Performance evaluation of orifice plate assemblies under non-standard conditions using CFD

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The performance characteristics of small diameter orificemeters (diameter = 40 mm NB) for four diameter ratios with higher plate thicknesses are evaluated using computational fluid dynamics (CFD) under the non-standard conditions using water as the working fluid. The plate thicknesses considered are larger than the maximum allowed thickness specified in standard, ISO- 5167. The study has shown the effect of plate thickness on the value of \( C_d \) to be only marginal up to twice the allowable plate thickness for all the diameter ratios studied. The value of the \( C_d \) marginally decreases with increase in the plate thickness for lower \( \beta \) values, whereas the value of \( C_d \) increases with increase in plate thicknesses for higher \( \beta \) values.

Keywords: Orificemeter, Discharge coefficient, Computational fluid dynamics, Reynolds number, Plate thickness, Diameter ratio

The orificemeter is one of the oldest and most widely used conventional differential pressure flow meter, and it is preferred in industries due to ease in fabrication, installation and maintenance. Although it is a precise flow meter, but its use is restricted by guidelines specified by different standards; BS 1042 and ISO 5167. ISO 5167 prescribes the standard design of the orificemeter with respect to various parameters, namely, flow-rate, differential pressure, pipe diameter, diameter ratio, plate thickness and working fluid. The performance of the orificemeter is quite sensitive to installation conditions like upstream pipe fittings, valves and other flow disturbances. The upstream flow disturbance causes the distortion in the velocity profile that subsequently leads to adverse effect on the performance of orificemeter. In order to overcome these, minimum upstream and downstream straight pipeline lengths are recommended in the standards for installation of the orificemeter. Extensive and systematic experiments have been carried out over the years to evaluate the performance of the orificemeter. Theory, calibration and installation requirement of the orificemeter are also well documented. Reader-Harris et al. have proposed the equation for the calculation of the discharge coefficient of the orificemeter for different pressure tappings and diameter ratios. Hobbs and Humphreys investigated the effect of upstream sharpness of the orifice plate on the discharge coefficient, and also traced the standard requirement for high edge sharpness. Morrison et al. observed the significant effect of the upstream velocity profile on the performance of the orificemeter, and showed that the increase of the centerline velocity causes the reduction in the differential pressure across the orifice plate, which results in increase of discharge coefficient. Martin analyzed the types of disturbances created by pipe fittings and valves, and their effects on the performance of the orificemeter. Irving calculated the discharge coefficient for upstream disturbed flows and different pressure tappings. He observed that the errors in measurement are less sensitive to flow disturbance, and subsequently recommended the use of standard tappings to rectify the errors in the flow measurements. Singh et al. have proposed the design of a self-adjusting variable area orificemeter with conical body. The flow rate is a linear function of the displacement of cone, and the differential pressure in the meter remains nearly constant over a wide range of flow rates. Husain and Goodson have reported the effect of the plate thickness and bevel angle on
the discharge coefficient of the orificemeter having 50 mm pipe diameter. Kim et al.\textsuperscript{11} observed the significant effects of the plate thickness on the performance of the orificemeter at low diameter ratio ($\beta = 0.10$), but at higher $\beta$, no significant conclusions were drawn. Consequently, they suggested the need of more studies to investigate the effect of the plate thicknesses for a wide range of diameter ratios. Recently, Singh et al.\textsuperscript{12} have also studied the performance of the orifice plate at non-standard condition and emphasized the need of more studies to quantify the effect of plate thickness.

