Numerical investigation of a thick plate restriction orifice on the pressure drop performance

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Abstract. This paper presents a numerical study on the thick plate restriction orifice on the pressure drop performance due to various orifice ratio, $\beta$. The restriction orifice was investigated using commercial software package namely, ANSYS. The restriction orifice was modelled using built-in modeler and simulated using Fluent module. The orifice ratio, $\beta$ was varied in the range of 0.5 to 0.75. Various flow velocities were applied from 50 to 420 m/s. The fluid flow in the constriction was hydrocarbon in vapour phase. The preliminary results of discharge coefficients were compared with literature and theoretical values between a sharp-edged and thick plate orifice to show a consistent trend. The results yielded that as the Reynolds number, $Re$ increases, the pressure drop performance increases exponentially. This is more prominent at high $Re$ of $8.75 \times 10^6$ where the pressure drop increases by 63% from the baseline of $Re$ number, $1.08 \times 10^6$ based on orifice ratio of 0.75. For a hydrocarbon with low rheological properties and high Re number, the orifice ratio, $\beta$ is best at higher range of 0.75 which show a pressure drop of 0.18 MPa.

1. Introduction
A restriction orifice is commonly used as the fundamental of orifice meters for many decades, and until recently, the restriction orifice is utilized as pressure-reducing device to reduce the acoustic induced-vibration in a piping system. Typically, the piping system in a high pressure letdown scenario of a typical process industry is usually installed with a pressure control valve for gas or vapour flow. The downstream flow after the pressure control valve has very high pressure which causes vibration in the pipe and subsequently, induce an undesirable high frequency noise. In addition to noise generation, the vibration eventually causes dynamic failure in the pipe over a short period of time.

Some of the restriction orifices [1-7] are: (a) sharp-edged, (b) square-edged (thick plate), (c) quadrant-edged and (d) conical-edged orifice as shown in figure 1, respectively. The shape of the orifice affects the flow behaviour at the constriction and subsequently, affects the local pressure drop and velocity at the exit of the orifice which creates a vena contracta shown in figure 1. The orifice ratio, $\beta$ is defined as the diameter of the orifice, $d$ to the inner diameter of the pipe, $D$ also shown in figure 1.
Figure 1. Restriction orifice for (a) sharp-edged, (b) square-edged, (c) quadrant-edged and (d) conical-edged [4,5,7].

There are two measurements of the pressure drop performance [7] for a restriction orifice which is defined at three pressure locations, conveniently labelled as $P_1$, $P_2$ and $P_3$ as shown in figure 2. These locations, $P_1$, $P_2$ and $P_3$ are called upstream pressure, throat pressure and final recovery pressure, respectively. The pressure drop between $P_1$ and $P_2$ is called the orifice pressure drop, $\Delta P_{orif}$. On the other hand, the pressure drop between $P_1$ and $P_3$ is called the permanent pressure drop, $\Delta P_{perm}$.

The hydraulic performance of the orifice is defined by the discharge coefficient, $C_D$ which is shown in equation (1) is based on [7]. The discharge coefficient is dependent on the volume flowrate, orifice ratio, orifice area and the pressure drop [4,5].

Discharge coefficient, $C_D$

$$C_D = \frac{Q_v\sqrt{1-\beta^4}}{A_o \sqrt{2 \Delta P_{orif}/\rho}} \quad (1)$$

where $C_D$ is the discharge coefficient (-), $Q_v$ is the volumetric flowrate (m$^3$/s), $\varepsilon$ is the expansibility factor (1 for incompressible fluid (-)), $\Delta P_{orif}$ is the pressure drop across orifice ($P_1 - P_2$) (Pa), $\rho$ is the fluid density (kg/m$^3$), $\beta$ is the orifice diameter to inner pipe diameter (-) and $A_o$ is the cross-sectional area of the orifice (m$^2$).
In these papers from Kumar [3] and Huang [7], they studied the effect of various orifice design on the pressure drop performance based on discharge coefficient, $C_D$, and compared the results to a standard flat orifice using air as single-phase gas fluid. In Kumar’s study [3], they investigated numerically the effect of slotted orifice on the discharge coefficient, $C_D$, by using air as two-phase flow, liquid-vapor medium. They varied the aspect ratios of slotted orifices from 1.5 to 3.0 and fixed at orifice ratio, $\beta$ of 0.4. They found that for a slotted orifice, a low orifice ratio is more sensitive to liquid presence in the stream and hence, is preferable for wet gas metering.

