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Development of novel predictive equations for local flow asymmetry within control valves using a distribution parameter-based method under multiphase conditions

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Abstract

When designing control valves for multiphase applications using conventional valve sizing methodologies, it is assumed that the flow properties such as mixture density and volume fraction of phases are uniform throughout the valve trim. In reality, however, the properties may be different due to the difference in the gas and liquid densities as well as velocities within the valve which cause phase stratification. Furthermore, complex geometric effects can also cause phase non-uniformity within the valve. Current valve sizing methodologies are based on global parameters such as overall valve coefficient (Cv) which do not consider local phase non-uniformities within the trim during the design phase. Among these methodologies, it has previously been shown that the harmonised Cv method is the most accurate method of sizing control valves under multiphase flow conditions across a wide range of flow regimes. The current study provides additional equations to be used with the harmonised Cv method to ensure that the local flow quality compliance is achieved, which seems to be the major weakness of the existing method. In order to obtain local gas and liquid flow rates, pressure drops and phase fractions, well-validated computational fluid dynamics (CFD) simulations were carried out. Two valve opening positions of 60% and 100% were considered each with 5, 10, and 15% inlet air volume fractions to simulate real life conditions. The results show that there is severe non-uniformity in the local gas and liquid distributions within the valve trim. To quantify the phase non-uniformities observed, a distribution parameter ($C_\text{o}$) based on the drift-flux model was used. The harmonised $C_\text{v}$ method was used to calculate the local Cv's and the local equivalent area factor ($\Psi$). A new equation was derived that considers the local variation of the $\Psi$ factor within the trim and which incorporates the distribution parameter. Based on the foregoing analyses, additional flow distribution equations have been proposed that when used with the harmonised Cv's valve design method for multiphase flow applications will ensure that the deviation in performance of the valve is minimal from the design conditions.

Keywords: Computational fluid dynamics, control valves, distribution parameter, flow capacity, two-phase flow, valve trim.

1 Introduction

Flow control in industrial pipelines with high pressure difference requirements is carried out using severe-service control valves. These are commonly found in the process, power generation, as well as
oil & gas industries. Sizing of severe-service control valves is carefully carried out to ensure controlled performance of the valves under critical service applications. Control valve sizing methodologies are well established for single-phase flows and are as stipulated in the standard BS60534-2-1 [1]. For the case of gas–liquid multiphase flows, an “expansion factor” is used to account for the effect of gas compressibility and expansion as well as its fraction in the overall mixture. Also, researchers and companies have devised methods of calculating an appropriate mixture density for use in the flow coefficient \( C_v \) design equation to take care of multiphase effects. There are three such methodologies (summarised in Table 1). The first, known as the “sum of Cvs” or “addition” method, is the simplest method for multiphase valve sizing. Essentially, the liquid and gas phase Cvs are calculated separately using the single-phase equations present in the standard BS60534-2-1 [1], and then summed. This is given by Equation (1) in the table. Usage of this method is limited to low gas fractions as Diener et al. [2]–[4] noted that the valve flow coefficient calculated can deviate at higher gas volume flow rates, as the effect of interfacial slip between the gas and liquid becomes increasingly significant. The second method is known as the “effective density” method (Equation (2)), and it considers the gas and the liquid phases as a homogeneous fluid with its own density and thermodynamic properties. The mixture density is estimated using an effective density approach [5]. The effective density method treats the multiphase system as a homogenous or frozen fluid and does not properly account for local differences in the gas and liquid velocities.

Table 1: Current methods of calculating the valve flow coefficient in gas–liquid multiphase conditions

<table>
<thead>
<tr>
<th>Method</th>
<th>Equation</th>
<th>Equation number</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sum of ( C_v )s ([1], [4], [6])</td>
<td>( C_{v,\text{value}} = 0.0366W \left[ \frac{X_L}{\sqrt{\Delta P \cdot \rho_{IL}}} + \frac{1 - X_L}{\sqrt{\Delta P \cdot \rho_{IG} \cdot Y^2}} \right] )</td>
<td>(1)</td>
</tr>
<tr>
<td>Effective density ([5], [6])</td>
<td>( C_{v,\text{value}} = 0.0366W \left[ \frac{\left( X_L + \frac{1 - X_L}{\rho_{IG} Y^2} \right)}{\rho_0} \right] \sqrt{\frac{\Delta P}{\rho_0}} )</td>
<td>(2)</td>
</tr>
<tr>
<td>Sum of harmonised Cvs ([6])</td>
<td>( C_{v,\text{value}} = 0.0366W \left[ \frac{X_L}{\sqrt{\Delta P \cdot \rho_{IL} \cdot N_L}} + \frac{1 - X_L}{\sqrt{\Delta P \cdot \rho_{IG} \cdot Y^2 \cdot N_G}} \right] )</td>
<td>(3)</td>
</tr>
</tbody>
</table>

Where \( N_L \) and \( N_G \) are the harmonised Cvs liquid and gas phase correction factors, the corresponding equations are given in the Appendix; \( \rho_{IL} \) and \( \rho_{IG} \) are the inlet liquid and gas densities (in kg/m³); \( \Delta P \) is the pressure drop (in bar); \( W \) is total inlet mass flow rate (in kg/h), \( X_L \) is the liquid mass fraction; and \( Y \) is the expansion factor.