Computational fluid dynamics (CFD) has emerged as an effective alternative tool to rigorous and sophisticated experiments because of its diverse applications in many industries. Over the last two decades, the use of commercial codes has increased tremendously because of their convenience and time efficiency in industries\textsuperscript{13}. The CFD simulation of any flowmeter is still complex, and careful selection of the numerical technique is required for accurate and precise numerical simulations. Bückle et al.\textsuperscript{14} have demonstrated the capability of CFD in the design improvement of the rotameter, and emphasized that CFD gives a better insight of the flow structure, particularly in the region of existing strong velocity gradient and recirculation zone above the float. Davis and Mattingly\textsuperscript{15} modeled the orificemeter of different diameter ratios and evaluated the performance over a wide range of Reynolds numbers. They found excellent agreement between the experimental and predicted values of the discharge coefficient. Erdal and Anderson\textsuperscript{16} have shown the limitation of the Standard $k$-$\varepsilon$ model for the performance prediction of the orificemeter, and emphasized the use of more advanced turbulence model to improve the numerical predictions. More recently, Ganiev et al.\textsuperscript{17} also discussed the choice of turbulence models to compute the discharge coefficient of standard orificemeter, and showed the suitability of Reynolds stress model (RSM) at high Reynolds number.

The literature shows that the CFD is a useful tool for design improvement of the orifice plate by providing the detailed flow features for the flowmeter. The present work extends the use of CFD tool for the performance evaluation of the orificemeter at non-standard conditions, as suggested by Singh et al.\textsuperscript{12} and Kim et al.\textsuperscript{11}. The existence of high pressure in the pipeline in many industrial applications results in the need of higher plate thicknesses for ensuring mechanical strength. In the present study, the effect of plate thickness on the performance of the orifice meter has been investigated. The design parameters of the orifice plate as given in codes are 0.05D for plate thickness, 0.02D for throat edge thickness and bevel angle in range of 30 to 45°.

**Mathematical Modeling**

**Governing equations**

The commercially available CFD code FLUENT\textsuperscript{18} has been used in the present study and is briefly described for completeness. The governing equations for steady and incompressible turbulent flow are given as:

$$\rho \frac{\partial u_i}{\partial x_i} = S_m \quad \ldots (1)$$

$$\frac{\partial}{\partial x_j} \left( \rho u_i u_j \right) = \frac{\partial P_i}{\partial x_j} + \frac{\partial \tau_{ij}}{\partial x_j} + \rho g_i + F_i + \frac{\partial}{\partial x_j} \left( -\rho u_i u_j \right) \quad \ldots (2)$$

The term $-\rho u_i u_j$ in the above equation is the Reynolds stress, and it needs to be modeled for closure solutions. The Boussinesq hypothesis is used to relate the Reynolds stresses to the mean velocity gradient, and it is computed as:

$$-\rho u_i u_j = \mu_t \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \frac{2}{3} \left( \rho k + \mu_t \frac{\partial u_i}{\partial x_i} \right) \delta_{ij} \quad \ldots (3)$$

In the present investigation, the Reynolds stress model (RSM)\textsuperscript{19} has been used for the closure solution for the set of equations.

The transport equations for the turbulence kinetic energy and its dissipation are required to obtain the boundary conditions for Reynolds stresses, which are given as:

$$\rho u_i \frac{\partial k}{\partial x_i} = \frac{\partial}{\partial x_i} \left[ \alpha_k u_{eff} \frac{\partial k}{\partial x_i} \right] + G_k - \rho \varepsilon \quad \ldots (4)$$

$$\rho u_i \frac{\partial \varepsilon}{\partial x_i} = \frac{\partial}{\partial x_i} \left[ \alpha_{\varepsilon} u_{eff} \frac{\partial \varepsilon}{\partial x_i} \right] + C_{1\varepsilon} \frac{\varepsilon}{k} (G_k) - C_{2\varepsilon} \frac{\rho \varepsilon^2}{k} - R \quad \ldots (5)$$
where, $G_k$ is the generation of turbulent kinetic energy due to the mean velocity gradient, and is calculated as:

$$G_k = -\rho u_i u_j \frac{\partial u_j}{\partial x_i}$$ … (6)

The eddy viscosity, $\mu_t$ is computed as:

$$\mu_t = \rho C_{\mu} \frac{k^2}{\varepsilon}$$ … (7)

where $C_{\mu}$ is a constant.

The values of the constants used in the turbulence model are the standard values reported in the literature.

