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COMPUTATIONAL STUDY OF THE AERODYNAMIC BEHAVIOR OF THE LAWS VANED PLATE FLOW CONDITIONER

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ABSTRACT

Flow conditioners are widely used in industry to generate fully developed turbulent pipe flow condition in relatively short distances for metering purposes. In the present study computational fluid dynamics technique was used to investigate the flow development after a flow conditioner developed by Laws and Ouazzane (1994).

INTRODUCTION

All flow meters are affected by the quality of the flow approaching the meter (mean flow distortions, swirl, turbulent fluctuations and pulsating flow). Numerous research workers have investigated experimentally and computationally these effects with reference to the shift in flow meter discharge coefficient. Aichouni et al. (1996,1998) reported errors up to ±30% on a Venturi flow meter while Yeh and Matingly (1996) reported errors up to ±17% on an ultrasonic flow meter when the meter is subject to distorted flow conditions. Given that most industrial installations include bends, valves, expanders and reducers, which are sources of both swirl, asymmetries and turbulence distortions, ensuring that fully developed flow in terms of both mean flow and turbulence structure approaches the meter is difficult to achieve in practice. Either very long lengths of straight pipe work upstream of the meter must be provided (and these may need to be of the order of 80 to 100 pipe diameters in length), or some form of flow conditioning device must be introduced into the system, combined with a shortened straight length of piping duct.

The flow conditioners used in the oil and gas industry, such as the Etoile, tube bundle, Sprenkle, Zinker and the perforated plate, have been described in the international standards (ISO 5167). Recently Laws and Ouazzane (1994) developed the 'Laws vaned plate' flow conditioner. Based on hot wire measurements of the mean velocity profiles and the axial turbulence intensity, the device has been shown to be efficient in the production of the fully developed condition within 10 pipe diameters of flow development.

In the present study, computational fluid dynamics techniques have been used to study the aerodynamic behavior of the Laws vaned plate flow conditioner. The numerical method which was described in early work by Aichouni and Laws (1991-1993) is based on the solution of the Navier-Stokes equations, associated with the two equation K-ε turbulence model. Different test cases have been studied numerically and the predictions are compared to the experimental results for different flow conditions (valve fully and 50 % open) obtained from Laws and Ouazzane (1994).

NUMERICAL APPROACH

The time mean averaged equations for conservation of mass and momentum are expressed in the general tensor notation by:

The continuity equation of mass conservation:

$$\frac{\partial U_i}{\partial X_i} = 0 \qquad (1)$$

The momentum equation:

$$\frac{\partial}{\partial X_j}(\rho U_i U_j) = -\frac{\partial P}{\partial X_i} + \frac{\partial}{\partial X_j}\left(\mu\left(\frac{\partial U_i}{\partial X_j} + \frac{\partial U_j}{\partial X_i}\right) - \rho\overline{u_i u_j}\right) \qquad (2)$$

The turbulent correlation term ($-\rho u_i u_j$) which represents the transport of the momentum due to the fluctuating motion, must be modeled to close the above set of differential equations.

The Standard K-ε turbulence model

The Standard K-ε turbulence model is based on the hypothesis that the turbulent stresses, like their viscous counterparts are linearly related to the rates of strain through the Boussinesq eddy-viscosity concept which can be expressed as:

$$-\rho\overline{u_i u_j} = \mu_t\left(\frac{\partial U_i}{\partial X_j} + \frac{\partial U_j}{\partial X_i}\right) - \frac{2}{3}\delta_{ij}\,\rho K \qquad (3)$$

The turbulent eddy viscosity μ_t is related to the turbulent kinetic energy, K, and its rate of dissipation, ε, by the relation:

$$\mu_t = C_\mu \rho \frac{K^2}{\varepsilon} \tag{4}$$

The turbulent kinetic energy, K, and its dissipation rate, ε, are determined from their modeled transport equations which can be written as:

$$\frac{\partial}{\partial X_j}(\rho U_j K) = \frac{\partial}{\partial X_j}\left(\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial K}{\partial X_j}\right) - \rho\overline{u_i u_j}\frac{\partial U_i}{\partial X_j} - \rho\varepsilon \tag{5}$$

$$\frac{\partial}{\partial X_j}(\rho U_j \varepsilon) = \frac{\partial}{\partial X_j}\left((\mu + \frac{\mu_t}{\sigma_k})\frac{\partial \varepsilon}{\partial X_j}\right) - C_1 \frac{\varepsilon}{K}\rho\overline{u_i u_j}\frac{\partial U_i}{\partial X_j} - C_2 \rho \frac{\varepsilon^2}{K} \tag{6}$$

The empirical constants appearing in the equations are those recommended by Launder and Spalding (1974). The model constants are $C\mu=0.09$, $C_1=1.44$ and $C_2=1.92$ and $\sigma_k=1.0$ and $\sigma_\varepsilon=1.3$ are the turbulent Prandtl numbers for K and ε respectively.

