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# Critical Pressure Ratio and Critical Flow Rate in a Safety Valve

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## Abstract

Subcooled water critical flow phenomena in a safety valve were investigated experimentally at various subcoolings with 3 different disk lifts. The valve inlet test condition is about 10 bar and subcoolings are between 10℃ and 125℃. Flow patterns are picturized at the critical status and pressure/flow characteristics in the test section are also reviewed. It turns out that critical flowrate and critical pressure ratio are considerably affected by different subcoolings while the effect of disk lifts on them are relatively small. Non-dimensional disk lift, subcooling and pressure are selected to develop the correlation on critical pressure ratio, non-equilibrium factor, and critical mass flux. The non-equilibrium critical flow correlation for the safety valve is developed based on Fauske´s non-equilibrium model and the present experimental data. Its root-mean-square error is within 5% compared with the experimental data.

## I. Introduction

Physical phenomenon of flowrate and chocking is not very well understood when a gas and a liquid phase are flowing simultaneously through a safety valve. In the case of safety valve, the geometrical and subcooling parameters are very influential on the mass flowrate and on the pressure topology. Several studies have been conducted to clarify the two-phase critical flow for various geometrical and thermodynamic parameters[1,2]. The thermodynamic nonequilibrium state affected by the geometry is not clear. As the critical condition in a valve strongly depends on the nonequilibrium factor, the observed phenomenon becomes so complicated. Henry and Fauske developed the nonequilibrium homogeneous two-phase flow critical model[3]. In the model, the critical flowrate can be obtained from the energy equation frozen at the inlet quality and it is assumed that the vapor which is formed is saturated at the local pressure. This research focuses on the visualization of flow patterns, critical pressure ratios, and choked flowrate in a safety valve, including the effect of variable receiver pressures, disk lifts and subcoolings. The developed flow model based on the experimental results accounting for the nonequilibrium effect is compared with other analytical critical flow models of safety valve.

## II. Thermal Non-equilibrium and Subcooled Flow Model

Flashing of liquid into vapor, if thermal equilibrium is maintained, occur as soon as the liquid moves into a region at a pressure lower than its saturation pressure. Flashing, however, could be delayed because of the lack of nuclei about which vapor bubbles may form, surface tension which retards their formation, due to heat transfer problems, and other reasons. When this happens, a case of meta-stability is said to occur. Meta-stability occurs in rapid expansion, particularly in short flow channels, nozzles and orifices. The case of short channels have not been completely investigated analytically. For sharp edged orifices, the pipe length to diameter ratio($L/D$) is 0, the experimental data showed that because residence time is short, flashing occurred outside the orifice and no critical pressure existed. The flow is determined from the incompressible flow orifice equation:

$$G_{cr,Pout} = K_d \sqrt{2\rho_f(p_o - p_b)} \quad \text{for } L/D = 0, \tag{1}$$

where $K_d$ is a valve discharge coefficient (0.61 is recommended for the case of $L/D$ = 0), and $\rho_f$ $p_o$ and $p_b$ denote inlet fluid density, valve inlet pressure and back pressure, respectively. Flow calculation results on a safety valve can be severely affected by the valve coefficient and this coefficient is usually provided by valve manufacturer using their own experimental results. Fauske performed various experiment on the 0.25 inch inner diameter channels with sharp edged enterance for $L/D$, between 0(an orifice) and 40. The critical pressure ratio was found to be 0.55 for long channels in which the $L/D$ ratio exceeds 12. For $L/D$ values greater than zero, he recommends to use the following equation:

$$G_{cr,Pcr} = K_d \sqrt{2\rho_f(p_o - p_{cr})} \quad \text{for } L/D > 0, \tag{2}$$

where $p_{cr}$ is the throat critical pressure determined by experimental data.

