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NUMERICAL VALIDATION OF ISOTHERMAL AND NON-ISOTHERMAL TURBULENT SWIRLING FLOWS IN ANNULI

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SUMMARY

Turbulent swirling flows in annuli are analysed numerically using the computer package FLUENT. The predictions are compared with experimental time mean velocities for both isothermal and non-isothermal flows. The geometry studied is a stationary annulus formed by two co-axial pipes of different diameter but of equal length with one wholly inside the other. Predictions are made for velocity and temperature distributions at various axial locations. Two turbulence models, namely the k-ϵ model and the Algebraic Stress Model (ASM), are used in the validation. A satisfactory agreement was found between the predictions and the measurements of axial and tangential velocity profiles. However, the agreement is less satisfactory for the temperature profiles, particularly in the core region.

NOTATION

C	turbulence model coefficients
d_h	hydraulic diameter of annulus
l	length of annulus
p	gauge pressure
r	radial distance
r_i	inner radius of annulus
r_o	outer radius of annulus
Re_h	Reynolds number, $\rho U_{av} d_h/\mu$
T	temperature
T_a	ambient temperature
U	time mean axial velocity
u	fluctuating axial velocity
U_{av}	average axial velocity
V	time mean radial velocity
v	fluctuating radial velocity
W	time mean tangential velocity
w	fluctuating tangential velocity
x	axial distance
α	radius ratio, r_i/r_o
μ	dynamic viscosity of fluid
ν	kinematic viscosity
ρ	density of fluid
σ	turbulence model coefficients
ψ	swirl angle
∇	Laplacian operator

INTRODUCTION

Swirling flows are found in a wide range of engineering applications such as cyclone separators, turbomachinery and heat exchangers. Swirling motion has favourable effects in some situations, while in others it is undesirable. Therefore it is important to have a good understanding of the general characteristics of such a flow. Recently, in the metallurgical industry, swirling gas flow inside an annular lance was found to improve the performance of lances used in top submerged gas injection systems (Conochie et al., 1984).

Despite the importance of annular swirl flows, only a few studies have been reported. The experimental studies on turbulent swirling flows in axisymmetric annuli, which are of interest to the present investigation are the work reported by Yeh (1958); Scott and Rask (1973); Scott and Bartelt (1976); Morsi (1983); Yowakim (1985) and Solnordal (1992). The majority of the reported theoretical work has concentrated on isothermal flows in simple geometries such as pipes and very little theoretical work has been published on turbulent swirling flows in annuli. Literature reveals that the major difficulty lies in finding a way to represent Reynolds stresses due to the complex structure of near wall shear stresses. In annulus flow, the shear stress on the convex wall decreases with axial distance, while on the concave wall it remains either constant or increases slightly. This phenomenon was initially discussed by Yeh (1958) and later measured by Morsi (1983).

Different approaches have been made by various workers to solve the governing equations pertaining to swirling flow in an annulus. Morsi and Clayton (1986) used a finite difference technique, originally developed by Gosman et al. (1969), to solve the Navier-Stokes equations. Turbulence was modelled using both a mixing length and a k-ϵ model, and a good agreement was found between theory and experiments. Morsi and Clayton (1987) gave an asymptotic solution to the annulus swirling flow problem, which could reduce computation time and cost. However, this method is limited to fully developed flow.

Holland and Fletcher (1986) used the parabolised type of Navier-Stokes equations coupled with the concept of eddy viscosity and mixing length to predict non-swirling pipe flows. A good agreement with the published data was reported. Reddy et al. (1987) solved the parabolised Navier-Stokes equations for swirling flow in an annulus using an implicit finite difference method. The suitability of several eddy viscosity/mixing length type turbulence models was assessed. The predictions of Reddy et al. only showed good agreement with experiment in near-wall regions.

Despite the importance of heated flow inside annuli, very little attention has been paid to this in the literature. However, recently Solnordal (1992) has carried out a comprehensive study on heated swirling gas flow inside an annulus.

