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NON-EQUILIBRIUM DISCHARGING FLOW FROM SAFETY VALVES

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ABSTRACT

The discharging flow rate from the safety valve at different lifts and disk materials was verified with the simple disk-type flow contraction. The water discharging flow rate was measured and compared with the Bernoulli model. The discharging coefficient of water was approximately 0.6 in spite of the lift, which was mainly due to the vena contraction at the minimum flow area. The air discharging flow rate was also measured and compared with the non-equilibrium expansion delay model. The air flow rate could be well predicted with the vena contraction ratio as same as the water flow and the expansion delay factor. Furthermore the discharging flow rate with flashing was measured at the different inlet subcooling or two-phase quality. The non-equilibrium flashing flow model directly using the steam table was proposed to obtain the critical mass flux at the vena contraction. The comparison of the experimental results with the non-equilibrium model indicated the significant non-equilibrium in the flashing flow at the vena contraction in spite of the disk material. The subcooled liquid and two-phase discharging behavior was carefully observed by using the transparent disk. The observation also supported the significant non-equilibrium. The two-phase discharging flow rate at the strong non-equilibrium was well correlated with the simple pressure loss correlation using the specific volume of water.

KEY WORDS: Two-phase/Multiphase flow, Boiling and evaporation, Energy and environmental systems, Non-Equilibrium model, Flashing flow, Safety valve, Vena contraction, Discharging coefficient

1. INTRODUCTION

The safety or relief valves to depressurize the pressure vessel are recognized as the most important safety devices for boilers and nuclear facilities. In the severe accident at Fukushima nuclear power plant of 2011, the discharging flow rate from the safety valves significantly affected the accident sequence and the corresponding operation. The increased pressure of containment vessel prevented the opening of valves and the ensuing partial opening reduced the discharging flow rate. It was pointed out that the precise estimation of the flow rate not only at the full opening of valve but also the partial opening is the key issue of the severe accident.

The discharging flow is usually restricted with the vena contraction of minimum flow area between disk and seat. The flow rate is significantly affected with the vena contraction, the delay of expansion and the non-equilibrium. In the previous study [1], the non-equilibrium flashing flow model using the approximation of steam condition was proposed by present author. Schmidt [2] proposed the correlation for non-equilibrium factor by using the huge data of European valve experiments. Recently this correlation was adopted as the regulation formula in the international standard, ISO [3], and Japanese industrial standard, JIS [4], for safety valves. But the detail of expanding or flashing flow through the vena contraction of safety valve is not well understood due to the sophisticated non-equilibrium phenomena. So the discharging flow rate of water, air and flashing two-phase through the simple disk-type flow contraction simulating the safety valves was measured. Furthermore the non-equilibrium flashing flow model directly using the steam table was proposed to obtain the critical mass flux at

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the vena contraction. The more precious prediction can be available compared to the previous model [1] using the approximation of steam condition.

When safety or relief valves are open, a fluid is discharged through the disk-type flow contraction. The contraction is due to the minimum flow area between disk and seat (pipe end), which is called as the curtain area of valve as shown in Fig.1. The curtain area A is defined as,

$$A = Ld\pi \tag{1}$$

where L is the valve lift and d is the valve inlet diameter. The valve seat is defined as the surface of pipe end which contacts to the disk at the closing of valve. The flow rate of discharging fluid is restricted with the curtain area when the lift L is less than $d/4$. The discharging flow rate is usually smaller than the flow rate calculated with the curtain area due to the vena contraction. The vena contraction is strongly affected with the valve lift and the seat configuration [5].

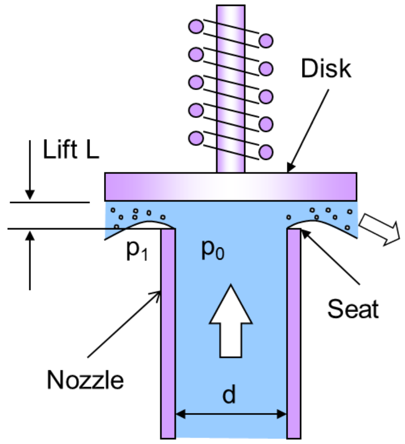

Fig.1 Disk-type flow contraction

The vena contraction also can be seen in the flow through orifice. Generally when a fluid flows through orifice, the flow rate is approximately 60% of the calculation using the minimum flow area in the subsonic region. So in the actual design of orifice, the discharging coefficient of approximately 0.6 is usually used to calculate the flow rate. However the vena contraction is mitigated in the thick orifice. In such the cases, the discharging coefficient becomes larger than 0.6 [5]. The difference between the orifice and disk-type contraction is the flow-directional change at the contraction. The discharging coefficient of approximately 0.6 is also obtained in the disk-type contraction but the longer passage between seat and disk like the thick orifice gives the larger values than 0.6 [5]. It has a possibility that the rapid flow-directional change enhances the non-equilibrium behaviour of disk-type contraction.

When the subcooled liquid or two-phase flow enters into the contraction, the phenomena become much more complicated due to the phase change. The phase changing flow rate was affected not only with the vena contraction but also with the sophisticated critical flow rate which is different that of single-phase flow. Furthermore the critical flow rate is strongly affected with the non-equilibrium as known as the delay of boiling.

The discharging coefficient obtained in single-phase flow can be used in the prediction for the flashing flow rate of subcooled liquid from safety valves [1]. The international standard, ISO, recommends the discharging coefficient of 0.65 both for subcooled liquid flow [3]. For two-phase flow, the higher value than the liquid flow is recommended because the discharging coefficient of gas is usually larger than liquid. However, the regulation by American Petroleum Institute, API, gives 0.65 for single-phase liquid and 0.8 for two-phase flow[6]. It is suggested that the vena contraction for two-phase was different from that for single-phase liquid flow. It should be noted that the non-equilibrium behaviour and the vena contraction in the disk-type contraction is not well understood. The disk-type contraction has the flow-directional change at the contraction which is different from the usual flow-directional contraction such as the orifice. For the correct regulation of safety and relief valves, it is very important to investigate the behaviour of discharging single and two-phase flow under the clear boundary conditions in this study.