Wall modelling

The proper modeling of the wall region is a vital step for wall-bounded turbulent flows. The flow in the region close to the wall has to be properly modeled, since viscous effects are pronounced there. In the present study, walls are modelled using the standard wall function with equilibrium turbulent boundary layer assumptions.

The effect of roughness is incorporated using the roughness constant, and the velocity profile is given as:

$$u^+ = \frac{\bar{u}}{u^*} = 1 + \frac{1}{K} \ln \left( \frac{E u^* K_h}{\nu} \right) - \Delta B$$ … (8)

where, $K$ (von-Karman constant) = 0.41, $E = 9.8$ and $\Delta B = 5$, $K_h$ is roughness height, $u^*$ is the frictional velocity.

Solution scheme

2D axi-symmetry flow simulations were carried out using the CFD Code “FLUENT”, which is based on the cell centered finite volume approach. The second order scheme was used for discretization of all governing equations, since grids consist of tetrahedral cells. The under-relaxation factors were used for all flow parameters to satisfy the Scarborough condition for convergence of the solution. The coupling between the pressure and velocity field was established using the SIMPLE algorithm. The discretized equations were solved by the segregated solver with an implicit solution scheme. The Algebraic Multigrid (AMG) solver was used for faster convergence of the solution. The double precision solver was used in computation, and solutions were converged until the normalized residual of all variables were less than $10^{-6}$.

Validation of the Code

Validation of the CFD code establishes the selection of appropriate turbulence models and extent of accuracy and reliability of the numerical simulations. The CFD predicted results are only accepted after the validation of the numerical scheme. The CFD code was validated against the experimental results of Singh et al. for orifice plate ($\beta = 0.50$) at non-standard conditions. The geometry of the orifice plate (Fig. 1) used for the validation is a standard orifice plate with 45° bevel angle. The upstream and downstream lengths of the pipeline were kept as 5$D$ and 40$D$ respectively in the simulation. The longer downstream length was chosen in order to ensure the proper specifications of the boundary condition at the exit of the pipe line. The schematic layout of the geometry of the computational flow domain that includes the orifice plate with upstream and downstream straight pipeline is shown in Fig. 2. The geometry of the flow domain was modeled using the bottom-up approach in GAMBIT, the CAD tool of FLUENT18, according to dimensions in the drawing. The process of grid generation is very crucial for accuracy, stability and economy of the predictions. A fine grid leads to better accuracy in the region of steep velocity gradient, and it is necessary to generate reasonably finer grid there. For regions where smooth flow exists coarser mesh could be used. The computational flow domain was meshed with structured grids. For efficient discretization, the
geometry was divided into three parts: upstream, downstream, and central part comprising the orifice plate and pressure taps. The upstream and downstream sections of the flow domains were meshed with reasonably fine grid. The sizes of grid were kept very fine in the central region to account for the expected steep velocity gradient and recirculation zone near the orifice plate. The discretization scheme used for meshing of the different sections of the flow domain are depicted in Fig. 3. For each case, the optimum number of grids was decided after grid independency tests. The grid independency tests were carried out by grid adaptation and comparison of the value of $C_d$ for different grid sizes over a wide range of Reynolds number. Figure 4a shows the grid independence tests, and it is seen that $C_d$ values corresponding to 226,920 and 318,370 cells have almost identical value in the range of Reynolds numbers studied. A very slight deviation in $C_d$ is seen in the lower range of Reynolds numbers, and hence we have chosen 226,920 as optimum number of cells for subsequent simulations using different turbulence models with water as the

**Table 1—Percentage deviation of the predicted values of $C_d$ from the experimental values (0.5973) with different turbulence models**

<table>
<thead>
<tr>
<th>Turbulence model</th>
<th>Discharge coefficient</th>
<th>Percentage deviations</th>
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</thead>
<tbody>
<tr>
<td>Realizable $k$-$\varepsilon$ model</td>
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<tr>
<td>Standard $k$-$\varepsilon$ model</td>
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<td>4.88</td>
</tr>
<tr>
<td>RNG $k$-$\varepsilon$ model</td>
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</tr>
<tr>
<td>Reynolds stress model</td>
<td>0.6132</td>
<td>2.67</td>
</tr>
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</table>