In the current study, we examined the capability of the computer package, FLUENT (1990), to predict velocity and temperature distributions of turbulent swirling flows inside an annulus. The numerical predictions are then compared with the published experimental data of Morsi (1983) and Solnordal (1992).

GOVERNING EQUATIONS AND TURBULENCE MODELS

For incompressible axisymmetric steady turbulent flow, the Navier-Stokes equations and the continuity equation can be written as

$$U\frac{dU}{dx}+V\frac{dU}{dr}=-\frac{1}{\rho}\frac{dp}{dx}+\nu\nabla^2 U-[\frac{d\overline{u^2}}{dx}+\frac{1}{r}\frac{d}{dr}(\overline{ruv})] \qquad (1)$$

$$V\frac{dV}{dr}+U\frac{dV}{dx}-\frac{W^2}{r}=-\frac{1}{\rho}\frac{dp}{dr}+\nu[\nabla^2 V-\frac{V}{r^2}]-[\frac{d}{dx}(\overline{uv})+\frac{1}{r}\frac{d}{dr}(\overline{rv^2})-\frac{\overline{w^2}}{r}] \qquad (2)$$

$$U\frac{dW}{dx}+V\frac{dW}{dr}+\frac{VW}{r}=\nu[\nabla^2 W-\frac{W}{r^2}]-[\frac{d}{dx}(\overline{uw})+\frac{d}{dr}(\overline{vw})+2\frac{\overline{vw}}{r}] \qquad (3)$$

$$\frac{dU}{dx}+\frac{V}{r}+\frac{dV}{dr}=0 \qquad (4)$$

where the over bar is used to denote time-averaged quantities.

A complete solution to the above equations is needed in order to obtain an accurate prediction. However, a turbulence closure has to be used to solve these equations. Furthermore, in the case of heated flow the energy equation must also be solved. For confined flows, the k-ϵ model of Launder and Spalding (1972) has been used quite extensively by various authors. The k-ϵ model is known to give poor predictions for strong swirling flows (Kobayashi and Yoda, 1987). The Algebraic Stress Model (ASM), which accounts for the non-isotropy of turbulent viscosity is believed to give more accurate predictions for strongly swirling flows.

The turbulence models used in this study are the k-ϵ model and the Algebraic Stress Model. The model coefficients used here with the k-ϵ Model are: C_1= 1.44; C_2= 1.92; C_μ= 0.09; σ_k= 1.0; σ_ϵ= 1.3; σ_h= 0.7. The model coefficients used with the Algebraic Stress Model are: C_D= 0.55; $C_{2,ASM}$= 1.0; C_3= 2.2.

COMPUTATIONAL PROCEDURE

FLUENT allows the use of either the k-ϵ model or the Algebraic Stress Model to solve the equations 1 to 4. The flow was considered as a 2D axisymmetric problem and a variable grid arrangement was used. The grid size gradually increased from each wall towards the core region as shown in Figure 1.

The flow geometry considered in this work is an annulus formed by two co-axial cylinders of different diameter but of equal length with one wholly inside the other.

The inlet velocity and temperature profiles were taken from the experimental data of Morsi (1983) and Solnordal (1992).

DISCUSSION OF RESULTS

The two experimental studies used for validation are those by Morsi (1983) and Solnordal (1992). The selected conditions used in the validation are summarised in Table 1 and the axial location of the measuring stations are given in Table 2. In the experimental part of the work by Morsi (1983), guide vanes were used to obtain different swirl intensities. Solnordal (1992) used a similar technique to produce swirled gas flow, with a number of heating elements to investigate the effect of heat input on the swirl characteristics along the annulus.

In Figures 2 to 5, the velocities and radial distances are normalised by the average axial velocity at that station (U_{av}) and the annular gap (r_o-r_i) respectively. In each graph, successive velocity profiles have been translated vertically by 0.2 units, so that they can be distinguished.

