Turbulence Modelling of Single Phase Flow in a Spray Dryer

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ABSTRACT

Highly swirling, recirculating flow, such as occurs in spray dryers is a challenge to model accurately. Both the standard k-ε and Reynolds Stress Transport (RST) turbulence models have been applied to spray dryer flows with mixed success. The k-ε model is known to give poor predictions in many strongly swirling flows but its implementation is less computer intensive than the more accurate RST model. A modified form of the k-ε model which seeks to account for the anisotropy of turbulence in swirling flow has been used with success for flow in cyclone separators. Here a comparison of modifications to the standard k-ε model is made for flow in a spray dryer. It is found that, although the modifications improved the prediction of cyclone and other flows, there is no consistent improvement for the spray dryer flow considered in this paper.

NOMENCLATURE

C_μ, C_2	constants in the k-ε turbulence model
D_{eq}	equivalent diameter of the inlet
k	turbulent kinetic energy
$l_{r\theta}$	constant in equations 3 and 4
u, v, w	velocity components in the x, r and θ directions
$\lvert U \rvert$	inlet air velocity
x, r, θ	axial, radial and azimuthal co-ordinates

Greek letters

ε	turbulence dissipation rate
$\mu_{r\theta}$	$r\theta$ component of eddy viscosity
ρ	density
σ_ε	effective Prandtl number
τ'_{ij}	scalar components of the Reynolds stress tensor

1. INTRODUCTION

Spray drying is a process whereby a liquid slurry is dried to a powder by spraying into a hot and usually swirling drying gas. In a co-current dryer the droplets are introduced into the spray chamber by a centrally located atomizer in the chamber ceiling. Drying air enters through an annulus surrounding the nozzle and both air and dried product exit at the bottom of the conical section of the chamber. This process is analogous to some other types of processes where an interaction between a swirling continuous and particulate phase exists such as in fuel combustors and cyclones.

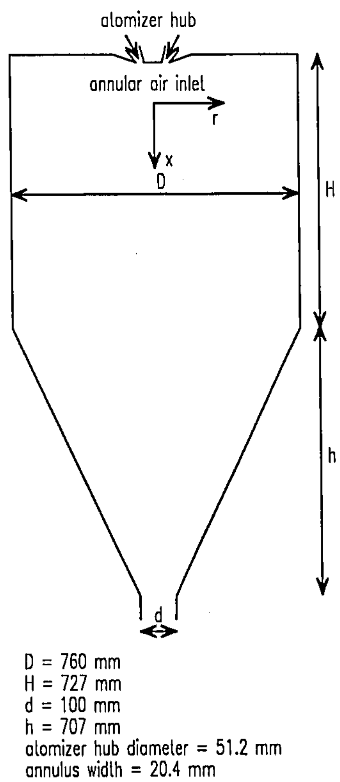

Figure 1 - Spray Dryer

The generation of swirl in a fluid has a significant effect on the fluid turbulence which becomes anisotropic (Sloan et al., 1986). This anisotropy is exemplified by the difference between the rx and $r\theta$ components of eddy viscosity measured by Lilley and Chigier (1971). The most commonly used turbulence model, the k-ε model, is based on an assumption of isotropic turbulence and a single eddy viscosity for all three components of the velocity vector. This model has been shown to be inaccurate for many flows with swirl, including spray drying, particularly well downstream of the inlet (Oakley and Bahu, 1993). In particular the k-ε model is too diffusive, resulting in a spreading of swirl (and axial) velocity profiles which is not found in experimental results. The RST model allows for anisotropy by solving for each individual Reynolds stress separately. This model has been found to be more accurate than the k-ε model for swirling flows but results in a markedly increased computational time (Speziale, 1987). Oakley and Bahu (1993) and Oakley (1994) have also found the RST model to have poorer convergence and to be more grid dependent than the k-ε model.

Several extensions to the k-ε turbulence model have been presented in the literature which attempt to improve the predictions for confined swirling flows, however none have proved to have general applicability. Here two modifications to the k-ε turbulence model (Abujelala and Lilley, 1984, Duggins and Frith, 1987) are investigated. These are applied to single phase flow in a co-current spray dryer as presented in unpublished work by Oakley and Lagarde.

2. NUMERICAL CONSIDERATIONS

Modelling of the spray dryer was carried out using the commercial fluid flow computer program CFX F3D version 4.1 (AEA Technology, 1995). Modifications to the momentum and k and ε equations were implemented via user Fortran code. Simulations were carried out in cylindrical co-ordinates (x, r, θ) and the flow in the spray dryer was assumed to be axisymmetric and steady state.