In the absence of significant frictional losses, Fauske proposed the following correlation considering the non-equilibrium factor in SI unit:

$$G_{cr,corr} = \frac{h_{fg}}{v_{fg}}\sqrt{\frac{1}{F(NE)Tc_f}}\,, \tag{3}$$

where $F(NE)$ = non-equilibrium factor, $h_{fg}$ = vaporization enthalpy(J/kg), $v_{fg}$ = change in specific volumn(m$^3$/kg), $T$ = absolute temperature (K), $c_f$ = specific heat of the liquid(J/kg K).

Another critical flow correlation considering subcooling effect was suggested by Fauske. The effect of subcooling on the discharge rate is simply obtained by accounting for the increased single-phase pressure drop $[p_o - p(T_o)]$ resulting from the subcooling, where the subscript $o$ refers to stagnation conditions. The critical flow rate can be stated as

$$G_{cr,Fauske} = \sqrt{2\rho_f K_d^2[p_o - p(T_o)] + G^2_{cr,Pcr}} \quad \text{for } 0 < L/D < 3. \tag{4}$$

When the value of $L/D$ is smaller than 3, $G^2_{cr,Pcr}$ calculated by Eq. (2) and $p(T_o)$ denotes stagnation saturation pressure. If we know the critical pressure and the enthalpy of the water-steam mixture, the critical flow can be calculated by above four equations but there are two unknown values: critical pressure and non-equilibrium factor. These values can be determined from experimental data.

## III. Experimental Apparatus and Procedure

The experimental main apparatus consist of boiler tank, collection tank, nitrogen gas delivery system and test section. A scheme of the experimental facility and test section is shown in Fig. 1.

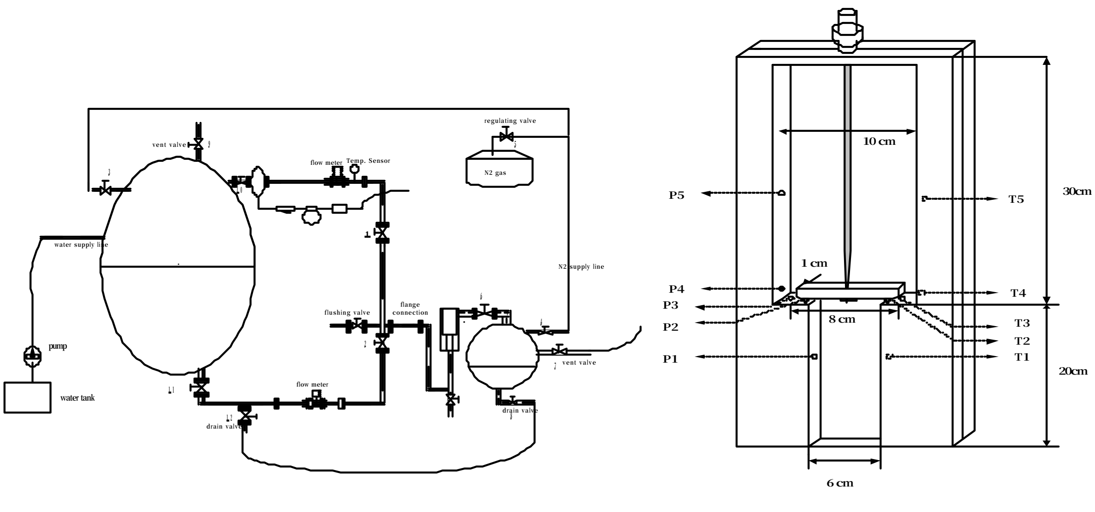


Fig. 1 Schematic drawing of experimental facility and test section

These main equipments are connected by 1 inch valves and pipings. The capacity of boiler tank is 1.3 cubic meters and 100 kW heater submerged in the tank bottom area. Water temperature of the tank can be controlled automatically by setting the temperature of controller. The test section enables to visualize the flow patterns at the inlet and outlet of the valve disk. The disk position of the valve model can be fixed to any desired lift from 0 to 6 mm by using the top located micrometer. The boiler tank is filled with cold water and it is heated up to the required temperature by the submerged heater. The tank pressure is maintained constant during the experiment by means of compressed nitrogen. The pipe line is preheated by means of slow flow of water originating from the boiler. The mass-flow rate is measured by the hot water flow transmitter located at the horizontal pipe line below the boiler tank. Temperatures are measured by 5 temperature transmitters which located in the test section, and the boiler tank and the collection tank temperature monitored also by temperature gages. Pressure measurements are made by 5 pressure transmitters in the test section and two pressure taps are holed at the inlet and outlet pipes of the test section. In this experiment, instrumentation on hot water flow, pressure and temperature is summarized as Table 1.

