless</td>
</tr>
<tr>
<td>(f)</td>
<td>Local Darcy friction factor; ((f = 4f'_D) for circular pipes), dimensionless</td>
</tr>
<tr>
<td>({f'_1, f'_2})</td>
<td>Functions defined by Equations (16) through (19)</td>
</tr>
<tr>
<td>({f'_3, f'_4})</td>
<td>Functions defined by Equations (16) through (19)</td>
</tr>
<tr>
<td>G</td>
<td>Mass flow rate per unit area, abbreviated &quot;mass flow rate,&quot; (\text{lb}_m/\text{sec-ft}^2)</td>
</tr>
<tr>
<td>(g_c)</td>
<td>Newton's constant, (32.2\ \text{lb}_{\text{m}}/\text{ft-lb}/\text{sec}^2)</td>
</tr>
<tr>
<td>h</td>
<td>Specific enthalpy, (\text{Btu/lb}_{\text{m}})</td>
</tr>
<tr>
<td>J</td>
<td>Mechanical equivalent of heat, (778 \text{ ft-lb}/\text{Btu})</td>
</tr>
<tr>
<td>J'</td>
<td>(J/144, 5.403 \text{ ft}^3/\text{lb}_m^2) - (\text{Btu})</td>
</tr>
<tr>
<td>K</td>
<td>Slip ratio, dimensionless</td>
</tr>
<tr>
<td>L</td>
<td>Pipe length, (\text{ft})</td>
</tr>
<tr>
<td>(l)</td>
<td>Downstream distance from pipe inlet, (\text{ft})</td>
</tr>
<tr>
<td>M</td>
<td>Mass, (\text{lb}_m)</td>
</tr>
<tr>
<td>(M^*)</td>
<td>Mass fraction defined by Equation (34), dimensionless</td>
</tr>
<tr>
<td>P</td>
<td>Pressure, (\text{psia})</td>
</tr>
<tr>
<td>(P_w)</td>
<td>Wetted perimeter, (\text{ft}).</td>
</tr>
<tr>
<td>(R_{cL})</td>
<td>Liquid Reynolds number defined by Equation (10), dimensionless</td>
</tr>
<tr>
<td>S</td>
<td>Specific entropy, (\text{Btu/lb}_m - \text{F}^\circ)</td>
</tr>
<tr>
<td>(t^*)</td>
<td>Dimensional time defined by Equation (37), (\text{ft}^2 - \text{sec}/\text{lb}_m)</td>
</tr>
<tr>
<td>(u)</td>
<td>Velocity, (\text{ft/sec})</td>
</tr>
<tr>
<td>V</td>
<td>Volume, (\text{ft}^3)</td>
</tr>
<tr>
<td>(v)</td>
<td>Specific volume, (\text{ft}^3/\text{lb}_m)</td>
</tr>
<tr>
<td>W</td>
<td>Mass flow rate, (\text{lb}_m/\text{sec})</td>
</tr>
<tr>
<td>X</td>
<td>Quality (vapor mass flow fraction), dimensionless</td>
</tr>
</tbody>
</table>
Major Symbols (Continued)

\( \alpha \) = Vapor volume fraction, dimensionless

\( \Gamma \) = Function defined by Equation (23), \( \text{in}^2/\text{lb}_f \)

\( \gamma \) = Isentropic exponent, dimensionless

\( \mu \) = Dynamic viscosity, \( \text{lb}_m/\text{ft} \cdot \text{sec} \)

\( \rho \) = Density, \( \text{lb}_m/\text{ft}^3 \)

\( \tau_w \) = Wall shear stress, \( \text{lb}_f/\text{ft}^2 \)

\( Q \) = Momentum flow rate, \( \text{lb}_m \cdot \text{ft/sec}^2 \)

Subscripts

\( E \) = Blowdown escape value

\( f \) = Saturated liquid

\( fg \) = Vaporization

\( g \) = Saturated vapor

\( GSP \) = Gas in single phase flow

\( HQ \) = High Quality

\( i \) = Initial value

\( L \) = Liquid

\( LSP \) = Liquid in single phase flow

\( M \) = Property at maximum flow rate

\( M \) = Mixture

\( HQ \) = High quality mixture

\( 0 \) = Value at stagnation condition

\( R \) = Receiver

\( 1 \) = Pipe entrance

\( 2 \) = Pipe discharge end

Special Symbols

\( (-) \) = Average value

\( (\ ') \) = Derivative with respect to pressure
REFERENCES


REFERENCES (Continued)


ACKNOWLEDGMENT

Particular gratitude is expressed to D. E. Williams for obtaining machine solutions from which flow rates in Figure 2-4 were obtained. R. E. Allen helped considerably in calculations for Figure 2-7 and preparation of graphs.
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<td>Thermodynamics</td>
<td>65 APE 4</td>
<td>April 20, 1965</td>
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### Title

**Maximum Two-Phase Vessel Blowdown from Pipes**

### Abstract

A theoretical model is proposed for maximum two-phase flow from a constant area, adiabatic pipe with friction. Graphs are given for maximum steam/water flow rate in terms of pipe geometry and entrance stagnation properties. The proposed model is used to obtain theoretical blowdown transients from saturated water systems initially at 1000, 1250, and 2000 psia. Estimated pipe flow rates and blowdown transients are compared with steam/water data. Reasonable agreement is shown.

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