Measurement and modelling of the air flow pattern in a pilot-plant spray dryer

Citation for published version (APA):

Kieviet, F. G., Raaij, van, J., Moor, de, P-PEA., & Kerkhof, P. J. A. M. (1997). Measurement and modelling of the air flow pattern in a pilot-plant spray dryer. *Chemical Engineering Research and Design*, 75(3), 321-328.
<https://doi.org/10.1205/026387697523778>

DOI:

[10.1205/026387697523778](https://doi.org/10.1205/026387697523778)

Document status and date:

Published: 01/01/1997

Document Version:

Publisher's PDF, also known as Version of Record (includes final page, issue and volume numbers)

Please check the document version of this publication:

- A submitted manuscript is the version of the article upon submission and before peer-review. There can be important differences between the submitted version and the official published version of record. People interested in the research are advised to contact the author for the final version of the publication, or visit the DOI to the publisher's website.
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- The final published version features the final layout of the paper including the volume, issue and page numbers.

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MEASUREMENT AND MODELLING OF THE AIR FLOW PATTERN IN A PILOT-PLANT SPRAY DRYER

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The air flow pattern in a co-current pilot plant spray dryer (diameter 2.2 metres) was modelled and measured for the case without spray. The swirl angle was zero and the modelling was done with a computational fluid dynamics package (FLOW3D by CFDS). The boundary conditions for the CFD-model (velocity and turbulence quantities at the inlet) were derived from measurements with a hot-wire probe. To validate the CFD model, air velocity magnitudes were measured at numerous locations in the spray drying chamber. To interpret the data, a novel approach was developed based on the interpretation of velocity distributions rather than time-averaging the signals. This was necessary because of flow reversals and large fluctuations in the air velocity. The measurements were compared with the CFD model results and the agreement between model and measurements was reasonable.

Keywords: Spray dryer; air flow; hot-wire anemometry; CFD

INTRODUCTION

Spray dryers are used to obtain a dry powder from a liquid feed. Although the process equipment is very bulky and operation is expensive, it is an ideal process for drying heat sensitive materials. Spray dryers have been used for nearly a century now, but it is still very difficult to model the performance of this type of process equipment, especially with respect to the quality of the dried product.

The research described in this publication is part of a project that aims to develop a model which can be used to predict product quality. Product quality is directly related to the temperature and humidity experienced by particles during drying. These factors can be derived from a combination of the particle trajectories and the temperature and humidity pattern in the drying chamber. It is obvious that the particle trajectories, and thus the temperature and humidity pattern, depend directly on the airflow pattern.

The airflow pattern is also important for operational concerns such as caking: the airflow pattern determines the particle trajectories and thus where and when (i.e. at what moisture content) the particles hit the wall. Modelling the air flow pattern is therefore an essential part of every model that aims to predict the performance of a spray dryer.

Since the introduction of fast computers and computational fluid dynamics (CFD) software, it is now possible to predict the airflow pattern in spray dryers. CFD software is still very much under development, and results should not be used without validation.

In the open literature, a small number of model studies are described. Most of these studies were carried out on spray dryers that were an order of magnitude smaller than the dryer described in this study (10.3 m³). For instance, Oakley *et al.*^{1,2,3} used a 0.453 m³ dryer, Langrish and Zbicinski⁴ used a 0.779 m³ dryer. Only Livesley *et al.*⁵ performed studies on industrial size spray dryers, but, unfortunately,

little detail is presented in their publication due to industrial confidentiality.

As a first step, this publication describes the modelling and measurement of the airflow pattern in the absence of spray. The second step will be the modelling and measurement of temperature and humidity in the drying chamber under operational conditions, i.e. with a spray present; this will be described in a subsequent article.

In the following section the CFD model is described. Like any model, this CFD model requires the specification of input parameters (boundary conditions). These include trivial quantities like temperature, geometry and so on, as well as the less trivial turbulence quantities at the air inlet.

The next section is concerned with a description of how these turbulence quantities can be determined as well as the theory of the measurement technique that was used. The experimental setup is then described, followed by measurements of the inlet conditions and subsequently the measurement of the airflow pattern.