**Fig. 2—Schematic layout of the flow domain showing the position of the orifice plate**

**Fig. 3—Discretized geometry of the flow domain showing the fine meshing in the central zone**

**Fig. 4—FE SEM analysis of SiC abrasive practical**

صغر عنصر الإكليل في اللغة العربية: 

**الجدول 1—الاختلاف النسبة للقيمة المقدرة في $C_d$ من القيمة التعبيري (0.5973) مع موديلات التурбуلتيفية المختلفة**

<table>
<thead>
<tr>
<th>موديل التوربولاينج</th>
<th>عامل التدفق</th>
<th>اختلاف النسبة (%)</th>
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<tr>
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<td>موديل $k$-$\varepsilon$ القياسي</td>
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<td>موديل RNG $k$-$\varepsilon$</td>
<td>0.6178</td>
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<tr>
<td>موديل التوتر Reynolds</td>
<td>0.6132</td>
<td>2.67</td>
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</tbody>
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*صورة 2—النمط المصممي لمنطقة التدفق يوضح موقع الأشكال* 

*صورة 3—نمط التوزيع المفصل للمنطقة التدفق يوضح التغليف الفائق في المنطقة المركزية* 

*صورة 4—تحليل SEM لحبيبات SiC الأصلية*
working fluid. The turbulence models used are two equation $k$-$\varepsilon$ model and six equation Reynolds stress model (see Table 1). Figure 4b shows the comparison of the experimental values and predicted values of the discharge coefficient with different turbulence models. It can be seen that the Reynolds stress model and RNG $k$-$\varepsilon$ model give similar matching with experimental results, but Reynolds stress model gives the closest matching with experimental values over a wide range of Reynolds number. The Standard $k$-$\varepsilon$ model predicts higher values for $C_d$ among all turbulence models used. Erdal and Andersson\textsuperscript{16} also concluded that the Standard $k$-$\varepsilon$ model fails to describe flow features of the flowmeter accurately. The deviation between the experimental and predicted values of the discharge coefficients are of the same order as the overall uncertainties in experimental and computational analysis. Table 1 shows the deviation between predicted and experimental values of discharge coefficients with different turbulence models. These deviations could be attributed to the limitations of the turbulence model, uncertainty in the experimental results, roughness of the pipe, etc. The maximum difference between the experimental and predicted values of the discharge coefficients using the RSM turbulence model is 2.70\%, which also establishes the accuracy of the CFD methodology.

Range of Geometry, Boundary Conditions and Parameters

Flow predictions were carried out for concentric orificemeter having pipe diameter of 40 mm NB (Nominal Bore) and four diameter ratios (0.40, 0.50, 0.60 and 0.70). The orifice meter simulated had a 30\degree bevel angle and 1.0 mm edge thickness. The thickness of the plate was also varied from 3.50 to 9.00 mm, which are higher than the specified thickness (3.20 mm) in ISO-5167\textsuperscript{3}. The grid independency tests were carried out for each case but plots are not presented for sake of brevity. The optimum number of meshes for each case are presented in Table 2.

The boundary conditions used for flow analysis through the orifice meter include the specifications of fully developed turbulent flow (following $1/7$\textsuperscript{th} power law) at the inlet and pressure outlet with zero gauge pressure at the exit of the pipeline. The centerline of the pipe was specified as axis for axi-symmetry flow simulations. Remaining faces of the flow domain including pipe wall and surfaces of the orifice plate

<table>
<thead>
<tr>
<th>Sl. No.</th>
<th>Diameter ratio</th>
<th>Plate thickness (mm)</th>
<th>Cells</th>
<th>Faces</th>
<th>Nodes</th>
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<td>338346</td>
<td>674658</td>
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were specified as wall with no slip condition. The surface of the orifice plate was specified as fully smooth whereas surface of the pipeline was modelled as rough wall with 1.0 mm roughness height and 0.50 roughness constant. The turbulence intensity ($I = 0.16 \times (Re)^{1/8}$) and hydraulic diameter ($H = 4A/P$) were specified as turbulence quantities at the inlet and outlet boundaries of the flow domain. The flow simulations were carried out for four diameter ratios and four plate thicknesses over a wide range of Reynolds numbers ($1.5 \times 10^4 - 1.0 \times 10^6$).