Table 1. Instrumentation

| Variable | Range | Accuracy | Type | Number |
|---|---|---|---|---|
| Hot Water flow | 300 - 6000 ℓ/hr<br>(0.3 - 6 $m^3$/hr) | 1 % | Vortex | 1 |
| Pressure | 0 - 20 bar | 0.1 % | D. P | 7 |
| Temperature | 0 - 200℃ | 0.1 % | K-Type | 5 |

The maximum disk lift is limited by hot water flow-meter. To analyze the disc lift and subcooling effects on the critical flow parametrically, disc inlet pressures maintain as close as 10 barg, subcoolings ranged between 10℃ and 125℃, and disc lifts selected between 1 and 3 mm. Table 2 shows the hot water test matrix.

Table 2. Test matrix and name of each experiment

| Lift \ Temperature | 55℃ | 125℃ | 145℃ | 165℃ |
|---|---|---|---|---|
| 1mm | 1mm-55℃ | 1mm-125℃ | 1mm-145℃ | 1mm-165℃ |
| 2mm | 2mm-55℃ | 2mm-125℃ | 2mm-145℃ | 2mm-165℃ |
| 3mm | 3mm-55℃ | 3mm-125℃ | 3mm-145℃ | 3mm-165℃ |

## IV. Experimental Results and Discussion

## 1. Hot Water Flow Trend for the Pressure Ratio of $P_{out}/P_{in}$ and $P_{nozzle}/P_{in}$

One important critical phenomenon is related to the flow trend for the pressure ratio of $P_{out}/P_{in}$ and $P_{nozzle}/P_{in}$ Detailed review of pressure and flow trend, two chocking conditions were established for the hot water experiments. In these experiment, the inlet pressure was held at a value greater than saturation pressure of the water. Figures 2 and 3 show flow and pressure ratio trend of two typical different subcooled water tests.