2. EXPERIMENTAL APPARATUS AND METHOD

Shown in Fig.2 is the schematic of experimental apparatus. Water is supplied from the water tank to the test section after depressurized through the control valve. When the depressurization exceeds the saturation pressure, flashing takes place and two-phase flow is supplied to the test section. The quality of two-phase flow can be obtained with the calculation assuming a constant enthalpy through the control valve. The water tank is connected with steam boiler and air compressor, and can be pressurized up to 0.6 MPa. The flow rate through the test section is measured with the electromagnetic flow meter of which measurement error is within ±0.5%. The meter without a contraction upstream of control valve can prevent the flashing. The pressure is measured with pressure gages of which measurement error is within ±1.25 Pa. T-type sheath thermocouple of 1mm in diameter is used to measure the temperature. The water tank has a water level indicator to confirm the discharging flow rate.

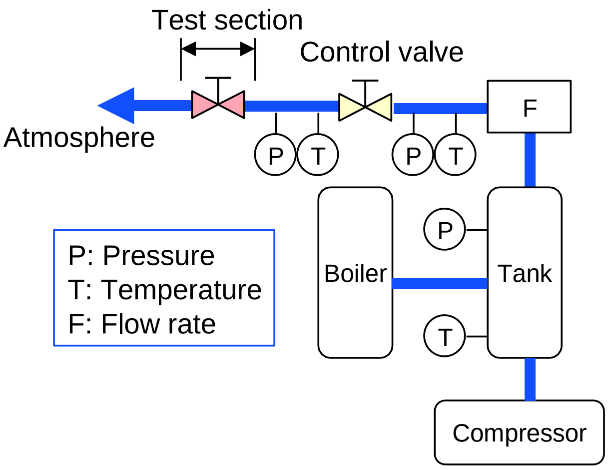

Fig.2 Experimental apparatus

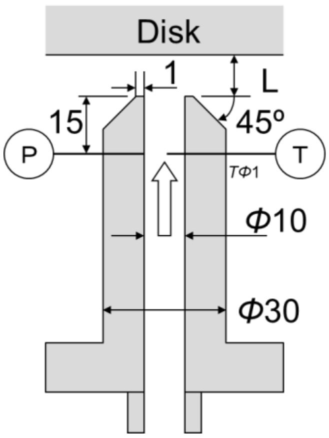

Fig.3 Test section

The test section is the simple disk-type contraction consisted of pipe and disk as shown in Fig.3. The vena contraction is strongly affected with the valve lift and seat configuration [5]. The present test section assures the stable discharging behavior at the present range of lifts. The inner and outer diameters of pipe are 10 and 30 mm, respectively. Thermocouple and pressure gage are installed just before the curtain area as shown in Fig.3. The measured pressure and temperature are used to calculate the discharging coefficient and non-equilibrium. The outside of the curtain area is open to atmosphere. The valve lift between the seat and disk is measured with a narrow gage or a laser distance meter. The valve lift is set at between 0.3 and 2.5 mm in the present experiment. So the ratio of lift to diameter is between 0.03 and 0.25 where the minimum flow area exists at the curtain area. As the material of disk, brass, Teflon and transparent glass are used to provide the different boiling ability and the observation from the disk backside.

In the single-phase discharging experiments, the temperature of water or air was approximately 20°C and the discharging pressure was controlled between 0.1 to 0.6 MPa. On the other hand, the discharging pressure of 0.18 to 0.54 MPa was maintained at the experiment of subcooled water. The subcooling was controlled between 0 and 40 K. The discharging pressure of 0.22 to 0.49 MPa was maintained at the two-phase experiment. The quality was controlled between 0 and 0.055. Basically the experiments were conducted with changing the discharging pressure and temperature step by step. The discharging behavior was recorded with video camera. The observation of discharging flow was conducted both from the side and the backside of transparent disk. The observation from the disk backside is expected to observe the boiling from cavitation or flashing two-phase flow just around the curtain area.

3. EXPERIMENTAL RESULTS

3.1 Single-phase discharging behavior The mass flow rate at the curtain area can be expressed with the discharging coefficient c_v and the enthalpy difference Δh between valve inlet and outlet as,

$$G = c_v \sqrt{2\Delta h / v_1} \tag{1}$$

where v_1 is the specific volume at the outlet. The discharging coefficient approximately indicates the vena contraction ratio at the curtain area which is the minimum flow area. The enthalpy drop can be obtained with the integration of specific volume assuming the isentropic change as,

$$\Delta h = -\int_{p_0}^{p_1} v dp \tag{2}$$

When the fluid is non-compressible as a liquid, the specific volume v_0 is constant and integration simply gives

$$\Delta h = v_0 (p_0 - p_1) \tag{3}$$

So the non-dimensional mass flux is

$$G^* = \frac{G}{\sqrt{p_0 / v_0}} = c_v \left[2(1-\eta_1)\right]^{1/2} \tag{4}$$

where η_1 is the pressure ratio defined as

$$\eta_1 = p_1 / p_0 \tag{5}$$

When the fluid is compressible as a gas and the expansion delay exists, the specific volume v is expressed with the expansion delay factor N as

$$v = N v_e + (1-N) v_0 \tag{6}$$

where v_e is the equilibrium specific volume and v_0 is the inlet specific volume. This non-equilibrium concept is also adapted in the previous non-equilibrium model [1] and ensuing ISO regulation[3]. Assuming the isentropic change as,

$$p v_e^{\kappa} = const. = p_0 v_0^{\kappa} \tag{7}$$

Equation (2) becomes

$$\Delta h = N v_0 p_0 \frac{1}{1-1/\kappa}\left(1-\eta_1^{1-1/\kappa}\right) + (1-N) v_0 p_0 (1-\eta_1) \tag{8}$$

By using the outlet specific volume of

$$v_1 = N v_0 \left(\frac{1}{\eta_1}\right)^{1/\kappa} + (1-N) v_0 \tag{9}$$

in Eq. (1) and (8), the non-dimensional mass flux can be obtained as

$$G^{*} = \frac{c_v\sqrt{2}}{N(1/\eta_1)^{1/\kappa} + 1 - N}\left[N\frac{1}{1-1/\kappa}\left(1-\eta_1^{\;1-1/\kappa}\right) + (1-N)(1-\eta_1)\right]^{1/2} \tag{10}$$

When N=1 in Eq.(10) is for the compressible fluid and N=0 in Eq.(10) is for the non-compressible fluid that is coincident with Eq.(4).