On the other hand, Huang [7] studied experimentally on a perforated orifice which shown a higher discharge coefficient, $C_D$ of 22.5 to 25.6% than a standard orifice using water as single phase, liquid medium. Their perforated orifice had at least 6 orifices in a diameter as compared to a single orifice. They varied different orifice ratio, $\beta$ for perforated orifices from 0.338 to 0.668 and also varied mass flow rates from 0 to 4000 kg/hr.

In another paper by Buker [8], they studied the Reynolds number dependence on a sharp-edged orifice and compared with two different designs of orifice with attached protrusion on the bottom and top as shown in figure 3. They found that the Reynolds number for a sharp-edge orifice have insignificant effect on the discharge coefficient, $C_D$ at lower range of orifice ratio from 0.1 to 0.3. An inflection point on the discharge coefficient, $C_D$ can be observed when the orifice ratios are increased from 0.5 to 0.65 for different Reynolds number.

There is one paper by Geng [9] also studied the slotted orifice plate and a standard flat orifice using numerical and experimental method for a gas flow. The standard flat orifice was simulated under one single orifice ratio for different mass flow rates. The slotted orifice shown a higher drag coefficient by about 5% as compared to a standard flat orifice. The results were compared experimentally to show a relative error within $\pm$ 10% for a slotted orifice.

A review on recent papers on orifice plates shows that most of the researchers are keen on irregular shapes such as slotted [10] and concentric shape [11] where both papers used tomography method and numerical method to observe the flow in the orifice qualitatively, respectively. The tomography method [10] was performed on a two-phase flow in the slotted orifice to show an optimum homogeneous flow at $x/D$ of 1.5 and 2.5. The numerical results of concentric shape [11] was compared with theoretical discharge coefficients and found that the relative errors agreed within $\pm$ 10.5%. Flow separation was observed at the bottom part of the upstream and downstream of concentric plate, and not present for an eccentric orifice plate.
All of the papers reviewed [3-11] show that the standard flat orifice used were at limited range and variations which were reported at a single parameters of geometrical design, constriction ratio, flow rate and fluid, i.e. water or gas. Other information pertaining to the standard orifice data is very scarce and difficult to obtain due to confidentiality and proprietary from the manufacturer. Millers [6] coefficient of discharge, \( C_D \) are widely used in the theoretical calculations for orifice meters, flow nozzles and Venturi meters. Thus, in this paper, the aim is to study the effect of various Reynolds number, \( Re \) and orifice ratio, \( \beta \) on the pressure drop performance and drag coefficients of a thick plate restriction orifice.

2. Methodology
The numerical software package by Fluent, ANSYS (Version 17) was utilized to solve the fluid complex equations of continuity, momentum and energy as shown in equation (2), (3) and (4), respectively. A square-edged restriction orifice was modelled in 3-dimensional in a cylindrical pipe. In the setup of the model, some assumptions made were: (a) steady-state flow, (b) isothermal condition across the flow boundary, (c) no-slip boundary condition, (d) no penetration from the pipe wall region to the gas flow margin, (e) no wall roughness, and (f) incompressible flow.