For evaluating the performance of orifice meter under different conditions, around 350 converged runs were performed. One run with 350,000 nodes required approximately 24 h for full convergence on an Intel Core2 Quad Processor with 4GB RAM.

**Results and Discussion**

After successful validation of the CFD code, the flow simulations were carried out to study the effect of the plate thickness on the discharge coefficient of the orificemeter over a wide range of Reynolds numbers. The flow simulations were carried out with water as the working fluid for different $\beta$ ratios. The discharge coefficients of the orificemeter were calculated using the standard formulae for incompressible fluid flow.

$$Q = \frac{C_d \pi d^2}{\sqrt{1 - \beta^2}} \times \sqrt{2 \rho \Delta P}$$  \hspace{1cm} (4)

where, $\Delta P$ is the differential pressure across the orifice plate.

Statistical analysis for each case was also carried out to evaluate the mean value, standard deviation and standard error in discharge coefficients using standard formula.

**Effect of plate thickness**

The discharge coefficients of the orifice plate were calculated for standard flange tappings. The effect of the plate thickness on the discharge coefficients for four $\beta$ values (0.4, 0.5, 0.6 and 0.7) are shown in Figs 5-8.

![Fig. 5–Variation of discharge coefficient with Reynolds number for different plate thicknesses for $\beta = 0.40 (D = 40 \text{ mm})$](image5)

![Fig. 6–Variation of discharge coefficient with Reynolds number for different plate thicknesses for $\beta = 0.50 (D = 40 \text{ mm})$](image6)

![Fig. 7–Variation of discharge coefficient with Reynolds number for different plate thicknesses for $\beta = 0.60 (D = 40 \text{ mm})$](image7)

![Fig. 8–Variation of discharge coefficient with Reynolds number for different plate thicknesses for $\beta = 0.70 (D = 40 \text{ mm})$](image8)
Figures 5 and 6 show the variation of the discharge coefficient of orificemeter with Reynolds number for four plate thicknesses of the orificemeter with $\beta = 0.40$ and 0.50 respectively. The value of the discharge coefficient is higher at low Reynolds numbers and it slightly reduces with increase in the Reynolds number up to $Re = 10^5$, and then it becomes nearly constant depicting a very weak dependence on Reynolds number. This variation of the discharge coefficient with Reynolds number has already been shown by earlier researchers. The variation of the predicted value of the discharge coefficients is seen to be parallel to the computed values of the discharge coefficients from the equations prescribed by ISO-5167. The predicted values of the discharge coefficient are slightly lower than the corresponding codal values and the deviation between the predicted and theoretical values of the discharge coefficient is about 2%. This may be due to the non-standard design of the orifice plate in the simulations. Further, it is also seen that the variation in values of the discharge coefficient are small with maximum deviation of 0.08% for the plate thickness of 7.00 mm (Table 3) (twice the thickness recommended by ISO-5167). The increase in plate thickness leads to the small reduction in the average value of discharge coefficient for the orificemeter of $\beta=0.4$, however, uncertainties in the discharge coefficients are nearly same. This phenomenon can be explained from the size of the recirculation zone formed at the downstream of the orifice plate, which is altered by plate thickness. The size of the recirculating zone increases in this case and higher losses occur at the downstream of the orifice plate (see Fig. 9). The variation of the plate thickness has also similar effect on the discharge coefficient of the orifice plate with $\beta = 0.50$ (Fig. 6), and it reduces slightly with increase of plate thickness. The orifice meter having plate thickness of 9 mm has the least value of average discharge coefficient (0.5980) whereas the thinnest