To overcome the shortcomings of the previous methods, the third method namely the “sum of harmonised Cvs” method treats both phases separately and introduces correction factors that are used in integrating the effects of the volume and velocity ratios between the phases (Equation (3)). This method has been identified as the most accurate of the three [7] since it more comprehensively accounts for interactions between the gas and liquid as well as phase slip. Nevertheless, the sizing is done using global flow variables such as the pressure drop across the entire valve system for a particular gas–liquid mixture flow rate. Also, it is assumed that flow variables such as mixture density and volume fractions remain uniform whereas in actual multiphase flow through complex geometries severe non-uniformity in these variables is observed. Hence the designs developed based on uniform
distribution assumptions may give rise to problems like cavitation and wear at the local level. Therefore, it is important to consider the local variations of pressure, flow rate and gas void fractions in the design process. Few studies are reported in the literature that utilise local parameters within the valve and its trim [8]–[11] for the evaluation of their performance. Apart from very few investigations such as Singh et al. [11] etc., the other studies investigated single-phase liquids flowing through the control valve [12], [13]. The main reasons for the lack of inclusion of local quantities in multiphase valve sizing are the limited data availability of the local multiphase properties and knowledge on the local flow mechanisms governing two-phase flow in such valves. These quantities include the local pressure, phase velocities, and volume fraction distributions within complex valve and trim geometries. The study by Singh et al. [11] involved using computational fluid dynamics (CFD) to investigate the local flow features of a severe service control valve with a flowing multiphase gas and liquid mixture. The effect of inlet gas volume fractions (5, 10, and 15%) as well as valve opening position (VOP, 100 and 60%) were quantified. The results showed the existence of high levels of non-uniformity in the local water and gas void fraction and velocity distributions within the control valve trim. They derived equations for local $C_v$ and the distribution parameter $C_o$ which quantified phase non-uniformities in various parts of the valve. However, no information was presented on how these non-uniformity indicators could be used alongside harmonised $C_v$s method for improving it. Furthermore, they did not include effect of geometry change on these parameters.

To further improve the accuracy of prediction in the flow variables in a designed valve, the current paper combines the harmonised $C_v$s method with the novel predictive equations based on data obtained from Singh et al. [11] and the data obtained in the current study. Essentially, local pressure, mixture density and phase fractions extracted from a well-validated CFD model of a control valve trim are used to calculate the $C_v$ in rows, discs, and quarters of the trim. A method has been developed for correlating inlet volume fraction and the distribution parameter with location within the valve trim. An overall $C_v$ value was obtained by appropriately summing local $C_v$ values. The distribution parameter, which captures the combined effect of local phase velocities and volume fractions and which allows local flow distribution to be quantified, can now be computed. Its inclusion in the relationship for $C_v$ ensures that local flow mechanisms are inherently captured in control valve sizing methodology to ensure operational quality compliance.

2 CFD modelling and validation

Computational fluid dynamics is a useful tool in the analysis of process systems and has been used for simulating detailed flow behaviour in valves and other process equipment [8]–[10], [14]–[16]. In order to obtain the local flow parameters such as the local gas volume fraction, pressure, and velocity distribution, numerical modelling of the control valve with a continuous-resistance trim was carried out using the commercial CFD code ANSYS version 19.2. The geometry of the control valve and connecting horizontal pipes is as shown in Figure 1(a). The control valve modelled here is widely used for flow control in the process industry. The length of the inlet and outlet pipes are kept at 2D and 6D respectively with 4D extra length is added to allow for the flow development. These are in accordance with the BS EN Standards [1], [17], [18]. The internal diameter D of the inlet and outlet pipes is the same and the exit section is longer because there is need to allow for flow development before pressure and other measurements are recorded. Figure 1 (b) gives the geometry of the trim showing the stack of discs (numbered 1–11 from the valve seat at the bottom to the top) and external row of cylinders forming the obstruction to flow and giving rise to the flow paths. Five such rows in staggered
formation are present from the periphery to the centre of the trim and their diameter progressively decreases accordingly in the direction of flow towards the centre of the trim, to manage pressure drop in a staged manner. For post-processing the results after CFD simulations, planes A–F (Figure 1 c) were constructed and data for fluid velocities, phase fractions, and physical properties e.g., phase and mixture densities were extracted to further analyse the distribution of these local parameters in order to apply the harmonised Cvs method locally. These planes have been constructed to obtain the local parameters at the inlet and outlet of each row of cylinders in different quarters of the trim. These are shown in Figure 1 (c). Appropriate meshing of the flow domain was carried out, and a mesh independency study was carried out to determine the optimum mesh size. The simulations were carried using the RANS formulation of the Navier-Stokes equations. The two-equation k-ω Shear Stress Transport (SST) model was used for turbulence modelling as it has been shown to have superior capability in modelling flows with severe velocity and pressure gradients, which are expected due to the complex trim geometry. For the boundary conditions, velocity inlet and pressure outlet conditions were used. The inlet liquid velocity used was 1.22 m/s based on experiments and allowable process conditions. The air velocities were calculated to be 2.5, 5.3, and 8.4 m/s and these give 5, 10, and 15% gas volume fractions to simulate real life conditions beyond which the slip between the phases becomes dominant and mixture density calculation becomes increasingly inaccurate.
To ensure that the simulation results are not dependent on the mesh size, a mesh independency study was carried out. Different mesh arrangements with 5.1 million, 5.4 million and 6.1 million elements were used for simulations where the gas volume fraction was monitored at the outlet as well as at the valve seat. The results indicate that there is a maximum of ±6% difference between the outlet volume fractions for the three mesh sizes at various volume fractions. At maximum volume fraction of 15%, the difference in outlet volume fraction was 1.5% between the 5.4 million element case and the 6.1 million element case. For reasons of computational efficiency, the mesh with 5.4 million was used for the simulations hereafter in this study. A flow loop with the same physical dimensions and flow rates as used in the simulations were used to carry out experiments to validate the simulations. Well-calibrated pressure transducers were installed at 2D and 6D locations before and after the valve respectively (Figure 1 d). They were used to measure the pressure drop that was used to calculate the valve $C_v$ which was then compared with those from the CFD simulations at the three flow rate conditions. The absolute percentage differences obtained were between 3 and 7%, which is acceptable considering the existence of random errors in conducting experiments and convergence/other numerical uncertainties of CFD modelling. For more details on the modelling, meshing, setup and validation, refer to Singh et al. [11]