Equation (4) is called as Bernoulli equation for non-compressible fluid. Shown in Fig.4 is the relation of liquid discharging coefficient c_v and pressure difference $\Delta p(=p_0-p_1)$ in the disk-type contraction of this study. The discharging coefficient c_v is obtained with Eq.(4). The dash-dotted line is c_v=0.61 that is same as the ideal fluid flows through a hole of thin orifice. All data at the lift of 0.3 to 2.5mm agree well with c_v=0.61 for the present seat and nozzle configuration. It should be noted that the present test section at the lift of 0.3 to 2.5mm can provide the free flow and vena contraction as same as the thin orifice.

Shown in Fig.5 is the relation of air discharging coefficient c_v and pressure ratio η_1 in the disk-type contraction of this study. The discharging coefficients are calculated with two methods from the experimental data at the lift of 0.5 to 1.2 mm. The close keys are calculated by Eq.(10) with N of 0 and the open keys are calculated with N of 1. The former discharging coefficient is obtained with the assumption of non-compressible fluid and the latter is obtained with the assumption of the compressible fluid. In the calculation of the compressible fluid, the critical non-dimensional mass flux of 0.684 is used at the pressure ratio less than the critical value of 0.528. The coefficient assuming the compressible fluid gradually increases with decreasing the pressure ratio and exceeds 1 at the smaller pressure ratio. The discharging coefficient that is larger than 1 suggests the non-equilibrium discharging behavior. On the other hand, the coefficient assuming the non-compressible fluid is approximately 0.65 that also suggests the strong non-equilibrium or expansion delay. It should be noted that air flows like a non-compressible fluid such as water through the present contraction. It has a possibility that the rapid flow-directional change enhances the non-equilibrium behavior of disk-type contraction.

In ISO regulation [3], the discharging coefficient of gas for the safety valves is mentioned to be higher than that of liquid. The equation for the equilibrium compressible fluid gives the higher value than that of liquid in the present study as shown in Fig.5. But by using the equation for the non-compressible fluid, approximately the same discharging coefficient as that of liquid can be obtained.

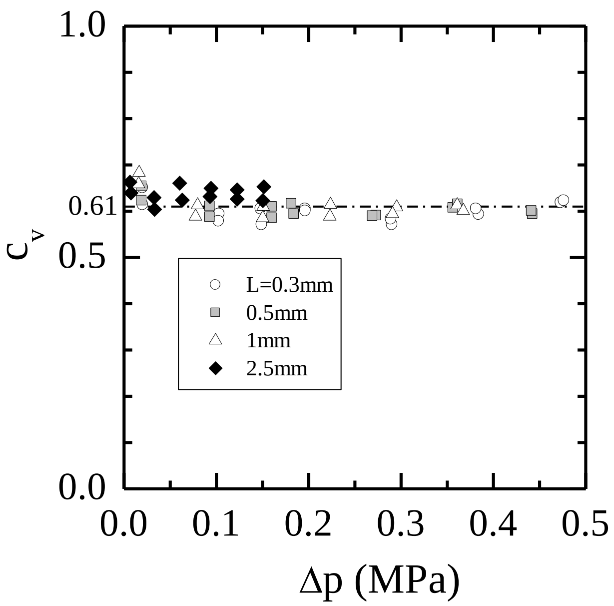

Fig.4 Discharging coefficient of water at different lifts

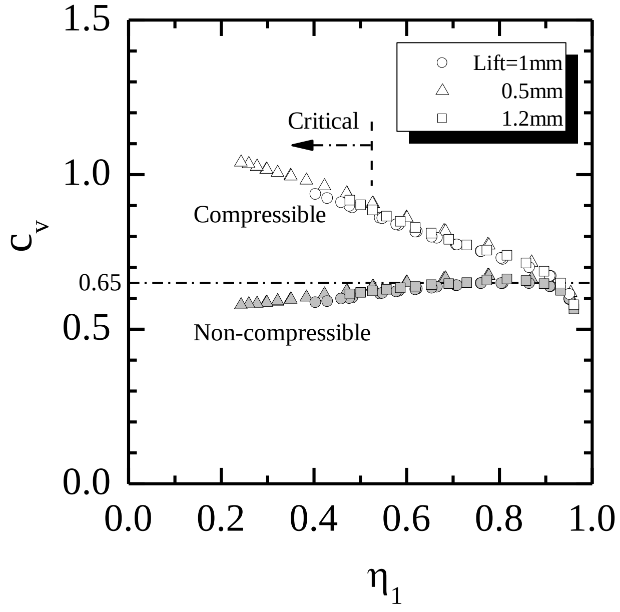

Fig.5 Discharging coefficients defined with compressible and non-compressible equations

Shown in Fig.6 is the relation of non-dimensional mass flux G^* and pressure ratio η_1 in the air discharging experiment at the lift of 0.5 to 1.2 mm. The lines are prediction with Eq.(10) where the discharging coefficient c_v of 0.65 and N of 1 to 0.1 is used. Reducing N in Eq.(10) that means the increase of non-equilibrium, the maximum flow rate increases and the corresponding pressure ratio decreases. The experimental data agree well with the prediction of N=0.1 in spite of the lift. The air flow rate could be well predicted with the discharging coefficient as approximately the same as the water flow and the expansion delay factor. The discharging coefficient is fixed instead of changing the vena contraction ratio in the present model. The effect of the vena contraction can be included in the expansion delay factor. This assumption is preferable at the subsonic flow region when the pressure ratio is higher than the critical value.