MODELLING THE AIRFLOW PATTERN

Present Knowledge in the Literature

In most spray dryers the air inlet is constructed such that the air enters with a tangential velocity component called swirl. The degree of swirl is expressed in the so-called swirl angle, which is defined as the angle between the axial and tangential velocity components of the drying air at the inlet.

The swirl angle has great influence on the airflow pattern. At small swirl angles, the airflow pattern consists of a fast flowing core with a slow circulation around that core. When the swirl angle is increased, at a certain value (critical swirl angle) vortex breakdown will occur. In this type of flow, the direction of the velocity in the centre of the original vortex will be reversed⁶⁻⁸.

Another aspect of the airflow pattern are fluctuations in velocity. Langrish *et al.*⁹ and Usui *et al.*¹⁰ observed slow fluctuations in the airflow, with frequencies of the order magnitude of 1 Hz. Oakley *et al.*¹ noticed the presence of oscillations of almost the same frequency in the CFD models of the airflow pattern. Langrish *et al.*⁹ noticed that the fluctuations were stronger in the case of high swirl and associated this observation with the occurrence of vortex breakdown. No explanation was given of the cause of the fluctuations in the case of low-swirl.

In flow visualization studies using video and particle image velocimetry, Stafford *et al.*¹¹ observed large velocity fluctuations in their spray dryer. The length scale of the disturbances was of the same order of magnitude as the radius of the dryer.

Theory of CFD Modelling

The basic equations of fluid flow are the Navier-Stokes equations (conservation of momentum), and the continuity equation. These equations can be found in every standard work on fluid flow modelling, e.g. Bird *et al.*¹². For the case of laminar flow, these equations can be solved analytically. Turbulent flow is strongly and rapidly fluctuating by nature. This, and the wide range of eddy sizes make it impossible to calculate the flow using present day computers; the calculation times as well as the memory required would be enormous. That is why one has to make all kinds of approximations to make computation feasible.

All current turbulence models are based on the splitting up of dependent quantities into a time averaged part and a fluctuating part. This is the so called Reynolds decomposition. For example, for the U velocity:

$$\tilde{U} = U + u \quad (1)$$

The mean of the fluctuating component equals zero by definition. When the Reynolds decomposition is applied to the Navier-Stokes equations, new unknowns emerge, e.g. the so-called Reynolds stresses:

$$\overline{uv}, \overline{uw}, \overline{vw} \quad (2)$$

Because of this, there are more unknowns than there are equations. When new equations are derived for the Reynolds stresses, new unknowns are introduced and so on. This makes it impossible to solve the equations (the so-called closure problem of turbulence). The only solution for this is to make use of assumptions using closure models.

The most widely used closure models are the k - ϵ model, the algebraic stress model (ASM) and the Reynolds stress model (RSM). The latter are more accurate in the case of anisotropic turbulence, which is the case in a high-swirl situation. The k - ϵ model is the most commonly used in engineering practice because it converges considerably better than the ASM and RSM model and requires less computational effort. Since the ASM and k - ϵ model yielded the same results for the situation described in this article, the k - ϵ model was used.

To calculate an approximated solution of the Navier-Stokes and continuity equation, the equations have to be made discrete. For this, the flow domain is divided into a number of control volumes. This is called the grid. For every grid cell, approximate solutions for the discrete Navier-Stokes and continuity equations are calculated.

Characteristics of the Flow Solver Used

The flow solver FLOW3D has been used, version 3.3 by CFDS (Computational Fluid Dynamics Services, Harwell, UK). This is a finite difference solver that uses body fitted co-ordinates which allows for the grid cells to be non-rectangular. The flow solver is based on the SIMPLEC algorithm. The solver has ten differencing schemes built in, of which the Upwind Differencing Scheme proved to be the most effective. The algorithms used for approximating the velocities were Block Stone's method; for the pressure, the Algebraic Multigrid method; and for the turbulence quantities k and ϵ , the Line Relaxation method. The values of the constants in the k - ϵ method were: $C_1 = 1.440$; $C_2 = 1.920$; $C_\mu = 0.090$; $\kappa = 0.419$.

On all edges of the flow domain the boundary conditions have to be specified. At the inlet the velocities, the turbulence intensity and dissipation all have to be specified. The boundary conditions at the outlet are such that the mass flow rate through the outlet equals the mass flow rate through the inlet.