The computational grid used to represent the spray dryer chamber was chosen to be 49x25. Work by previous authors (Oakley et al., 1988, Langrish et al., 1993) has shown that use of a finer grid results in numerical instability due to a natural oscillation of the flow. This is also the experience of the present authors. Simulations were carried out with several finer grids in order to determine if the results were grid independent; however no satisfactory convergence could be obtained using any of these grids, with the residuals persistently oscillating.

Inlet air velocities were set based on air inlet flowrate and geometry. These were u = 19.8 m/s, v = -4.73 m/s and w = 11.43 m/s. A turbulence intensity of I = 0.2 was assumed at the air inlet although simulations were carried out for turbulence intensities of 0.15, 0.2 and 0.25 to determine the sensitivity of the results to this parameter. Inlet values of k and ε were then calculated according to:

$$k_{inlet} = 1.5(|U|I)^2 \quad (1)$$

$$\varepsilon_{inlet} = \frac{k_{inlet}^{3/2}}{0.333D_{eq}} \quad (2)$$

where the value of D_{eq} was assumed to be twice the width of the annulus.

For the outlet of the dryer a mass flow boundary condition is set with zero axial gradient. No-slip boundary conditions are set for the walls of the chamber.

3. MODIFIED k-ε TURBULENCE MODELS

3.1 Abujelala and Lilley Approach

Abujelala and Lilley (1984) attempted to obtain optimum values of the turbulence parameters C_μ, C_2 and σ_ε by minimizing the maximum absolute discrepancy between predicted axial and swirl velocities and the corresponding measured values for confined swirling pipe flow. The experimental data used for comparison in the optimization was that taken using a five-hole pitot tube (Yoon and Lilley, 1983). General predictions of moderately and strongly swirling flows were found to be more accurate using the optimized constants than predictions with the standard k-ε model. This optimization is not attempted here; rather the values of the constants determined by Abujelala and Lilley are used.

Langrish and Zbicinski (1994) have used these modified constants for the prediction of wall deposition rates in spray dryers and found that the wall deposition rate was predicted more accurately using the optimized constants than with the standard k-ε model constants. It was suggested by Langrish and Zbicinski that the constants proposed by Abujelala and Lilley were more appropriate for 'high recirculation' flows such as those commonly found in spray drying, while the standard constants were more appropriate for 'low recirculation' flows.

The Abujelala and Lilley constants are: C_μ = 0.125, C_2 = 1.6, σ_ε =1.1949. For comparison the standard values (Launder and Spalding, 1972) are: C_μ = 0.09, C_2 = 1.92, σ_ε =1.3.

Langrish and Zbicinski (1994) performed a simulation where the assumed inlet turbulent kinetic energy was doubled and the turbulence dissipation rate adjusted accordingly. This was to determine whether differences between predicted and measured wall deposition rates was due to differences between assumed and actual inlet turbulence intensity. They found a small improvement in the accuracy of the predictions using the higher inlet turbulence parameters; a calculation using these parameters is therefore included here.

3.2 Duggins and Frith Approach

Duggins and Frith (1987) modelled the anisotropic character of the turbulence by the use of two different eddy viscosities. The standard k-ε eddy viscosity was used in the axial and radial momentum equations, while the eddy viscosity used in the azimuthal momentum equation is given by a mixing length model.

Duggins and Frith (1987) set:

$$\mu_{r\theta} = \rho l_{r\theta}^2 r^2 \left| \frac{\partial w}{\partial r} - \frac{w}{r} \right| \qquad (3)$$

for the azimuthal eddy viscosity. Here we use the more general form:

$$\mu_{r\theta} = \rho l_{r\theta}^2 r^2 \left[\left(\frac{\partial}{\partial r} u \right)^2 + \left(r \frac{\partial}{\partial r} \frac{w}{r} \right)^2 \right]^{1/2} \qquad (4)$$

This eddy viscosity is then used to calculate the $r\theta$ and θx components of the Reynolds stresses according to:

$$\tau'_{r\theta} = \tau'_{\theta r} = \mu_{r\theta} r \frac{\partial}{\partial r}\left(\frac{w}{r} \right) \qquad (5)$$

$$\tau'_{\theta x} = \tau'_{x\theta} = \mu_{r\theta} \frac{\partial}{\partial x}(w) \qquad (6)$$

The other Reynolds stresses are then calculated using the k-ε eddy viscosity.

The above change of azimuthal eddy viscosity is derived from an analysis of data of turbulent swirling jets provided in Lilley and Chigier (1971). In that paper the anisotropy of swirling flows was demonstrated by calculating values of effective viscosities from velocity measurements. The rx and $r\theta$ components of eddy viscosity were shown to be significantly different from each other. The rx component started from a maximum at the axis and decayed toward the boundary of the jet, while the $r\theta$ component was zero at the axis and had a peak value at an intermediate radius then decreasing again toward the wall.