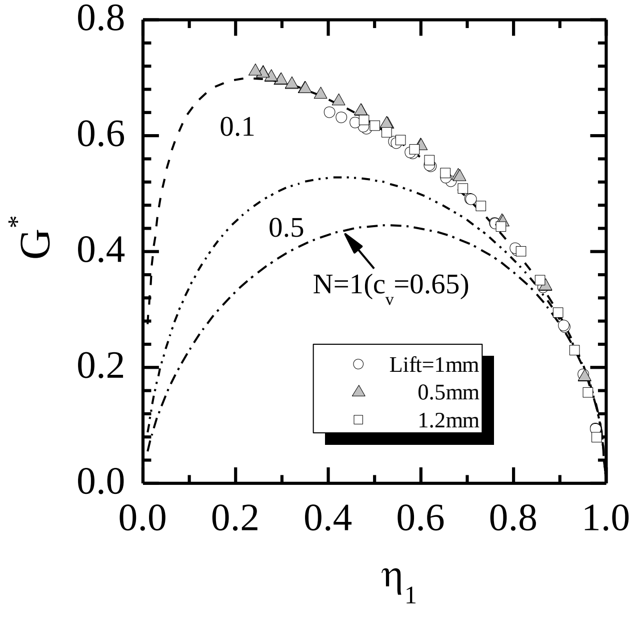

Fig.6 Discharging non-dimensional mass flux of air

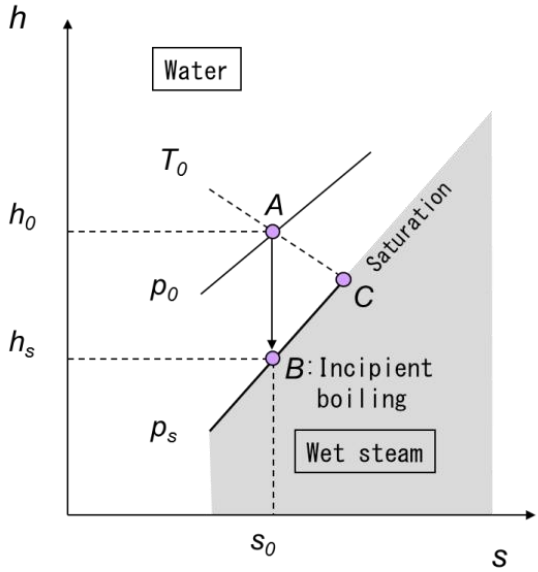

Fig.7 Isentropic change from point A

3.2 Two-phase discharging behavior When the subcooled liquid or two-phase flow enters into the contraction, the phenomena become much more complicated due to the phase change. The phase changing flow rate is affected not only with the vena contraction but also with the sophisticated critical flow condition. Furthermore the critical flow rate is strongly affected with the non-equilibrium as known as the delay of boiling.

When the subcooled liquid at the pressure p_0 and temperature T_0 indicated as a point A in Fig.7 is depressurized adiabatically, the incipient boiling takes place at the point B and the void fraction increases along with the isentropic vertical line. The pressure p_s of incipient boiling can be estimated as the saturated pressure corresponding to the initial temperature T_0. The saturated or incipient boiling pressure p_s at the point B can be approximately estimated with the initial temperature T_0. It should to be noted that the point C of the pressure p_s and the temperature T_0 is different from the point B of incipient boiling. The temperature of B is a little lower than the temperature T_0.

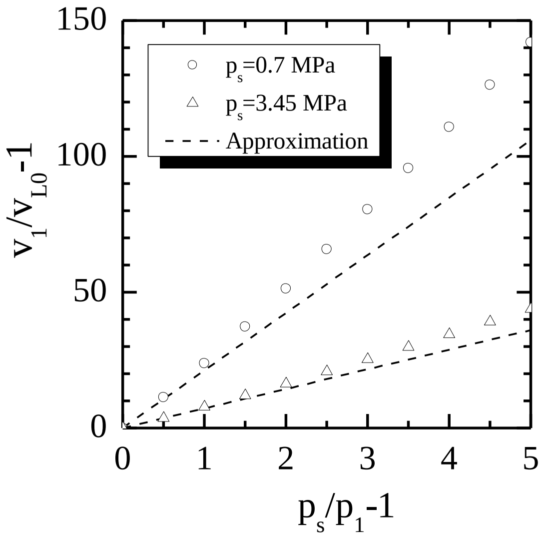

Fig.8 Approximation of equilibrium steam condition (N=1)

The depressurization below the saturation pressure results as the rapid expansion of fluids due to the steam generation. In the previous non-equilibrium model [1], the specific volume v_1 at the pressure p_1 below the saturation pressure p_s is approximated as,

$$\frac{v_1}{v_{L0}} = N\omega\left(\frac{p_s}{p_1} - 1\right) + 1 \tag{11}$$

$$\text{where } \omega = \frac{c_{p0} T_0 p_s}{v_{L0}} \left(\frac{v_{G0} - v_{L0}}{h_{LG0}}\right)^2 \tag{12}$$

The above physical values in Eqs.(11)and (12) can be estimated as the saturation values corresponding to the inlet temperature T_0. So the values at the point C can be used as an approximation. The other physical values are c_{p0} is isobaric specific heat of saturated water, v_{L0} is specific volume of saturated water, v_{G0} is specific volume of saturated steam, h_{LG} is latent heat. Shown in Fig.8 is the comparison of approximation by Eq.(11) and the exact values obtained with steam table when the non-equilibrium parameter N=1. When the pressure drop is small, the approximation agrees well with the steam table. Especially the error is large when the inlet pressure is relatively low.