3 Description of local $C_v$ calculation using the harmonised $C_v$s method

In this section the method used for calculating local $C_v$s at each row of the disc quarter is described, and the method used is based on the BS standard [1]. The inlet pressure ($P_1$), outlet pressure ($P_2$), mass flow rates ($W_G, W_L$) and the volume fractions of air and water ($f_G, f_L$) were extracted at planes created at the inlet and outlet of each row as shown in Figure 1 (c). The subscripts G and L denote the gas and liquid phase respectively. The mass fractions were calculated dividing the mass flow rate of air or water by the total mass flow rate ($W$). The critical liquid pressure ratio factor ($F_F$) is calculated using Eqn. (1) which is function of the vapour and critical pressure ratios:

$$ F_F = 0.96 - 0.28 \frac{P_v}{P_c} $$

where, $P_v$ is the vapour pressure and $P_c$ is the critical pressure of the liquid phase and is readily available in steam tables or other reference texts with physical property tables e.g., Green & Southard [19]. The limiting pressure drop for the liquid phase is calculated by utilising correction factors for the pressure recovery and piping geometry. Here, $F_L$ is the pressure recovery factor and $F_p$ is the piping geometry factor. If the pressure drop ($\Delta P_{valve}$) across the valve is less than the limiting pressure drop, then the pressure drop is used as the liquid sizing pressure drop ($\Delta P_{LS}$) in $C_v$ calculation otherwise the limiting pressure drop is used. Hence,

$$ \Delta P_{LL} = \left( \frac{F_L}{F_p} \right)^2 (P_{inlet} - F_F P_V) $$

The limiting pressure drop for the gas phase is given by:
\[
\Delta P_{LG} = F_y X_T P_{inlet}
\]

Where, \( F_y \) is calculated by dividing the specific heat ratio of the gas phase by 1.4 and \( X_T \) is the compressible flow coefficient which is a factor dependent on the valve trim. If the pressure drop at the row is less than the limiting pressure drop, then the pressure drop is used as the gas sizing pressure drop (\( \Delta P_{GS} \)) for \( C_v \) calculation otherwise the limiting pressure drop is used. Based on volume fraction ratio (\( \frac{f_G}{f_L} \)) harmonised Cvs method correction factors (see Appendix for the details of the conditions) were obtained. The pressure drop in the valve in this study is less than both the gas and liquid limiting pressure drops, therefore, \( N_L \) and \( N_G \) are used in this study. The expansion factor (\( Y \)) is calculated \cite{1} using:

\[
Y = 1 - \frac{x_{sizing}}{3 x_{choked}}
\]

where, \( x_{choked} = F_y X_T \), and

\[
x_{sizing} = \begin{cases} 
\frac{\Delta P}{p_1}, & \text{if } \frac{\Delta P}{p_1} < x_{choked} \\
x_{choked}, & \text{if } \frac{\Delta P}{p_1} \geq x_{choked}
\end{cases}
\]

If \( \frac{\Delta P}{p_1} \geq 0.963 \), then this method should not be used.

As mentioned in the introduction section, the sum of harmonised Cvs method does not consider the local variations of pressure, volume fraction and flow rates. There can be a significant difference in these variables depending on the location within the geometry so in order to account for this, local Cvs have been determined at each row of cylinders in all the quarters. Thus, the existing equation has been applied to a local level to improve generality of the equations for diverse flow conditions. The local row Cvs have been calculated using Equation 8 which also includes the correction factors \( N_L \) and \( N_G \) for the harmonised Cvs method.

\[
C_{v_row} = 0.0366 W \left[ \frac{X_L}{\sqrt{\Delta P \cdot \rho_L \cdot N_L}} + \frac{1 - X_L}{\sqrt{\Delta P \cdot \rho_G \cdot Y^2 \cdot N_G}} \right]
\]

Expressions for the correction factors \( N_L \) and \( N_G \) and the expansion factor \( Y \) are given in the Appendix. To obtain \( C_v \) in each quarter, we consider that the rows are in series. As a result, \( C_{v_{quarter}} \) is calculated as follows:

\[
C_{v_{quarter}} = \sqrt{\frac{1}{\sum_{i=1}^{5} C_{v_{row_i}}}}
\]

Now the total \( C_v \) can be computed by

\[
C_{v_{disc}} = \sum_{quarter=1}^{4} C_{v_{quarter}}
\]

In order to understand the local variations of phase distributions, velocities and pressure, data has been extracted from the CFD simulation at different rows and quarters of the valve trim. Because of
the local variations in these quantities, the local $C_v$ is highly non-uniform. In this section a detailed analysis of the local variations in the phase fraction, velocities and pressure has been carried out.

3.1 Flow streamlines within valve trim

Figure 2 shows the streamlines of the flowing mixture at the top disc (for 100% VOP) and the middle disc (for 60% VOP) at 5, 10, and 15% gas volume fraction. The streamlines are coloured by the magnitude of the gas volume fraction. The arrows at 5% gas volume fraction and 100% VOF show the flowing direction and is the same for all the other conditions. It is seen in all the cases studied and in all quadrants of the trim that the local flow distribution as depicted by the gas volume fraction values is not uniform or symmetric but significant non-uniformity is exhibited. The local gas volume fraction is as high as 70% in some quadrants and as lower than 5% in others, especially at 10 and 15% air inlet volume fractions at 100% VOP.

![Figure 2: Path lines of the flowing mixture coloured by the air volume fraction (black arrows in the first figure show the flow direction)](image)
There is also non-uniformity at the 60% VOP, but not as much as the 100% VOP which could be attributed to the increased compression experienced by the gas caused by the smaller area available for flow for the same inlet conditions. There is flow recirculation at the exit of the quadrants shown with the red circles at both 5% gas volume fraction conditions. It is observed that the recirculation produces a more symmetric structure at the 100% VOP than at the 60% VOP cases where there seems to be more turbulence. This is expected due to the increased velocity occasioned by the smaller available flow area. For more a more comprehensive discussion on the phase/velocity distributions, the reader is referred to Singh et al. [11] where detailed contours for the local gas volume fraction, velocity, and pressure across the top, middle and bottom discs have been provided.

