

# Single-phase discharging behavior from safety valves

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## Abstract

The discharging flow rate from safety or relief valves is restricted with the minimum flow area between disk and seat. The flow rate is usually smaller than the calculation with the minimum flow area due to the vena contraction. The clarification of vena contraction is very important but actually difficult to visualize as the rapid radial expansion occurs just after the minimum flow area. It has a possibility that the rapid flow-directional change of disk-seat contraction enhances the non-equilibrium behavior. The simple disk-seat flow contraction was fabricated to measure the discharging flow rate at different valve lifts, seat and nozzle configurations. Furthermore the two kinds of nozzles used in actual safety valves were installed instead of the simple straight nozzle to verify the effect on vena contraction. The discharging flow rate of water was affected with the lift, seat and nozzle configuration due to the different formation of vena contraction. The adequate shape of nozzle was effective to increase the vena contraction ratio to the minimum flow area. The air flow rate could be well predicted with the expansion delay model by using the vena contraction ratio as same as the water flow and the adequate expansion delay factor.

**Key words** : Safety Valve, Discharging Flow, Single-Phase, Vena Contraction, Radial Expansion, Lift, Seat, Nozzle, Expansion Delay

## 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.

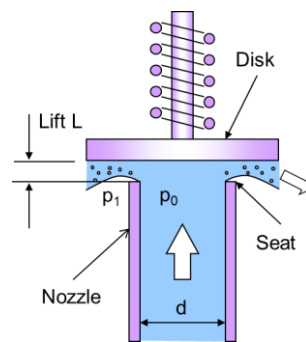
The discharging flow is usually restricted with the vena contraction of minimum flow area between disk and seat. The flow rate is significantly affected with the vena contraction and the non-equilibrium. In the previous study (osakabe et al., 1996), the non-equilibrium flashing flow model using the approximation of steam condition was proposed by present author. Schmidt (2007) 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 (2008), and Japanese industrial standard, JIS (2013), for safety valves. But the detail of expanding or flashing flow through the safety valve is not well understood due to the sophisticated formation of vena contraction and non-equilibrium phenomena. So the discharging flow rate of single-phase without the phase change through the simple disk-seat flow contraction simulating the safety valves was measured. The clarification of vena contraction is very important but actually difficult to visualize as the rapid radial expansion occurs just after the minimum flow area. It has a possibility that the rapid flow-directional change of disk-seat contraction enhances the non-equilibrium behavior. In the present model, the expansion delay is introduced instead of changing the vena contraction ratio to the minimum flow area.

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

$$A = Ld\pi \quad (1)$$

where  $L$  is the valve lift and  $d$  is the nozzle diameter. The valve seat is defined as the surface of nozzle 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  $d/4$ . Furthermore, 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 (Chiba et al., 2012).

The vena contraction also can be seen in the flow through orifice (Hirahara, 1989). Generally when a fluid flows through thin 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 thin 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. The difference between the orifice and disk-seat contraction is the flow-directional change at the contraction. The discharging coefficient of approximately 0.6 is also obtained in the disk-seat contraction but the longer passage between seat and disk like the thick orifice gives the larger values than 0.6 (Chiba et al., 2012). For the compressible fluid, it has a possibility that the rapid flow-directional change enhances the non-equilibrium behavior of disk-seat contraction.



**Fig.1** Disk-seat flow 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 from that of liquid flow. Furthermore the critical flow rate is strongly affected with the non-equilibrium as known as the delay of boiling.

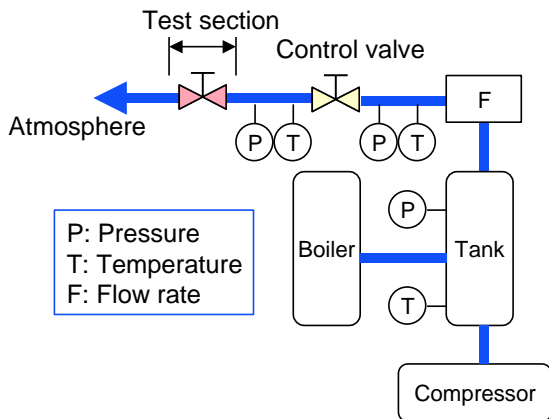
The discharging coefficient obtained in liquid flow was tentatively used in the prediction for the flashing flow rate of subcooled liquid from safety valves (Osakabe et al. 1996). The international standard, ISO (2008), recommends the discharging coefficient of 0.65 for subcooled liquid flow. For two-phase flow, the higher value than the liquid flow is recommended because the discharging coefficient of critical flow is usually larger than liquid. The regulation by American Petroleum Institute, API (2000), also gives 0.65 for liquid flow and 0.8 for two-phase flow. It is suggested that the vena contraction for critical flow is different from that for liquid flow. It should be noted that the non-equilibrium behavior and the vena contraction in the disk-seat contraction is not well understood. The disk-seat contraction has the flow-directional change at the contraction which is different from the usual flow-directional contraction such as the orifice. For the further understanding, it is very important to investigate the behavior of discharging single-phase flow without phase change under the clear boundary conditions as in this study.

