Steam Flow Through Safety Valve Vent Pipes

H. E. BRANDMAIER

M. E. KNEBEL

INTRODUCTION

The large volume released through a steam safety valve in a power plant must be discharged to the atmosphere without causing damage to equipment or injury to personnel. In a typical installation the steam flows from a valve discharge elbow, hereafter referred to as the valve pipe, through a suitable length of piping, referred to as a vent pipe, and is finally exhausted to the atmosphere. In the common vent system design, shown in Figure 1, in which the vent

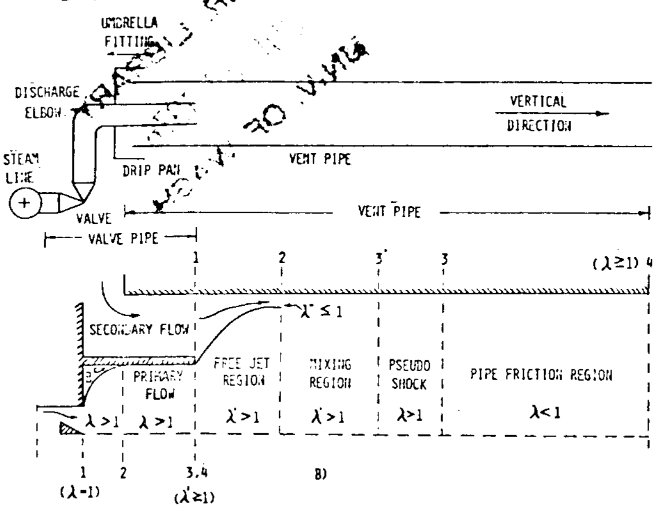

Fig. 1 Steam Safety Valve System

(A) Schematic
(B) One-dimensional Model

pipe inlet forms an "umbrella fitting" over the valve pipe outlet, the valve outlet extends into the vent pipe with sufficient clearance so that thermal expansion or reaction induced movement of either valve or vent pipe will not affect the other component.

Proper design of the vent pipe is necessary to assure that steam will not be released into the power plant interior. This can occur by blowback of steam at the clearance between valve pipe outlet and vent pipe because of an undersized vent pipe for the existing steam conditions.

Thus, given steam conditions and vent pipe length, the design problem consists of the determination of the minimum vent pipe diameter to eliminate the possibility of blowback.

An additional requirement to assure the integrity of the vent system is the determination of the reaction forces acting on the valve pipe and the vent pipe for proper support of these members. Failure of either component could release more steam into the surroundings than from blowback. The force acting upon the valve pipe is frequently determined from valve manufacturer's data, but may also be calculated from steam conditions at that point. The forces acting on the vent pipe may be determined from steam conditions at the vent pipe inlet and outlet.

Two methods frequently employed in the design of safety valve vent systems are cited in (1). The first is based upon a procedure established almost 35 years ago by Benjamin (2); the second is a more recent industry developed procedure.

The former method assumes a value for nozzle efficiency in the conversion of available enthalpy drop into velocity to determine the steam velocity at the valve pipe outlet. Assuming sonic flow at the vent pipe outlet the static pressure at the vent pipe inlet is calculated considering friction losses in a long pipe. Blowback is assumed not to occur if the velocity head of the steam jet leaving the safety valve is equal to or greater than the calculated static pressure inside the vent pipe inlet.

An examination of these methods shows deficiencies in each. The former method requires the assumption of a nozzle efficienc for expansion through the safety valve. No rational basis exists for selection of the efficiency value, other than the observation that a value of 25 to 30% results in what appears to be an adequate vent pipe design.

The latter method calculate steam conditions at the vent pipe inlet by working backwards from the vent pipe outlet sonic conditions. The resulting calculated pressure and velocity at the vent pipe inlet may not be the same as the pressure and velocity

NOMENCLATURE

A - flow area

a - sonic velocity

D'_1 - primary flow diameter at vent pipe inlet

f - Darcy friction factor

F - mass flow function

G - function defined by equation (3)

I - impulse or momentum function

L - duct length

M - Mach Number

$\dot{m}$ - mass flow

P - pressure

p - ratio of secondary to primary stagnation pressure

R - gas constant

S - entropy

T - temperature

t - square root of ratio of secondary to primary stagnation temperatures

W - structural load on pipe

α - vent pipe area ratio, $A_3/A'_1 = A_4/A'_1$

β - valve pipe area ratio, $A_3/A_1 = A_4/A_1$

γ - ratio of specific heats

λ - velocity ratio

μ - ratio of secondary to primary mass flow

SUPERSCRIPT

$'$ primary flow

$''$ secondary flow

$*$ quantity at sonic condition

SUBSCRIPT

0 stagnation condition

1 initial section

2 section at which flow fills pipe

3 section at which shock system produces subsonic flow

3' section at which mixing of primary and secondary flows is complete in vent pipe

4 pipe exit

a atmospheric conditions

Σ stagnant corner region in valve pipe calculated working forward from the sonic conditions at the safety valve pipe outlet. Thus the procedure does not directly tie together conditions at these two points.'

It appears then that while either method may result in a satisfactory vent pipe sizing, the result is fortuitous. The analysis described in the following sections, based upon the application of modern fluid dynamics, provides a sounder basis for vent pipe design. The description of the flow processes uses the similarity between the flow in the vent system with that in free-jet wind tunnels and supersonic ejectors.