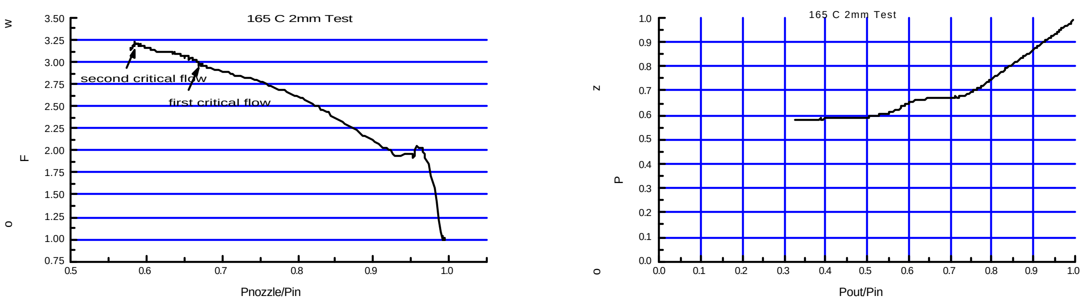


Fig. 2 The characteristics of flow and pressure ratios of 165℃ hot water

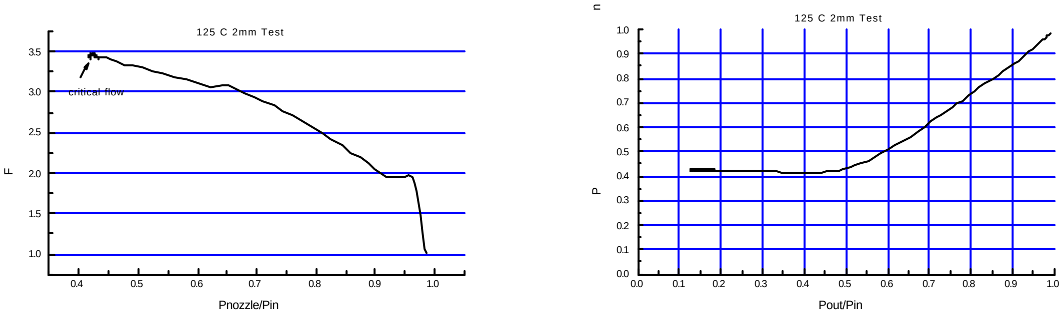


Fig. 3 The characteristics of flow and pressure ratios of 125℃ water

From these pictures, it was evident that cold waters(below 125 ℃) have only one critical pressure ratio but hot waters(above 145℃) have two critical pressure ratios. These phenomena give some important conclusions that for the low subcooling or saturated waters. One constant nozzle pressure characteristic does not exist as that occurred in a single phase flow and we could find second constant nozzle pressure characteristic with the decreasing of back pressure. Even though nozzle pressure values remains constant as the outlet pressure decreases, the flow rate increases with the decreasing of back pressure

$P^{*}= P_{out}/P_{in}$(bar), (0.10 ~0.33)

$T^{*}= T_{sub}/T_{in}$(K), (0.03 ~0.38) (5)

$L^{*}$= Disk Lift / Seat Length(mm), (0.2 ~0.6)

where $P_{out}$ = disk outlet pressure(1 ~3 bar), $P_{in}$ = disk inlet pressure(8.5 ~10 bar), $T_{in}$ = valve disk inlet water temperature(50 ~165℃), $T_{sub}$ = $P_{in}$ saturation temperature minus $T_{in}$(10 ~125℃). During the test, the constant nozzle pressure characteristic exists when the downstream pressure decreases sufficiently. This pressure is called as the critical pressure( $P_{cr}$). The critical pressure ratios of $P_{cr}/ P_{in}$ for the different $T^{*}$ and $L^{*}$ are shown in Fig. 5.

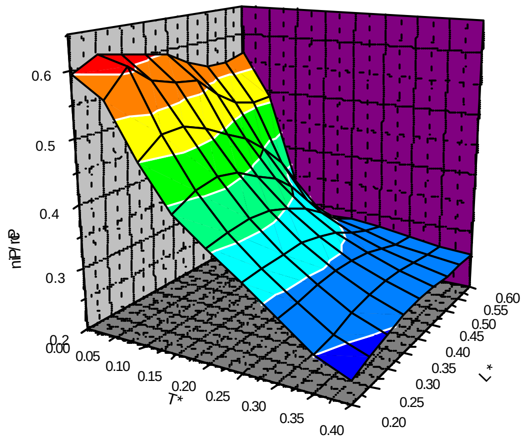


Fig. 5 The critical pressure ratios of $P_{cr}/ P_{in}$ for the different $T^{*}$ and $L^{*}$

These pressure ratios of high subcooled water( $T^{*}$=0.4) were approximately 0.25 and low subcooled water( $T^{*}$=0.05) were about 0.6. These values increase as the subcooling margin or disk lifts decrease and low subcooled water values of $P_{cr}/ P_{in}$ are almost the same with the theoretical critical pressure ratio of wet saturated steam as 0.58[5]. Figure 5 also shows these critical pressure ratios with respect to the disk lift. The critical pressure ratios of low lift cases are much higher than those of the high lift case. When 3 mm test results are compared with 1 or 2 mm test results, that phenomenon is clearly shown. However, the effect of disk lift on the critical pressure ratio is relatively smaller than that of the subcooling. Using the experimental results, the correlation of the critical pressure ratio is developed as follows:

$$P_{cr}/P_{in} = 0.04828(P^{+})^{-0.4605}(T^{+})^{-0.5646}(L^{+})^{-0.0818} \quad (6)$$

Figure 6 shows the comparison of the critical pressure ratio correlation with measured values.