3.2 Local profiles

Figure 3 shows the variations of the local gas void fraction, non-dimensional water velocity and non-dimensional pressure profiles. These were non-dimensionalised using the respective maximum velocity or pressure for that inlet condition at 100% VOP. Non dimensionalisation was done to achieve like for like comparison on a scale of 0 to 1 for the different cases. Each quantity was extracted from the CFD results along planes A–F shown in Figure 1 (c). Row 1 values correspond to the values extracted at plane A and averaged, while row 2 values correspond to those extracted from plane B and averaged, and so on. The gas void fraction distribution is plotted on a logarithmic scale due to the very small quantities of air at the bottom disc and at the middle disc of the valve trim. For all inlet gas volume fractions, it can be seen that the void fraction distributions at the middle and bottom locations (i.e., discs 6 and 1) are highly asymmetric. Also, the dimensionless liquid velocity and pressure profiles exhibit more regular profiles with symmetry about two axes. In the case of the velocity, the highest local velocities occurred at the second quarter, which coincides with the location of the inlet pipe to the valve. The localised velocities are higher for the bottom disc at 5% inlet gas volume fraction. At 10, and 15% inlet gas volume fractions however, the velocities at the 2nd quarter of the top disc are the highest due to the higher-pressure conditions with increased gas fraction. On a row basis, for all volume fractions and quadrants, the velocities are seen to increase from row 1, peak at row 3 and after this, it almost remained constant. For the pressure profiles, the pattern is seen to be highly symmetric with cylinder rows for all quadrants and all discs. As the trim is a continuous resistance trim, the staged pressure drop is along the expected lines with the highest pressures at the inlet at the trim circumference, decreasing towards the centre of the trim where the gas–liquid mixture exits the valve. It can also be seen that the pressure increases from the top to the bottom of the trim. Contours showing these volume fraction trends for each of the three quantities have been comprehensively presented in a previous article published by the authors (see Singh et al. [11]).
Figure 3: Distributions of local gas void fraction, water velocity and pressure at row and disc location for VOP = 100%. The notation on the circumferential axis denotes Quarter.row. For example, 1.3 represents Quarter 1, Row 3. Note that the radial axis of the local volume fraction is logarithmic.

Flow behaviour in control valves have been shown to display different characteristics in partial opening conditions. Figure 4 shows the local distribution of $\alpha$, $V_w/V_{w\ max}$, and $P/P_{max}$ at 60% VOP. As the valve is only partially open, disc 1 has no flow and hence only the flow profiles in discs 6 and 11 are available and shown in the figure 3. For all inlet gas volume fractions, it is seen that the void fraction distribution at discs 6 and 11 are highly asymmetric. The asymmetry increases with increasing inlet gas volume fraction. The dimensionless liquid velocity and pressure profiles in contrast, exhibit more symmetric profiles just as seen in the case of the 100% VOP cases. For the liquid velocity, the highest local values occurred at the second quarter, which is exactly the location facing the inlet pipe. The magnitude of these velocities can be observed to be higher for the bottom disc at 5% inlet gas volume fraction. When considered by rows, for all volume fractions and quadrants, the velocities are seen to progressively increase from row 1 to row 5, which is slightly different than that observed for the fully open valve condition where velocity peaking occurs at row 3. The difference in peaking behaviour for the two VOP cases may be as a result of the 60% VOP having a lower area available for
flow which delays velocity peaking as well as because of the effect of asymmetric gas volume fraction. For the pressure profiles, similar profiles occur as observed in the fully open valve condition i.e., highly symmetric profiles with respect to the rows for all quadrants and all discs. The only difference is that the pressure magnitudes at the middle and bottom are more closely matched than in the 100% VOP condition. This may be attributed to lesser vertical distance at 60% VOP as compared to 100% VOP that the flow needs to travel. As a result, a lesser top to bottom pressure drop is experienced in the partial valve opening condition. It can also be seen that the pressure continuously drops from the top to the bottom of the trim.

![Diagram](image)

**Figure 4**: Distributions of local gas void fraction, water velocity and pressure at row and disc locations for VOP = 60%. Similarly, the notation on the circumferential axis denotes Quarter.row for example 2.3 represents Quarter 2, Row 3. Note that the radial axis of the local volume fraction is logarithmic.

The valves are designed for overall Cv value provided by the customers. It is therefore necessary to ensure that the designed valve provides satisfactory performance at the local level. Thus, local Cv values have been calculated as described earlier. The variation of $C_v$ corresponding to different discs
in the trim and different rows within the discs for full and partial valve opening conditions at the three inlet gas volume fractions are shown in Figure 5. The $C_v$ is calculated using the harmonised Cvs method outlined in Section 3 and relevant values obtained from the CFD simulations. It is seen that Quarter 2 which is situated at the valve entrance has the highest magnitude of $C_v$ values in all cases. This is expected and it is consistent with the observed high void fraction and water velocities in that quarter as shown in the profiles in Figure 3. Furthermore, the $C_v$ values are observed to generally decrease with trim depth as flow moves from Disc 11 at the top to Disc 1 at the bottom. Such a behaviour of the $C_v$ can be attributed to the combined effect of relatively low pressure drop and higher flow rates through the top disc as compared to the bottom disc. In general, $C_v$ asymmetry changes depending upon the location of the quarter with respect to the inlet flow direction. It can be seen that each quarter is having a range of Cvs varying from 1.5 to about 3. Row wise distribution of Cv indicates the values change depending upon local phase velocities and volume fraction.
Figure 5: $C_v$ profiles at 100% (left) and 60% VOP (right) at row and disc locations. The notation on the circumferential axis denotes Quarter.row for example 2.3 represents Quarter 2, Row 3.

3.3 Quantification of asymmetry in the profiles

The previous section has clearly shown that assumption of symmetric flow characteristics may not be appropriate for valves used for severe service applications. The design process largely assumes symmetric profiles for the flow variables like volume fraction and mixture density. To ascertain extent of non-uniformity in the flow distribution, a new indicator named asymmetry ratio ($\zeta$) has been developed. The asymmetry ratio has been defined as the ratio of a flow variable at the same radial centre from the trim centre for quarters facing each other. For example, Quarter 2 is the one that faces the flow from the inlet pipe. The quarter 4 is the face just opposite the quarter 2. Hence, the asymmetry ratio will be ratio of the value of the flow variable at a distance $r$ in quarter 4 and the value of the variable at the same distance $r$ in quarter 2. Also, in perpendicular direction same ratio has been calculated for quarter 3 and quarter 1. Mathematically, the asymmetry ratio is represented as $\zeta = \frac{(q_{u+2,r})}{V_{u+2,r}}$, with Var denoting the variable (which can be pressure, velocity, gas volume fraction or $C_v$), Qu is quarter, and $i = 1$ or 2. This ratio has then been plotted as a function of non-dimensional distance. In this trim the flow enters from outside and goes towards the centre. Hence, the non-dimensional radius ($r/R$) will be higher at the inlet and lower at the outlet.