## 2. Experimental apparatus and method

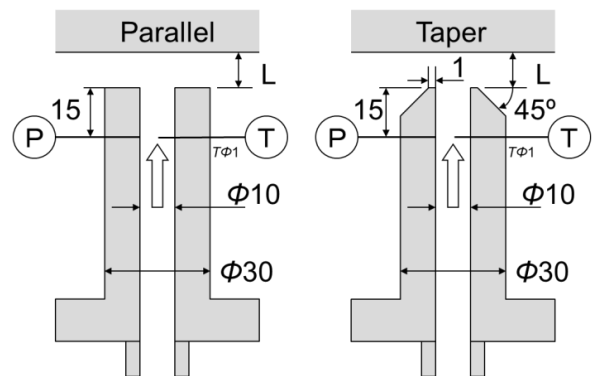
Shown in Fig.2 is the schematic of experimental apparatus. Water is supplied from the tank to the test section after depressurized through the control valve. When the depressurization exceeds the saturation pressure, flashing takes place and two-phase flow is supplied to the test section. The quality of two-phase flow can be obtained with the

calculation assuming a constant enthalpy through the control valve. The tank is connected with steam boiler and air compressor, and can be pressurized up to 0.6 MPa. The water flow rate through the test section is measured with the electromagnetic flow meter of which measurement error is within  $\pm 0.5\%$ . This meter without a contraction upstream of control valve can prevent the flashing. For the low water flow rate, the eddy flow meter of which measurement error is within  $\pm 3\%$  is used. The air flow rate can be measure with the hot wire flow meter of which measurement error is within  $\pm 2\%$ . The pressure is measured with pressure gages of which measurement error is within  $\pm 1.25$  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.

The vena contraction is strongly affected with the valve lift and the seat. The taper and parallel seats were used in the present experiment of straight nozzle to study the effect of seat configuration. The straight nozzle is defined as the inner diameter is constant. The test section is the simple disk-seat contraction consists of straight nozzle and disk as shown in Fig.3. The inner and outer diameters of nozzle 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 flow rate and discharging coefficient. 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 and transparent glass to provide the observation from the disk backside are used.



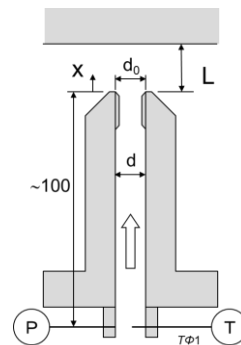
**Fig.2** Experimental apparatus



**Fig.3** Test sections of straight nozzle



**Fig.4** Photograph of actual nozzle

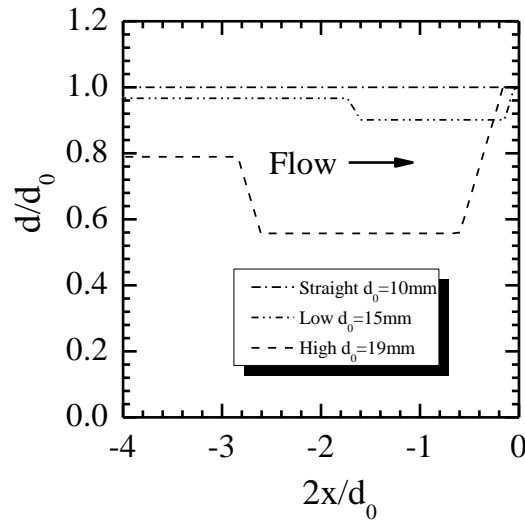


**Fig.5** Test section of actual nozzle

In actual safety valves, the inner diameter of nozzle is sometimes not constant. To obtain the higher discharging coefficient, the inner shape has been determined empirically. Shown in Fig.4 are the nozzles used in the actual safety valves. The discharging flow suddenly expands at the curtain area of straight nozzle but gradually expands before the curtain area in the actual valves.

Shown in Fig.5 is the schematic of actual nozzle. The flow-directional coordinate  $x$  is defined from the outlet of nozzle. The top of nozzle is at  $x$  of 0. The inner profile of nozzle is not straight. Shown in Fig 6 is the profile of actual nozzle. The nozzles have the special contractions before the curtain area at the top of nozzle. The inner contraction contributes to the gradual expansion of flow at the curtain area. Two kinds of actual nozzles with the high and low

contraction are used in the present study. The high or low contraction of the nozzle inner profile can be recognized in Fig.6. The diameter  $d_o$  at the curtain area are 19 mm in the high contraction nozzle and 15mm in the low contraction nozzle.