The non-equilibrium flashing flow model directly using the steam table was proposed to obtain the critical mass flux at the vena contraction in the present study. The more precious prediction can be available compared to the previous model [1] using the approximation of steam condition. In the method directly to use the steam table, the following procedure is taken. First the entropy at the depressurized pressure p_1 is described as,

$$s_1 = x_1 s_{Gs} + (1 - x_1) s_{Ls} \tag{13}$$

where x_1 is quality, s_{Gs} *and* s_{Ls} are the entropy of saturated steam and water, respectively. As this entropy is equal to the inlet entropy s_0 under the isentropic change, the quality can be obtained as,

$$x_1 = \frac{s_0 - s_{Ls}}{s_{Gs} - s_{Ls}} \tag{14}$$

The obtained quality gives the equilibrium specific volume and enthalpy as

$$v_{e1} = x_1 v_{Gs} + (1 - x_1) v_{Ls} \tag{15}$$

$$h_1 = x_1 h_{Gs} + (1 - x_1) h_{Ls} \tag{16}$$

where v_{Gs} *and* v_{Ls} are the specific volumes of saturated steam and water, respectively, at the depressurized pressure p_1. The enthalpy h_{Gs} *and* h_{Ls} are those of saturated steam and water, respectively. So when the depressurized pressure is given, the specific volume and enthalpy of equilibrium state can be obtained.

The enthalpy difference Δh between valve inlet and outlet is described with Eq.(2). In the case of subcooled liquid, the integration can be divided at the saturation pressure p_s into two parts.

$$\Delta h = -\int_{p_0}^{p_s} v_{L0}\,dp - \int_{p_s}^{p_1} v\,dp \tag{17}$$

where v is the specific volume of actual wet steam. As the specific volume v_{L0} of water is constant until the pressure p_s,

$$\Delta h = -v_{L0}(p_s - p_0) - \int_{p_s}^{p_1} v\,dp \tag{18}$$

This equation is for $p_1 < p_s$. At $p_1 > p_s$, the second terms of Eq.(18) becomes 0 as,

$$\Delta h = v_{L0}(p_0 - p_1) \tag{19}$$

When the depressurization is rapid, the boiling delay is considered to exist like the expansion delay of gas. The specific volume v is expressed with the boiling delay factor N as

$$v = N(v_e - v_s) + v_s \tag{20}$$

where N is non-equilibrium (boiling delay) factor, v_e is the equilibrium specific volume, v_s is the specific volume at incipient boiling. The flashing takes place at the complete equilibrium at N=1 but the flashing does not take place at N=0. Substituting Eq. (20) into Eq.(18) yields

$$\Delta h = v_{L0}(p_0 - p_s) + N(h_s - h_1) + (1 - N)v_s(p_s - p_1) \tag{21}$$

When the inlet condition is the two-phase flow, the specific volume after the depressurization can be expressed as,

$$v = N(v_e - v_0) + v_0 \tag{22}$$

The inlet specific volume v_0 is the homogeneous two-phase specific volume using the inlet quality x_0 defined as,

$$v_0 = (1 - x_0)v_{L0} + x_0 v_{G0} \tag{23}$$

So enthalpy drop can be expressed as,

$$\Delta h = N(h_0 - h_1) + (1 - N)v_0(p_0 - p_1) \tag{24}$$

The mass flux G can be obtained with Eq.(1) by substituting the enthalpy drop obtained above. The non-dimensional mass flux G^* is defined with Eq.(3).

The relation of G^* and η_1 when the non-equilibrium parameter N is 1 is shown as the solid line in Fig.9. The model prediction is performed at the subcooling of 10K under the inlet pressure of 0.7MPa. The broken line shows the sonic mass flux calculated with the equation derived from the approximation of steam table[1]. Reducing the pressure ratio η_1, the mass flux increases upto the maximum value at η_s which already exceeds the sonic velocity of two-phase. When the flasing takes place at η_s, the single-phase flow changes to the two-phase flow accompanied with the reduction of the sonic velocity. So at the flashing inception, the subsonic velocity under the single-phase suddenly becomes the surpersonic velocity under the two-phase flow. This sopisticated critical flow condition can be seen specially in the inlet subcooling case. It should be noted that the critical mass flux can always be obtained as the maximum of mass flux.

Shown in Fig.10 is the relation of non-dimensional critical mass flux and inlet subcooling in a safety valve at the inlet pressure of 0.69 MPa.. The experimental data were obtained with the actual safety valve [1]. The lines are predictions with the non-equilibrium model using the steam table where the non-equilibrium parameter N is assumed as 1 to 0.01. Increasing the inlet subcooling, the prediction lines of different N merge into one curve. The non-equilibrium significantly affects the mass flux at the lower subcooling. When the subcooling is large, the effect of non-equilibrium is relatively small. The experimental data agree well with the prediction of N=0.035.