Set-Up of the Problem in the Flow Solver

The spray dryer used in this research project is a pilot-plant co-current spray dryer by Niro Atomizer. The geometry is depicted in Figure 1. The drying air enters the drying chamber through an annulus with the nozzle as its centre (see Figure 2). The angle between the air inlet vector and the vertical is 35° , pointing to the centre of the annulus. The outlet of the spray dryer is a pipe mounted through the wall of the cone, bent downwards in the centre of the tower.

Figure 1. Geometry of the spray dryer.

Figure 2. Details of the geometry of the air inlet section (units in cm).

The geometry of the spray dryer was converted to a grid of 40 by 55 cells. Rotational symmetry was assumed. The grid is depicted in Figure 3. The grid is refined at the inlet so that it spans a range of five cells. Because of the rotational symmetry, the horizontal part of the outlet pipe cannot be modelled and is neglected.

Besides the physical properties of the flowing medium (air), the following boundary conditions were specified: the inlet velocity vector U, V, W (respectively the velocity in axial (x), radial (y) and tangential direction (z)) and the turbulence intensity k and the rate of dissipation ε . These quantities were measured and were found to be $(U, V, W) = (6.03, -4.22, 0.00 \text{ m s}^{-1})$; $k = 0.027 \text{ m}^2 \text{ s}^{-2}$, $\varepsilon = 0.37 \text{ m}^2 \text{ s}^{-3}$. These measurements are discussed in the next section.

Figure 3. The grid used in the CFD simulations; due to the rotational symmetry only half of the spray dryer is depicted. The air inlet covers a range of five cells. The horizontal part of the outlet pipe is not modelled.

The Calculated Flow Field

The flow solver was executed with the problem set-up as described above; vector plots of the calculated flow field are depicted in Figure 4. A measure of the quality of the convergence of the solution is the sum of the absolute values of the mass residuals of all grid cells. This sum was 0.002% of the total flow rate through the dryer. To check whether the solution was dependent on the grid chosen, the grid was refined twice in both the x and y -direction (80 by 110 cells) and the flow field for this grid was calculated. The difference between the two solutions was negligible.

THE DETERMINATION OF THE INLET CONDITIONS AND THE AIR FLOW PATTERN

Flow Conditions at the Air Inlet

The turbulent kinetic energy k and the dissipation rate ε at the inlet are boundary conditions that are required for the CFD model. There are a number of approaches to obtain values for these quantities. Oakley *et al.*¹ regarded k and ε as fit parameters in the CFD model. In a subsequent article², the use of CFD to model the inlet section of the dryer was described. The inlet section consisted of an annulus with swirl vanes, guiding ambient air into the drying chamber. Using the CFD technique, k and ε could be estimated.

Langrish *et al.*⁹ estimated the turbulent kinetic energy k

Figure 4. Vector plots (vector length represents speed (left figure) and normalized vectors (right figure)) of the calculated solution of the flow pattern.

from measurements using this equation:

$$k \equiv \frac{1}{2} \overline{u_i u_i} \approx \frac{3}{2} \overline{u^2} \quad (3)$$

The dissipation rate ε was estimated using the following equation:

$$\varepsilon = \frac{c_u^{3/4} k^{3/2}}{d} \quad (4)$$

Here, d is a characteristic length scale of turbulence and was taken to be equal to the distance between the swirl vanes. In the publication of Usui *et al.*¹⁰ and Langrish *et al.*⁴, a similar approach was followed.

Another approach is to derive both the turbulent kinetic energy k and the dissipation rate ε from measurements. This can be done as follows: the dissipation rate of turbulence energy is defined as¹³:

$$\varepsilon = \nu \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \frac{\partial u_i}{\partial x_i} \quad (5)$$

The dissipation mainly takes place in eddies of small length scales. These eddies of small length scales are inclined to isotropy ('local isotropy'). In the case of pure isotropic turbulence, the dissipation rate can be calculated using the Taylor micro scale λ ¹³:

$$\varepsilon = 30\nu \frac{\overline{u^2}}{\lambda^2} \quad (6)$$

Here, the Taylor micro scale is defined as:

$$\frac{1}{\lambda^2} \equiv \frac{1}{2\overline{u_i^2}} \left[\frac{\partial u_i}{\partial x_2} \right]_{x_1=0}^2 \quad (7)$$