Continuity equation:
\[
\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathbf{u}) = 0
\]  
Momentum equation:
\[
\frac{\partial (\rho \mathbf{u})}{\partial t} + \nabla \cdot (\rho \mathbf{u} \mathbf{u}) = -\nabla p + \nabla \tau
\]  
Energy equation:
\[
\frac{\partial (\rho E)}{\partial t} + \nabla \cdot (\rho \mathbf{u} E + \rho p \mathbf{u}) = 0
\]

where \( \rho \) is the density, \( u \) is the velocity, \( P \) is the pressure, \( \tau \) is the viscous stress tensor, \( E \) is the energy. All cases were simulated in turbulence flow regime (\( Re > 1 \times 10^5 \)) using standard \( k-\varepsilon \) model based on the recommendation by Kumar et al. [3]. The equations for \( k \) and \( \varepsilon \) are as follows:

\[
\frac{\partial}{\partial t} \left( \rho k \right) + \frac{\partial}{\partial x_j} \left( \rho k \mathbf{u}_j \right) = \frac{\partial}{\partial x_j} \left[ \left( \mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] + G_k - \rho \varepsilon
\]  
\[
\frac{\partial}{\partial t} \left( \rho \varepsilon \right) + \frac{\partial}{\partial x_j} \left( \rho \varepsilon \mathbf{u}_j \right) = \frac{\partial}{\partial x_j} \left[ \left( \mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} \right] + C_{1\varepsilon} \frac{\varepsilon}{k} G_k - C_{2\varepsilon} \rho \frac{\varepsilon^2}{k}
\]

where \( G_k \) is the generation of turbulence kinetic energy due to mean velocity gradients and \( C_{1\varepsilon} \), \( C_{2\varepsilon} \), \( \sigma_k \), \( \sigma_\varepsilon \) are the standard \( k-\varepsilon \) model constants, and \( \mu_t \) is the turbulent viscosity \( \left( \rho C_{\mu} \frac{k^2}{\varepsilon} \right) \).
The geometrical dimensions for restriction orifice followed from design standard BS1042, and manufacturer technical sheet from Fisher (6011, US) as shown in figure 4. The entrance and exit length from the orifice was based on the recommendation from Kumar [3] which was 5D before and 10D after the orifice to ensure the flow was fully developed. The diameter of the pipe, D was at 0.3 m. Hence, the length of the pipe was 1.5 m and 3.0 m for entrance length and exit length, respectively.

The fluid used throughout the investigation was a hydrocarbon in gas phase in a process line where the properties of the process gas are summarized in table 1. The variable parameters for simulation are summarized in table 2. The velocities and orifice ratio selected were arbitrarily and within the recommended range as proposed by an industry.

The mass flow rate, $Q$, can be calculated using standard orifice meter shown in equation (7) based on the re-arrangement of equation (1). The calculated theoretical flow rate as a function of Reynolds number, $Re$ is based on pipe diameter, D given in equation (8).

$$ Q_v = \frac{C_D A'_0 \varepsilon \sqrt{\Delta P_{orif}/\rho}}{\sqrt{1-\beta^4}} $$

$$ Re_D = \frac{\rho v D}{\mu} $$

where $\rho$ is the fluid density (kg/m$^3$), $v$ is the fluid velocity (m/s), $D$ is the pipe diameter (m) and $\mu$ is the fluid dynamic viscosity (Pa.s).

Pressure drop, $\Delta P$ of an internal flow due fluid viscous effect can be calculated in equation (9) based on the Darcy friction factor.

$$ \Delta P = \frac{f L \rho v^2}{2} $$

where $f$ is the Darcy friction factor (-), $L$ is the pipe length (m), $D$ is the pipe diameter (m) and $\rho v^2/2$ is the dynamic pressure (Pa)

![Figure 4. 3-D model of orifice with boundary conditions.](image)

**Table 1. Fluid properties of the hydrocarbon gas.**

| Properties       | Value     |
|------------------|-----------|
| Density (kg/m$^3$) | $9.07 \times 10^{-1}$ |
| Viscosity (cP)   | $1.30 \times 10^{-2}$ |
| Molecular Weight (g/mol) | 21.3 |

**Table 2. Variable parameters for simulation.**

| Parameter          | Value               |
|--------------------|---------------------|
| Velocity, $v$ (m/s) | 50, 100, 200, 300, 420 |
| Orifice ratio, $\beta$ (-) | 0.5, 0.7, 0.75 |
The downstream velocity, $v_2$ can be calculated using theoretical formula based on the orifice ratio as shown in equation (10).

$$v_2 = \sqrt{\frac{2\Delta P_{perm}}{\rho(1-\beta^4)}}$$

(10)

where $\Delta P_{perm}$ is the permanent pressure drop ($P_1 - P_3$) (Pa), $\rho$ is the fluid density (kg/m$^3$) and $\beta$ is the orifice ratio (-).