Two cases were run, these were with the value of $l_{r\theta}$ set to 0.034 and 0.068. A third case with $l_{r\theta}$ = 0.017 was attempted however satisfactory convergence could not be achieved with this value. Comparison was then made with the pitot tube data and predictions using the standard k-ε model.

4. RESULTS AND DISCUSSION

4.1 Inlet Turbulence

It was necessary to determine if the simulations were sensitive to inlet values of turbulence intensity because no experimental data was available to specify the inlet turbulence with accuracy. Simulations were first performed using the standard k-ε turbulence model with different values of inlet turbulence intensity. A comparison of radial profiles of axial and swirl velocities was made at four axial locations in the cylinder section of the dryer for three turbulence intensities: 0.15, 0.2 and 0.25; and with pitot tube data. Comparison was made at 120, 200, 300 and 600 mm below the ceiling of the dryer. In addition a simulation where the inlet turbulence dissipation rate was approximately seven times larger was also included in the comparison.

The inlet turbulence intensity in this case did not make a significant difference to the velocity profiles. When the inlet turbulence dissipation rate was increased by a factor of seven there was considerable variance in the velocity profiles predicted, however this did not result in more accurate prediction of the pitot tube data.

Hereafter, a turbulence intensity of 0.2 is chosen with the inlet values of k and ε calculated according to equations 1 and 2. These values are of the same order as those used by Oakley and Lagarde (unpublished report).

4.2 Predictions of the Modified Turbulence Models

Close to the inlet of the dryer the differences between measurements and predictions were not significant. The k-ε model provided the most accurate predictions for the measurements at 120 mm and 200 mm down stream of the air inlet, however the Duggins and Frith model was slightly improved in predicting velocities near the wall even at 120 mm although this improvement is best seen from 200 mm plus.

Figures 2-5 show a comparison of simulations based on the Duggins and Frith, and Abujelala and Lilley models, with predictions of the standard k-ε model and the pitot tube data of Oakley and Lagarde. The plots show that none of the turbulence models predict the experimental data with accuracy at 300 mm and 600 mm downstream of the air inlet. Predictions using the k-ε and RST model also simulated by Oakley and Lagarde (unpublished report) but not shown here were similarly inaccurate with the RST model results not showing any significant improvement over the k-ε model.

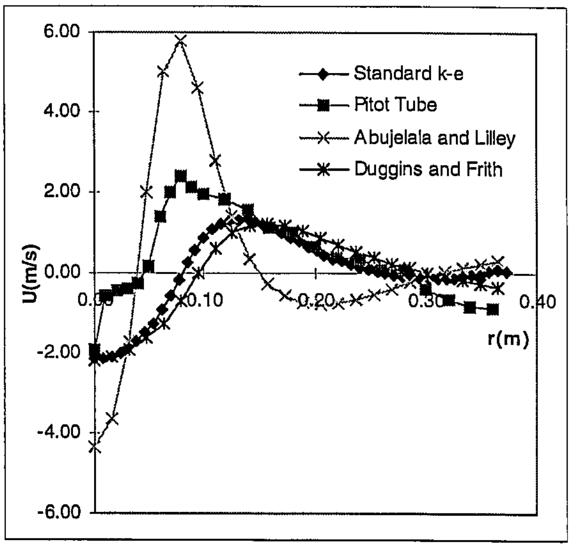

Figure 2 - Measurements and predictions of radial profiles of axial velocity at x=300mm.

In figure 2 all models predict the recirculation zone near the axis, however the magnitude and width of this zone varies considerably for each model. Only the Duggins and Frith model predicts the reverse flow near to the wall. The Abujelala and Lilley model is the most inaccurate, predicting a peak velocity about three times larger than that actually measured. Apart from predicting the negative axial velocity near the wall the Duggins and Frith model does not vary significantly from the standard k-ε in this case.

Figure 3 demonstrates the slight improvement in accuracy that the Duggins and Frith model has compared to the standard k-ε. A peak velocity is predicted closer to the axis and also closer to the actual peak value of the pitot tube data. Near to the wall of the spray chamber the Duggins and Frith model also proves to be the most accurate predicting a lower velocity which is closer than any of the other models to the actual velocity measured. The Abujelala and Lilley model over-predicts the swirl velocity for this case.