**Fig.6** Inner profile of actual nozzle

In the single-phase discharging experiments of present study, 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. Discharging model

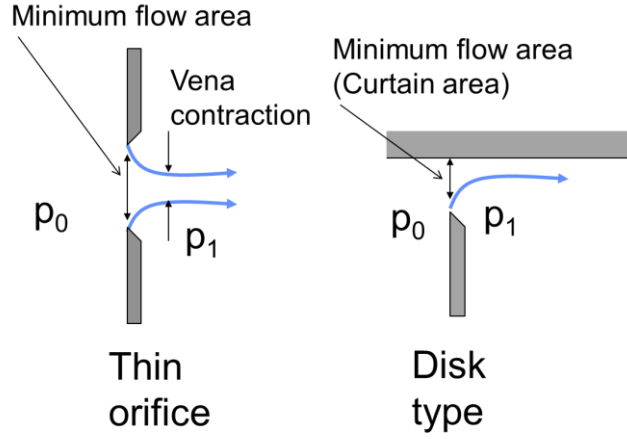
Shown in Fig.7 is the minimum flow area and vena contraction in thin orifice and disk-seat contraction. The flow through the minimum flow area further accelerates towards the vena contraction with the depressurization in the orifice flow. The flow rate is used to be calculated assuming that the vena contraction at the downstream of minimum flow area is the outlet pressure. The actual mass flux can be obtained with multiplying the discharging coefficient  $c_v$  which is approximately equal to the ratio of vena contraction to the minimum flow area. If the compressible flow of slow subsonic velocity is considered, the discharging coefficient should be approximately the same as the incompressible flow. But when the sonic velocity is attained at the minimum flow area, the flow cannot accelerate with the vena contraction. So the flow rate is restricted at the minimum flow area of critical pressure and the vena contraction would disappear. The previous experimental observation of orifice flow supports this disappearance of vena contraction (Hirahara, 1989).

The vena contraction cannot be recognized but surely exists also in the disk-seat contraction. In this geometry, the rapid radial expansion occurs just after the minimum flow area called as the curtain area. It has a possibility that the rapid flow-directional change of disk-seat contraction enhances the non-equilibrium behavior. In the present model, the expansion delay is introduced instead of changing the vena contraction ratio expressed as the discharging coefficient. The mass flow rate at the curtain area can be expressed with the discharging coefficient  $c_v$  and the enthalpy difference  $\Delta h$  between valve inlet and outlet as,

$$G = c_v \sqrt{2\Delta h} / v_1 \quad (2)$$

where  $v_l$  is the specific volume at the outlet. The enthalpy drop can be obtained with the integration of specific volume assuming the isentropic change as,

$$\Delta h = - \int_{p_0}^{p_1} v dp \quad (3)$$



**Fig.7** Minimum flow area and vena contraction

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

$$\Delta h = v_0(p_0 - p_1) \quad (4)$$

So the non-dimensional mass flux is

$$G^* = \frac{G}{\sqrt{p_0/v_0}} = c_v [2(1-\eta_1)]^{1/2} \quad (5)$$

where  $\eta_1$  is the pressure ratio defined as

$$\eta_1 = p_1 / p_0 \quad (6)$$

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

$$v = Nv_e + (1-N)v_0 \quad (7)$$

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 (osakabe et al., 1996) and the ensuing ISO regulation (2008). Assuming the isentropic change as,

$$pv_e^\kappa = const. = p_0v_0^\kappa \quad (8)$$

Equation (3) becomes

$$\Delta h = Nv_0p_0 \frac{1}{1-1/\kappa} (1-\eta_1^{1-1/\kappa}) + (1-N)v_0p_0(1-\eta_1) \quad (9)$$

By using the outlet specific volume of

$$v_1 = Nv_0 \left( \frac{1}{\eta_1} \right)^{1/\kappa} + (1-N)v_0 \quad (10)$$

in Eq. (2) and (9), the non-dimensional mass flux can be obtained as

$$G^* = \frac{c_v \sqrt{2}}{N(1/\eta_1)^{1/\kappa} + 1 - N} \left[ N \frac{1}{1-1/\kappa} (1-\eta_1^{1-1/\kappa}) + (1-N)(1-\eta_1) \right]^{1/2} \quad (11)$$

When  $N=1$  in Eq.(11) is for the compressible fluid and  $N=0$  in Eq.(11) is for the non-compressible fluid that is coincident with Eq.(5). The first and second terms in the right hand side bracket of Eq.(11) are the compressible and non-compressible terms, respectively. If the flow at the low pressure ratio is considered, two terms become same as the flow can be treated as the incompressible flow and the effect of  $N$  can be negligible.