3.3.1 Asymmetry in volume fraction profiles

Figure 5 (a) shows the variation of the asymmetry ratio for volume fraction with non-dimensional distance for the three discs within the trim, namely the top disc, the middle disc, and the bottom disc for multiphase flow with low input volume fraction of 5% by volume. It can be seen from the figure that at the top disc, in the flow direction (quarters 2 and 4), the flow is quite symmetric with asymmetry value of 1.0. In the perpendicular direction (quarters 1 and 3) the asymmetry ratio is less than one. The nature of the flow significantly changes for the middle and bottom discs. For the middle disc the asymmetry ratio in the inlet flow direction is significantly below 1, indicating an order of magnitude difference in volume fraction. Similar trend is seen for bottom disc. In the perpendicular direction the asymmetry ratio has a value six or seven orders of magnitudes smaller, indicating severe maldistribution of phases as indicated in the earlier section.

The above discussion has clearly indicated that volume fraction varies considerably within the trim, and a different value of asymmetry ratio is obtained for each disc, and also, these values change for some discs considerably as a function of non-dimensional distance from inlet. Also, within a disc, a different value is seen for each quarter. This clearly indicates that uniform volume fraction-based design methods may need to consider these variations within the design methods to meet all the flow quality requirements in severe service valves.
Figure 6: Variation of the asymmetry ratio $\xi$ in gas volume fraction for inlet gas volume fraction of (a) 5%, (b) 10%, (c) 15% at VOP = 100% and (d) $\xi$ in gas volume fraction of 10% at VOP = 60% for comparison purposes.

To evaluate asymmetry ratio at higher volume fractions, Figure 5 (b) has been drawn that shows asymmetry ratio at the input volume fraction of 0.1. The trends seen in Figure 5 (a) seem to be present in Figure 5 (b) as well. The only difference is that the asymmetry ratio for middle disc in the direction of inlet flow is fairly close to the top disc unlike at the volume fraction of 0.05. At the maximum volume fraction of 15% used in the current investigation, the volume fraction distribution seems to be mixed, although overall trends are similar to the one seen at 5% and 10%. Major outcomes from the Figure 5 (a, b, and c) are that at the top disc, the flow is almost symmetric in both the flow direction and perpendicular to it, whereas the volume fraction is asymmetric both for the middle and bottom discs in both the directions. However, the level of asymmetry is higher in the perpendicular to the inlet flow direction.

To establish effect of valve opening on asymmetry ratio, Figure 5 (d) has been drawn to show the asymmetry ratio at 60% valve opening, and thus only asymmetry ratio corresponding to the middle and the bottom discs have been shown for a representative input volume fraction of 10%. It can be seen that the asymmetry ratio is nearer one (meaning symmetric flow) in the inlet flow direction as compared to the perpendicular to the inlet flow direction. After establishing the asymmetry in volume fraction as a function of disc position, the input volume fraction and valve opening position, the remaining flow variables have been analysed in the following sections.

3.3.2 Asymmetry in velocity profiles

The flow geometry influences the velocity field considerably. To demonstrate this effect asymmetry ratio in water velocity profiles has been depicted in Figure 6. It can be seen in Figure 6 (a) that at low volume fractions, the asymmetry in velocity values is limited to 50% at the exit. At the inlet the maximum asymmetry is about 30%. Asymmetry in the inlet flow direction is seen to be smaller than
that observed for perpendicular to the inlet flow direction. With increasing volume fraction (Figure 6 b) the asymmetry is about 25% at the inlet and about 30% at the exit. Also, in Figures 6 (a) and (b), it was noticed that maximum asymmetry is noticed for the bottom disc.

Figure 7: Variation of the asymmetry ratio $\xi$ in water velocity for inlet gas volume fraction of (a) 5%, (b) 10%, (c) 15% at VOP = 100% and (d) $\xi$ in water velocity for inlet gas volume fraction of 10% at VOP = 60% for comparison.

For the maximum volume fraction of 0.15%, the trend seems to have changed. Now the asymmetry is in the range of ±20%. At the top disc asymmetry is -20% whereas it is +20% for middle and bottom discs in the inlet direction of flow. In the perpendicular to the inlet direction, the velocity profiles seem to be symmetric. To establish the effect of valve opening position on the asymmetry ratio for velocity profiles, data was collected at a 60% valve opening position. It can be seen that in both the directions the asymmetry is much reduced and is limited to ±8% and most of the asymmetry is seen nearer the middle disc as compared with the bottom disc.

The above discussion clearly indicates that because of multiphase conditions and given that the valve geometry may be complex, considerable flow redistribution takes place resulting in asymmetry as high as 50% and assumption of symmetric flow distribution may be a source of problem often seen in severe service valve applications for multiphase flows.

3.3.3 Asymmetry in pressure profiles

To further explore the causes of flow redistribution, local asymmetry in pressure profiles has been investigated. Typically, these trims are designed to have staged pressure drop in the direction of flow. Asymmetry in the pressure values indicate higher loss of energy because of excessive three dimensionality in the flow. At low volume fraction of 5% (Figure 7 (a)), the pressure profiles are mostly symmetric except at the bottom disc in the perpendicular direction to the inlet flow.
Figure 8: Variation of the asymmetry ratio $\xi$ in pressure for inlet gas volume fractions of (a) 5%, (b) 10%, (c) 15% at VOP = 100% and (d) $\xi$ in pressure for inlet gas volume fraction of 10% at VOP = 60% for comparison.

Similar trend is noticed at higher volume fraction of 10% (Figure 7 (b)) except that now the pressure profile is asymmetric at the bottom disc in the direction of inlet flow as well. Similar trend is noticed at 15% volume fraction as well. At valve opening position of 60% and the inlet volume fraction of 10%, it can be seen in the figure 7(d), that the profiles are only marginally asymmetric. From the above discussion it is clear that pressure profiles are largely symmetric and only at bottom disc significant levels of asymmetry is seen.

