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# SUBCOOLED AND SATURATED LIQUID FLOW THROUGH VALVES AND NOZZLES

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## ABSTRACT:

This article discusses the decrease in flow capacity of valves and nozzles as the temperature of the flowing liquid increases. It was found that the mass flow capacities of both, valves and nozzles will typically decrease to one-third of their value when going from a highly subcooled liquid flow to saturated liquid flow. Test results of subcooled and saturated liquid flow through industrial valves and nozzles are presented. A new mass flow rate model for flow capacity prediction of valves and nozzles exposed to subcooled and saturated liquid flow is developed.

## INTRODUCTION:

Compressed liquefied gases such as propane, chlorine and ammonia, for example, are transported and stored in a large variety of pressure vessels and containers. These vessels and containers are equipped with safety or pressure relief valves for protection against overpressurization. Typically the pressure vessels are not completely filled with liquid in order to accommodate liquid expansion. The safety and pressure relief valves are installed on top of the pressure vessel, i.e., in the vapor space of the partially liquid-full container. The valves are sized and set (adjusted) for vapor flow. Under certain accident conditions, for instance, when the tank or vessel tips over, the state of the substance flowing through the valve may not be vapor but is saturated or subcooled liquid. For safety or other design calculations it is important to be able to predict the flow rates of subcooled and saturated liquids through valves. Flow capacity ratings for liquid flow through such valves are often determined on the basis of cold water tests. Cold water, however, is a highly subcooled liquid and is unlikely to flash into vapor during its passage through the valve or nozzle. Flow capacity ratings obtained from cold water tests are erroneous, often yielding over predictions of mass flow rates by a factor of more than two, if such predictions are applied to saturated or slightly subcooled liquid flows.

This article presents experimental results which quantify the decrease in liquid flow capacity of safety and pressure relief valves as the temperature of the flowing liquid is increased from highly subcooled states to the saturation state. An equation for the prediction of the mass flow rate of saturated and subcooled liquids is derived and compared to the experimentally found values. In addition the effect of the nozzle length to nozzle diameter ratio on the flow capacity was experimentally investigated.

## DESCRIPTION OF VALVES AND NOZZLES:

The valves used in the experiments were typical nozzle-type safety relief valves (see Figure 1). The smallest cross-sectional area which the fluid passes when flowing through such a valve is the

so-called orifice. The orifice is the valve component which determines the flow capacity rating. For nozzle-type safety relief valves the so-called orifice would be more appropriately described as nozzle as is seen in Figure 1.
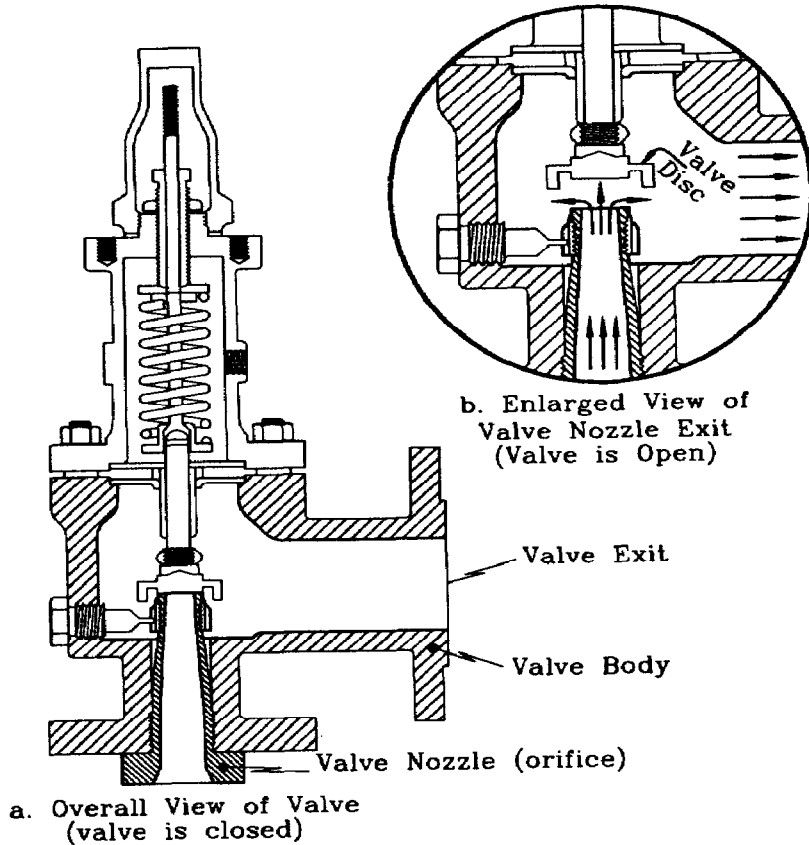


b. Enlarged View of
Valve Nozzle Exit
(Valve is Open)