## 4. Experimental results

### 4.1 Straight nozzle

Equation (5) is called as Bernoulli equation for non-compressible fluid. Shown in Fig.8 is the relation of liquid discharging coefficient  $c_v$  and pressure difference  $\Delta p (=p_0 - p_1)$  in the disk-seat contraction of straight nozzle. The discharging coefficient  $c_v$  is obtained with Eq.(5). 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 lift of 0.3 to 2.5mm agree well with  $c_v=0.61$  for the taper seat. It is considered that the taper seat can provide the free flow and vena contraction as same as the thin orifice. On the other hand, the discharge coefficient is larger than 0.61 at the lift less than 1mm for the parallel seat when the pressure difference is small. The coefficient gradually decreases as the pressure difference increases. The coefficient larger than 0.61 suggests the free flow like orifice cannot be obtained. It is clear that the discharging flow is affected with the seat configuration. Increasing the pressure difference and discharging velocity, the coefficient suddenly drops to  $c_v=0.61$  as shown in Fig.8. This sudden drop indicates the change of vena contraction.

The pressure difference at which the discharging coefficient becomes  $c_v=0.61$  is larger at the smaller lift. The larger pressure difference and the higher velocity are needed at the smaller lift to make a free flow like the orifice flow. It is considered that the seat configuration can affect the vena contraction at the smaller lift condition. On the other hand, the taper surface always gives a flow like the orifice.

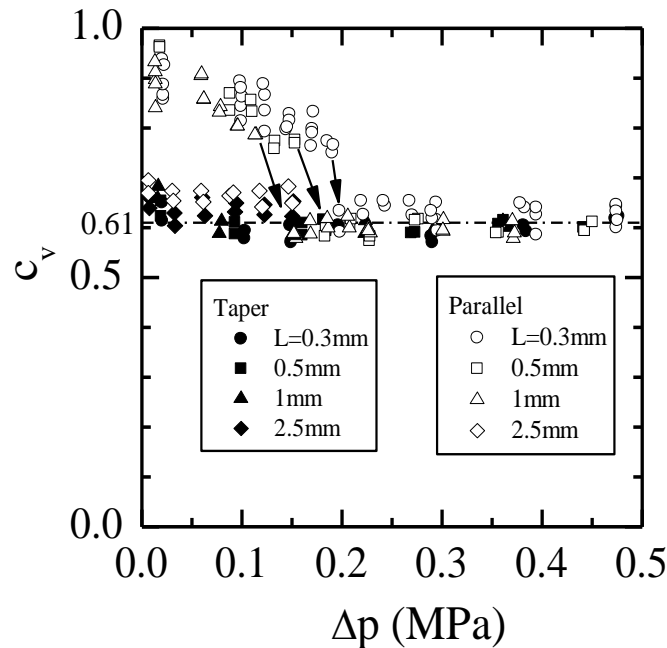
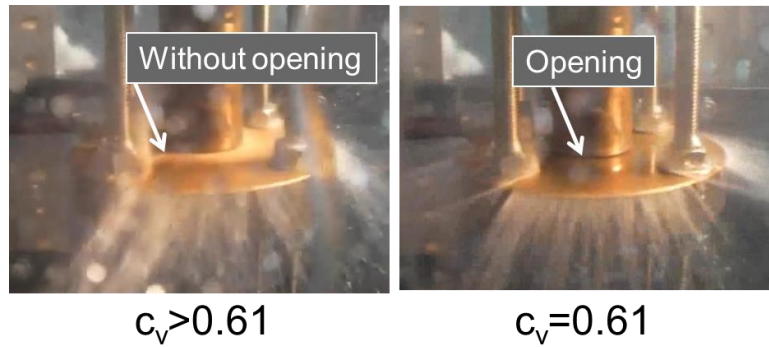
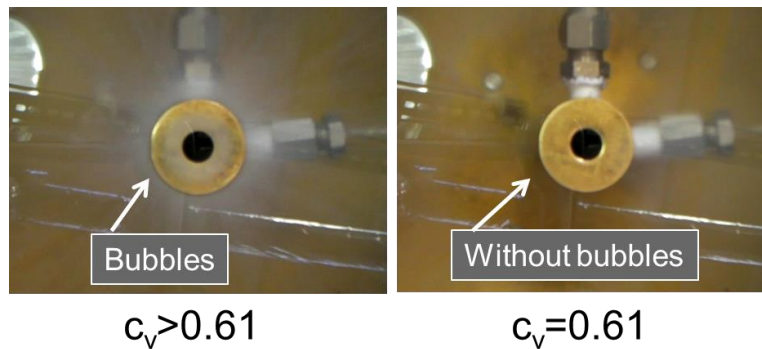