![Fig. 9–Plots of the recirculation zone in the downstream region of the orifice plate showing the variation of its extent with plate thicknesses for $\beta = 0.40$](image-url)
plate (3.50 mm) has the highest value (0.6011). The uncertainty in computations of the discharge coefficients over the range of Reynolds number studied is nearly same for all cases. Husain and Goodson\textsuperscript{10} have also shown that the effect of plate thicknesses is marginal for orificemeter at lower \( \beta \) value (0.30) for a 50 mm pipe diameter. In the experimental study, the orificemeter with 6.34 mm plate thickness has nearly same discharge coefficient as compared to the reference plate thickness (3.18 mm).

The variations of the discharge coefficient with plate thickness of the orificemeter at higher diameter ratios (0.60 and 0.70) are presented in Figs 7 and 8. The trends are reversed for the variation of the discharge coefficient as compared to those at low \( \beta \) (0.40 and 0.50). The value of \( C_d \) increases marginally with an increase in the plate thickness at both the diameter ratios. Discharge coefficients have no significant variation with change of the plate thickness of the orifice plate for \( \beta = 0.60 \), and it is nearly same for all cases. There is a marginal increase (0.10\%) of the discharge coefficient for 9.00 mm plate thickness corresponding to the base plate thickness (3.50 mm). Percentage uncertainty also increases, but it can be assumed to be constant within a small limit. The variation of the discharge coefficient is seen to be significant for the orifice plate with \( \beta = 0.70 \). The orificemeter with plate thicknesses of 3.50 and 5.10 mm have nearly same values (0.6271 and 0.6277 respectively) of the discharge coefficient. The percentage uncertainty and standard deviation are slightly higher at both \( \beta \) values, but by only a small margin. The effect of the plate thickness is visible at higher plate thicknesses (7.00 and 9.00 mm), and the value of discharge coefficient increases by 0.83\% of the value of the discharge coefficient for reference plate thickness (~3.50 mm) as per ISO-5167. The increase in the value of discharge coefficients for \( \beta = 0.70 \) at higher plate thicknesses (7.00 mm and

<table>
<thead>
<tr>
<th>Sl. No.</th>
<th>Diameter ratio</th>
<th>Plate thickness</th>
<th>Mean value</th>
<th>Standard deviation</th>
<th>Percentage errors</th>
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<td>0.5925</td>
<td>0.0020</td>
<td>0.36</td>
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<tr>
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<td>0.0020</td>
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<td>0.0069</td>
<td>1.17</td>
</tr>
<tr>
<td></td>
<td></td>
<td>5.10 mm</td>
<td>0.6277</td>
<td>0.0072</td>
<td>1.23</td>
</tr>
<tr>
<td></td>
<td></td>
<td>7.00 mm</td>
<td>0.6323</td>
<td>0.0076</td>
<td>1.28</td>
</tr>
<tr>
<td></td>
<td></td>
<td>9.00 mm</td>
<td>0.6323</td>
<td>0.0081</td>
<td>1.37</td>
</tr>
</tbody>
</table>

Table 4– Comparison of the discharge coefficients of orificemeter (\( \beta = 0.50 \)) at two bevel angles of the plate

<table>
<thead>
<tr>
<th>Plate thickness</th>
<th>Mean value</th>
<th>Standard deviation</th>
<th>Percentage errors</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>30°</td>
<td>45°</td>
<td>30°</td>
</tr>
<tr>
<td>3.50 mm</td>
<td>0.6011</td>
<td>0.6135</td>
<td>0.0038</td>
</tr>
<tr>
<td>5.10 mm</td>
<td>0.6009</td>
<td>0.6084</td>
<td>0.0040</td>
</tr>
<tr>
<td>7.00 mm</td>
<td>0.6006</td>
<td>0.6077</td>
<td>0.0041</td>
</tr>
<tr>
<td>9.00 mm</td>
<td>0.5980</td>
<td>0.6040</td>
<td>0.0037</td>
</tr>
</tbody>
</table>
above) could be attributed to the nozzle effect created as a result of the thickness. Subsequently, downstream pressure taps lies in the region where pressure starts recovering. The computed values of the average discharge coefficient, standard deviations and percentage uncertainty for different plate thicknesses are tabulated in Table 3. The effect of plate thickness has negligible impact on the percentage uncertainty and it is nearly constant.