FLOW DESCRIPTION

The combination of a valve pipe and vent pipe is very complex, from a fluid-dynamic viewpoint, due to the annular valve orifice, whose axis is generally perpendicular to the flow, and the number of bends in the piping required to conduct the steam to atmosphere. For mathematical tractability, the model to be discussed consists of a one-dimensional valve orifice in the flow direction and straight piping with inlets and outlets normal to the pipe axis as shown in Figure 1. In addition, it is assumed that transient effects due to valve opening and closing are small and that the thermodynamic properties of steam are approximately those of a perfect gas with an appropriate isentropic exponent.

The flow in the valve pipe is similar to that in free-jet wind tunnels (3-5); in cylindrical tube rocket launchers (6); and in ejectors operating at zero secondary flow (7). A major difference between these primarily aerospace problems and the present stationary power plant problem is that the initial velocity is generally supersonic in the former case and sonic in the latter.

Referring to Figure 2a, the flow leaving the valve orifice can be characterized as an underexpanded jet (8-10) exhausting into a larger diameter cylindrical pipe. The flow expands across a series of expansion waves to supersonic velocities until its static pressure equals the pressure surrounding the jet; the corresponding flow direction defines the jet boundary. An oblique shock wave is generated at the intersection of the jet boundary and the pipe wall which is required to turn the flow parallel to the wall. The subsequent flow in the valve pipe then consists of a series of intersecting oblique shocks which decrease in intensity due to their interaction with each other and with the pipe boundary layer. If the valve pipe is long enough the flow eventually becomes subsonic and behaves as a subsonic flow in a long pipe with friction (11,12).

The stagnant corner region at the orifice is essentially isolated. Its pressure is not atmospheric and is determined by the balance between the flow leaving it by

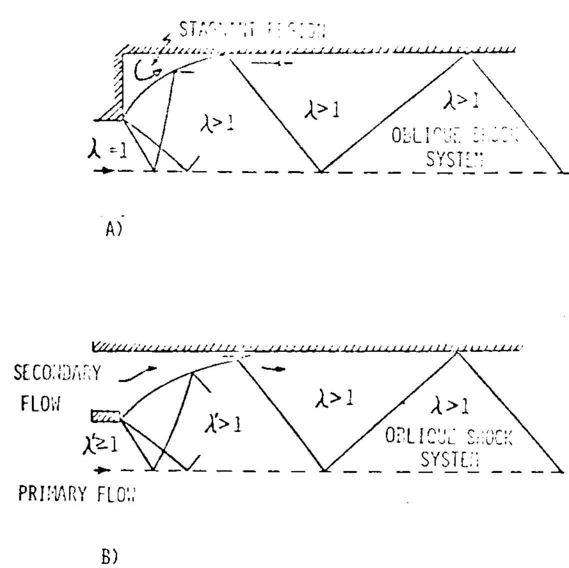

Fig. 2 Underexpanded Flows

(A) Valve Pipe
(B) Vent Pipe

entrainment in the supersonic jet and that entering it through the subsonic boundary layer under the adverse pressure gradient d to the oblique shock.

Flow in the vent pipe is similar to th flow in the valve pipe. However in the ven pipe, depending on the design and operating conditions, there may be either a secondary flow of air into the vent pipe from the surrounding atmosphere, as in an ejector; no secondary flow; or blowback of steam from t vent pipe system into the powerplant. The objective of the vent pipe deisgn is to determine the minimum vent pipe area required prevent blowback of steam.

Figure 2b depicts the flow field when there is a secondary flow from the surrounding atmosphere into the vent pipe.

Downstream of the oblique shock wave, primary and secondary flows mix in the cons diameter pipe until at some section the flo are completely mixed and the intersecting shocks have reduced the initially supersoni flow to a subsonic flow. Depending on the vent pipe length, the flow at the vent pipe outlet may be sonic with a static pressure greater than atmospheric; supersonic if the vent pipe is very short; or subsonic in whi case the outlet pressure must equal atmosphe The limiting condition for no blowback corresponds to zero secondary flow and a pipe length for which the flow accelerates to so conditions at the pipe outlet. At greater lengths the shock system moves upstream int the free jet region, the pressure rises and blowback occurs.

Early analyses of the flow in ejectors and free-jet wind tunnels were based on a o dimensional flow model (4,7,13). While useful, this model is incapable of providing information on the three-dimensional flow in the underexpanded jet region, on details of the mixing of primary and secondary flows, and on the statis pressure in the corner region of the valve pipe. The development of methods r the analysis of three-dimensional, supersonic, compressible flows containing shock waves coupled with increased digital computer capacity and speed has resulted in more precise determinations of the flow characteristics for these devices. This approach has been successfully applied to free jets (8,10), ejectors (14-16) and ducts with abrupt changes in cross-section (6). Progress has also been made in determining the static pressure in the corner region at an abrupt change in flow area (17,18).

Emphasis in the cited studies was on the underexpanded jet portion between the primary jet exit plane - the valve orifice or the valve pipe exit in the present case - and the vicinity of the intersection of the jet boundary and the pipe wall. The subsequent flow containing shock-shock and shock-wall interactions appears to have been only investigated experimentally (6,17).

Due to the possibility of two-phase flow and condensation shocks when the steam is initially close to saturation conditions, the flow in powerplant safety valve piping is far more complex than flow in the free-jets, ejectors and other fluid devices on which the preceding discussion was based. The difficulty of analyzing this problem, and the time and cost required to develop suitable computer programs has left the powerplant designer with essentially the Benjamin method (2) developed over thirty years ago. Although still one-dimensional, the analysis and results discussed in the following sections will extend the power plant designer's knowledge of this complex flow problem and present the data in a simple, easily useable form.

FLOW ANALYSIS

Figure 1b is the general model used to analyze flow in the valve pipe and vent pipe. The flow characteristics are as described in the preceding section. Although mixing and flow deceleration occur simultaneously, this analysis assumes that the primary and secondary flows first mix to yield a uniform supersonic flow. This is followed by deceleration through the oblique shock system which is replaced in the analytical model by a normal shock - the pseudo-shock created by Crocco (19). While these phenomena may actually require many pipe diameters (19,20), if not the entire pipe, for completion it is assumed that they are completed within a small fraction of the pipe length.

Considering the vent pipe first, the impulse function, $I = PA + \dot{m}A$, can be expressed as (21)

$$I = ((\gamma + 1)/2\gamma)\, \dot{m}a^* \,(\lambda + 1/\lambda) \qquad (1)$$

In deriving equation (1), conservation of mass and energy and the equation of state for a perfect gas were used. The velocity ratio λ is the ratio of the fluid velocity to the critical sonic velocity attained by isentropic expansion from the local stagnation state. Alternatively, Mach Number M can be used; however the equivalent expression is more complex. In addition, the range of λ is narrow i.e., $0 < \lambda \leq ((\gamma+1)/(\gamma-1))^{1/2}$ compared to the range of M i.e., $0 < M \leq \infty$.

Applying equation (1) between the inlet section 1 and section 3 where the primary and secondary flows are completely mixed results in the momentum conservation equation,

$$(\dot{m}' + \dot{m}'')a_3^* G_3 = \dot{m}' a^{*\prime} G_1' + \dot{m}'' a^{*\prime\prime} G_1'' \qquad (2)$$

in which the primary flow is denoted by a single prime and the secondary flow by a double prime and

$$G = \lambda + 1/\lambda \qquad (3)$$

Equation (2) assumes continuity of static pressure across the boundary between primary and secondary flows, although the average pressures may differ widely; the isentropic exponent is the same for the primary and secondary flow; and wall friction for the secondary flow is negligible.

The mixed flow stagnation temperature can be expressed in terms of the primary and secondary stagnation temperatures, using conservation of energy, as,

$$(\dot{m}' + \dot{m}'')T_{03} = \dot{m}' T_0' + \dot{m}'' T_0'' \qquad (4)$$

Letting $\mu = \dot{m}''/\dot{m}'$ and $t^2 = T_0''/T_0'$, and combining equations results in,

$$[(1 + \mu)(1 + \mu t^2)]^{1/2} G_3 = G_1' + \mu t G_1'' \qquad (5)$$

Knowing the steam temperature upstream of the safety valve T_0' and the ambient temperature T_0'' defines t. In the case of the valve pipe, $\lambda_1' = 1$ corresponding to sonic conditions at the valve orifice; in the case of the vent pipe, $\lambda_1' \geq 1$ and is determined by the valve pipe analysis. λ_3 can then be determined from equation (5) as a function of $\lambda_1'' \leq 1$ for specified values of $\mu \geq 0$. However, equation (5) has two solutions for λ; the supersonic solution which corresponds to conditions upstream of the normal shock and the subsonic solution which corresponds to conditions downstream of the normal shock. These are given by,

$$\lambda_3 = (G_3/2) - ((G_3/2)^2 - 1)^{1/2} \qquad (6)$$

$$\lambda_3' = (G_3/2) + ((G_3/2)^2 - 1)^{1/2} \qquad (7)$$

it follows from equations (6) and (7) that the product of the sub- and supersonic values of λ_3 equals 1.

The ratio of the pipe area to the primary flow area at section 1 is determined from conservation of mass in the form

$$\dot{m} = \gamma(2/(\gamma+1))^{\frac{1}{2}} A(P_0/a_0)\lambda\ (1-((\gamma-1)/(\gamma+1))\lambda^2)^{1/(\gamma-1)} \tag{8}$$

where,

$$a_0 = (\gamma R T_0)^{\frac{1}{2}} \tag{9}$$

Letting

$$F = \lambda(1-(\gamma-1)\lambda^2/(\gamma+1))^{1/(\gamma-1)} \tag{10}$$

and $p = P_0''/P_0'$, and applying equation (8) to both primary and secondary flows leads to the following equation from which the vent pipe area ratio $\alpha = A_3/A'_1$ can be determined,

$$\mu = (\alpha-1)_p F_1'' / tF_1' \tag{11}$$

The value of fL/D_3 corresponding to the limiting condition of sonic flow at the vent pipe exit and λ_3 calculated using equation (6) is determined from the friction equation for flow in long pipes (21).

$$fL/D_3 = ((\gamma+1)/2\gamma)(\ln\lambda_3^2 + 1/\lambda_3^2 - 1) \tag{12}$$

Although equations (5), (6) and (10)-(12) are sufficient to determine the vent pipe area for given values of L, λ_1' and μ, there are physical limits which must be considered in their application.