For all simulations, the velocity profiles predicted using the two turbulence models are distinctly different. The axial velocity profiles predicted using the k-ϵ model are flat across the bulk of the flow passage, and nearly symmetrical within the flow passage. The results of Morsi (1983) given in F 2a are not well predicted in the outer wall region, suggesting that the logarithmic law of the wall is not suited to confined swirling flows. The inlet axial velocity profile had no wall boundary layers, since the experimental apparatus used by Morsi (1983) allowed their removal at the inlet. Therefore, the mathematical model should predict the growth of these layers. The results indicate that this is not achieved. The inlet profiles for Solnordal's (1992) work (Figure 3 and 4) were taken within the confined flow passage, so that boundary layers were already present. Therefore, prediction of such growth is less critical, and agreement with experimental data is good.

Axial velocity profiles predicted using the ASM show no symmetry within the flow passage. Instead, the profiles show a distinct region of reduced axial velocity near the inner annular wall. Such decreases are common for highly swirling flows, and can lead to regions of reversed flow. However, the experimental results show no evidence of this behaviour. The prediction of reduced axial velocity near the inner wall suggests that the formulation of the ASM equations can predict turbulence distributions typical of more strongly swirling flows.

The shape of the predicted tangential velocity profiles is again dependent on the choice of turbulence model. Figures 2a, 3b and 4b show the k-ϵ model predicts a rapid decay to a forced vortex flow ($W \propto r$), whereas the ASM predicts a decay to a combined vortex.

The forced vortex profile predicted using the k-ϵ model is in good agreement with experimental data well downstream of the inlet ($x/d_h \geq 28.0$, Figures 3b and 4b). In this region, the flow is approaching axial flow conditions where turbulence is well modelled using an isotropic turbulent viscosity. Therefore, the k-ϵ model, which assumes a locally isotropic turbulent viscosity, is expected to predict downstream conditions with reasonable accuracy.

The combined vortex structure predicted by the ASM shows poor quantitative agreement with all experimental data. However, the data of Morsi (1983) and the upstream data of Solnordal (1992) all show a distinctly combined vortex structure. Therefore, the upstream behaviour is qualitatively predicted using the ASM. The downstream decay is, however, not simulated.

The addition of heat to the system has no detectable effect on the mean velocity profiles as shown by both the experimental data and the model predictions (Figures 3 and 4). Although the predicted air temperature distributions of Figure 5 show the expected variation from a high temperature near the outer (heated) wall to a reduced temperature at the inner wall, the shape of the profiles is somewhat different from the measured ones. However, the accuracy of the predictions is still within ±5°C.

CONCLUSIONS

The capability of the finite difference computer code FLUENT in predicting the isothermal and non-isothermal annular swirl flow was assessed. It can be said that the prediction of the axial flow distributions is satisfactory. The Algebraic Stress Model predicts the tangential velocity well in the entrance region of the annulus whereas the k-ϵ model predictions are better in the fully developed flow region. Further study is called for to assess the effect of turbulent characteristics, particularly near the convex wall, on the prediction of developing and developed regions of swirling flow of different intensity.

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Author	Flow	Re_h	α	r_o (mm)	l (m)	ψ(°)
Morsi (1983)	Isothermal	14,745	0.51	54.4	0.9	15
Solnordal (1992)	Isothermal	150,000	0.40	41.4	3.5	45
Solnordal (1992)	Heated	125,000	0.40	41.4	3.5	45

Table 1 Experimental Conditions

Morsi (1983) (r_i/r_o= .51)	Axial station	1	2	4	8	10
	x/d_h	0	1.74	5.14	11.92	15.31
Solnordal (1992) (r_i/r_o= .40)	Axial station	1	2	3	4	5
	x/d_h	5.4	9.5	15.8	28.0	38.0