Substitution of equations (3) and (4) into (2), (5) and (6) form a closed set of equations. Solution of this set of equations enables the time mean and turbulent flow parameters (U_i, p, $-\rho u_i u_j$, K, and ε) to be calculated.

Initial and Boundary Conditions

In the present study no attempt was made to compute the flow through the flow conditioner. The solution was started by taking as the duct inlet condition the measured profiles at the first axial station 0.5 D downstream of the flow conditioner. Profiles of the axial velocity and the turbulence intensity components measured by Laws and Ouazzane (1994) were used to prescribe the initial conditions required for the solution. The turbulent kinetic energy, K, is calculated assuming isotropic turbulence, while the dissipation rate, ε, is evaluated from:

$$\varepsilon = C\mu^{3/4} K^{3/2} / l_m \tag{7}$$

Where l_m is a characteristic length scale of turbulence. Morisson et al. (1992) assumed it to be constant across the pipe section and equal to 0.07R. In the present study, the l_m distribution was assumed to follow a ramp function as suggested by Laws and Aichouni (1993). The no-slip boundary conditions are imposed at the pipe walls.

The turbulence model discussed above represents a closed system of equations in describing high Reynolds number flows. However, the no-slip conditions at the wall create a region immediately adjacent to the wall in which viscous effects increase in importance and turbulence length scales decrease as the wall is approached. Consequently a special treatment is required to provide the bridge between the numerically treated region and the viscous near-wall region. Since wall functions have proved successful in many calculations of straight and curved boundary layer flows, this method is adopted in the present work. This method is based on the assumption that the Log law can be applied in the wall region:

$$\frac{U}{U_\tau} = \frac{1}{\kappa}\ln\left(\frac{\rho E y u_\tau}{\mu}\right) = \frac{1}{\kappa}\ln(E y^+) \tag{8}$$

Where κ=0.42 is the Von-Karman constant and the roughness parameter E is specified as 9.0 for smooth pipes.

Numerical Procedure

The governing equations are discretized using the conservative control volume approach. In general, pipe flow is characterized by a predominant direction, which makes the diffusion in the stream wise direction negligible, while it is more important than the convection in the cross-stream direction. The convection process is approximated using the upwind scheme in the stream wise direction and the central differencing scheme in the cross-stream direction. Such formulation permits both accuracy and stability to be maintained in the numerical solution. The procedure for calculation of the flow field is based on the SIMPLE algorithm of Patankar (1980). The pressure field is determined by first calculating an intermediate velocity field based on a guessed pressure field and then obtaining a correction so as to satisfy the continuity equation.

The grid distribution in the calculation domain is non-uniform in the cross-stream direction. In the present calculations, it was found very important to maintain an acceptable cell aspect ratio Δz/Δy, which was maintained proportional to the grid width. To ensure that the solution was grid independent, different grids were used. The tested grids contain 30, 46 and 60 nodes in the cross-stream direction. The axial mean velocity profile was used to check the grid independence of the computational results. Figure 1 shows the calculated mean velocity (U/Um) at Z/D=20. It is clear that no differences are noticed except in the near wall region. The grid with 46 nodes in the radial direction was chosen to carry out the computations, together with a downstream step length equal to 4 percent of the computational domain.