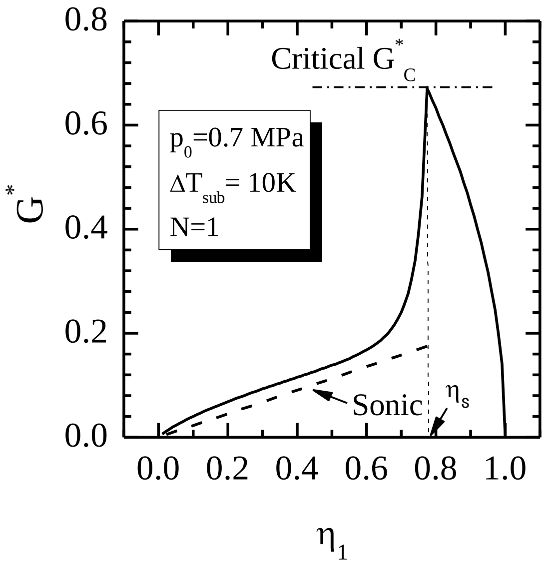

Fig.9 Relation of non-dimensional mass flux and pressure ratio

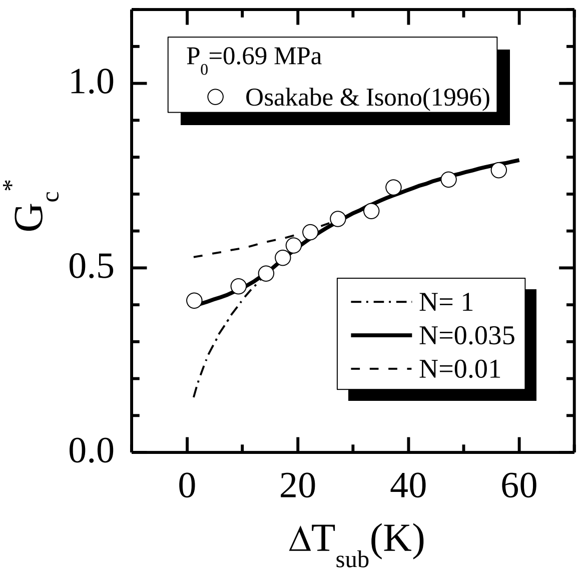

Fig.10 Comparison with previous valve experiment

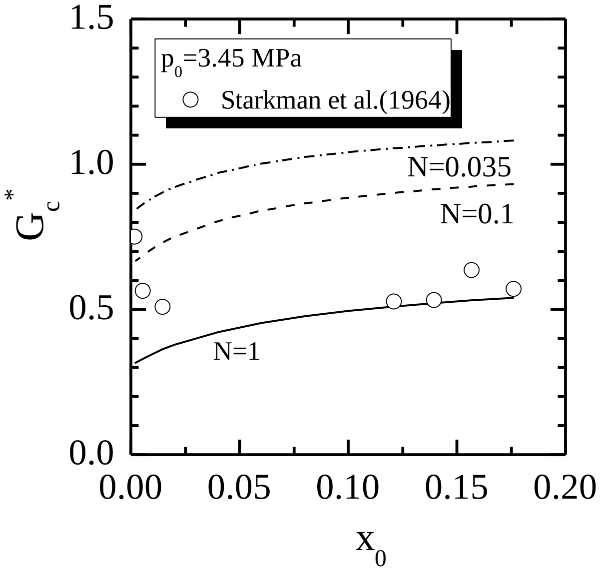

Fig.11 Comparison with previous nozzle experiment

Shown in Fig.11 is the relation of non-dimensional critical mass flux and inlet quality in a nozzle at the inlet pressure of 3.45 MPa. The experimental data were obtained with the typical convergent-divergent nozzles [6]. The discharging coefficient c_v of nozzle can be considered to be approximately 1.The lines are predictions with the non-equilibrium model using the steam table where the non-equilibrium parameter N is assumed as 1 to 0.035. The critical mass flux increases with the decrease of N, which corresponds to the increase of non-equilibrium. The experimental data agree well with the prediction of N=1 at the higher inlet quality and gradually increases at the lower quality. It is considered that the boiling delay is mitigated with the increase of quality as the interfacial surface to generate the steam increases. When the quality is approximately 0, the data tend to agree with the prediction of N=0.035 as same as the flashing of subcooled liquid from the actual valve in Fig.10.

Shown in Fig.12 is the relation of non-dimensional critical mass flux and inlet subcooling. The experimental data were obtained at the lift of 1 mm with the disk of different material at the inlet pressure of 0.18 to 0.58 MPa. Teflon disk is expected to enhance the boiling due to the existence of many cavities on the surface. Transparent glass disk is for the observation of discharging flow from the backside. The lines are predictions with the non-equilibrium model using the steam table where the non-equilibrium parameter N is assumed as 1 to 0.001. In the model, the discharging coefficient of 0.61 was used. The effect of the inlet pressure is relatively small on the non-dimensional mass flux and the inlet pressure of 0.35 MPa was used in the calculation. The most of experimental data agree well with the prediction of N=0.001 and some data near the subcooling of 0 agree with the prediction of N=0.035. The significant difference due to the disk material cannot be observed.

Shown in Fig.13 are the observations of flashing from the backside of transparent disk. The nozzle hole and seat can be seen from the backside of transparent disk. The left photograph is at the pressure of 0.44 MPa and the subcooling of 3.3 K. The white circle indicating the vigorous boiling bubbles locates far from the nozzle hole suggesting the strong non-equilibrium. In the complete equilibrium, the white circle should exist just at the outlet of nozzle hole. This non-dimensional mass flux agrees well with the prediction of N=0.001. The right photograph is at the pressure of 0.41 MPa and the subcooling of 1.2 K. The white bubbles can be seen just after the nozzle hole suggesting the weak non-equilibrium. This non-dimensional mass flux agrees well with the prediction of N=0.035.It is considered that the bubbling behaviour clearly corresponds to the non-equilibrium.