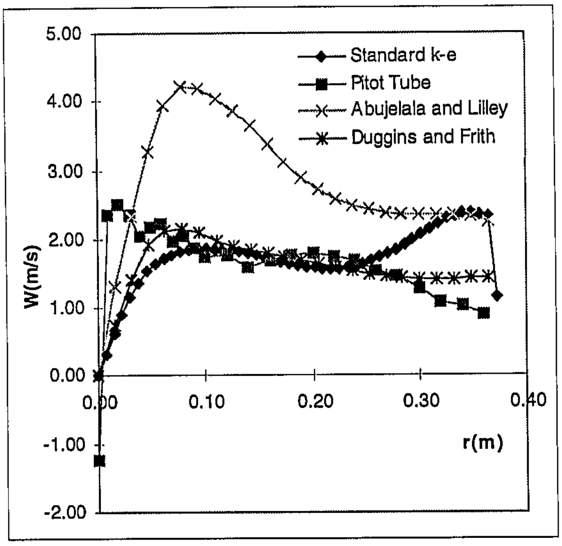

Figure 3 - Measurements and predictions of radial profiles of swirl velocity at x=300mm.

In figure 4 the large discrepancy between the predicted velocity and experimental data when using the k-ε turbulence model can be seen. The Duggins and Frith model is virtually indistinguishable from the standard k-ε in this case. The Abujelala and Lilley model is also inaccurate predicting a negative velocity at the axis when a positive velocity is observed. This model however does predict the magnitude of the peak velocity and also the recirculation zone between a radius of 0.2 and 0.35 m shown by the experimental data.

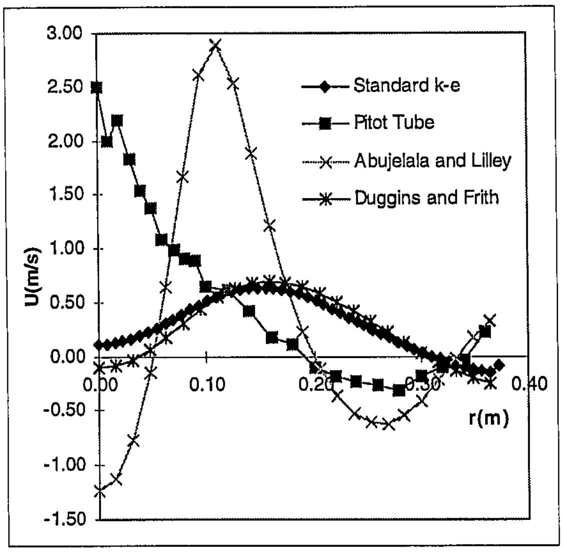

Figure 4 - Measurements and predictions of radial profiles of axial velocity at x=600mm.

In figure 5, the Duggins and Frith model demonstrates the same improvement shown in figure 3 over the standard k-ε for predicting the swirl velocity both closer to the axis and to the wall, however the improvement is small and the model is still very poor. At this axial distance for the swirl velocity the Abujelala and Lilley model predicts the pitot tube data better than the standard k-ε model however the peak value is still lower and further away from the axis than shown by the pitot tube data.

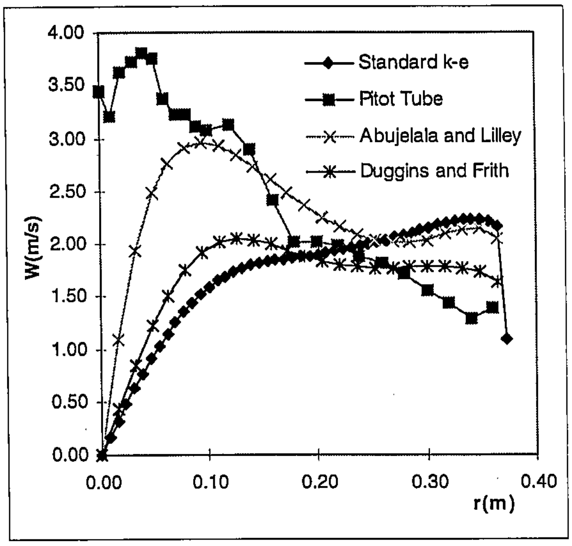

Figure 5 - Measurements and predictions of radial profiles of swirl velocity at x=600mm.

The effect on the predictions, based on the Abujelala and Lilley constants, of varying the inlet turbulence parameters according to the method used by Langrish and Zbicinski (1994) was found to be small even well down in the spray chamber. The predictions of the Duggins of Frith model were found to be more accurate using a value of 0.068r for the mixing length, rather than the 0.034r suggested by Duggins and Frith.