### Effect of bevel angle

The effect of the bevel angle on the discharge coefficient within the specified plate thickness has also been investigated for an orificemeter having $\beta=0.50$. With all other parameters being kept the same. The average value of the discharge coefficient as a function of the plate thickness is plotted in Fig. 10 and is given in Table 4. It is seen that the orificemeter corresponding to a 30° bevel angle has small variation in the value of discharge coefficients, and has a weak dependence on the plate thickness up to 7.00 mm beyond which the value of $C_d$ reduces with increase in plate thickness. The uncertainty in the value of discharge coefficients is also nearly the same for all the plate thicknesses (Table 4). Husain and Goodson$^{10}$ have also shown that the effect of the plate thickness reduces for a 30° bevel angle. Discharge coefficient has little variation for a 45° bevel angle, and it also slightly reduces with increase in the plate thickness. Uncertainty in the discharge coefficients also increases with increase of plate thicknesses (0.65% for 3.5 mm and 0.78% for 9.00 mm).

### Conclusions

Parametric investigations carried out to study the performance characteristics of a 40 mm NB orifice plate assembly with higher plate thickness has proved the adequate capability of the six equation Reynolds stress turbulence model for performance prediction of the orificemeters. The predicted values of the discharge coefficients match fairly well with experimental values over a wide range of Reynolds numbers. At lower $\beta$ values (0.40 and 0.50), the discharge coefficient of an orificemeter decreases marginally with increase in plate thickness. For higher diameter ratio (0.60 and 0.70), the discharge coefficient increases slightly with increase in the plate thickness. The value of discharge coefficient is altered for the plate thickness beyond 7.00 mm (twice recommended value of plate thickness by ISO-5167). The bevel angle of the orifice plate has marginal impact on the discharge coefficient of orifice plate for different plate thicknesses. The average value of the discharge coefficient at 45° bevel angle is higher than the corresponding values at 30° bevel angle.

### Nomenclatures

- $D$: pipe diameter, mm
- $d$: orifice hole diameter, mm
- $\beta$: diameter ratio = $d/D$
- $Re$: Reynolds number
- $\rho$: density, kg/m$^3$
- $p$: static pressure, Pa
- $Y$: expansionability factor
- $C_d$: discharge coefficient
- $\Delta P$: differential pressure across the pressure tappings, Pa
- $Q$: mass flow rate, kg/s
- $\mu$: dynamic viscosity of the fluid, Pa-S
- $C_{1e}$, $C_{2e}$, $C_\mu$: empirical constants of turbulence model
- $\sigma_k$, $\sigma_\epsilon$: turbulent Prandtl number based on $k$ and $\epsilon$
- $G_k$: generation term (kinetic energy)
- $Y_{id}$: contribution of the fluctuation dilation
- $k$: turbulent kinetic energy
- $M$: number of dependent variable
- $R_i$: sum residual for a dependent variable
- $S_m$: mass added to the continuous phase
- $S_{Np}$: normalizing factor
- $U$: mass average inlet velocity
- $u$: instantaneous velocity
- $\bar{u}$: time averaged mean velocity
- $u'$: velocity perturbation
- $u_b$: bulk velocity
- $V$: cell volume
- $V_f$: mass flux (velocity) through the face
- $X$: longitudinal coordinate
- $o$: outlet
- $in$: inlet
- $i,j,k$: tensorial notations

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Fig. 10–Variation of the average value of the discharge coefficient with Reynolds number at two bevel angles for $\beta = 0.50$
References