3.3.4 Asymmetry in $C_v$

A typical valve is designed for a given $C_v$ obtained from process requirements. Total $C_v$ of a valve is made of $C_v$ of the trim, $C_v$ of the body and $C_v$ of the seat. $C_v$ of the trim in turn depends on local $C_v$ values corresponding to each disc which in turn depends on Cv values of different rows. Typically, in design process it is assumed that $C_v$ of a row is largely independent of angular direction. This assumption helps in the design process as now the equivalent area of flow can be calculated for a given $C_v$ value. We have seen that the volume fraction field and the velocity fields are largely asymmetric and hence we need to investigate whether local $C_v$ values corresponding to rows in each quarter are a function of angular direction.
Figure 9: Variation of the asymmetry ratio \( \xi \) in \( C_v \) for inlet gas volume fractions of (a) 5\%, (b) 10\%, (c) 15\% at VOP = 100\% and (d) \( \xi \) in \( C_v \) for inlet gas volume fraction of 10\% at VOP = 60\% for comparison.

Figure 8 (a) shows the asymmetry ratio for low volume fraction flow (5\%). It can be seen that asymmetry in the row average \( C_v \) corresponding to the quarters in the inlet flow direction and perpendicular to the inlet flow direction is significant for the top disc (of about 10\%). At higher volume fraction of 10\%, this asymmetry is about at 60\% in the inlet flow direction and about 30\% perpendicular to the flow direction. At the highest volume fraction, the asymmetry is of the order of 100\% in the perpendicular to the inlet flow direction and about 40\% in the inlet flow direction. At the valve opening position of 60\% and volume fraction of 10\%, asymmetry of about 20\% can be observed in the perpendicular to the inlet flow direction and of about -4\% in the inlet flow direction both for the bottom disc.

To summarise the foregoing discussion, the local analysis has shown that the \( C_v \) at different quarters and discs is highly non-uniform. When valves are designed, it is assumed that the \( C_v \) is constant for all the quarters and for different discs. Typically, area required for the flow is calculated using Eqn. (11) for a given \( C_v \), where \( \psi \) is a coefficient dependent on the trim and \( A_{eq} \) (the equivalent flow area for each disc quadrant shown in Figure 10):

\[
\psi = \frac{C_v}{A_{eq}} \tag{11}
\]

Due to the non-uniformity of \( C_v \) within the trim, as shown in Figure 4, it is imperative to derive an expression for \( C_v \) as a function of location (radial and axial), operating conditions (inlet gas volume fraction and VOP), as well as local gas volume fraction and local mixture velocity. For the latter two, we have previously shown [11] that they can be conveniently represented by a so-called “distribution parameter” which captures local gas volume fraction non-uniformities within the trim.
As the equivalent area into each trim quarter is the same (as shown in Figure 10), any non-uniformity in Cv value will impact the flow conditions during the actual conditions of operation. To explore this further, the $C_v$ has been implicitly correlated via $\Psi$ to quantify the impact of non-uniformity in $C_v$. The newly developed correlations could be used with the conventional valve design process to ascertain variations in the expected flow performance of the valve.

3.4 $\Psi$ correlation method

In order to describe the effect of non-uniformity of the liquid and gas phases, local distribution parameter $C_o$ was used. The distribution parameter is a variable that describes the effect of non-uniformities in the flow and indicates the difference in cross-sectional area averaged liquid and gas velocities with local velocities [20]. It was initially proposed by Zuber and Findlay [21] and applied to gas–liquid two-phase pipe flow. $C_o$ is calculated using Eqn. (12) at each quarter of the bottom, middle and top discs of the trim. This parameter takes the radial air volume fraction and the phase velocities into consideration [11]:

$$C_{o\,\text{quarter}} = \frac{\frac{1}{N_{\text{rows}} \sum_{N_{\text{rows}}}^{N_{\text{rows}}} \alpha_i v_{\text{mix},i}}}{\frac{1}{N_{\text{rows}} \sum_{N_{\text{rows}}}^{N_{\text{rows}}} v_{\text{mix},i}} \frac{1}{N_{\text{rows}} \sum_{N_{\text{rows}}}^{N_{\text{rows}}} \alpha_i}}$$  

(12)

where, $v_{\text{mix}} = v_{sg} + v_{sl}$ is the local volumetric flux density of the mixture (mixture velocity); $v_{sg} = \alpha v_g$ and $v_{sl} = (1 - \alpha) v_l$, where $v_g$ and $v_l$ are the local gas and liquid velocities respectively. $v_{sg}$ and $v_{sl}$ are known as superficial velocities of gas and liquid phase respectively, defined as the velocity of the phase if it flows alone in the domain. $N_{\text{rows}}$ is the number of rows in the trim disc, which is 5 in this case. Using non-linear regression, the distribution parameter was correlated to the localised position based on the normalised height ($h' = \frac{h}{H}$) of the disc from the valve seat, normalised angle ($\phi = \frac{\theta}{360}$) relative to the valve inlet, and the operating conditions: valve opening position ($VOP$) and the valve air inlet volume fraction ($\alpha_{inlet}$). The resultant expression is as given as:

$$C_o = 1.04 \cdot VOP^{-0.03} \cdot h'^{-0.04} \cdot \phi^{0.035} \cdot \alpha_{inlet}^{0.006}$$  

(13)

The value of $\theta$ for quarters 1, 2, 3 and 4 is 90, 180, 270 and 360 degrees respectively. It takes the values of 0.25, 0.5, 0.75 and 1.0 when normalised and it is represented with $\Phi$. Figure 11 (a) shows the comparison of the predicted $C_o$ from Eqn. (13) with actual $C_o$. The actual $C_o$ plotted on the
horizontal axis is calculated from the CFD data. As seen from the figure, these correlate quite well with the predicted \( C_o \), as the deviations between them are well within the ±10% error band. The average absolute percentage error is 4.2% and \( R^2 = 0.7632 \) which shows a good agreement of the general trend of the dataset. The local \( C_v \) for each quarter on the top, middle and bottom discs was calculated using the harmonised Cvs method using Eqns. (8) and (9). Local \( \psi \) values were then calculated from the \( C_v \) values obtained from CFD and effective area using Eqn. (11).