The Taylor micro scale can be obtained from the *spatial* autocorrelation function of the velocity:

$$\rho(x_1) = 1 - \frac{x_1^2}{\lambda^2} \quad (x_1 \text{ around } 0) \quad (8)$$

This yields:

$$\lambda = \sqrt{\frac{-2}{\left. \frac{\partial^2 \rho(x)}{\partial x^2} \right|_{x=0}}} \quad (9)$$

The dissipation rate can also be calculated from the autocorrelation function of a velocity signal in *time* (a so-called time record) at one location, using the Taylor hypothesis of 'frozen turbulence':

$$\frac{\partial}{\partial t} = -U \frac{\partial}{\partial x_1} \quad (10)$$

This hypothesis is valid only if the mean velocity is constant and if the turbulence intensity is small.

For isotropic turbulence, the autocorrelation function is symmetrical around zero and its shape can be approximated with a parabola:

$$\rho(\tau) = 1 + a\tau^2 \quad (11)$$

The combination of equation (9) and (10) and the substitution of equation (11) yields:

$$\lambda = \sqrt{\frac{-2U^2}{\left. \frac{\partial^2 \rho(\tau)}{\partial \tau^2} \right|_{\tau=0}}} = U \sqrt{-\frac{1}{a}} \quad (12)$$

Now the dissipation rate ε can be calculated using equation (6).

For calculating the autocorrelation function to determine the turbulence dissipation rate, the measurement equipment must meet two requirements: the response time must be very short and the sensitivity must be high in order to be able to measure the fast and small velocity fluctuations due to turbulence. Two measuring techniques meet these requirements: laser Doppler anemometry (LDA) and hot-wire anemometry (HWA). In this research project, an HWA system was available.

Hot-Wire Anemometry

The hot-wire probe consists of a thin platinum coated silica wire supported by two prongs and the wire is heated by an electrical current. The heat transfer from the wire to the air depends on the velocity of the air. The electrical resistance (equivalent to the temperature) of the wire is kept constant by an electrical circuit. The bridge voltage of this circuit is a measure of the effective cooling velocity¹⁴ of the air:

$$E^2 = D + F(U_e)^n \quad (13)$$

The parameters D , F and n are determined by calibration.

The effective cooling velocity depends on the angle between the wire and the direction of the flowing air¹⁵:

$$U_e^2 = U_n^2 + k^2 U_p^2 + h^2 U_{bn}^2 \quad (14)$$

The indices n , p and bn mean: n = normal to wire, parallel to prongs; bn = normal to wire and normal to prongs and p = parallel to wire.

Because only heat transfer is measured, it is impossible to distinguish between two opposite flow directions. This is called the forward reverse ambiguity¹⁶. By calculating a velocity distribution of the measured effective cooling velocities, it is possible to check for flow reversals: in stable flow, the velocities *should* be distributed normally. If the distribution lies against the frequency axis so that it appears to overlap with velocities < 0 , flow reversals have occurred. In this case, the arithmetic mean of the cooling velocity i.e. the first moment of the velocity distribution can overestimate the true mean significantly.

A better way to estimate the mean in this case was found to be as follows: since the part of the velocity distribution at negative velocity is added to the positive counterpart, the true velocity distribution can be described with:

$$P(U) = \frac{1}{\sigma \sqrt{2\pi}} \left[\exp\left(-\frac{(U-\mu)^2}{2\sigma^2}\right) + \exp\left(-\frac{(U+\mu)^2}{2\sigma^2}\right) \right] \quad (15)$$

In this equation it is assumed that the true velocity distribution is a normal probability function. The mean velocity can now be estimated by fitting this function to the measured normalized velocity distribution.