2.1. Mesh Generation

The model was generated with unstructured tetrahedral mesh for simple boundaries. For complex regions such as the constriction area in the restriction orifice were generated with structured hexahedral with at least mesh size of 0.001 mm as shown in the cross-section in figure 5.

![Cross-section of the orifice](image)

**Figure 5.** Mesh generation of restriction orifice.

2.2. Convergence Criterion

The convergence criterion of the simulation was based on the residual value of calculated variables such as mass, velocity components, turbulent kinetic energy, $k$, and turbulent kinetic energy dissipation rate, $\varepsilon$. The simulation was considered as converged when all other variables residual achieved a value of less than $10^{-4}$ when the plots reached a minimum plateau.

3. Grid Independence Study

The number of mesh generated was from $0.56 \times 10^6$ to $1.10 \times 10^6$ and the acceptable mesh size was chosen at $0.96 \times 10^6$ based on the trend that plateau after that mesh size. The result of the grid independence study is plotted in figure 6. Based on this selected mesh size, all other models due to different orifice ratios, $\beta$ were mesh generated within this range.

![Grid independence study](image)

**Figure 6.** Grid independence study on the restriction orifice.
4. Results and Discussions

The results from the numerical study such as drag coefficient, $C_D$, pressure distribution, $P$, orifice pressure drop, $\Delta P_{orif}$, velocity contour, and pressure contour for different orifice ratio, $\beta$ are discussed. As a direct comparison with similar setup and conditions are difficult as aforementioned in the literature review, the closest possible range of conditions, fluid and geometry are compared with Kumar [3], Miller [6] and Buker [9]. The drag coefficients, $C_D$ are shown in figure 7, 8 and 9. A theoretical comparison is also demonstrated using internal flow equation of volume flow rate from the pressure drop performance is shown in figure 9. The plots of pressure distribution against axial distance, $x$ from the pipe entrance to the exit where total length is 4.5 m, are plotted in figure 10.

Shown in figure 7, the shape of the orifice affects the discharge coefficient by about 64% as compared between a sharp-edged and a thick plate orifice. The thick plate orifice has lower discharge coefficient than the sharp-edged orifice. Similarly, both have an increasing trend when the orifice ratio increase. However, based on the theoretical results by Buker [9], the discharge coefficient, $C_D$ for a sharp-edged orifice reaches to a peak at orifice ratio, $\beta$ of 0.63 and $C_D$ reduces beyond that orifice ratio. An extrapolation of the theoretical results by Buker [9] shows that an intersection at orifice ratio of 0.71 between thick plate and sharp-edged orifice.

The results of discharge coefficients, $C_D$ for different Reynolds number, $Re$ are plotted in Figure 8 where a comparison with Buker [9] for orifice ratio of 0.5 for sharp-edged orifice. Buker [9] results were of higher margin than the current study due to the shape of the orifice. A thick plate orifice has lower discharge coefficients due to larger drag area than the sharp-edge orifice, and also due to the rheological properties of the hydrocarbon fluid where density and viscosity is lower than air by about 26.3% and 27.3%, respectively. The flow performance of downstream velocities were calculated based on the permanent pressure drop, $\Delta P_{perm}$ from simulation using equation (9) and compared with theoretical equation (10) which shown a deviation of 6.63% to 20.2% in figure 9. This is expected due to the absence of discharge coefficient, $C_D$ in the theoretical flow equation, and a similar error margin of 10.5% was also demonstrated in paper [11].