The local \( \psi \) values were then correlated with the normalised height, normalised angle, valve opening position and \( C_o \) obtained from Eqn. (13). The resulting correlation is shown in Eqn. (14). The comparison of predicted \( \psi \) with the actual \( \psi \) calculated from CFD is shown in Figure 11 (b). It can be seen that the predicted values of \( \psi \) are within the ±10% error band. The average absolute percentage error is 2.7% and \( R^2 = 0.7146 \) which shows a good agreement of the general trend of the dataset. The equation for \( \psi \) obtained using non-linear least squares regression is expressed as follows:

\[
\psi_{h,\theta} = 3.55 \, VOP^{-0.23} \, C_o^{0.42} \left( \frac{h'}{\phi} \right)^{0.34}
\]  

(14)

In actual practice, \( \psi \) is considered to be constant. However, the equations (13 and 14) indicate that \( \psi \) varies for discs and quarters, and thus it indicates deviation from the standard design conditions. Furthermore, this indicates that there will be a certain level of variation in the performance of the valve designed based on harmonised Cvs method. As long as this variation can be minimised or managed, the valve performance may be assumed to be satisfactory. Just to demonstrate this is calculated, for example, in order to obtain the \( \psi \) value at the 6th disc of the trim (from the valve seat) and the third quarter at 100% VOP for the 0.05 inlet gas volume fraction condition, \( h' = 0.5, \alpha_{inlet} = 0.05, \) VOP = 1, and \( \phi = 0.75 \), these will first be substituted to Eqn. (13) to obtain \( C_o = 1.04 \). With the exception of \( \alpha_{inlet} \), these will all be substituted in Eqn. (14) to calculate \( \psi_{h,\theta} = 0.87 \) at the specified conditions, which is only 1.64% more than the actual value. This is an acceptable level of deviation. The calculated value of \( \psi \) can then be used to calculate \( C_v \) for each quarter from which \( C_v \) for the entire valve may be calculated. This will allow to ensure whether the calculated \( C_v \) is equal to the \( C_v \) supplied by the customer before the design process. A good match in the \( C_v \) will indicate satisfactory design as far as global performance is considered. For local performance, the variations in the \( \psi \) value
will be an indicator. For a good design, the $\Psi$ value should vary by only a small amount. While the predictive equations were developed for continuous resistance trim presented in this study, the relevant equations can be written in general for other types of continuous resistance control valve trims as below:

\[ C_o = A_1 V O P b_1 h c_1 \phi d_1 a_{inlet} \]  \hspace{1cm} (15)

\[ \Psi = A_2 V O P b_2 c_o \phi e_2 \]  \hspace{1cm} (16)

Similar studies can be carried out for other valve geometries as well for multiphase flows and predictive equations can be developed to estimate their performance.

The process to use the equations is as follows:

1. Design the valve as per the process conditions ($C_v$ and other parameters) from the customer
2. Depending on the likely inlet gas fraction ($\alpha$), calculate the distribution parameter $C_o$ for the trim at different heights and quarters using Eqn. (13)
3. Calculate $\Psi$ for different quarters using Eqn. (14)
4. Using the effective area ($A_{eq}$) of each quarter and $\Psi$ values calculated in step 2 above, calculate $C_v$ of each quarter Eqn. (11)
5. As the flow in the disc quarters is in parallel, calculate $C_v$ of the disc using Eqn. (17):

\[ C_{v_{disc}} = \sum_{quarter=1}^{4} C_{v_{quarter}} \]  \hspace{1cm} (17)

6. The discs of the trim are in parallel, the $C_v$ of the trim with n discs can be calculated using Eqn. (18):

\[ C_{v_{trim}} = \sum_{disc=1}^{n} C_{v_{disc}} \]  \hspace{1cm} (18)

7. Compare calculated $C_v$ with the original $C_v$ and establish the performance of the valve.

As an example, suppose we want to calculate the $C_v$ of the trim used in this study, which is composed of 11 discs and the gas volume fraction is 5%. Using the steps presented above, the $C_v$ of the trim was calculated as 43.98 which is close to the experimentally obtained $C_v$ of 42.79. This is an error of 2.8% which shows the effectiveness of the method. The local $\Psi$ values obtained from Eqn. (14) for this case are shown in Figure 12 which illustrates the variation of the $\Psi$ factor with height from the seat and disc quarters. $\Psi$ has been non-dimensionalised using the maximum $\Psi$, value within the trim, i.e., $\Psi_{max}$. It is seen that $\Psi$ is lowest at the bottom disc, gradually increasing up to the highest value at the top disc. Additionally, $\Psi$ is the highest in quarter 1 and the lowest in quarter 4. As $\Psi$ is directly proportional to $C_v$, these trends match the variations in the $C_v$ profiles observed in Figure 5.
The above has clearly indicated that the multiphase flow valves have considerable asymmetry in the flow profiles which affect the local flow conditions significantly. From Figure 12, it is clear that the performance of the valve under extreme conditions may be affected by up to 25%. Especially for severe service applications, this may result in several operational problems such as cavitation, flashing, and vibration. Information about the level of asymmetry present will help designers in incorporating changes that may minimise this level of asymmetry. This paper presents three novel equations that can be embedded in existing design methodologies for the development of better performing valves.

4 Conclusions

In this study, we have presented a well-validated CFD study to investigate the local flow characteristics in a severe-service control valve trim under multiphase gas and liquid flow. Three different air inlet volume fractions (5, 10, and 15%) were tested at 60% and 100% valve opening positions. It was found that there is significant maldistribution in the flow within the valve trim. In order to quantify the phase non-uniformities observed, data extracted from specified planes were used to obtain a relationship for the distribution parameter in each disc quarter as a function of the non-dimensional height from the seat and the angular position. Usually, when designing the valves, $\Psi$ is taken to be a constant value. However, the results presented in this study showed that it is different depending upon the height from the seat and the angular position as compared to the valve inlet. The harmonised Cvs method was used to calculate the local $C_v$ and local $\Psi$ factor was calculated. Subsequently, a novel equation was derived that considers the local variation of the $\Psi$ factor within the trim that incorporates the local phase fractions and velocities through the distribution parameter $C_o$. Based on the analysis presented in this paper, a method for linking $C_v$ values at a local level to the global $C_v$ of the valve for multiphase flow applications has been proposed to enable compliance of global flow characteristics.