Valve Exit

Valve Body

Valve Nozzle (orifice)

a. Overall View of Valve
(valve is closed)

Fig. 1. Typical Nozzle-Type Safety Valve

The valves tested had the orifice sizes $D$ and $F$ and the nozzles tested had the orifice sizes $D$, $E$ and $F$. The industrial standard orifice sizes of $D$, $E$ and $F$ refer to nominal orifice diameters of 0.375 inch (9.53 mm), 0.500 inch (12.70 mm) and 0.625 inch (15.88mm). The corresponding actual smallest areas within the nozzles were 0.123 inch$^2$ (0.738 cm$^2$) for the $D$ nozzle, 0.196 inch$^2$ (1.176 cm$^2$) for the $E$ nozzle and 0.342 inch$^2$ (2.053 cm$^2$) for the $F$ nozzle. The stated nominal diameter is always somewhat larger than the actual diameter. While the nozzles of size $D$ and size $F$ were those taken directly from the respective commercial safety valves, the $E$ sized nozzle was specially made and had the additional feature that its length to diameter ratio could be varied. This nozzle was tested for flow capacity varying the nozzle's length to diameter ratio from $L/D = 5$ to

*L/D* = 20. The inner surface of the nozzles as supplied by the valve manufacturer had a mirror like surface finish . The effect of surface roughness was tested by destroying the smooth surface finish with coarse sand paper. A pattern of scratches up to 1/64 inch (0.4mm) deep resulted.

## TEST APPARATUS AND PROCEDURE:

The valve test facility used permits testing of valves, nozzles, orifices, and short pipes with compressed air, or steam, or air-water mixtures or steam-water mixtures. Other, non toxic substances may also be used as a test fluid, up to a pressure of 200 psia (1.38 MPa).
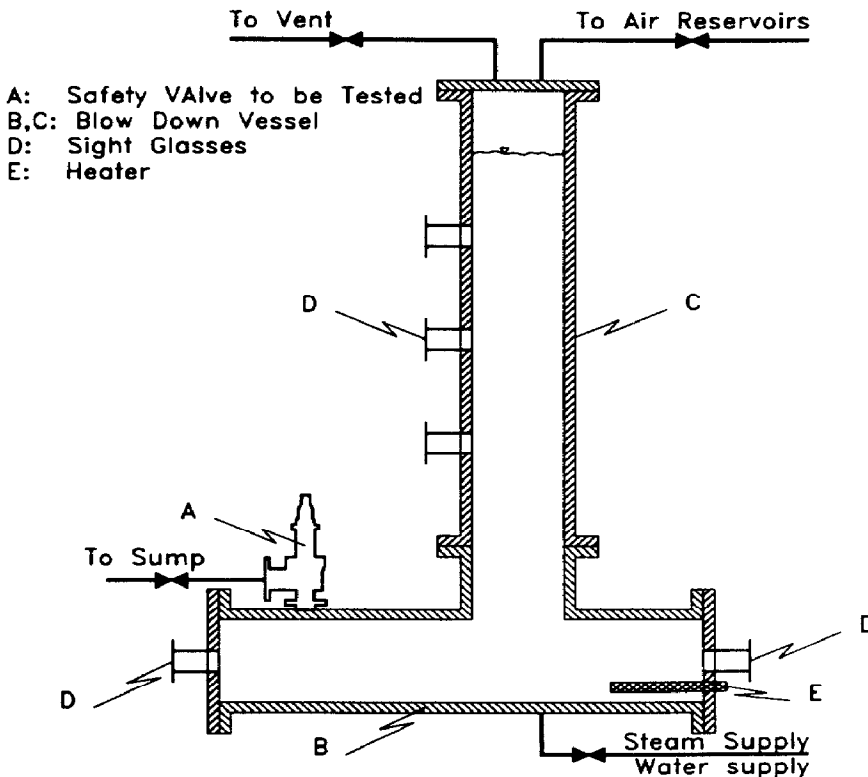
Fig. 2. Sketch of Test Assembly

The valve to be tested (see item *A*, Figure 2) is mounted to a blowdown vessel which consists of a vertical eight inch nominal (approximately 20 cm) stainless steel pipe, 96 inches (244 cm) high (item *B*, Figure 2), with a 36 inch (91 cm) long tee (item *C*, Figure 2) attached to the bottom flange of the vertical pipe. The vessel is equipped with 12 sight glasses (items *D*, Figure 2) for observing the fluid in the vessel and the flow into the valve. The valve itself is mounted vertically upward (as it should be) on a four inch N.P.T. flange welded to the tee. The volume used for blow down is approximately 2.5 cubic feet (0.07 m³).

A thermocouple and pressure transducer are located near the valve inlet, a pressure transducer is connected to the valve body (opposite the valve disc), a differential pressure transducer is connected to the top and bottom of the blow-down tank (the stand pipe), and a displacement transducer is connected to the valve stem. Each transducer is connected to a data acquisition system which is controlled by a desk-top computer. Each transducer voltage is read once a second and stored in computer memory.

All tests reported in this articles were performed with subcooled or saturated water as the flowing medium. Three pressure-temperature ranges were selected for the present investigation, namely:

Pressure $P_1$ = 60 psia (413.7 kPa)

Temperature range: 60 °F - 293 °F ( 15.6 °C - 145.0 °C)

Pressure $P_2$ = 80 psia (551.6 kPa)

Temperature range: 60 °F - 312 °F( 15.6 °C - 155.6 °C)

Pressure $P_3$ = 100 psia (689.5 kPa)

Temperature range: 60 °F - 328 °F ( 15.6 °C - 164.4 °C)

Before the start of a typical blow-down test additional air reservoirs are filled with compressed air and the gag is placed on the valve stem. The gag is a simple locking bracket to keep the valve closed. The additional reservoirs have a total volume of 70 ft$^3$ (2 m$^3$) and are connected to the to of the blow-down vessel. Water is added to the blow-down tank and steam is simultaneously injected. Both water and steam are added until the tank is full and approximately saturated at atmospheric pressure. Heating is continued with consequent rise in pressure until the desired temperature is reached. Then the steam is shut off, the vent is closed, and an air pressure (generally within 5 psia (34.5 kPa) of the saturation pressure) from the additional reservoirs is applied to the blow-down vessel. The air pressure prevents the vessel pressure from dropping too quickly below a preset level and thereby insures steady state tests of sufficient duration. A check valve prevents flow of liquid from the blow-down vessel to the air reservoirs. The gag of the valve is knocked off and the timer is started. During the blowdown phase an oscilliscope trace is made of the valve displacement and observations of valve chatter are made. When the liquid level reaches the lower most sight glass the timer is stopped. Subcooled liquid flow results by setting the pressure in the additional air reservoirs above the saturation pressure of the liquid water in the blowdown vessel. A low quality mixture up to a quality of 1 to 1.5% can be formed in the blowdown vessel by setting the pressure in the additional air reservoirs below the saturation pressure. The quality can be directly calculated from the volume flow and mass flow measurements taken. The estimated accuracy of the measurements taken is as follows: pressure ±1.5 psi (10.4 kPa), temperature ±0.5 °F (0.3 °C), cold water flow rate ±1%.

EXPERIMENTAL RESULTS:

Figures 3 through 6 present the experimentally found decrease in mass flow rate as the temperature of the flowing liquid increases to the saturation temperature. For the test series described in this article no air was injected into the water of the blow down vessel.
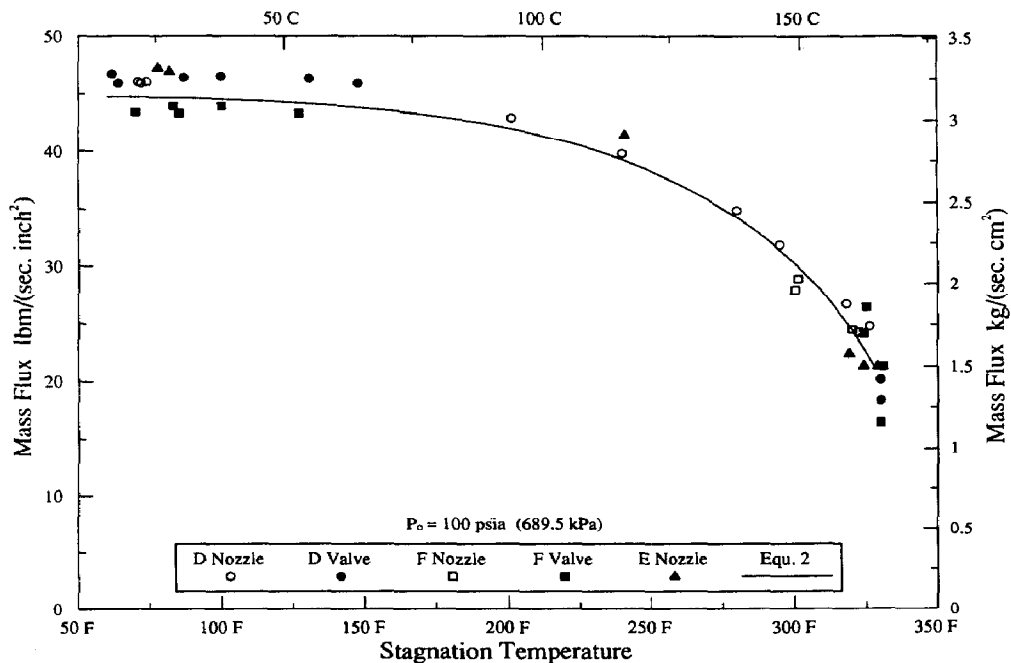
Fig. 3. Measured Water Flow Through Nozzles and Valves; Nominal Vessel Pressure: 100 psia
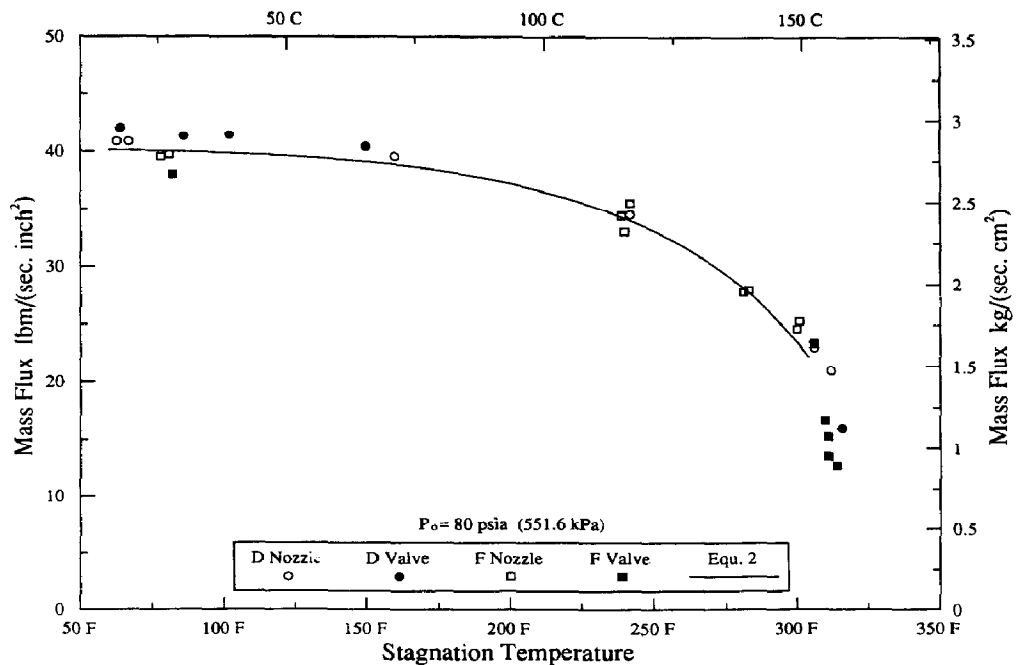


Fig. 4. Measured Water Flow Through Nozzles and Valves; Nominal Vessel Pressure: 80 psia
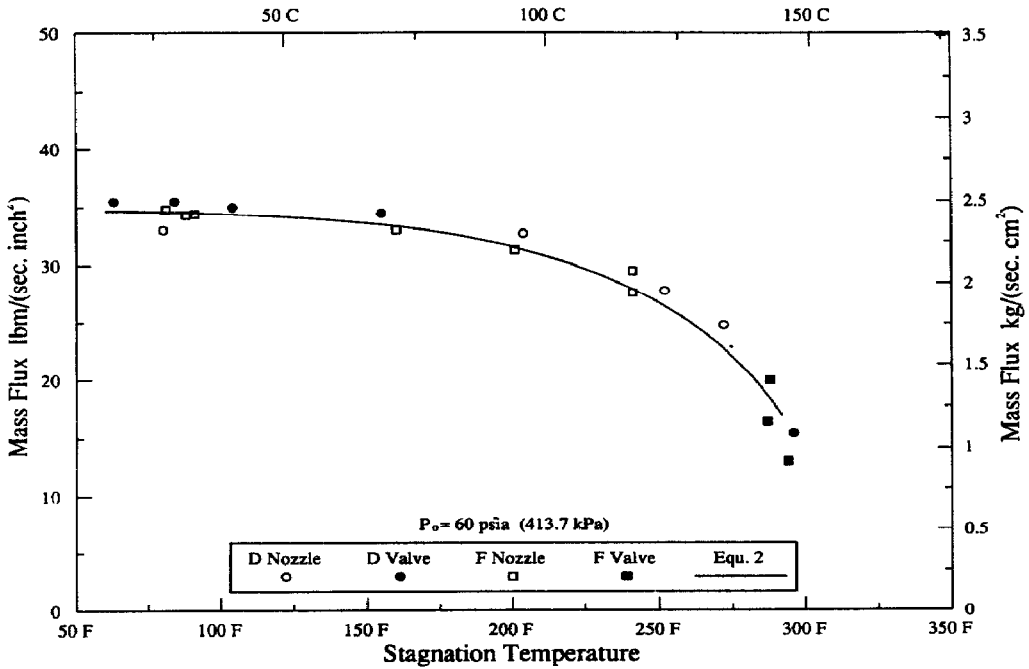
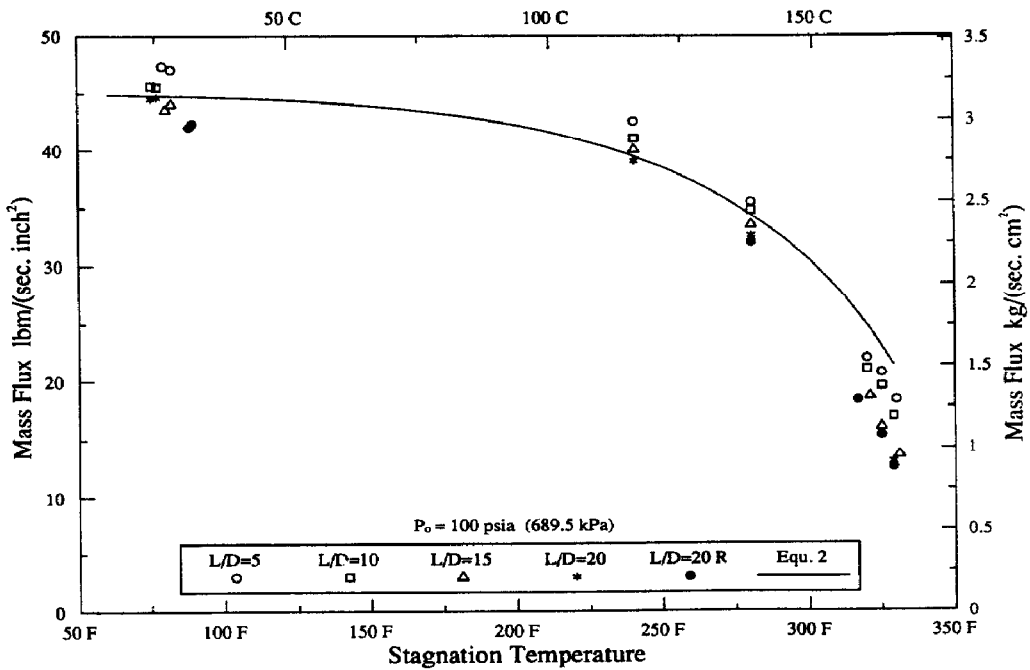Fig. 5. Measured Water Flow Through Nozzles and Valves; Nominal Vessel Pressure: 60 psia



Fig. 6. Effect of Relative Nozzle Length *L/D* on Flow Rate; Nominal Vessel Pressure: 100 psia

Figure 3 gives the measured mass flow rates of liquid water from a vessel at pressure at $P_o$ = 100 psia (689.5 kPa) to a downstream sump at atmospheric pressure. The water temperature in the vessel was varied from 60 °F (15.6 °C) to 328 °F (164.4 °C). The experimental results for flow from vessels with pressures of $P_o$ = 80 psia (551.6 kPa) and of $P_o$ = 60 psia (413.7 kPa) are given in Figures 4 and 5.

The solid lines in Figures 3 through 6 were calculated with equation 2, using the same flow coefficient of $C_v$ = 0.85 for both, valves and nozzles and using a $K$ value of 0.75. The use of this particular $K$-value with Equation 2 was seen to lead to the best representation of the experimental data. The flow coefficient $C_v$ was established in a series of auxiliary valve tests with cold water. The respective temperature ranges are 60 °F (15.6 °C) to 312 °F (155.6 °C) and 60 °F (15.6 °C) to 293 °F (145.0 °C) for Figures 4 and 5.

FLOW MODEL DEVELOPMENT:

The mass flow rate of and incompressible fluid through a nozzle or valve is given by the equation

$$\rho A V = C_v A \sqrt{2g_c \rho (P_o - P_e)} \tag{1}$$

where $\rho$ is the density in kg/m³ (lbm/ft³); $V$ is the velocity in m/s (ft/s), $P$ is the pressure in Pa (lbf/ft²), $g_c$ is 1 (Ns²/(kg·m)) (32.2 lbf·s²/(lbm·ft)), $A$ is the flow cross-sectional area in m² (ft²) and $C_v$ is the flow coefficient for the nozzle of valve (dimensionless).

In equation (1) the pressure $P_o$ refers to the vessel or upstream stagnation pressure and the pressure $P_e$ is the static pressure of the fluid where its velocity is $V$ and the cross-sectional flow area is $A$.

Suppose the flowing medium is not an incompressible, non-cavitating fluid but is a real liquid such as water or liquefied propane. Real liquids will cavitate (boil) once the pressure decreases to the saturation pressure $P_{sat}$ corresponding to the temperature of the given liquid. In other words, the downstream pressure $P_e$ in equation (1) can not be arbitrarily low but should reflect the saturation pressure of flowing liquid. Rewriting equation (1) for the mass flow rate per unit area $G$ = $\rho V$ [kg/(s·m²)] or [lbm/s·ft²)] and accounting for the limiting pressure by setting $P_e = KP_{sat}$ yields

$$G = C_v \sqrt{2g_c \rho_o (P_o - K P_{sat})} \tag{2}$$

where $P_{sat}$ is the saturation pressure corresponding to the temperature of the liquid in the upstream vessel and $\rho_o$ is the density at vessel conditions.

For the determination of the factor $K$ the flow of boiling (saturated) liquid through nozzles and valves is considered. From saturated liquid conditions the maximum obtainable mass flux is calculated by expanding isentropically to lower and lower pressures until the so -called critical mass flux $G_l^*$ is reached. A further decrease in downstream pressure does not increase the mass flux. This type of flow model is called the *HEM* model, (see Ref. 1). The calculation of $G_l^*$ according to the *HEM* two-phase flow model, however, stipulated the liquid-vapor mixture to be in thermal equilibrium. Only flow through long pipes permits thermal equilibrium to be established. The flow through nozzles and valves is usually so rapid, that so-called frozen flow exists, i.e.,

thermal non-equilibrium is dominant (see for instance Ref. 2). This accounts for the fact that when a boiling liquid is vented to the atmosphere through a short nozzle the mass flow rates are 2 to 3 times as high compared to venting through a pipe with the same effective inside diameter. To account for the thermal non-equilibrium conditions a multiplier $N$ is introduced, namely:

$$G^*_{non-equ.} = NG^*_l \tag{3}$$

where $G^*_l$ is the critical mass flux from the saturated liquid state calculated by isentropic expansion as described in the two-phase flow literature (e.g. see Ref. 1). The condition that subcooled liquid mass flow rates through nozzles and valves decrease with increasing temperatures to the non-equilibrium mass flow rates $NG^*_l$ at saturation gives the following expression for $K$,

$$K = \frac{P_o}{P_{sat.}} - \frac{N^2(G^*_l)^2}{2g_c \rho_o P_{sat.}} \tag{4}$$

The calculation of the critical mass flux $G^*_l$ is cumbersome for two reasons. One, it is iterative since the pressure at which $G^*_l$ occurs is not appriori known. Two, the available thermodynamic data to generate the necessary mass flux data is for most substances too inaccurate and not of sufficient precision to permit the determination of $G^*_l$. For example, employing published and widely used thermodynamic data for such substances as propane or propylene may yield errors in the calculation of the critical mass flux $G^*_l$ exceeding 100% in magnitude, as discussed by Sallet et al., (Ref. 3). A recently developed correlation (see Ref. 2), however, is sufficiently accurated to be applied here, namely

$$G^*_l = G^*_v[\sqrt{(\rho_{ol}/\rho_{ov})P_R}/(1.22T_R^{3/2})] \tag{5}$$

where $\rho_{ol}$ is the density of the saturated liquid, (lbm/ft$^3$) or (kg/m$^3$); $\rho_{ov}$ is the density of the saturated vapor, (lbm/ft$^3$) or (kg/m$^3$); $T_r$ is the reduced temperature $T_R = T_o/T_c$; $T_o$ is the saturation temperature; $T_c$ is the thermodynamic critical temperature, $(R)$ or $(K)$ and $G^*_v$ is the critical mass flux for flow from saturated vapor stagnation conditions, [lbm/(s·ft$^2$)] or [kg/(s·m$^2$)].

The vapor critical mass flux of a large variety of substances including propane, water, ammonia, chlorine vinyl chloride propylene is accurately given by the following correlation, see Sallet (Ref. 4):

$$G^*_v = \sqrt{\left[\frac{P_c M}{RT_c}\right]P_c g_c[0.167(P_R)^5 + 0.534(P_R)^{1.95}]} \tag{6}$$

where $G^*_v$ is the critical mass flux reached from saturated vapor stagnation conditions, [lbm/(s·ft$^2$)] or [kg/(s·m$^2$)]; M is the molecular weight of the substance $P_c$ is the thermodynamic critical pressure, (lbf/ft$^2$) or (Pa); $P_R$ is the reduced pressure, $P_R = P_o/P_c$, (dimensionless); $P_o$ is the stagnation pressure (saturation pressure), (lbf/ft$^2$)-or (Pa); R is the universal gas constant:

$$R = 1545.0\frac{ft \cdot lbf}{lbmol \cdot R} \text{ or } R = 8314.4\frac{J}{kmol \cdot K}$$

$T_c$ is the thermodynamic critical temperature, $(R)$ or $(K)$ and

$$g_c = 32.174\frac{lbm \cdot ft}{lbf \cdot s^2} \text{ or } 1\frac{kg \cdot m}{N \cdot s^2}$$

## DISCUSSION:

The idea to predict saturated liquid flow by an equation similar to that of equation (2) originally stems from Burnell (Ref. 5). Burnell suggests that

$$K = 1 - 0.264 \frac{\sigma(\text{at} T = T_{sat})}{\sigma_{ref.}} \tag{7}$$

where $\sigma$ is the surface tension of water at the saturation temperature corresponding to the upstream vessel pressure, $\sigma_{ref}$ is the surface tension of saturated water at $P = 175$ psia (1.21 MPa) and the factor 0.264 is and experimentally found constant. Burnell's equation is only applicable to water flow and is valid over the pressure range from ambient to $P_o = 175$ psia (1.21 MPa). Although experiments and flow prediction development were concerned only with boiling (saturated) water flow within the give pressure range, Burnell's equation can also be used to predict water flow in the subcooled regime for pressures less than 175 psia (1.21 MPa). In the present article the factor $K$ was related to the general critical mass flux $G_l^*$ in order to develop a flow rating equation which is applicable to the flow of commonly used industrial liquefied gases in addition to water flow rating and which is applicable to a much larger pressure range and is also suitable for flow through valves in addition to nozzle flow.