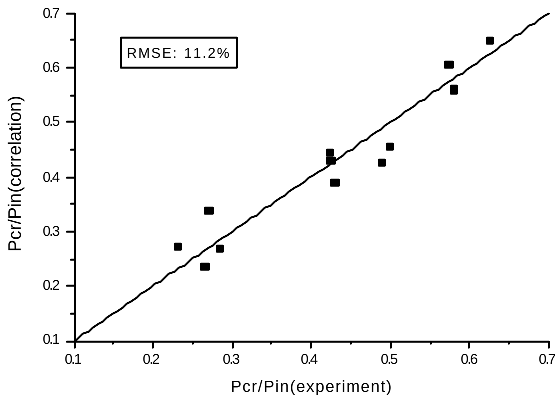


Fig. 6 Comparison between results from critical pressure ratio correlation with measured data

There is in good agreement between the results from the developed correlation and the experimental data with the root mean square error of 11.2%.

## 3. Non-equilibrium Factors and Critical Mass Flux

The critical flowrate depends on inlet pressure, outlet pressure, subcooling and disk lift. The expected mass flowrate will be increased with the increasing of inlet pressure and subcooling margin. These effects can be explained by non-equilibrium factors of fluid. The Fauske model, which accounts for nonequilibrium vapor generation by introducing an empirically found non-equilibrium coefficient $F(NE)$, is considered here. The constant $F(NE)$ describes the partial phase change at the throat. The critical flow equation considering the non-equilibrium factor is described in Eq. (3). In this equation the unknown parameter is only $F(NE)$ value and the other values are determined by the inlet fluid conditions. Therefore, using the experimental results and this equation, we can easily get the non-equilibrium factors of $F(NE)$ values as follows:

$$F(NE) = \left(\frac{h_{fg}}{v_{fg}}\right)^2 \frac{1}{Tc_f} \frac{1}{(G_{cr,exp})^2} \quad (7)$$

Where $G_{cr,exp}$ is the measured maximum mass flux. Using this equation, the measured non-equilibrium values for the different $T^*$ and $L^*$ are suggested in Fig. 7.

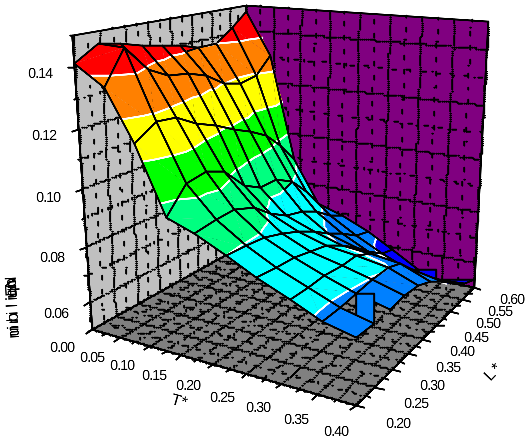


Fig. 7 Non-equilibrium values according to the $T^*$ and $L^*$

This picture also shows that the disk lift effect on the non-equilibrium factor is relatively smaller than the subcooling effect. By using the same non-dimensional parameters, the non-equilibrium function $F(NE)$ of experiment is obtained by

$$F(NE) = 0.01519(P^*)^{-0.3186}(T^*)^{-0.5047}(L^*)^{-0.1826}. \quad (8)$$

The root mean square error of this correlation is 9.4%. Therefore, $F(NE)$ equation also shows good agreement with the experimental results. Figure 8 denotes the comparison of the non-equilibrium factor correlation with the measured data.