CRediT statement

D. Singh: Software, Data Curation, Experimentation, Analysis, writing – original draft, review and editing. M. Charlton: Writing – review and editing A. M. Aliyu: Writing – original draft, review and editing.
5 References


[16] T. Asim and R. Mishra, “Computational fluid dynamics based optimal design of


6 Appendix

The correction factors for the liquid phase are calculated according to the following conditions:

\( N_L \) – When the valve pressure drop does not exceed the limit sizing pressure drop of any phase.

**Condition 1)** When: \( \frac{f_G}{f_L} \leq 0.120 \)

\[ N_L = 1 - 3.666 \frac{f_G}{f_L} \]

**Condition 2)** When: \( \frac{f_G}{f_L} > 0.120 \) \( \text{and} \leq 0.300 \)

\[ N_L = 0.687 - 1.055 \frac{f_G}{f_L} \]

**Condition 3)** When: \( \frac{f_G}{f_L} > 0.300 \) \( \text{and} \leq 9.500 \)

\[ N_L = 0.360 + 0.031 \frac{f_G}{f_L} \]

**Condition 4)** When: \( \frac{f_G}{f_L} > 9.500 \) \( \text{and} \leq 16.300 \)

\[ N_L = 0.422 + 0.025 \frac{f_G}{f_L} \]

**Condition 5)** When: \( \frac{f_G}{f_L} > 16.300 \)

\[ N_L = 0.552 + 0.017 \frac{f_G}{f_L} \]

\( N_{LL} \) – When the valve service pressure drop exceeds the liquid limiting sizing pressure drop by 16.7% \( (\Delta P_V = 1.167 \Delta P_{LL}) \) – but the gas sizing limiting pressure drop is not exceeded.

**Condition 1)** When: \( \frac{f_G}{f_L} \leq 0.120 \)

\[ N_{LL} = 1 - 2.892 \frac{f_G}{f_L} \]

**Condition 2)** When: \( \frac{f_G}{f_L} > 0.120 \) \( \text{and} \leq 0.300 \)

\[ N_{LL} = 0.801 - 1.233 \frac{f_G}{f_L} \]

**Condition 3)** When: \( \frac{f_G}{f_L} > 0.300 \) \( \text{and} \leq 9.500 \)

\[ N_{LL} = 0.420 + 0.037 \frac{f_G}{f_L} \]

**Condition 4)** When: \( \frac{f_G}{f_L} > 9.500 \) \( \text{and} \leq 16.300 \)

\[ N_{LL} = 0.494 + 0.029 \frac{f_G}{f_L} \]

**Condition 5)** When: \( \frac{f_G}{f_L} > 16.300 \)

\[ N_{LL} = 0.644 + 0.020 \frac{f_G}{f_L} \]

\( N_{LG} \) – When the valve service pressure drop exceeds the gas limiting sizing pressure drop by 16.7% \( (\Delta P_V = 1.167 \Delta P_{LG}) \) – but the liquid sizing limiting pressure drop is not exceeded.
Condition 1) When: \( \frac{f_g}{f_L} \leq 0.120 \quad N_{LG} = 1 - 3.950 \frac{f_g}{f_L} 

Condition 2) When: \( \frac{f_g}{f_L} > 0.120 \text{ and } \leq 0.300 \quad N_{LG} = 0.646 - 0.993 \frac{f_g}{f_L} 

Condition 3) When: \( \frac{f_g}{f_L} > 0.300 \text{ and } \leq 9.500 \quad N_{LG} = 0.339 + 0.029 \frac{f_g}{f_L} 

Condition 4) When: \( \frac{f_g}{f_L} > 9.500 \text{ and } \leq 16.300 \quad N_{LG} = 0.397 + 0.023 \frac{f_g}{f_L} 

Condition 5) When: \( \frac{f_g}{f_L} > 16.300 \quad N_{LG} = 0.518 + 0.016 \frac{f_g}{f_L} 

The correction factors for the gas phase are calculated according to the following conditions:

\( N_G \) – When the valve pressure drop does not exceed the limit sizing pressure drop of any phase.

Condition 1) When: \( \frac{f_g}{f_L} \leq 2.70 \quad N_G = 0.430 + 0.086 \frac{f_g}{f_L} 

Condition 2) When: \( \frac{f_g}{f_L} > 2.70 \text{ and } \leq 17.80 \quad N_G = 0.605 + 0.021 \frac{f_g}{f_L} 

Condition 3) When: \( \frac{f_g}{f_L} > 17.80 \quad N_G = 0.942 + 0.002 \frac{f_g}{f_L} 

\( N_{GL} \) – When the valve service pressure drop exceeds the liquid limiting sizing pressure drop by 16.7% (\( \Delta P_V = 1.167 \Delta P_{LS} \)) – but the gas sizing limiting pressure drop is not exceeded.

Condition 1) When: \( \frac{f_g}{f_L} \leq 2.70 \quad N_{GL} = 0.405 + 0.081 \frac{f_g}{f_L} 

Condition 2) When: \( \frac{f_g}{f_L} > 2.70 \text{ and } \leq 17.80 \quad N_{GL} = 0.569 + 0.019 \frac{f_g}{f_L} 

Condition 3) When: \( \frac{f_g}{f_L} > 17.80 \quad N_{GL} = 0.886 + 0.001 \frac{f_g}{f_L} 

\( N_{GG} \) – When the valve service pressure drop exceeds the gas limiting sizing pressure drop by 16.7% (\( \Delta P_V = 1.167 \Delta P_{GS} \)) – but the liquid sizing limiting pressure drop is not exceeded.

In this case the \( N_{GG} \) values can be accepted as the same values as the ones used for \( N_G \).