Table 2 Location of Axial Measuring Stations

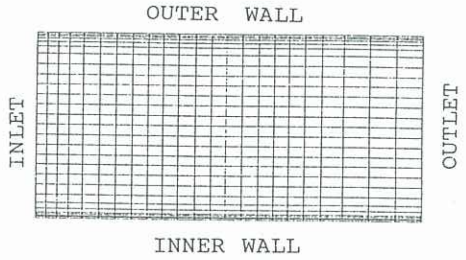
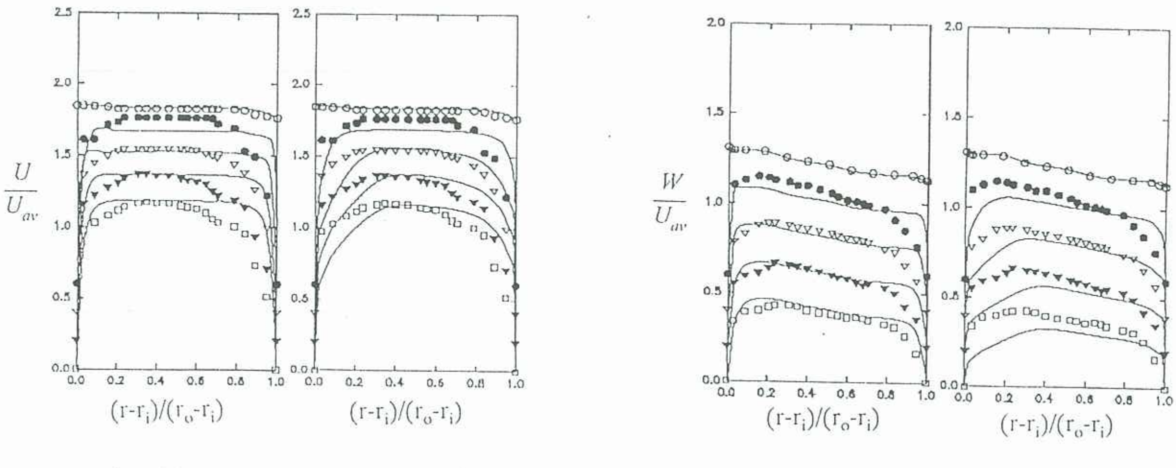


Figure 1 Computational Grid for the Annular Gap (Not to Scale)



k - ε Model Algebraic Stress Model

a. Axial Velocity Profiles

k - ε Model Algebraic Stress Model

b. Tangential Velocity Profiles

Figure 2 Comparison of FLUENT Predictions (——) with Experimental Data of Morsi (1983) (Re_h= 14,745, ψ=15°); ○ x/d_h= 0, ● x/d_h= 1.74, ▽ x/d_h= 5.14, ▼ x/d_h= 11.92, □ x/d_h= 15.31.

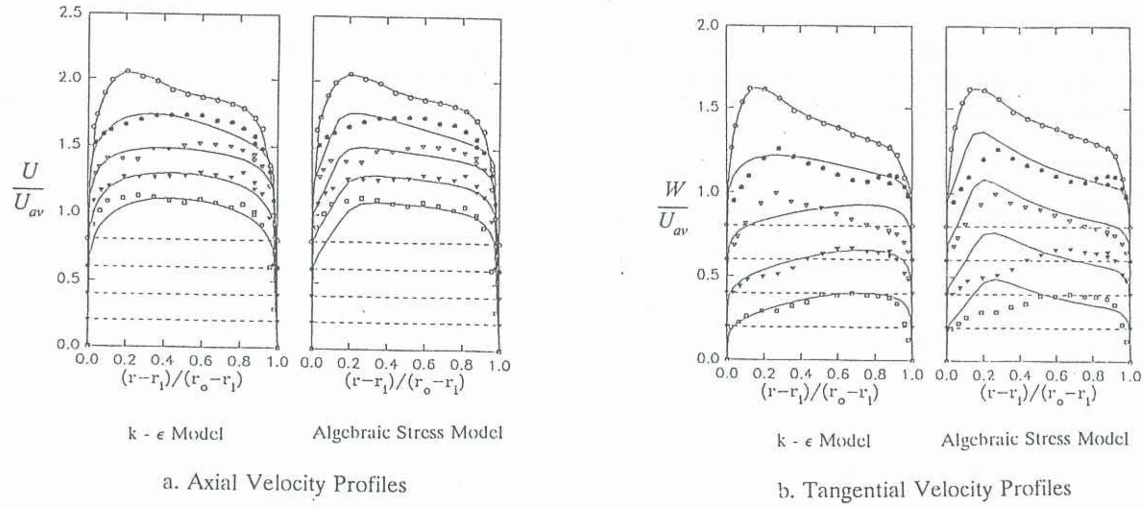


Figure 3 Comparison of FLUENT Predictions (——) with Experimental Data (Isothermal Flow) of Solnordal (1992) (Re_h= 150,000, ψ=45°); ○ x/d_h= 5.4, ● x/d_h= 9.5 , ▿ x/d_h= 15.8, ▾ x/d_h= 28.0, ▫ x/d_h= 38.0 .