In a critical status, the measured maximum mass flux for different $T^*$ and $L^*$ are shown in Fig 9. From Eqs. (8) and (3), the critical flow correlation considering the non-equilibrium factor can be written as

$$G_{cr,corr} = \frac{h_{fg}(P_{in})}{v_{fg}(P_{in})}\sqrt{\frac{1}{F(NE)\,T(P_{in})\,c_f}}\,. \quad (9)$$

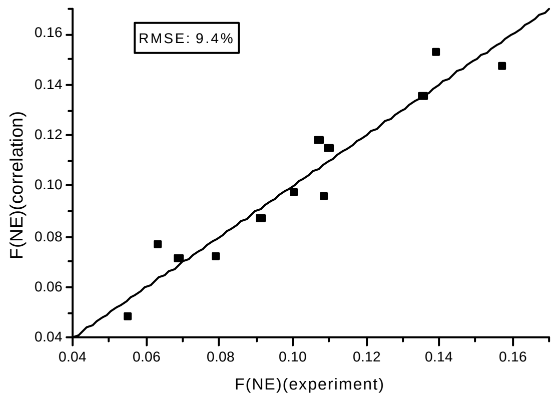


Fig. 8 Comparison between results from non-equilibrium factor correlation with measured values

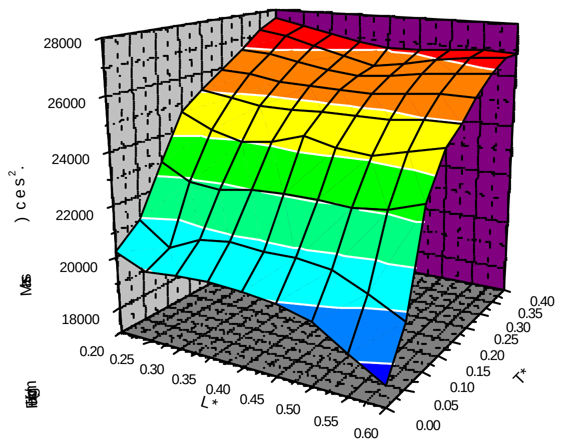


Fig. 9 Measured maximum mass flux for the different $T^*$ and $L^*$

Considering the hot water test ranges, the variation of specific heat of hot water is from 4.16 to 4.32 kJ/kgK. Therefore, the average value of $c_f$ is set to be approximately 4.2kJ/kgK. The uncertainty error of $c_f$ is from -1.38% to +0.47%. Calculation results from the present correlation and other correlations such as orifice flow correlations, Fauske's subcooled equation are compared with experimental data in Fig. 10. The orifice flow equation using the back pressure value( $G_{cr,Pout}$) estimates the critical flow higher than measured values for the low subcooling ranges. The orifice flow equation using the critical pressure value( $G_{cr,Pcr}$) gives the smallest values and predicts the critical flow

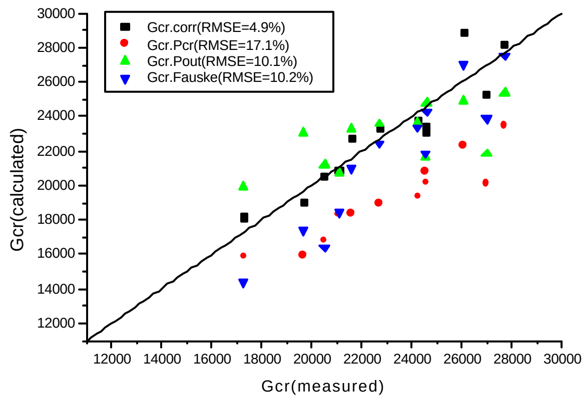


Fig. 10 Comparison between results from critical flow correlations with measured data

less than experimental data. Even though Fauske´s subcooled critical flow equation( $G_{Cr.Fauske}$) shows good agreement with experimental critical flow data but slightly underestimates them. The root mean square error of developed correlation ( $G_{Cr.corr}$) is 4.9 %. Therefore, this equation shows good agreement with the experimental results.

## V. Conclusions

This study is to provide new experimental results of subcooling water critical characteristics in a safety valve. The general critical characteristics of subcooling water for the different disk lifts could be summarized as follows.

(1) Voids are initiated in the disk surface areas for both low and high subcooling waters and critical pressure ratio as one of the indications of the critical status is characterized by constant nozzle pressure at the throat of the test section with the decreasing of back pressure. The critical pressure ratio is increased with the decreasing of subcoolings and disk lifts.

(2) Experimental results show that variation of mass flux for the different disk lift is negligible but subcooling effect should be considered in calculation of safety valve mass flux.