The first limit corresponds to sonic velocity at section 2 in the secondary flow. This will be referred to as the 'secondary flow sonic limit'. The supersonic primary flow area is a maximum at section 2 whereas the subsonic secondary flow area is a minimum. The flow conditions at section 2 are more difficult to determine than those at section 1 since neither the primary nor secondary velocities or flow areas are known. However, assuming no mixing of primary and secondary flows and therefore isentropic flow between sections 1 and 2, conservation of mass yields for the primary and secondary flows,

$$A_2'/A_1' = F_1'/F_2' \tag{13}$$

$$A_2''/A_1'' = F_1''/F_2'' \tag{14}$$

As the vent pipe area is constant,

$$A_1' + A_1'' = A_2' + A_2'' = A_3 \tag{15}$$

Using equations (13) and (14) to eliminate A_2' and A_2'' in equation (11) leads to,

$$(\alpha-1) = (1-(F_1'/F_2'))/((F_1'/F_2'')-1) \tag{16}$$

Substituting equation (11) for $(\alpha-1)$ and simplifying yields,

$$(1/F_1' - 1/F_2') + (1/F_1'' - 1/F_2'')(\mu t/P) = 0 \tag{17}$$

Applying equation (1) between sections 1 and 2 and neglecting wall friction for the secondary flow results in,

$$(G_1' - G_2') + (G_1'' - G_2'')\mu t = 0 \tag{18}$$

Finally, using equation (18) to eliminate the product μt in equation (17) gives,

$$(G_1' - G_2')/(1/F_1' - 1/F_2') = p(G_1'' - G_2'')/(1/F_1'' - 1/F_2'') \tag{19}$$

This equation can be numerically solved for λ''_1 in terms of λ'_2 for a given combination of p, λ_1', and λ_2''. The vent pipe area ratio is then obtained from equation (16) and the secondary to primary flow ratio μ obtained from equation (11) given t.

A satisfactory, simpler solution assumes incompressible secondary flow at section 1. Equation (19) then reduces to a quadratic equation for λ_1''. Figure 3 shows the results using this approach for p = 0.05. From the curves for λ_2'' = 1,0.5 and 0.1 the sonic limit is seen to be a lower limit

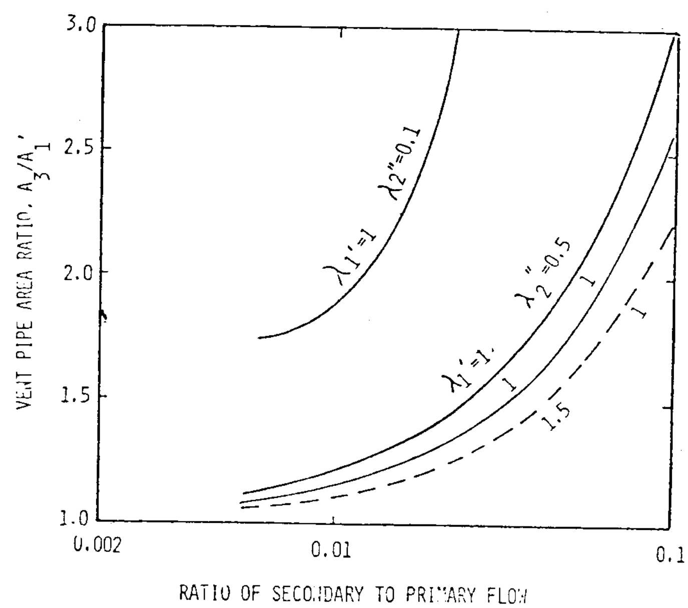

Fig. 3 Secondary Flow Sonic Limit

$\gamma = 1.3$, $t = 2/3$, $p = 0.05$

on the vent pipe area ratio α for a given value of secondary to primary flow ratio μ; at larger values of α, $\lambda''_2 < 1$. As λ'_1 increases to 1.5 in Figure 3, a smaller area ratio and therefore a smaller increase in primary flow velocity is required to accelerate the secondary flow to sonic conditions at section 2.

Once α has been calculated from equations (5), (6) and (10)-(12) for specified values of L, λ_1 and μ, Figure 3 can be used to estimate λ''_2, the secondary flow velocity ratio at section 2. For the range of α and μ considered herein, $\lambda''_2 << 1$ and therefore the 'secondary flow sonic limit' is not reached. Figure 3 is inapplicable to the steam blowback limit, $\mu = 0$.

The most interesting limit originates from the thermodynamic requirement that the entropy cannot decrease in the direction of flow. This thermodynamic limit, initially studied by Hermann (4), can be expressed in terms of the difference between the mixture entropy upstream of the normal shock at section 3' and the sum of the entropies of the primary and secondary flows at section 1. For an ideal gas, the entropy is expressed as,

$$\Delta S/R = (\gamma/(\gamma-1)) \ln T_0 - \ln P_0 \qquad (20)$$

Thus,

$$(\dot{m}' + \dot{m}'')(\Delta S/R)_{3'} \geq \dot{m}'(\Delta S/R)'_1 + \dot{m}''(\Delta S/R)''_1 \qquad (21)$$

or, after substituting equation (20) and simplifying,

$$-(1+\mu)(\ln (P_{03'}/P'_0)(T_{03}/T'_0)^{-\gamma/(\gamma-1)}) + \mu \ln(pt^{-2\gamma/(\gamma-1)}) \geq 0 \qquad (22)$$

Substituting equation (6) into mass conservation between sections 1 and 3', ie $\dot{m}_3 = (1+\mu)\dot{m}_1'$, results in

$$P_{03}'/P'_{01} = (1+\mu)(F'_1/F_3')(T_{03}/T'_0)^{\frac{1}{2}}(1/\alpha) \qquad (23)$$

From equation (4), the total temperature ratio is,

$$T_{03}/T'_0 = (1+\mu t^2)/(1+\mu) \qquad (24)$$

Thus each solution of equation (5) must satisfy the inequality expressed by equation (22). For each value of fL/D_3 this thermodynamic limit imposes an upper limit on the allowable pressure ratio p for a physically realizable flow. In the actual flow where primary and secondary flow mixing and shock interactions occur simultaneously and not in series as assumed in this model, this limit may not exist at all. This possibility depends on a more detailed analysis for evaluation.