Based on figure 10, the pressure distributions show a similar trend with Kumar [3] where a direct comparison is not possible due to the different fluid and velocities being used. The model was extended to orifice ratio, $\beta$ of 0.70 and 0.75 for velocities of 50 to 420 m/s to observe the trend. The pressure distribution can be observed that as the velocity increases the pressure decreases by about 460%, at the highest velocity of 420 m/s for orifice ratio of 0.5. This is expected as the orifice is so narrow that the passage will incur a high pressure drop. On the other hand, the orifice ratio of 0.70 and 0.75 yields about 87% and 48% in pressure reduction, respectively.

![Figure 7. Discharge coefficients, $C_D$ for different orifice ratio, $\beta$.](image-url)
**Figure 8.** Discharge coefficient, $C_D$ versus Reynolds number, $Re$ for different orifice ratio, $\beta$.

**Figure 9.** A comparison of flow performance between simulation and theoretical values for different orifice ratios, $\beta$.

**Figure 10.** Pressure profiles along the distance, $x$ for different orifice ratios, $\beta$ of (a) 0.5, (b) 0.7 and (c) 0.75.
The permanent pressure drop, $P_{\text{perm}} (P_1 - P_3)$ for different orifice ratios, $\beta$ and different Reynolds number, $Re$ are plotted in figure 11. As the orifice ratio increases, the pressure drop also increases exponentially from $Re$ of $1.04 \times 10^6$ to $8.75 \times 10^6$ as shown in the trend lines. All trend lines are plotted with best fit, $R^2$ of 0.99, on average. The increment in pressure drop is about 63% at orifice ratio of 0.75 from the same range of Reynolds number. A tremendous amount of penalty in pressure drop is observed when the orifice ratio reduces from 0.75 to 0.50 which yields to about 837% at the highest $Re$ of $8.75 \times 10^6$.

The velocity and pressure contour for different orifice ratios, $\beta$ is shown in figure 12 and figure 13, respectively. As orifice ratio of 0.70 is an intermediate result between ratio of 0.5 and 0.75, only two sets of results are shown here for orifice ratio of 0.50 and 0.75.

Based on figure 12 (a), the velocity contour at orifice ratio of 0.50 shows that the velocity vectors are widely scattered along axial distance demonstrating a trend of non-uniformity. On the other hand, orifice ratio of 0.75 shows velocity vector of more uniform distribution shown in figure 12 (b). This is similar to the pressure distribution shown in figure 13 (a) and (b) where orifice ratio of 0.75 is more uniform than 0.50. This can be seen in the darker blue regimes between orifice ratio of 0.75 and 0.50 for Reynolds number from $1.04 \times 10^6$ to $8.75 \times 10^6$. It is desirable to have uniform pressure distribution in a piping system than a non-uniform distribution for different Reynolds number.

![Figure 11](image_url)

**Figure 11.** Pressure drop, $\Delta P_{\text{perm}}$ vs. Reynolds number, $Re$ for different orifice ratios, $\beta$. 

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Figure 12. Velocity contour for orifice ratio of (a) 0.50 and (b) 0.75.
5. Conclusions
In this study, the velocity and pressure distribution due to different orifice ratios, $\beta$ from 0.50, 0.70 and 0.75 are compiled and analyzed for its pressure drop performance. Based on both contours, velocity and pressure, some conclusions are made:

a. Restriction orifice with flat protrusion has lower discharge coefficients that sharp-edged protrusion.
b. Thick plate orifice of 0.50 provides a tremendous penalty in pressure drop that yields to about 837%.
c. Thick plate orifice of 0.75 sufficiently provides a pressure drop of 63% at the highest Reynolds number.
d. Velocity and pressure contours shows that the thick plate orifice of 0.75 provides uniform distribution than orifice ratio of 0.50.

6. Recommendation and Future Studies
For future studies, it is recommended that: (a) an experimental study to verify the thick plate shape; (b) a numerical study of a thick plate in a piping system consisting of various components, and (c) a numerical study on the thick plate in comparison with various orifice shapes on the pressure drop performance.

Figure 13. Pressure contour for orifice ratio of (a) 0.50 and (b) 0.75.
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