Experience has shown that the maximum mass flux through short nozzles and valves of initially saturated liquids is about three times the critical mass flux $G_l^*$ which is calculated on the basis of the *HEM* (see refs 2 and 6).
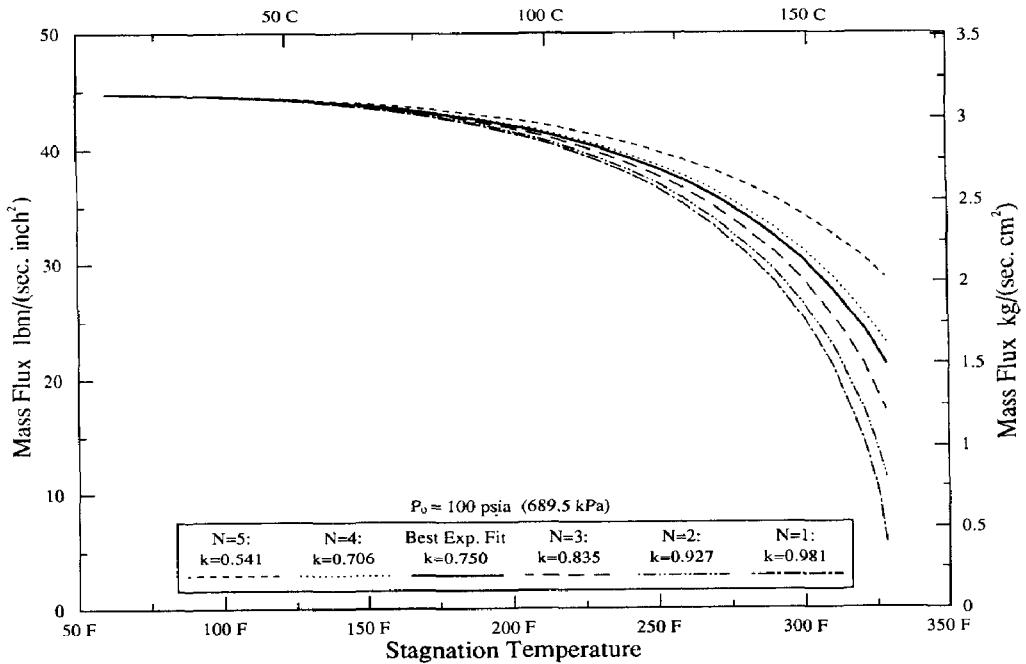


Fig. 7. Comparison of Calculated Mass Flux with Experimental Results.
Note: Water at $P_o$=100 psia (689.5 kPa); valve or nozzle coefficient $C_v$=0.85;

Figure 7 supports this rule of thumb. The solid line represents a good fit of the experimental data which were previously presented (see Figures 3 to 6). This line falls between the flow predictions for $N = 2$ and $N = 4$. (see equations 2 to 4). Chosing $N = 2$ is equivalent of saying that the non-equilibrium conditions cause the mass flux to be twice the value of the *HEM* prediction while $N = 4$ means that the non-equilibrium maximum mass flux is four times the *HEM* prediction.

The simple correlation for the *HEM* mass flux of diverse liquefied gases from saturated liquid conditions ($G_i^*$) given by Equation (5) is accurate to $\pm 11\%$. A more accurate correlation equation was recently developed by Sallet (Ref. 4) and is given by the expression

$$M_R = -0.5 \log P_R + 1.16 \tag{8}$$

where $M_R = (G_i^*/G_v^*)^{1/3} (\rho_{ol}/\rho_{ov})^{1/2} (T_R)^{-3/2}$

Equation (8) is valid over the pressure range $0.1 \leq P_R \leq 0.5$ and correlates critical flow data of the most commonly used liquefied gases within $\pm 5\%$. The critical mass flux correlation given by Equation (6) has a range of maximum error (depending upon the substance) between 1.4 and 6.0 percent. The average error range is from 0.7% for chlorodifluoromethane to 4.8% for the 23 points calculated, equally spaced within the reduced pressure range $0.1 \leq P_R \leq 0.9$.

The experimental findings presented in this article can be summarized as follows:

1.  The liquid mass flow rates through valves and nozzles decreased to approximately one third of their value when the temperature of the flowing liquid was increased from a highly subcooled state to saturation conditions.

2.  The size of the valve does not influence the mass flux (mass rate per unit area) for given upstream fluid conditions, provided the ratio of valve exit area to valve nozzle area is large enough.

3.  The mass flux through unrestricted flow nozzles and through valves which have as an integral part these identical flow nozzles is the same if the upstream fluid conditions are the same. The slight variations of mass flux for different valves and nozzles at the same temperature and nominal pressure are principally due to the fact that the nominal pressure is not exactly equal to the actual pressure at nozzle inlet. The actual pressure at nozzle inlet varied slightly from test run to test run and also varied slightly during a test run.

4.  The length to diameter ($L/D$) ratio had only a slight effect on the mass flux of saturated and subcooled water within the tested range of $5 \leq (L/D) \leq 20$. Increasing the roughness of the inner surface of the nozzle has no noticeable effect on the flow rates within the measured accuracy.

## ACKNOWLEDGEMENTS

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