Other limits include a vent pipe outlet pressure greater than atmospheric pressure, which is consistent with the assumption of sonic velocity at this section. Finally, there are limits to the velocity ratio λ_3 downstream of the normal shock. The maximum value of λ_3 corresponds to no primary flow acceleration between sections 1 and 2. As $\lambda_3 \lambda_3' = 1$ and $\lambda_3' > \lambda_1'$ it follows that $\lambda_3 \leq 1/\lambda_1'$; this is approximate since, as equation (5) shows, μ has an effect. The minimum value of λ_3 is determined by the maximum value of $\lambda_{3'} = ((\gamma+1)/(\gamma-1))^{\frac{1}{2}}$ upstream of the normal shock which corresponds to an infinite Mach Number. Thus $\lambda_3 \leq 1/\lambda_{3'}$, or $\lambda_3 \leq ((\gamma-1)/(\gamma+1))^{\frac{1}{2}}$. These limits produce limits to fL/D_3 in accordance with equation (12). In particular, a minimum $\lambda_3 = 0.3612$ for $\gamma = 1.3$ results in a maximum $fL/D_3 = 4.096$. This limit is, however, beyond the range of interest in Figures 5-7 where the abscissa is $fL/D_1' = \alpha^{\frac{1}{2}}(fL/D_3)$.

ZERO SECONDARY FLOW

The case of zero secondary flow corresponds both to the limiting case for no blowback, given the vent pipe length and pressure ratio p, and to the flow in the valve pipe. Setting $\mu = 0$ in the equations in the preceding section gives the equations applicable to this case.

Thus the momentum and thermodynamic limit equations reduce to, after dropping the superscripts.

$$G_3 = G_1 + (\alpha-1)\, p/F_1 \qquad (25)$$

$$\alpha F_{3'}/F_1 \geq 1 \qquad (26)$$

In the vent pipe case given λ_1 and $fL/D_{3'}$, which determines λ_3 from equation (12), equation (25) uniquely determines $(\alpha-1)$p. For $\alpha > 1$, $\lambda_3 < 1/\lambda_1$ i.e., the flow accelerates between sections 1 and 3'. However, it should be noted that since fL/D_3 for the vent duct depends on D_3 and therefore α, the solution is iterative. The thermodynamic limit, equation (26) must be checked for each solution of equation (25).

In the valve pipe case, the area ratio β is generally known or can be deduced from the valve manufacturer's literature. The unknown quantities in equation (25) are the static pressure in the corner, which determines p and the velocity at section 3 in the valve pipe. Either experimental data or a multidimensional analysis is required to determine the corner pressure, P_B in Figure 1b. As the valve pipe is much shorter than the vent pipe, the velocity changes primarily due to the oblique shock system. The minimum valve pipe outlet velocity is the sonic velocity. The maximum velocity is the supersonic velocity at section 2 which corresponds to the supersonic solution of equation (25) in the form

$$G_3 = 2 + (2/(\gamma+1))(\beta-1)\, P_B/P_1 = G_4 \qquad (27)$$

In equation (27) $\lambda_1 = 1$, $P_B/P_1 = (P_B/P_{01})(1 - (\gamma-1)\lambda_1^2/(\gamma+1))^{-\gamma/(\gamma-1)}$ and β is the ratio of valve pipe outlet area to valve orifice area, A_3/A_1.

From equation 8, the ratio of the stagnation pressure at the valve pipe outlet to the stagnation pressure at the valve orifice is,

$$P_{04}/P_{01} = (1/\beta)\ F_1/F_4 \qquad (28)$$

If sonic conditions exist at both sections, the stagnation pressure ratio is simply the inverse of the area ratio.

RESULTS

Figures 4-7 illustrate the effect of the major parameters on the minimum vent pipe area ratio α for no steam blowback. The initial velocity and stagnation pressure at section 1 of the vent pipe in Figure 1 depend on the stagnation pressure at the valve orifice and the geometry of the valve pipe. This is shown in Figure 4. The variation of the ratio of corner pressure to

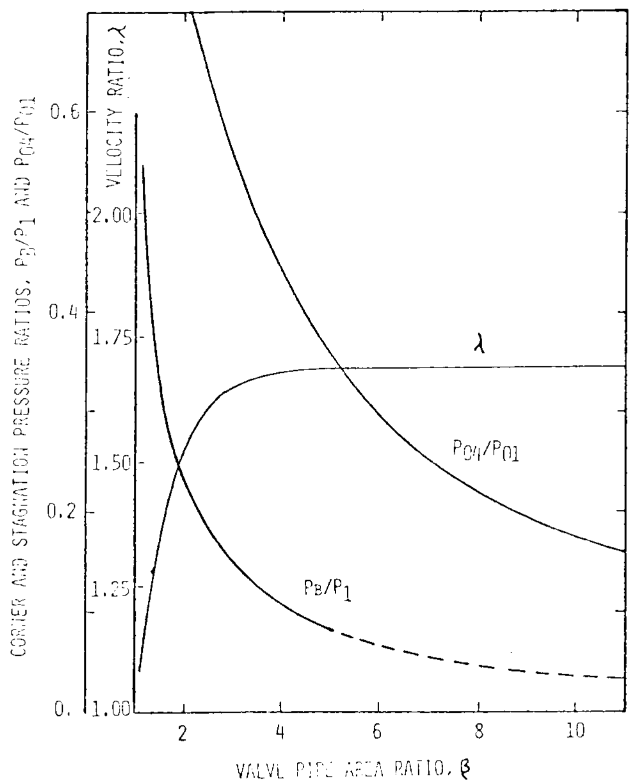

Fig. 4 Corner Pressure, Velocity and Stagnation Pressure Ratio as Functions of Valve Pipe Area Ratio

$\lambda_1 = 1$

static pressure in the flow P_B/P_1 is based on the analysis by Korst, Chow and Zumwalt (18); the dotted line is an extrapolation of their data. The maximum velocity $\lambda = \lambda_3 = \lambda_4$, corresponding to impingement of the flow on the valve pipe wall near the pipe outlet, was calculated from equation (27) and plotted in Figure 4. It approaches a constant at large ratios of valve pipe outlet to valve orifice area; for the assumed corner pressure data the maximum value of λ is about 1.7. The minimum velocity corresponds to sonic conditions at the outlet, $\lambda = 1$. The stagnation pressure ratio P_{04}/P_{01} shown in Figure 4 was calculated from equation (28). As $P_{04}/P_{01} < 1$, the entropy increases in the flow direction.

For this one-dimensional analysis the valve pipe outlet velocity, equal to the vent pipe inlet velocity, can only be determined within limits, eg for the assumed corner pressure data, $1 \leq \lambda_1 \leq 1.7$. The pressure ratio p is the ratio of the secondary flow stagnation pressure P_0'' to the primary flow stagnation pressure P_0' at the vent pipe inlet; thus

$$p = P_0''/P'_0 = (P''_0/P_{01})(P_{01}/P_0') = (P''_0/P_{01})(P_{01}/P_{04}) \qquad (29)$$

To a good approximation P_{01} is the steam pressure upstream of the valve orifice and P_0'' is atmospheric pressure.

Figure 5 is an example of the principal design data, calculated from equations (25) and (26) which correspond to the limiting case of zero secondary flow and the pipe length that produces sonic flow at the vent pipe outlet. The data is for superheated

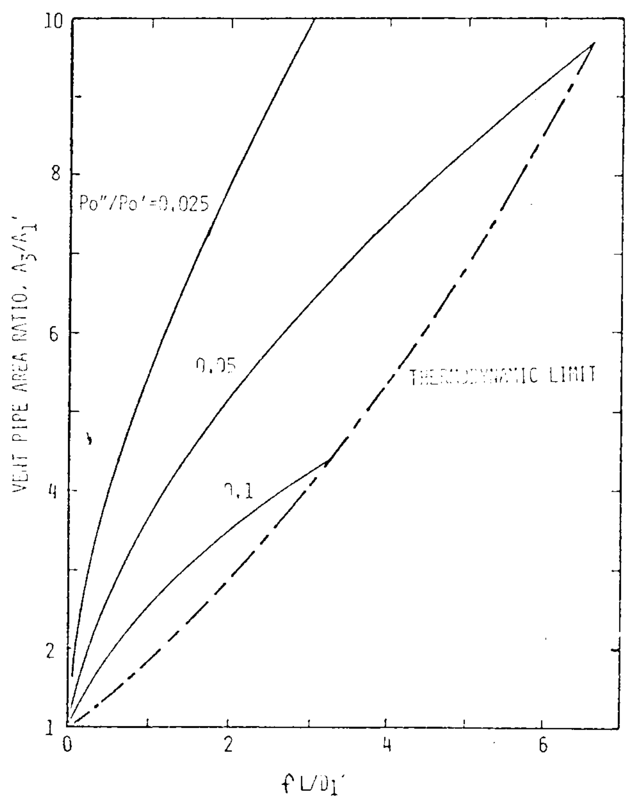

Fig. 5 Effect of Pressure Ratio on the Limiting Area Ratio

$\gamma = 1.3, \ \mu = 0, \ \lambda'_1 = 1$

steam and sonic flow at the vent pipe inlet. For each value of the product of friction and the ratio of vent pipe length to the diameter of the primary flow at the vent pipe inlet there exists a maximum pressure ratio p corresponding to the thermodynamic limit determined from equation (26) using the equal sign.

For each pressure ratio at or below this maximum pressure ratio, there is a minimum area ratio below which steam blowback occurs. As an example, for a valve pipe area ratio β of 10, sonic flow at the valve pipe outlet and a stagnation pressure P_{01} of 3000 psia (2.07×10^7 Pa) at the valve orifice, $p = 10(15/3000) = 0.05$. Thus, assuming $fL/D'_1 = 4$, Figure 5 shows that the minimum area ratio is about 7.45.

The vent pipe area ratio corresponding to the maximum value of λ'_1, at the vent pipe inlet can be determined from data such as shown in Figure 6. Here, the effect of inlet velocity, at a given pressure ratio,