The eddy viscosity calculated for the azimuthal momentum equation varied significantly from that calculated using the definition in the k-ε model. In general the standard k-ε definition for eddy viscosity started at a peak at the axis and then decayed toward the wall of the chamber. The eddy viscosity calculated using the mixing length model started at zero at the axis and then gradually increased reaching a maximum at the wall. These profiles of eddy viscosity resemble the actual eddy viscosity profiles measured by

Lilley and Chigier (1971) for turbulent swirling pipe flow.

5. CONCLUSIONS

For the swirling flows normally encountered in spray drying the standard k-ε model is not adequate in predicting velocity profiles well downstream of the inlet to the spray chamber. The standard model is too diffusive and results in much flatter velocity profiles than are typically measured in real dryers. However the RST model, despite taking account of the anisotropy of the turbulence, does not result in a significant improvement in accuracy. Two modifications to the standard k-ε model which seek to account for its inadequacy in modelling certain types of swirling flows have been applied to a spray dryer. The aim was to provide a model which accounts for the anisotropy of the turbulence without a major increase in computational cost.

Both modifications have been shown to give no consistent improvement in accuracy of the prediction of axial and swirl velocities for turbulent single phase flow in a spray dryer. Some improvement has been noted in specific areas, in particular near the wall for the Duggins and Frith model.

The accuracy of the pitot tube data used to validate the computational results is in doubt due to the inherent inaccuracies of this instrument and the observed variance of the data. More accurate data is needed to make a fair comparison and an experimental program to collect this data is about to begin.

It is possible that the non-linear k-ε model of Dyakowski and Williams (1993), which extends the Duggins and Frith (1987) model, may give more accurate predictions in the region closer to the axis and further down in the spray chamber. This will be explored in further work. Also a mixing length model of Davidson (1988) is currently being investigated for it's applicability to spray dryer flows.

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REFERENCES

Abujelala, M.T., and Lilley, D.G., 1984, Limitations and Extensions of the k-ε Model as Applied to Turbulent Confined Swirling Flows, *Chem. Eng. Commun.*, 31, pp. 223-236.

AEA Technology, 1995, *CFX 4.1 Flow Solver User Guide*, Release 4.1, AEA Technology, Harwell Laboratory, Oxfordshire, UK.

Davidson, M.R., 1988, Numerical calculations of flow in a hydrocyclone operating without an air core, *Appl. Math. Modelling*, 12, pp. 119-128.

Duggins, R.K., and Frith, P.C.W., 1987, Turbulence Anisotropy in Cyclones, *Filtration & Separation*, pp. 394-397.

Dyakowski, T., and Williams, R.A., 1993, Modelling Turbulent Flow Within a Small-Diameter Hydrocyclone, *Chem. Eng. Sci.*, 48(6), pp. 1143-1152.

Langrish, T.A.G., Oakley, D.E., Keey, R.B., Bahu, R.E., and Hutchinson, C.A., 1993, Time Dependent Flow Patterns in Spray Dryers, *Trans IChemE*, 71(A), pp. 355-360.

Langrish, T.A.G., and Zbicinski, I., 1994, The Effects of Air Inlet Geometry and Spray Cone Angle on the Wall Deposition Rate in Spray Dryers, *Trans IChemE*, 72(A), pp. 420-430.

Launder, B.E., and Spalding, D.B., 1972, Mathematical Models of Turbulence, Academic Press, London.

Lilley, D.G. and Chigier, N.A., 1971, Non-Isotropic Turbulent Stress Distribution in Swirling Flows from Mean Value Distributions, *Int. J. Heat & Mass Trans.*, 14, pp. 573-585.

Oakley, D.E., Bahu, R.E., and Reay, D., 1988, The Aerodynamics of Co-current Spray Dryers, *Proc. of the Sixth Int. Drying Symp.*, Versailles, pp. 359-364.

Oakley, D.E., and Bahu, R.E., 1993, Computational Modelling of Spray Dryers, *European Symposium on Computer Aided Process Engineering* - 2, pp. S493-S498.

Oakley, D.E., 1994, Scale up of Spray Dryers with the Aid of Computational Fluid Dynamics, *Drying Technology*, 12(1&2), pp. 217-233.

Sloan, D.G., Smith, P.J., and Smoot, L.D., 1986, Modelling of Swirl in Turbulent Flow Systems, *Prog. Energy Combust. Sci.*, 12, pp. 163-250.

Speziale, C.G., 1987, On Nonlinear k-l and k-ε Models of Turbulence. *J. Fluid Mech.*, 178, pp. 459-475.

Yoon, H.K.; and Lilley, D.G., 1983, Five-Hole Pitot Probe Time-Mean Velocity Measurements in Confined Swirling Flows, *Paper AIAA-83-0315*, Reno, Nevada.