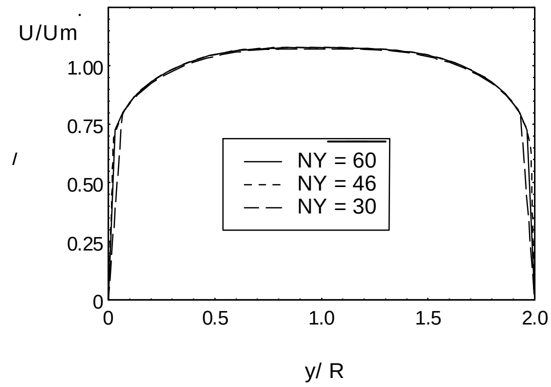

Figure 1 – Grid independence test – Velocity profile calculated at Z/D=20 in the case of valve fully open

RESULTS AND DISCUSSION

The present study is concerned with the prediction of the flow development after the Laws vaned plate described in Laws and Ouazzane (1994). Their first measured profiles at 0.5 diameter downstream of the unit have been taken as initial conditions for the present calculations. In the experimental study the flow conditioner was tested with an upstream valve set at 50, 70 and 100% open. Here, the K-ε predictions are compared to the experimental values at different axial stations.

Figure 2 – Comparison between experiments and predictions for the case of valve fully open.

Figure 3 – Comparison between experiments and predictions for the case of valve 50 % open.

Figures (2) show a comparison between the numerical simulations and the experimental profiles obtained at the axial stations Z/D=2.5 and 5.5 for the case of valve fully open. The figure shows that the mean velocity profiles are predicted correctly while the turbulence intensity profiles show some discrepancy between the simulations and the measurements. The experimental turbulence intensity profiles exhibit a central peak at the first measured station, which decays rapidly at some 5.5 diameters downstream of the flow conditioner unit.

The numerical profiles show a slower decay, which would need longer distances to develop.

Figure (3) show a comparison between the numerical simulations and the experimental profiles obtained respectively at the axial stations Z/D=2.5 and 5.5 for the case of valve 50 % open. The measured profiles present an initial asymmetry which decays to give a relative uniform profile by the axial station Z/D=5.5. The figure shows that the mean velocity profiles predicted by the turbulence model are in reasonable agreement with the measured profiles at all the axial stations. The measured and computed axial turbulence intensity profiles, which are in good agreement, present decay towards uniform profiles downstream of the flow conditioner.

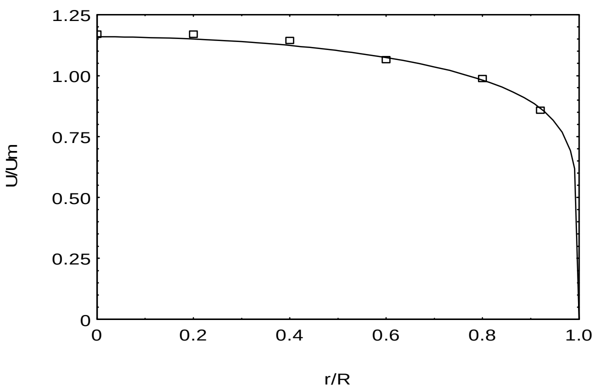

Figure 4 – Comparison of flow profiles obtained at the fully developed flow

In order to study the flow development further downstream of the conditioner, calculations have been carried out up to 120 diameters from the unit. At 100 pipe diameters the flow have reached completely the fully developed condition as shown in figure 4 where the computed profiles (the axial mean velocity and the Reynolds shear stress) are compared to the experimental values reported by Lawn (1971). The initial conditions do not have any effect at this late stage of flow development.

Figures 5 and 6 show the development of the flow after the conditioner unit with the valve fully open. The velocity profiles develops from a wake type flow at the initial stages. The velocity profiles appeared to be fully developed at 15 pipe diameters after which only small variations occur in the central flow region. The axial turbulence intensity shown in figure 6, which develops, from higher value at the initial stages seem to be developed after some 25 pipe diameters. The turbulence levels started very high in the central region (28%) and decreases rapidly to attend relatively constant levels (5%) at some 8 pipe diameters from the conditioner. This is in good agreement with the experimental results presented by Laws and Ouazzane (1994). This shows the efficiency of the conditioner in redistributing the turbulent energy between the flow regions and the removal of the flow distortion. Similar behavior has been obtained for the case of valve 50 and 70% open, which are not presented here for space limitations.