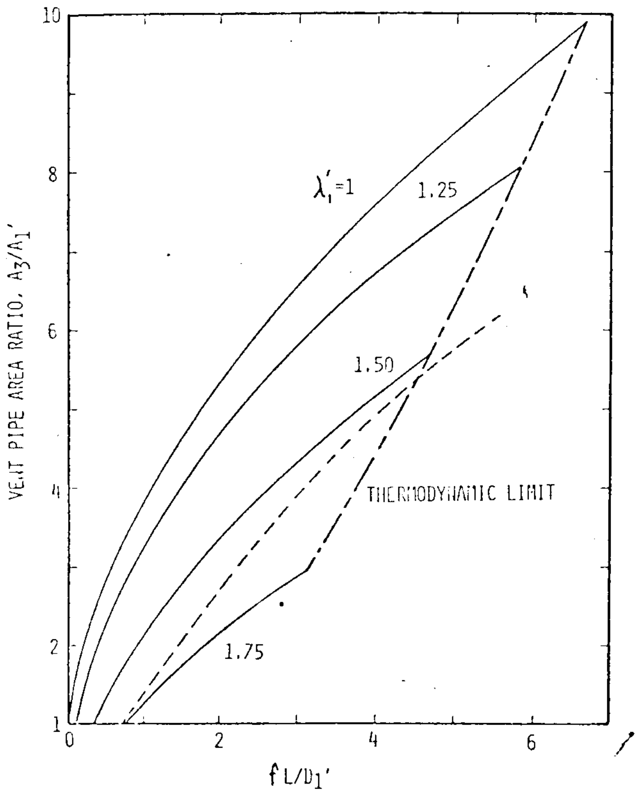

Fig. 6 Effect of Initial Velocity on Area Ratio

$\gamma = 1.3$, $\mu = 0$

———— $\lambda'_1 = 1$, $P''_0/P'_0 = 0.05$

-------- $\lambda'_1 = 1.7$, $P''_0/P'_0 = 0.02857$

on the minimum vent pipe area ratio is shown. The thermodynamic limit imposes an upper bound on the initial velocity for a specified value of fL/D'_1. The maximum allowable value of λ_3 as a function of λ'_1 results in the lower limit on fL/D'_1 shown in Figure 6.

From Figure 4, $\lambda'_1 = 1.7$ and $P''_0/P'_0 = 0.175$ for $\beta = 10$; thus for $P'_0 = 3000$ psia (2.07×10^7 Pa), $p = (15/3000)/0.175 = 0.02857$. From the dotted line for $\lambda'_1 = 1.7$ and $p = 0.02857$ in Figure 5, the minimum area ratio for $fL/D'_1 = 4$ is 4.9. Assuming $\lambda'_1 = 1$ and a specified P_{01}, provides a conservative area ratio for the vent pipe since, for larger values of λ'_1 and correspondingly lower values of p, the minimum vent pipe area ratio is less.

Figure 7 shows the effect of vent pipe area ratio and length on the secondary mass flow for $p = 0.05$ and $\lambda'_1 = 1$. In the

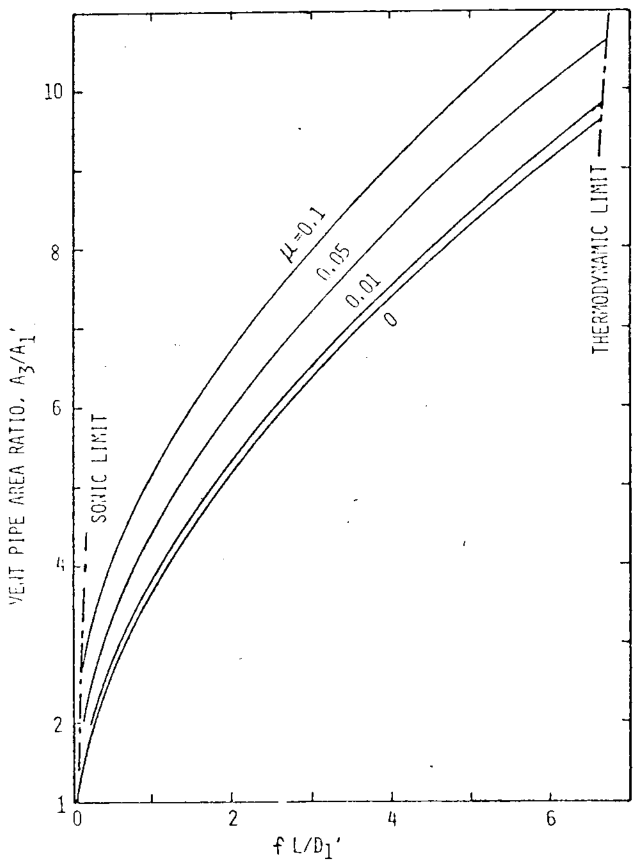

Fig. 7 Effect of Secondary Flow on Area Ratio

$\gamma = 1.3$, $P''_0/P'_0 = 0.05$ $\lambda'_1 = 1$

region to the right of the $\mu = 0$ curve, steam blowback occurs. In the region to the left of the $\mu = 0$ curve, a secondary flow of air is induced from the power plant into the vent pipe. Thus, for a constant value of fL/D'_1, increasing the vent pipe area ratio increases the secondary mass flow μ. Similarly, for a constant α, decreasing fL/D'_1 increases μ. Assuming a value of $\mu > 0$ for the vent pipe design provides a design margin for the vent pipe area ratio calculated as described herein.

SATURATED STEAM

The equations for determining the minimum vent pipe area ratio are based on the assumption that steam behaves as a perfect gas. In the calculations, an isentropic exponent γ of 1.3 in the superheated case and 1.1 in the saturated case was assumed. Although the area ratio is relatively insensitive to γ, the thermodynamic limit strongly depends on γ.

The flow analysis for saturated steam is more complex than for superheated steam due to non-equilibrium effects arising from the existence of two phases - vapor and liquid droplets. It is generally assumed that the liquid and vapor temperatures and velocities are the same; therefore non-equilibrium effects are confined to the changing vapor and liquid masses as steam condenses or the liquid droplets vaporize with the corresponding conversion of latent and sensible heats. In the case of nozzle flows, for which most analytical and experimental information is available (22-24), non-equilibrium results in supersaturation of the vapor, followed by rapid condensation and latent heat release at a discontinuity referred to as the condensation shock (23).

The one-dimensional flow equations including a phase change require, in addition to the perfect gas equation, and mass, momentum and energy conservation equations, an equation from which the liquid mass can be determined as a function of the flow parameters. In the non-equilibrium condensing flow case, this is an equation for the rate of formation of liquid droplets. If the phases are assumed to be in thermodynamic equilibrium, the equation is the Clausius-Clapeyron equation relating saturation pressure and temperature (23).