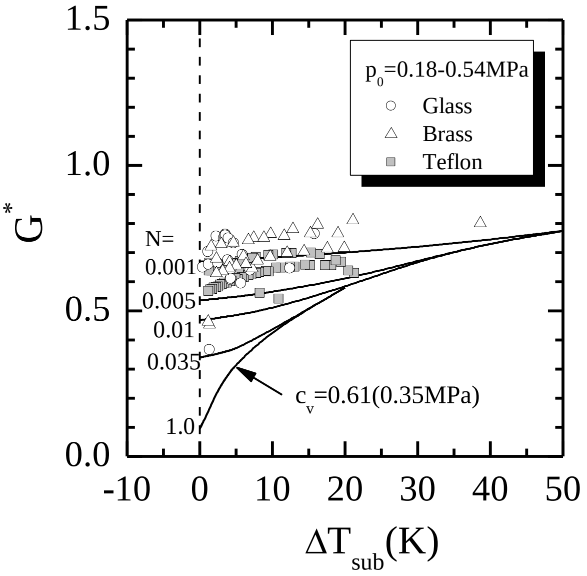

Fig.12 Relation of subcooling and critical mass flux

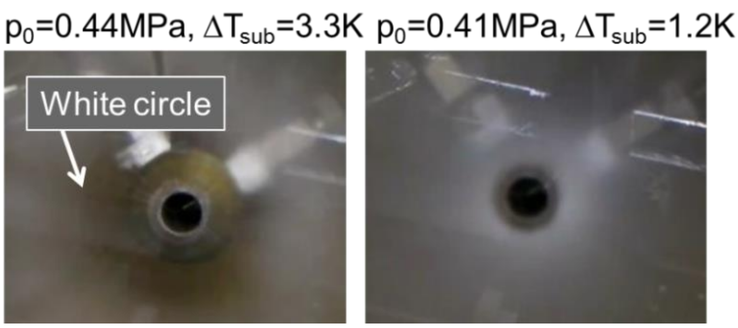

Fig.13 Flashing behaviour of subcooled liquid

When the non-equilibrium is significantly strong, the flashing does not take place at the vena contraction between the seat and disk. The water discharging coefficient is tentatively obtained from the single-phase Eq.(4). Shown in Fig.14 is the discharging coefficient assuming no flashing. The dash line is c_v=0.61 that is same as the water discharging flow. Most data at lift of 1mm agree well with c_v=0.61 but the some data are significantly lower than 0.61 suggesting the flashing.

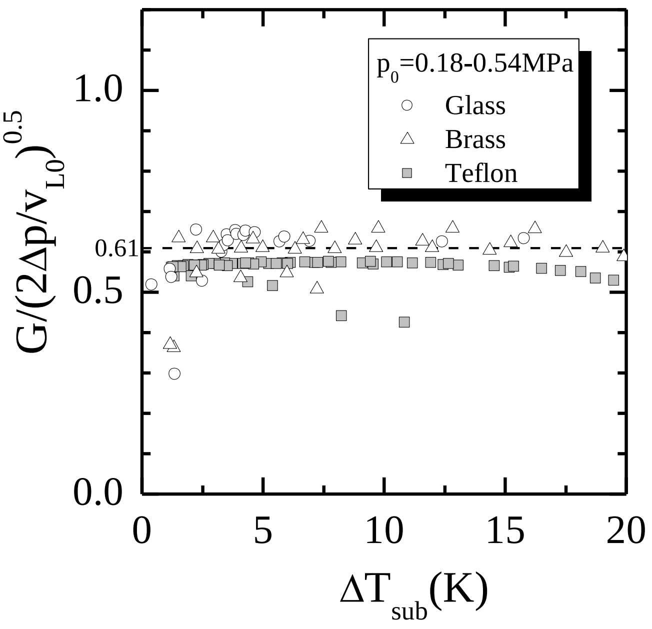

Fig.14 Discharging coefficient of subcooled liquid

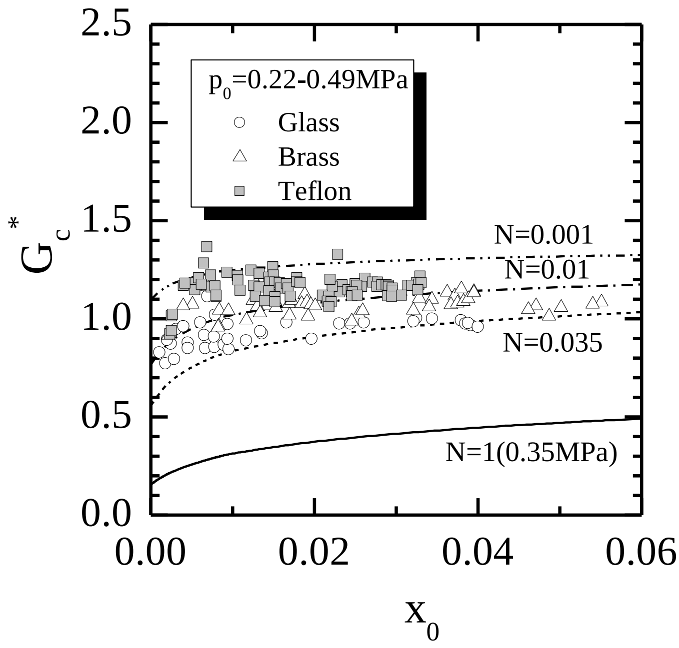

Fig.15 Relation of critical mass flux and inlet quality

Shown in Fig.15 is the relation of non-dimensional critical mass flux and inlet quality. The experimental data were obtained at the lift of 1 mm with the disk of different material at the inlet pressure of 0.22 to 0.49 MPa. The lines are predictions with the non-equilibrium model using the steam table where the non-equilibrium parameter N is assumed as 1 to 0.001. In the model, the discharging coefficient of 1 is used. It is considered that the vena contraction disappears in the oscillatory two-phase flow. ISO and API also recommend the higher value than that of water. The effect of the inlet pressure is relatively small on the non-dimensional mass flux and the inlet pressure of 0.35 MPa is used in the calculation. The most of experimental data locate between the prediction of N=0.001 and 0.035. The significant difference due to the disk material also cannot be observed. In the experiment, the bubbly flow from the nozzle hole was confirmed by using the transparent glass disk.

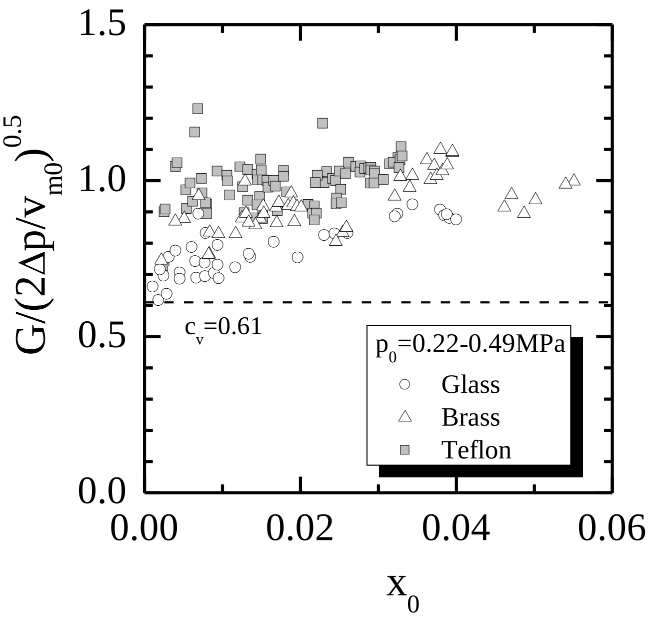

Fig.16 Relation of critical mass flux and inlet quality

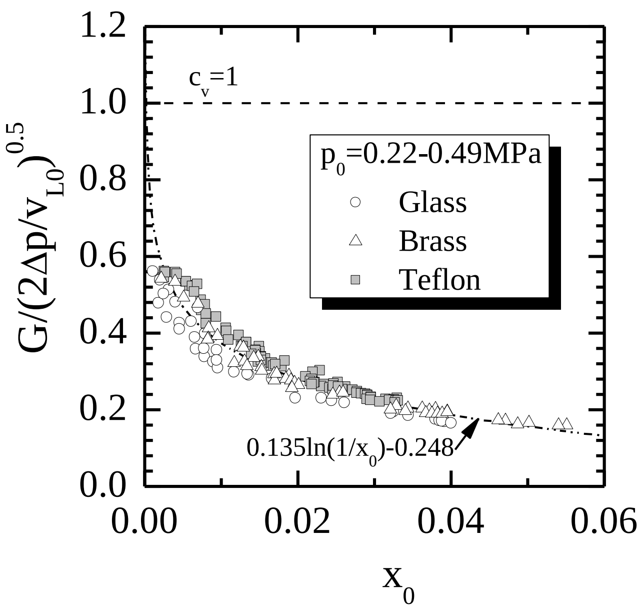

Fig.17 Relation of critical mass flux and inlet quality

When the non-equilibrium is significantly strong, the flashing or expansion does not take place at the vena contraction between the seat and disk. The discharging coefficient is tentatively obtained from the single-phase Eq.(4). Shown in Fig.16 is the discharging coefficient by using the homogeneous specific volume v_{m0} of two-phase flow. The dash line is c_v=0.61 that is same as the water discharging flow. Most data at the lift of 1mm is larger than c_v=0.61 and scatter widely. The discharging coefficient by using the homogeneous specific volume of two-phase flow is difficult to correlate.

Shown in Fig.17 is the discharging coefficient c_v by using the specific volume v_{L0} of water. The experimental data gradually decrease with increasing the inlet quality described as

$$c_v = \frac{G}{(2\Delta p / v_{L0})^{0.5}} = 0.135 \ln(1/x_0) - 0.248 \tag{25}$$

When the non-equilibrium is strong, the discharging mass flow rate can be described simply with the discharging equation of water. It is very interesting that Lockhart& Martinelli [8] also proposed the two-phase pressure loss correlation in pipes based on the single-phase correlation where the specific volume of gas or liquid is use.

4. CONCLUSIONS

The safety or relief valves to depressurize the pressure vessel are recognized as the most important safety devices for boiler and nuclear facilities. The discharging flow is usually restricted with the vena contraction of minimum flow area between valve disk and seat. The flow rate is significantly affected with the vena contraction, the delay of expansion and the non-equilibrium. In the previous study, the non-equilibrium flashing flow model using the approximation of steam condition was proposed by present author, and adopted as the regulation formula in ISO and JIS for safety valves. But the detail of flashing flow through the vena contraction of safety valve is not well understood due to the sophisticated non-equilibrium phenomena. The simple disk-type flow contraction was fabricated to verify the discharging flow rate at different valve lifts and disk materials. The followings are major results.

(1) The water discharging flow rate was measured and compared with the Bernoulli equation. The discharging coefficient of water was approximately 0.6 in spite of the lift, which approximately indicated the vena contraction ratio at the minimum flow area. The air discharging flow rate was also measured and compared with the non-equilibrium expansion delay model. The air flow rate could be well predicted with the vena contraction ratio as same as the water flow and the expansion delay factor.

(2) The non-equilibrium flashing flow model directly using the steam table was proposed to obtain the critical mass flux at the vena contraction. The more precious prediction can be available compared to the previous model using the approximation of steam condition which was adopted in ISO and JIS regulations.

(3) The discharging flow rate with flashing through the simple contraction between disk and seat was measured at the different inlet subcooling or two-phase quality. The comparison of the experimental results with the non-equilibrium model using the steam table indicated the significant non-equilibrium in the flashing flow at the vena contraction in spite of the disk material. The subcooled liquid and two-phase discharging behavior was carefully observed by using the transparent disk. The observation also supported the significant non-equilibrium.

(4) The two-phase discharging flow rate at the strong non-equilibrium was well correlated with the simple pressure loss equation using the inlet water density. But the discharging coefficient by using the homogeneous specific volume of two-phase flow scattered widely and was difficult to correlate.

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NOMENCLATURE

A	curtain area	(m^2)
c_v	discharging coefficient	(-)
c_p	isobaric specific heat	(J/kgK)
d	valve inlet diameter	(m)
G	mass flux	(kg/m^2s)
h	enthalpy	(J/kg)
Δh	enthalpy difference	(J/kg)
h_{LG}	*latent heat*	(J/kg)
L	lift	(m)
N	non-equilibrium factor	(-)
p	pressure	(Pa)
s	entropy	(J/kgK)
T	temperature	(°C)
ΔT_{sub}	subcooling	(K)
v	specific volume	(m^3/kg)
η	pressure ratio	(-)
κ	specific heat ratio	(-)
ω	omega parameter	(-)

Subscript

C	critical
L	liquid
m	homogeneous mixture
S	saturation
0	inlet
1	outlet

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