(3) The developed critical flow correlation considering the non-equilibrium factor estimates experimental critical flow data less than 5% errors when the inlet pressures are between 8.5~10 bar, disk lifts between 1~3 mm, and the subcoolings between 10~125℃.

**References**

[1] L. Bolle, P. Downar-Zapolski, J. Franco, J.M. Seynhaeve, "Experimental and Theoritical Analysis of Flashing Water Flow through a Safety Valve", J. Hazardous Materials 46(1996) 105-116.

[2] Osakabe,M, Isono,M, "Effect of Valve Lift and Disk Surface on Two-Phase Critical Flow at Hot Water Relief Valve", International Journal of Heat and Mass Tansfer, May 1,1996, v39, n8.

[3] R.E. Henry and H.K.Fauske, "The Two-Phase Critical Flow of One-Component Mixtures in Nozzles, Orifices, and Short Tubes", J. Heat Transfer, Trans. ASME, 95-179(1971).

[4] F. R. Zaloudek, "The Critical Flow of Hot Water through Short Tubes", HW-77594, 1963.

[5] D.W.Sallet, " Vapor Flow Rating of Valves Using Small Pressure Reservoirs", Nuclear Engineering and Design 72(1982) 321-327.

and this phenomenon can be seen more effectively for the low subcooling hot water experiments. This increasing trend can be explained by two effects. First, the inlet pressure is increased compared with initial condition and secondly, the water temperature is decreased compared with initial condition. During the test, inlet pressure and temperature are maintained to be constant values, so that these effects may be negligible. F. R. Zaloudek performed the critical flow test for the hot water through the tube shorter than 1.5 inch. His experiment showed a four-step flow trend and two constant flow regimes defined as first-step critical and second-step critical[4]. Another critical flow effect which we can consider is flow streamlines of disc outlet sections as shown Fig. 4.

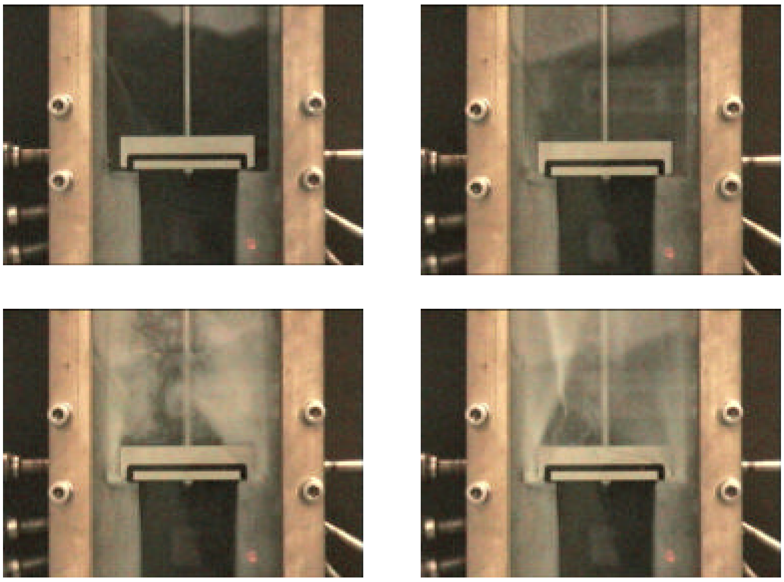

Fig. 4 Flow patterns of 163℃, 3mm hot water with respect to the different back-pressure

From this picture, it can be seen that the void fraction(or vapor velocity) at the downstream section is increased with the decreasing of back-pressure. Another important phenomenon is the throat outlet temperature become to higher than the saturation temperature of the throat pressure when the subcooling is small and the disk lift is high. Therefore, even though constant throat pressure condition is established, the critical flow rate is affected by these phenomena.

## 2. Critical Pressure Ratio

The measured critical pressure ratios can be defined by a function of valve inlet pressure, inlet temperature, pressure difference of valve in and out, and disk seat length and lift. The non-dimensional expression of these parameters are defined by the following equations: