Numerical Investigation of Beta Ratio and Reynolds Number Effect on Coefficient of Discharge of Venturimeter

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Abstract. The present work is intended to give information on the construction and performance of the venturimeter. It also indicates the need for more effective utilization of venturimeter and offers a conceptual design of a venturimeter for determining the flow rate of a fluid flowing through the pipe. The experiment was conducted with three different Beta ratios ($\beta$) of 0.3, 0.5, and 0.7 installed in a pipe of 2.54 cm diameter with water as a working fluid. The study mainly focused on the effects of beta ratio and Reynolds number on coefficient of discharge. A two-dimensional model of venturimeter with different Beta ratios ($\beta$) was also created to do simulation & compare the numerical results with the experiments and to further explore the capability of the CFD code Ansys Fluent© to visualize the flow patterns in the venturimeter, recognize eddies and wake regions at diverging part and to plot and compare velocity profiles and pressure distribution at selected zones in the venturimeter. It was seen that the change in beta ratio and Reynolds number has considerable impact on Coefficient of discharge.

Keywords: Venturimeter; Beta ratio; Reynolds number; Coefficient of discharge; Computational Fluid Dynamics.

NOMENCLATURE

| Symbol | Description |
|--------|-------------|
| $R$    | Reynolds number |
| $Cv$   | Convergent side ($Diameter$ is $D=25.4mm$) |
| $Dv$   | Divergent side ($Diameter$ is $D=25.4mm$) |
| $L_1$  | Length of convergent cone |
| $L_2$  | Length of divergent cone |
| $d$    | Diameter of the throat=Length of the throat |
| $\alpha_1$ | Angle of the convergent cone (21°) |
| $\alpha_2$ | Angle of the divergent cone (10°) |
| $\beta$ | Beta ratio |
1. Introduction

Accurate flow measurement is one of the greatest concerns in industries, as uncertainties in product flows can cost considerable losses (Hollingshead et al., 2011). And as most of the fluids of great utility (petrol, diesel, and water) are usually transmitted through pipe lines because of its suitability, durability and effectiveness. So an easy, reliable, accurate and economical flow measurement technique is essential for the distribution of these fluids in the pipelines. Though various types of devices are available to measure the flow rate through pipe lines. But venturimeter, orifice meter and elbow meter are preferred due to their simplicity in construction and cost effectiveness (Rahaman et al., 2009). Among them the venturimeter is one of the oldest and most reliable of the differential head meters. Certain angles of convergence and divergence must be observed for standard Venturi-meter behavior. The conduit walls should converge at about 21° and diverge on the downstream side at about 7 to 15°. The approach piping requirements will be similar to those of orifices; but they can be relaxed somewhat with few detrimental effects. A frequently used Venturi meter is the Herschel-type Venturi tube. It has a converging cone of 21° and a diverging cone of 10°. The throat length of these meters is equal to the throat diameter. This is considered by many users to be the “standard” or “classical” Venturimeter (Nikhil et al., 2014).

Substantial amount of research has been done concerning the coefficient of discharge of flow meters but very little of it has been dealt with extremely small Reynolds numbers. Depending on fluids viscosity the definition of small and large Reynolds numbers can vary. For water, small Reynolds numbers can be defined as $Re < 10,000$, while large Reynolds numbers are $Re > 1,000,000$ (Miller, 1996). Miller et al., 2009 conducted an experiment to evaluate the effect of an emulsion mixture through a Venturi meter. The test consisted of an emulsion mixture flow loop, where the mixture was pumped through the system at different Reynolds numbers to determine the effect of viscosity on the discharge coefficient. They concluded that the following equation can be used to estimate the discharge coefficient for different ranges of Reynolds numbers.

$$C_d = B + A \times \log(Re). \quad (1)$$

Even though this equation can be used to determine the discharge coefficient, there is still significant uncertainty with the values of $A$ and $B$ depending on the viscosity of the fluid being used. They observed that the resulting relative errors for $Re > 2000$, ranged from 2 to 4 %, while the uncertainty grew for values from $400 < Re < 2000$ to as much as 6 %. Harris et al., 2001 studied the effect of changing the convergent angle of Venturi meter on discharge coefficient of a high pressure gas Venturi tube at high Reynolds number. They concluded that the data in gas from the Venturi tubes with a convergent angle of 10.5° are much smoother than those from Venturi tubes with the standard or the higher convergent angle. Colter et al., 2011 numerically studied the discharge Coefficient performance of Venturi, Standard Concentric Orifice Plate, V-Cone, and Wedge Flow Meters at Small Reynolds Numbers using CFD code FLUENT®. They concluded that the various discharge coefficients decrease rapidly as the Reynolds number approaches 1 for each of the flow meters. Rahaman et al., 2009 conducted an experiment to investigate the effects of Beta Ratio (It is the ratio of diameter of throat of venturimeter to diameter of inlet of venturimeter) and Reynolds’s Number on Coefficient of Discharge of Orifice Meter. They used five orifice plates having beta ratios of 0.30, 0.35, 0.47, 0.59 and 0.71 which were installed concentrically in a pipe of 8.5 cm diameter with water as a working fluid. It was found that Discharge coefficient had a positive linear relationship with beta ratio where its dependency was stronger on $\beta$ than Reynolds number in case of low flow rates. Gill et al., 2011 examined field installations where constructed Venturi meters were used to measure flows over a range of magnitudes and for a variety of data collection methodologies. They presented Guidelines for construction and installation of these Pipe Venturi meters. Naveen et al., 2010 numerically studied the effect of beta ratio & Reynolds number on Coefficient of discharge of orifice meter with Non-Newtonian fluid. It was found that the coefficient of discharge increases with Reynolds number from 100 to 10000 and then it remains constant for further increase in Reynolds.
number. Nikhil et al. 2014 conducted an experimental and CFD analysis of flow through Venturimeter. They mainly prepared an efficient and accurate computational model for calibrating flow meters which avoids employing of costly and time consuming experimental methods. Vijay and Subrahmanyam, 2014 undertook a CFD simulation on different geometries of Venturimeter using water as a fluid under steady flow conditions. The velocity and pressure distributions were described and also graphs were plotted.

From the literature review it is clear that very few researchers studied the combined effect of beta ratio & Reynold’s number on the coefficient of discharge of venturimeter at low Reynolds number. While Miller et al., 2009 studied the effect of beta ratio & Reynolds number, his study mainly focused on fluids having high viscosities. Though Naveen et al., 2010 studied the effect of beta ratio & Reynolds number on Coefficient of discharge but their study was numerical using Commercial CFD software & mainly concentrated on Non-Newtonian fluids. The research on the effect of Beta ratio at low Reynolds number is very scarce. Also the problems which are associated with orifice meter/Venturimeter usually overestimate the discharge. Hence a discharge coefficient, $C_d$ is multiplied with theoretical discharge to get actual discharge. However, $C_d$ has no unique value and it varies both with $Re$ and Beta ratio. For most commercial orifice meters and high Reynolds number, the value of $C_d$ is usually constant, whereas for lower Reynolds number, it varies somewhat abruptly. On the other hand, it is seen that discharge coefficient increases with Beta ratio for a fixed Reynolds number, resulting in a decrease of differential pressure across the device. Too much decreasing this pressure head for extreme Beta ratio reduces the accuracy of measurement (Rahaman et al., 2009). Hence an optimum study of selection of Beta ratio has to be performed. So the objective of this paper is to perform an optimum study using Venturimeter of Beta ratio. This study also gives a better insight and help in the design of Venturimeter for low Reynolds number fluids.

To the author’s best of knowledge, the research onto the effect of beta ratio ranging from 0.3 to 0.7 and for low Reynolds flows is very scarce. Therefore this study is carried out to get a better insight into the effect of beta ratio especially at low Reynolds number. This will help in the future design of Venturimeter for specific applications of low Reynolds number flows.

2. Experimental setup and mathematical modelling

The Experiments were conducted in Fluid mechanics research laboratory of P.A College of Engineering, Mangalore on Venturimeter having three different Beta Ratio’s ranging from 0.3 to 0.7 with water as a flowing fluid. The flow rate of the fluid was computed by measuring the difference in pressure head at inlet and throat section of Venturimeter fitted with U-tube differential manometer with mercury as a fluid. The schematic of the experimental setup is shown below:

![Figure 1. Setup of Venturimeter](image-url)
2.1. Design of Venturimeter

The Venturimeter used in this project consists of a successive convergent, a Cylindrical throat and a divergent sections equipped with pressure taps at selected locations as shown below:

![Figure 2. Venturimeter](image)

Where
- \(Cv\) → Convergent side
- \(Dv\) → Divergent side
- \(L_1\) → Length of convergent cone
- \(L_2\) → Length of divergent cone
- \(d\) → Diameter of the throat=Length of the throat
- \(\alpha_1\) → Angle of the convergent cone (21°)
- \(\alpha_2\) → Angle of the divergent cone (10°)

The detailed calculation of the Venturi meter for different Beta ratio required for the design of Venturimeter is explained below:

2.1.1. Calculation of throat diameter

1. For Beta ratio (\(\beta\)) of 0.3
   i) Calculation of Throat Diameter
   - Convergent diameter (\(D\)) = 25.4mm
   - Convergent cone angle (\(\alpha_1\)) = 21°
   - Divergent cone angle (\(\alpha_2\)) = 10°
   - \(\beta = \frac{d}{D} \rightarrow 0.3 = \frac{d}{25.4}\)
   - \(d = 7.62\)mm
   - i.e. Throat diameter (\(d\)) = 7.62mm=Length of throat

   ii) Calculation of Length of Convergent & Divergent cone
   - \(\tan \alpha = \frac{(D-d)}{L} \rightarrow L_1 = \frac{(D-d)}{\tan \alpha_1} = \frac{25.47}{\tan 21^\circ} = 46.31\)mm.
   - i.e Length of convergent cone (\(L_1\)) = 46.31mm.
   - \(L_2 = \frac{(D-d)}{\tan \alpha_2} \rightarrow L_2 = \frac{25.4 - 7.62}{\tan 10^\circ} = 100.83\)mm.
   - i.e Length of divergent cone (\(L_2\)) = 100.83mm.
iii) Calculation of Area of throat
\[ A_1 = \frac{\pi}{4} \times D_2 \Rightarrow A_1 = \frac{\pi}{4} \times 25.42 \]
=506.70 mm²
i.e Area of the Convergent \(A_1\) =506.70 mm².

\[ A_2 = \frac{\pi}{4} \times d_2 \Rightarrow A_2 = \frac{\pi}{4} \times 7.622 \]
i.e Area of the throat \(A_2\) =45.03 mm²

According to the above design the schematic diagram of Venturi meter for Beta ratio 0.3 is shown below:

![Figure 3. Venturimeter with Beta ratio 0.3](image)

Similar calculations were carried for Beta ratio 0.5 and Beta ratio 0.7 as shown in the figure below:

![Figure 4. Venturimeter with Beta ratio 0.5](image)

![Figure 5. Venturimeter with Beta ratio 0.7](image)

2.2. Geometric Modelling and Mesh Generation

The ANSYS FLUENT-14.5© CFD model was used to model and simulate the flow through a Venturimeter. The analysis was done for beta ratios of 0.3, 0.5 and 0.7 for which the co-efficient of discharge and its relationship with Reynolds number were found and compared with the experimental results.
The geometry of the Venturimeter with $\beta=0.3$ is as shown in the figure 6 below:

![Geometric Model ($\beta=0.3$)](image)

**Figure 6.** Geometric Model ($\beta=0.3$)

In the next phase of the project, meshing and CFD analysis for the base case ($\beta=0$) and for different cases ($\beta=0.3$, 0.5, 0.7) were performed to carry out the optimization study. ANSYS Workbench consisting of ANSYS Meshing Tool, CFX Solver and ANSYS post processor were used for the simulation. CFD simulation was performed for all the cases and the results were presented and compared by using the ANSYS Post Processor. The Computational Mesh for the Venturimeter Geometry with $\beta=0.3$ in the ANSYS Meshing application is as shown in figure 7 below.

![Meshed Model ($\beta=0.3$)](image)

**Figure 7.** Meshed Model ($\beta=0.3$)
The details of the Mesh used in the ANSYS Meshing application are shown in Table 1.

**Table 1. Meshing Details (β=0.3)**

| Statistics   |       |
|--------------|-------|
| Nodes        | 149974|
| Elements     | 145200|

| Mesh Metric   |       |
|---------------|-------|
| Skewness      | 4.80E-13 |
| Min           | 6.41E+01 |
| Max           | 6.23E+02 |
| Average       | 1.23E+02 |
| Standard Deviation | 5.16E+02 |

Similarly from the experimental data, the geometry of the Venturimeter with Beta ratio of 0.5 is as shown in figure 8:

**Figure 8. Geometric Model (β=0.5)**

**Figure 9. Meshed Model (β=0.5)**
Table 2. Meshing Details (β=0.5)

| Mesh Metric |  
|-------------|
| Nodes | 144761 |
| Elements | 140000 |
| Skewness | 7.51349623998318E-02 |
| Min | 0.792647223907036 |
| Max | 0.131107429991224 |
| Standard Deviation | 7.53009462646034E-02 |

Also in the same format figure 9 below represents the geometry of the Venturimeter with beta ratios of 0.7 and figure 10 gives its Computational Mesh. The details of Mesh used are shown in table 3 below.

Figure 10. Geometric Model (β=0.7)

Figure 11. Meshed model (β=0.7)
3. Methodology

The calculation of steady, incompressible flow is done by solving the mass and momentum conservation equations given as:

\[ \rho \frac{\partial U_i}{\partial x_i} = 0 \]  \hspace{1cm} (1)

\[ \frac{\partial p}{\partial t} + \rho g_i \frac{\partial x_i}{\partial x_i} = \frac{\partial}{\partial x_i} \left( \frac{\partial p}{\partial x_i} + \rho g_i + \rho f_i \right) \]  \hspace{1cm} (2)

where \( p \) is the static pressure, \( \rho g \) is the gravitational body force and \( f_i \) is the external body force.

The Realizable k-\( \varepsilon \) turbulence model with enhanced wall treatment (Shih et al., 1995) and the Shear-Stress Transport (SST) k – \( \omega \) model developed by Menter (1994) were used for the closure solution for the above equation set. The Realizable k – \( \varepsilon \) model is a two-equation formulation which determines the turbulent length and time scales by solving two transport equations: one related to the turbulent kinetic energy \( k \) and its dissipation rate \( \varepsilon \). The Realizable k-\( \varepsilon \) turbulence model with enhanced wall treatment is superior to the Standard k-\( \varepsilon \) model for the prediction of separated flows. The SST k – \( \omega \) model is also classified as a two-equation model which blends the k – \( \varepsilon \) approach applied in the outer region of the boundary layer while solving the k equation and an additional transport equation for the turbulence kinetic energy specific dissipation rate \( \omega \) in the inner part of the boundary layer. The solution was computed in the commercial CFD code Ansys Fluent 14.5, in which the pressure based solver, was selected for this particular case. The heat transfer from the wall of the domain was neglected. The velocity inlet boundary condition was used to define the volume flow rate at the flow inlet and the outflow boundary condition at the outlet was set to 1. The hydraulic diameter and turbulent intensity at the inlet were specified as 0.0254m and 5% respectively. The adiabatic wall condition was used at the wall region. The governing equations were solved using a segregated solver in which the momentum equations are solved first, and then the continuity equation and finally the pressure and velocity are updated. The numbers of iterations specified were 10000. The pressure drop across the convergent portion of the Venturi was calculated using the code and these values were used to calculate the discharge coefficient using the simple Bernoulli’s equation. The discharge coefficient was calculated for different values of flow rates. The table 4 gives the details of the solver used for the solution of the problem.

| Statistics           |       |
|----------------------|-------|
| Nodes                | 155588|
| Elements             | 150800|
| Skewness             |       |
| Min.                 | 9.10451597946707E-02 |
| Max.                 | 0.612640652368515   |
| Average              | 0.1322062966355302  |
| Standard Deviation   | 7.66443128804725E-02|
Table 4. Details of the Solver used in Ansys Fluent

| Type Of Solve                  | Pressure Based Solver |
|-------------------------------|-----------------------|
| Energy Model                  | Off                   |
| Material                      | Water                 |
| Density Of Water              | 1000 kg/ m³           |
| Turbulent Intensity           | 5%                    |
| Hydraulic Diameter            | 0.0254m               |
| Area Of Measuring Tank        | 0.2772m²              |
| Solution Method               | SIMPLE Scheme         |
| Solution Initialization       | From Inlet            |

3.1. Grid independence and validation

As in many numerical simulations using commercial software’s, success of the analysis depends on mesh quality of the model. Three successful models were created through numerous trials and errors. As expected, it is observed that optimum grid sizes vary with Coefficient of discharge, \( C_d \). Thus, different sets of numerical experimentation have been conducted for different values of \( C_d \). Figure 12 shows the graph of \( C_d \) Vs. grid refinement level. It can be seen from these figures that irrespective of the grid size used, all the \( C_d \) values superimpose each other. However, taking into the consideration of better resolution, a grid size consisting of 150800 elements was chosen.

![Figure 12. Grid refinement level](image)

Table 5. Validation

| Reynolds Number | \( C_d \) ( ISO 5167-1) | \( C_d \) (CFD) |
|-----------------|-------------------------|---------------|
| 5x105           | 0.95                    | 0.93          |
4. Results and discussion

The experimental results were plotted in the form of graph and are shown below:

![Figure 13.1 Cd vs. Re curve for β=0.3](image)

**Figure 13.1** Cd vs. Re curve for β=0.3

The graphical representation of coefficient of discharge (Cd) Vs. Reynolds number (Re) for a Venturimeter with Beta ratio 0.3 is as shown in figure 13.1 above. It is very much clear that, as Reynolds number decreases, the value of coefficient of discharge decreases (Britton and Stark [11], Hollingshead et al., [1]). This is because of the viscous losses that occur at this Reynolds number. Therefore the prediction of correct value of Cd is very much essential when Venturimeter at low Reynolds number is used. Also because Cd varies abruptly at low Reynolds number, hence the flow rate also varies (Jana et al., 2008). In such applications an iterative process is necessary to arrive at actual flow rate. The range of Cd obtained was from 0.89 to 0.96 which was nearly equal to the Standard Venturimeter.

![Figure 13.2 Cd vs. Re curve for β=0.5](image)

**Figure 13.2** Cd vs. Re curve for β=0.5

The above figure 13.2 shows the effect of Re on Cd for Beta ratio 0.5. It can be seen that as Reynolds number increases, coefficient of discharge also increases (same trend as that for Beta ratio 0.3). The variation was low at the beginning and gradually it became high at high Reynolds number. Also at some Reynolds number a decrease in the value of Cd was also observed because of low pressure drop in the Venturimeter of Beta ratio 0.5 which reduces the flow rate. The range of Cd obtained for this Venturimeter was from 0.87 to 0.94 which is little less compared to Venturimeter of Beta ratio 0.3.
Figure 13.3 Cd vs. Re curve for β=0.7

Figure 13.3 shows the effect of Re on Cd for Beta ratio 0.7. Again the same trend was observed for Beta ratio 0.7 but the variation was very fast as compared for the other Beta ratios. The range of Cd obtained was from 0.84 to 0.90 which was very much less compared to the value of Cd for Venturimeter of Beta ratio 0.3. It can also be observed that the increase in the value of Cd decreased as Reynolds number increases. Thus there is an upper limiting value of Reynolds number beyond which Cd is unaffected by its increase.

By taking average of the near constant values of the Coefficient of discharge (Cd) and Reynolds number of each Venturimeter the relationship between Coefficient of discharge (Cd), Reynolds number and beta ratio it can be summarized as in table 6 below.

Table 6. Relationship between Cd and Re

| Beta Ratio | Average Cd  | Average Re |
|------------|-------------|------------|
| 0.3        | 0.964554    | 29624      |
| 0.5        | 0.938621    | 52075      |
| 0.7        | 0.875965    | 54538      |

Figure 14. Effect of Beta ratio on Average Cd

The above figure 14 shows the variation of Beta ratio on Average value of Cd (From Table.4). It is very much clear from this figure that the value of Cd decreases as Beta ratio increases. But the rate of decrease in the value of Cd decreases as Beta ratio increases. And the highest value of Cd obtained was for Beta ratio 0.3 and it was equal to that Standard commercial Venturimeter.
Figure 15. Streamlines (β=0.3)

Figure 16a. Velocity contours (β=0.3)

Figure 16b. Velocity plot (β=0.3)

The figure 15 and figure 16 above show the streamlines and velocity plots in the x direction of a Venturimeter of Beta ratio 0.3. It can be seen that the velocity of the fluid is high at the throat as expected because of small area at the throat. Also it can be seen the variation of velocity in the divergent section of the Venturimeter is gradual as compared to the convergent section. Similar trend was observed for Venturimeter of Beta ratio 0.5 and Beta ratio 0.7.
Figure 17a. Pressure contours (β=0.3)

Figure 17b. Absolute Pressure Plot (β=0.3)

Figure 17 above shows the streamlines, the contours and absolute pressure in the x-direction. It can be seen that the pressure goes on decreasing in convergent section and reaches a minimum value at the throat section. Also it is evident that the pressure regained after the divergent section is not equal to the initial pressure in the pipe. This shows that frictional losses occur in the Venturi section and reduces the pressure of the flow. Similar trend was observed for other Venturimeter of Beta ratio 0.5 and Beta ratio 0.7 (figure 18 and figure 19) but when compared to 0.3 the pressure drop was less for Venturimeter of Beta ratio 0.5 and very less for Venturimeter of Beta ratio 0.7. Hence the Cd value obtained was less as compared to Venturimeter of Beta ratio 0.3. This is in conformity with the experimental results.

Figure 18a. Streamlines (β=0.5)
Figure 18b. Velocity contours ($\beta=0.5$)

Figure 18c. Velocity plot ($\beta=0.5$)

Figure 18d. Pressure contours ($\beta=0.5$)

Figure 18e. Absolute Pressure Plot ($\beta=0.5$)
Figure 19a. Streamlines ($\beta=0.7$)

Figure 19b. Velocity contours ($\beta=0.7$)

Figure 19c. Velocity plot ($\beta=0.7$)

Figure 19d. Pressure contours ($\beta=0.7$)
5. Conclusion
1. The value of coefficient of discharge (Cd) decreases as Beta ratio (β) increases. The value of Cd observed was highest for Venturimeter of Beta ratio 0.3, nearly equal to that of a commercial standard Venturimeter.
2. The value of discharge coefficient decreases as Reynolds number decreases. For fixed Reynolds number, same trend was observed that as Beta ratio decreases, the value of Cd increases. And highest value of Cd was observed for a Beta ratio of 0.3.
3. Pressure drop was less for Venturimeter of Beta ratio 0.5 and very less for Venturimeter of Beta 0.7 as compared to Venturimeter of Beta ratio 0.3. The variation of pressure was not gradual in convergent section of all the Venturimeter considered. More variation of pressure was seen for Venturimeter of Beta Ratio 0.3.
4. The increase in the value of Cd decreased as Reynolds number increases. Thus there is an upper limiting value of Reynolds number beyond which Cd is unaffected by its increase.
5. Numerical results agreed well with experimental results.
6. Least velocity at the throat was observed for a Venturimeter of Beta Ratio 0.3.

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