Fig.8 Discharging coefficient in relation with differential pressure



**Fig.9** Discharging behavior in parallel seat ( $L=1.0\text{mm}$ ) observed from side

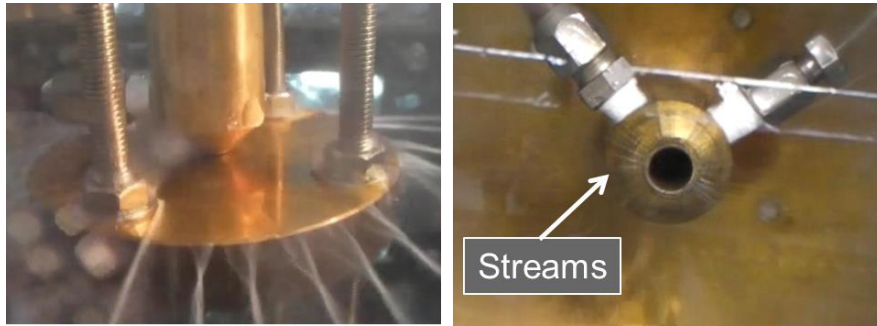


**Fig.10** Discharging behavior in parallel seat ( $L=1\text{mm}$ ) observed from disk back

The sudden drop of discharging coefficient can be seen only in the parallel seat. The flow behavior at the drop was observed from the side. The discharging behavior for the lift of 1mm and the parallel seat is shown in Fig.9. Before the sudden drop of coefficient, the discharging flow is contacting to the rim of parallel seat as shown in the left photo of Fig.9. The gap between seat and disk was filled with the discharging water. After the sudden drop of coefficient, the gap between seat and disk was not filled with water. The space between the seat and water film could be observed and the thickness of water film became thinner.

The flow change at the sudden drop of coefficient was observed from side as mentioned above. The clear observation was not obtained when the flow rate was relatively large. So the test section was set upside down and the transparent disk was installed for the observation from disk back. In the observation, the lift was fixed at 1 mm and the flow behavior affected with the seat configuration was investigated. The white annulus between the parallel seat and disk was observed from the disk back as shown in the left photo of Fig.10 when the pressure difference and discharging velocity is small. When the annulus existed, the noisy sound was recognized. The sound is considered to be due to cavitation of flow. Further increasing the pressure difference, the sudden drop of coefficient took place and the white annulus with the sound disappeared. The seat can be seen clearly as shown in the right photo of Fig.10. This indicates the gap between the seat and disk was filled with water.

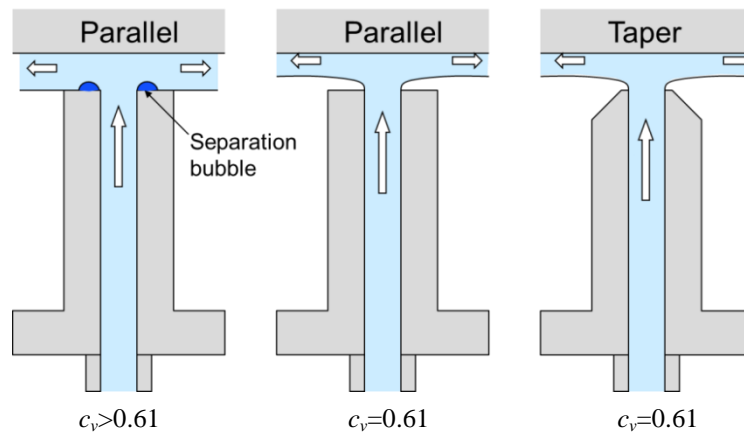
However there was no change of discharging flow in the taper seat even when the lift is less than 1mm as shown in Fig.8. The flow observation in the taper seat does not show the flow change as that in parallel seat. The thin water film is always discharged as shown in Fig.11. The discharging water film is thin as same as the condition after the sudden drop in the parallel seat. When the lift is 2.5 mm in taper and parallel seats, the space between the seat and water film was always observed and the sudden drop was not observed. All the data of discharge coefficient for the parallel seat are approximately 0.61 at lift of 2.5mm. When the lift is enough large compared to the radial length of seat, the vena contraction becomes as same as that for the taper seat.



**Fig.11** Discharging behavior in taper seat ( $L=0.3\text{mm}$ )

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 that is the surface of pipe end which contacts to the disk at closing. At the pressure difference and flow rate are small in the parallel seat, the flow after the vena contraction re-attach to the seat as shown in the left of Fig.12. The re-attachment makes a separation bubble just after the curtain area. So at the separation bubble region, the pressure decreases below the outlet pressure of atmosphere and the cavitation occurs. The cavitation is recognized as the white annulus in the left photo of Fig.10. It should be noted that the discharging coefficient becomes larger than 0.61 when the separation bubble region is formed. The vena contraction is mitigated with the re-attachment of flow to the seat as same as the thick orifice.

On the other hand, the larger lift prevents the re-attachment and formation of separation bubble. The gap between seat and disk was not filled with the flow as shown in the center of Fig.12. The larger pressure difference and the higher discharging velocity also prevent the re-attachment even at the smaller lift. In this case, the pressure smoothly decreases to atmosphere and the cavitation do not appear. Also the taper seat prevents the re-attachment and formation of separation bubble as shown in the right of Fig.12. In this case, the pressure also decreases to atmosphere smoothly and the cavitation bubbles do not appear.



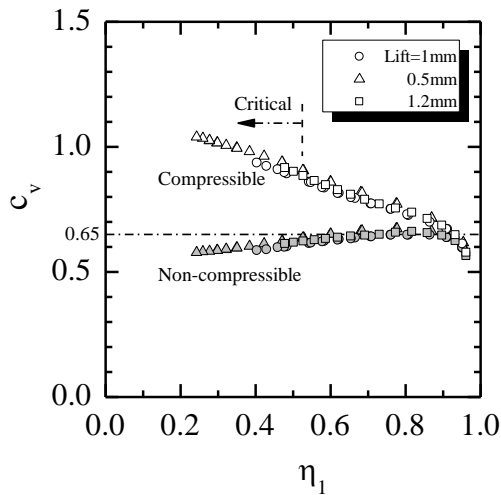
**Fig.12** Flow pattern in parallel and taper seats

Shown in Fig.13 is the relation of air discharging coefficient  $c_v$  and pressure ratio  $\eta_l$  in the disk-seat contraction of taper seat and straight nozzle. 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.(11) 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. These values can be calculated with the equilibrium theory of compressible fluid. The coefficient assuming the compressible fluid gradually increases with decreasing the pressure ratio and becomes 1 at the smaller pressure ratio. The discharging coefficient is nearly equal to the vena contraction ratio to the minimum flow area. The clarification of vena contraction is very important but actually difficult to visualize as the rapid radial expansion occurs just after the minimum flow

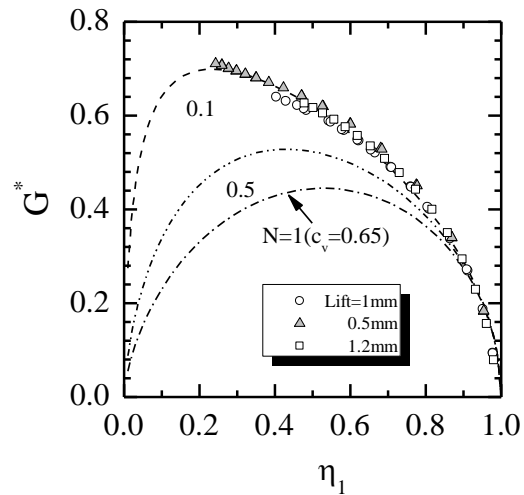


area. On the other hand, the coefficient assuming the non-compressible fluid is approximately 0.65. It should be noted that air mass flow through the present disk-seat contraction can be described like a non-compressible fluid such as water.

In ISO regulation (2008), 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.13. But by using the equation for the non-compressible fluid, approximately the same discharging coefficient as that of liquid can be obtained.



**Fig.13** Discharging coefficients defined with compressible and non-compressible equations

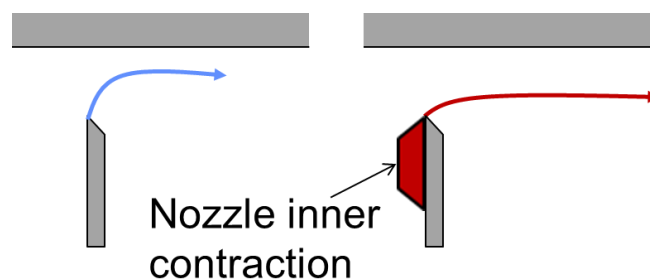


**Fig.14** Discharging non-dimensional mass flux of air

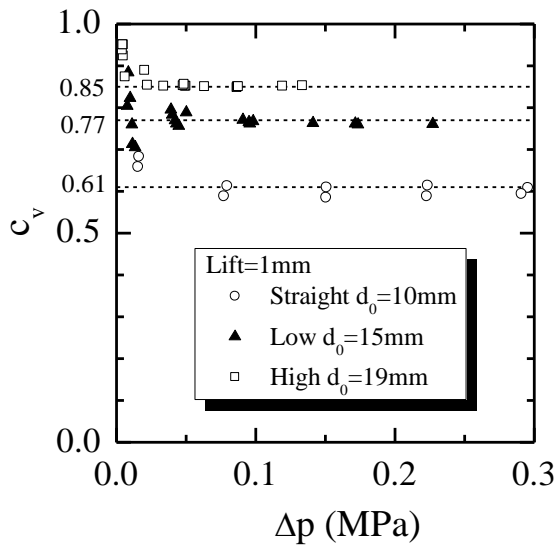
Shown in Fig.14 is the relation of non-dimensional mass flux  $G^*$  and pressure ratio  $\eta_l$  in the air discharging experiment of taper seat and straight nozzle at the lift of 0.5 to 1.2 mm. The lines are prediction with Eq.(11) where the discharging coefficient  $c_v$  of 0.65 and  $N$  of 1 to 0.1 is used. Reducing  $N$  in Eq.(11) that means the increase of expansion delay, 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 nearly equals to the vena contraction ratio is fixed in the present model. The effect of the vena contraction can be included in the expansion delay factor.

#### 4.2 Actual nozzle

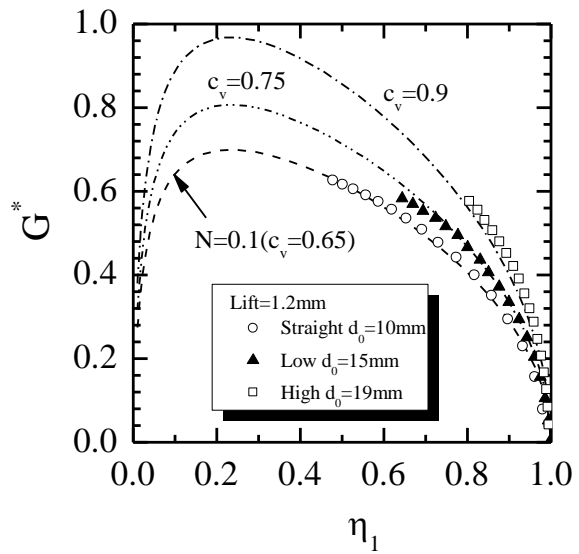
The discharging flow suddenly expands at the curtain area of straight nozzle but gradually expands before the curtain area in the actual valves. Shown in Fig.15 is the schematic difference between straight and actual nozzle. The actual nozzles have the special inner contractions before the curtain area. The inner contraction of nozzle contributes to the gradual expansion of flow at the curtain area as the right figure of Fig.15. The gradual expansion results as the larger vena contraction.



**Fig.15** Control of vena contraction



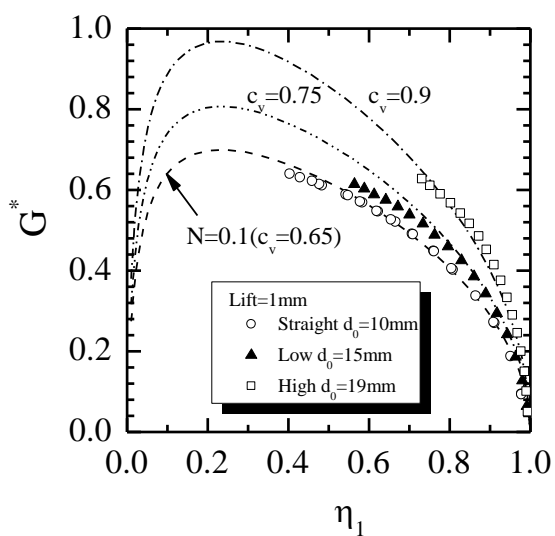
**Fig.16** Discharging coefficient of water



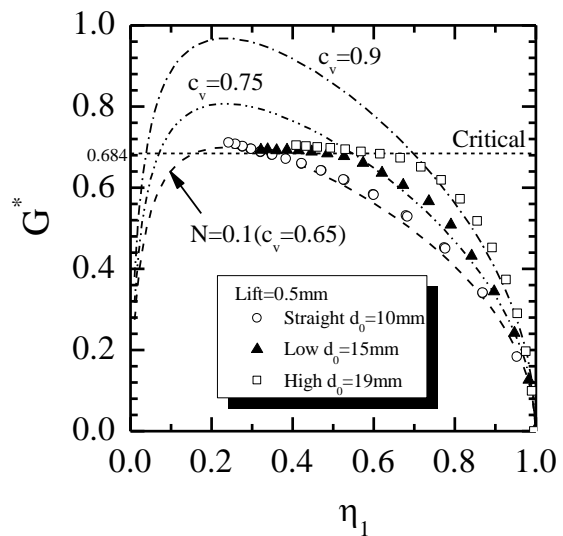
**Fig.17** Non-dimensional mass flux of air (L=1.2mm)

Shown in Fig.16 is the relation of liquid discharging coefficient  $c_v$  and pressure difference  $\Delta p (=p_0 - p_1)$  in the disk-seat contraction of actual nozzle. The data of taper seat and straight nozzle are compared in the figure. The discharging coefficient  $c_v$  is obtained with Eq.(5). Two kinds of actual nozzles with the high and low contraction are used. The higher discharging coefficient was obtained at the higher contraction. The higher contraction results as the higher discharging coefficient indicating the larger vena contraction.

Shown in Fig.17 and 18 is the relation of non-dimensional mass flux  $G^*$  and pressure ratio  $\eta_1$  in the air discharging experiment at lift of 1.2 and 1mm, respectively. The lines are prediction with Eq.(11) where the discharging coefficient  $c_v$  of approximately the same as that of water and  $N$  of 0.1 is used. The experimental data agree well with the prediction of  $N=0.1$  in spite of the nozzle. The data of taper seat and straight nozzle are compared in the figure. The data of the actual nozzle with low contraction becomes slightly lower than the predictions as the pressure ratio decreases. The air flow rate could be well predicted with the discharging coefficient as approximately the same as the water flow and the appropriate expansion delay factor.



**Fig.18** Non-dimensional mass flux of air (L=1mm)



**Fig.19** Non-dimensional mass flux of air (L=0.5mm)

Shown in Fig.19 is the relation of non-dimensional mass flux  $G^*$  and pressure ratio  $\eta_l$  in the air discharging experiment at the lift of 0.5 mm. The lines are prediction with Eq.(11) where the discharging coefficient  $c_v$  of approximately the same as that of water and  $N$  of 0.1 is used. The experimental data agree well with the prediction of  $N=0.1$  in spite of the nozzle when the mass flow rate is less than sonic limitation of the critical condition. The critical mass flux is

$$G^* = \left[ \kappa \left( \frac{2}{\kappa + 1} \right)^{\frac{\kappa+1}{\kappa-1}} \right]^{1/2} \quad (12)$$

Substitution of the specific heat ratio  $\kappa$  of 1.4 in Eq.(12) gives the critical non-dimensional mass flux of 0.684. After the prediction with Eq.(11) reaches the critical condition, the experimental data agree well with the critical mass flux with Eq.(12). When the sonic velocity is attained at the minimum flow area, the vena contraction disappears. The flow rate is restricted at the minimum flow area of critical pressure and the vena contraction cannot be observed. It should be noted that the expansion delay model is not adequate to calculate the critical condition. The critical condition should be correctly calculated at the actual minimum flow area.

## 5. Conclusions

The discharging flow rate from safety or relief valves is restricted with the minimum flow area between disk and seat. The flow rate is usually smaller than the calculation with the minimum flow area due to the vena contraction. The simple disk-seat flow contraction was fabricated to measure the discharging flow rate at different valve lifts, seat and nozzle configurations. Furthermore two different kinds of nozzles used in actual safety valves were installed instead of the simple straight nozzle to verify the effect on vena contraction. The followings are major results.

- (1) Increasing the pressure difference, the sudden drop of discharging coefficient can be seen in the parallel seat. Before the sudden drop of coefficient, the discharging flow is contacting to the rim of parallel seat. The flow after the vena contraction re-attaches to the seat and the gap between seat and disk is filled with water. After the sudden drop of coefficient, the gap between seat and disk is not filled with water. The space between the seat and thin water film could be observed.
- (2) When the pressure difference and flow rate are small in the parallel seat, the re-attachment makes a separation bubble just after the curtain area. The larger lift prevents the re-attachment and the formation of separation bubble. The larger pressure difference and the higher discharging velocity also prevent the re-attachment even at the smaller lift. Also the taper seat prevents the re-attachment and formation of separation bubble.
- (3) The discharging flow suddenly expands at the curtain area of straight nozzle but gradually expands before the curtain area in the actual valves. The actual nozzles have the spatial contractions before the curtain area. The inner contraction contributes to the gradual expansion of flow at the curtain area. The gradual expansion results as the larger vena contraction. The higher inner contraction of nozzle results as the higher discharging coefficient indicating the larger vena contraction
- (4) The air discharging flow rate is compared with the expansion delay model. The air flow rate can be well predicted with the vena contraction ratio as same as the water flow and the appropriate expansion delay factor. However, the expansion delay model is not adequate to calculate the critical condition. The critical condition should be correctly calculated with the sonic velocity at the actual minimum flow area.

## 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 <sup>2</sup> s )
$h$	enthalpy	( J/kg )
$\Delta h$	enthalpy difference	( J/kg )
$h_{LG}$	latent heat	( J/kg )
$L$	lift	( m )
$N$	expansion delay factor	( - )

$p$	pressure	(Pa)
$\Delta p$	pressure difference	(Pa)
$T$	temperature	(°C)
$\Delta T_{sub}$	subcooling	(K)
$v$	specific volume	(m <sup>3</sup> /kg)
$x$	flow directional position	(m)
$\eta$	pressure ratio	( - )
$\kappa$	specific heat ratio	( - )

#### Subscript

0	inlet
1	outlet

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