In addition to an increase in the number of equations to be solved, the speed of sound, and therefore Mach Number, depend on non-equilibrium effects. If the flow is in equilibrium, an equilibrium sound speed can be defined based on a homogeneous mixture of vapor and liquid. At the other extreme, the liquid droplets behave as inert particles and the vapor properties alone determine the speed of sound.

As a first step toward evaluating two-phase flow in the vent pipe a few equilibrium flow vent pipe calculations were made for saturated steam using the ASME steam tables (25). This was accomplished by expressing the mass, momentum and energy conservation equations in their most basic form and using the steam tables to determine the relationship between enthalpy, entropy, density, temperature and pressure. A limited number of hand calculations was made for saturated steam at one valve orifice stagnation pressure; the results indicated that the minimum vent pipe area ratio was smaller in the real gas case than in the ideal gas case. However, more extensive computerized calculations are required to verify that the ideal gas approach yields conservative area ratios over the range of saturated steam conditions of interest in power plants.

STRUCTURAL LOADS

Structural loads are applied to the valve and vent pipes and supports as a result of the change in magnitude and direction of the impulse function, equation (1). Thus considering the valve pipe the load or thrust W parallel to its axis is simply

$$W = I_4 - P_0'' A_4 \qquad (30)$$

where the subscripts refer to the valve pipe in Figure 1. The product $P_0'' A_4$ is the force at the lower end of the valve pipe in the usual case where there is an elbow immediately downstream of the valve orifice. In the vent pipe case, due to the possibility of bends in the pipe, the values of I at the inlet and outlet must be resolved into their components along coordinate axes. The differences, e.g., $I_{1x}-I_{2x}$ along the x-axis, are the corresponding loads on the support. In addition to these forces, moments are also produced. As the value of λ_4 and therefore I_4 depends on the pipe length, the limiting value of λ_1' that maximizes I_4 should be used in designing the support.

EXAMPLE

As an application of the preceding, consider a superheater safety valve for the following conditions:

P_{01}	2800 psia(1.930×10^7Pa abs)
T_0	1000°F(537.8°C)
rated $\dot{m}$	350,000 lb/hr(1.588×10^5kg/h)
Valve Size	6" sch 40
I.D.	6.065 in (15.40 cm)
(A_4)valve=(A_1)vent	28.9 in^2 (186.4 cm^2)

Since the ASME valve rating is 90% of capacity, the design flow will be increased to full capacity. For calculation purposes, the vent pipe will be assumed to be 50 ft. long. Design data are:

γ (superheated steam)	1.3
$\dot{m}$=1.11(rated $\dot{m}$)	107.9lb/sec(48.94 kg/s)
L	50 ft (15.24 m)
a_0 (eq 9)	2290 ft/sec(698 m/s)

The calculation will be performed for assumed vent pipe sizes of 12, 14, and 16 inches standard weight.

Vent pipe size, inches	12	14	16
Inside diameter, D_3 inches (cm)	12.0(30.48)	13.25(33.65)	15.25(38.74)
Flow area, A_4 in^2 (cm^2)	113.1(729.7)	137.9(889.7)	182.6(1178.)
Vent pipe area ratio, $\alpha = A_4/A_1$	3.91	4.77	6.32
f for full turbulence	0.0130	0.0128	0.0125
fL/D_3	0.648	0.580	0.493
λ_3 (eq. 12, sub-sonic solution)	0.604	0.618	0.638
F_1 (eq.10)	0.627	0.627	0.627
G_1 (eq.3, $\lambda_1 = 1$)	2.0	2.0	2.0
G_3 (eq.2)	2.260	2.236	2.205
$(^3\alpha - 1)p$ (eq.25)	0.163	0.148	0.128
Valve orifice area A_1 (eq.8), in^2(cm^2)	3.60(23.2)	3.60(23.2)	3.60(23.2)
Valve pipe area ratio, $\beta = A_4/A_1$	8.028	8.028	8.028
P_{04}/P_{01} (eq.28, $F_1 = F_4$)	0.1246	0.1246	0.1246
P_a/P_{01}	0.00525	0.00525	0.00525
p (eq.29)	0.0421	0.0421	0.0421
α	4.87	4.52	4.04
$\lambda_3' = 1/\lambda_3$	1.656	1.618	1.567
F_3' (eq.10)	0.379	0.402	0.433
$\alpha F_3'/F_1$	2.95	2.90	2.79

All vent pipe sizes satisfy the thermodynamic limit since, in all three cases, $\alpha F_3'/F_1 \geq 1$. Comparing the calculated values of α vs. A_3/A_1 for the assumed pipe size, the 14 inch vent pipe is the smallest vent pipe with adequate flow area to preclude blowback. Therefore, the 14 inch pipe would be selected.

CONCLUSIONS

The proposed vent pipe analysis provides a simple, convenient design method that is consistent with modern fluid dynamics. However, a one-dimensional analysis of the vent pipe system has certain limitations. A multi-dimensional analysis, in conjunction with experimental data to correlate the results, could provide the basis for establishing design margins. These would then be applied to the one-dimensional analysis results in the design of a safety valve piping system. Until such data becomes available careful judgment must be exercised in applying the one-dimensional procedures.

After completing this paper, the writers became aware of a similar analysis by Liao (26) for flow in a safety valve system. A brief review indicates that Liao combines real and perfect gas relationships in the saturated steam case, uses one-dimensional equations based on Mach Number rather than velocity ratio, and expresses the blowback criterion in terms of the momenta at vent pipe sections 1 and 2 in Figure 1.

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