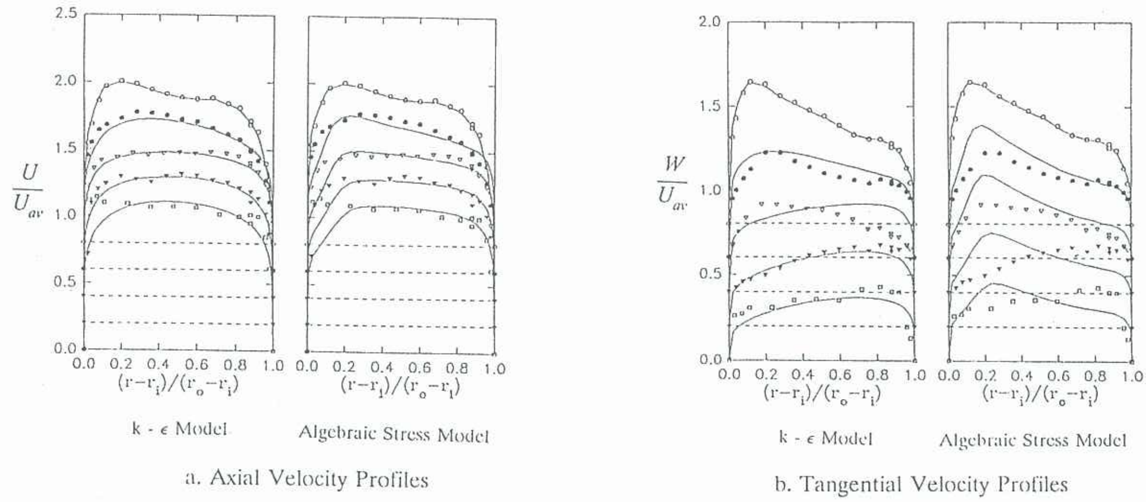


Figure 4 Comparison of FLUENT Predictions (——) with Experimental Data (Heated Flow) of Solnordal (1992) (Re_h= 125,000, ψ=45°, Heat input = 10.5 kW over 3m of pipe); ○ x/d_h= 5.4, ● x/d_h= 9.5 . ▿ x/d_h= 15.8, ▾ x/d_h= 28.0, ▫ x/d_h= 38.0 .

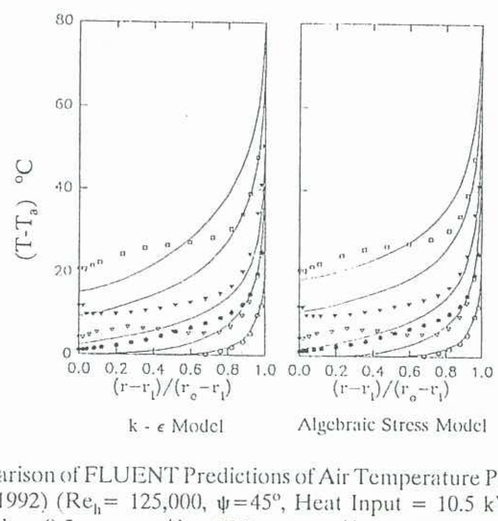


Figure 5 Comparison of FLUENT Predictions of Air Temperature Profiles (——) with Experimental Data of Solnordal (1992) (Re_h= 125,000, ψ=45°, Heat Input = 10.5 kW over 3m of pipe) ○ x/d_h= 5.4, ● x/d_h= 9.5 . ▿ x/d_h= 15.8. ▾ x/d_h= 28.0. ▫ x/d_h= 38.0.