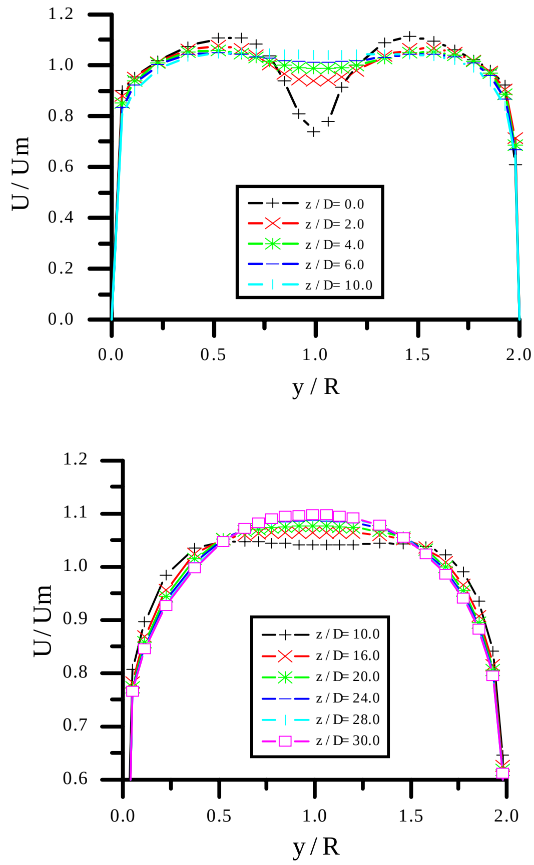

Figure 5 – Predictions of velocity profile development for the case of valve fully open

CONCLUSION

Computational fluid dynamics techniques were used to predict the flow development after a flow conditioner developed by Laws and Ouazzane (1994). The K-ε turbulence model predictions compared to the measured values are in reasonable agreement in the early flow stages where steep variations occur. The profiles obtained at the fully developed condition are compared to experimental results and the agreement is remarkable.

The numerical predictions were used to study the efficiency of the flow conditioner to remove flow distortions and produce the fully developed flow. The present computations confirm the experimental observations on the efficiency of the flow conditioner to produce fully developed condition within 10 to 15 pipe diameters in terms of mean velocity. The turbulence structure needs longer distances (about 25 D) to develop. Completely fully developed flow condition would require distances of the order of 40 pipe diameters.

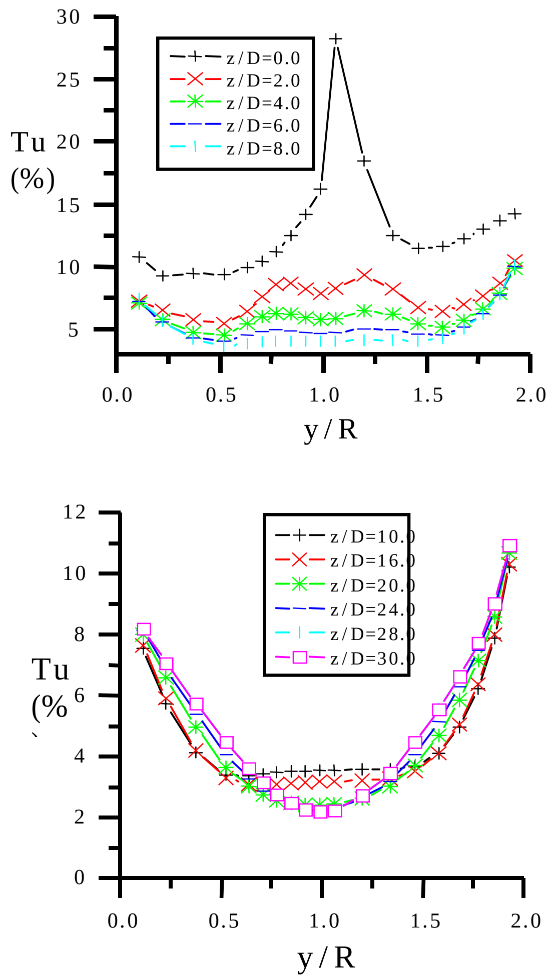

Figure 6 – Turbulence profile development for the case of valve fully open

NOMENCLATURE

D : pipe diameter
K : turbulent kinetic energy
p : pressure
T_u (%) : axial turbulence intensity
y , Z : radial and axial co-ordinates
U_i : mean velocity components
U_i : fluctuating velocity components
U_m : axial mean velocity
U_τ : friction velocity
ε : dissipation rate of the kinetic energy
μ : fluid dynamic viscosity
μ_t : turbulent eddy viscosity
ρ : fluid density
τ_w : wall shear stress

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