If the hot-wire probe(s) are positioned in three orthogonal directions (as depicted in Figure 5), it appears to be possible to determine the velocity magnitude as well as a number of

Figure 5. Positions of the hot-wire anemometer in three independent positions.

possible velocity directions:

$$\begin{pmatrix} \tilde{U}_1^2 \\ \tilde{U}_2^2 \\ \tilde{U}_3^2 \end{pmatrix} = \begin{pmatrix} k^2 & 1 & h^2 \\ 1 & k^2 & h^2 \\ h^2 & 1 & k^2 \end{pmatrix}^{-1} \begin{pmatrix} \tilde{U}_A^2 \\ \tilde{U}_B^2 \\ \tilde{U}_C^2 \end{pmatrix} \quad (16)$$

This is true only if the magnitude and direction of the flow is constant. This is not the case in the flow problem in this study. Therefore, this method cannot be used.

EXPERIMENTAL SET-UP AND RESULTS

Hot-Wire Set-Up

The measurements were done using a Thermo Systems Inc anemometer system consisting of a single normal probe type 1210, a monitor and power supply module type 1051-2 and, a constant temperature anemometer type 1054A. The wire is about 40 micrometers in diameter and about 2 millimetres in length.

The bridge voltage was sampled (up to 20 kHz) using a 12-bits AD-converter (DAS8PGA, Keithley Instruments) built into a PC. The data acquisition software converts the measured voltage in real time to the effective cooling velocity using equation (13). Besides recording and storing time records (short time intervals and 16384 samples), the software also calculates the mean, the standard deviation and the velocity distribution of the calculated cooling velocities (any time interval and number of samples unlimited).

The autocorrelation function of a time record was calculated from the inverse Fourier transform of the squared moduli of the Fourier spectrum (Wiener-Khinchin theorem). To enhance the accuracy of the Fourier transform, the time record was divided in seven half overlapping chunks that were all fast Fourier transformed individually. Before calculating the squared moduli of each chunk, the transform was multiplied with a Hanning window. The resulting transformed chunks were averaged and were inverse fast Fourier transformed yielding the autocorrelation function¹⁷.

For the calibration of the hot-wire, a Thermo Systems Inc model 1125 calibrator was used. This system consists of a calming chamber with a series of wire meshes and a rounded nozzle. By measuring the pressure difference (Hottinger Baldwin model PD1 + KWS 3073) over the nozzle, the velocity of the air could be calculated. The velocity profile was assumed to be flat with a low turbulence intensity. The pressure signal was recorded using a PC and the parameters D , F and n were obtained by non-linear regression of equation (13).

Figure 6. Velocity distribution of inlet velocities.

To position the hot-wire probe in the spray drying chamber, a vertical arm was used that can be moved manually along the wall of the cylindrical part of the dryer. To the bottom side of the arm, a horizontal arm was fixed with a length that equalled the radius of the spray dryer. On the roof of the dryer, a little motor was installed to rotate the vertical arm under PC control. In this way, all radial positions could be reached.

Measurements in the Air Inlet

At first the flow direction was determined using tufts. From this it appeared that the swirl angle was nearly zero; the exact value of the swirl angle could not be measured easily. The inflow of air was not the same at every location in the inlet. To be able to make a fair comparison of the CFD calculations and the air flow measurements, the top side of the air distributor was removed. This reduced the tangential velocity component of the inlet air to zero.

The hot-wire was positioned in the centre of the air inlet gap, adjacent to the spray dryer roof and normal to the airflow (wire parallel horizontal and prongs vertical). For this, the probe was mounted on a support that was fixed to a vertical arm positioned in the centre of the dryer. The vertical arm was rotated to measure at several positions in the annulus. The velocity signal was recorded with a sampling rate of 20 kHz.

In Figure 6 the velocity distribution is depicted. A time record at 20 kHz is shown in Figure 7. The turbulent kinetic energy k was calculated using the average of three time records and by means of equation (3), and had a value of $0.027 \text{ m}^2 \text{ s}^{-2}$. Because the turbulence intensity is low (2%) and the velocity is constant (see Figure 7) the Taylor hypothesis of 'frozen turbulence' can be applied. Therefore the dissipation rate ε can be calculated from the autocorrelation function and equations (6), (11) and (12). The value of ε was calculated to be $0.37 \text{ m}^2 \text{ s}^{-3}$.

Figure 7. Time record at inlet, measured at 20 kHz.

Figure 8. Some characteristic velocity distributions.

Measurement of the Airflow Pattern in the Drying Chamber

A multi position technique was attempted to obtain velocity magnitudes *and* directions according to equation (16) as a function of position in the drying chamber. After a number of experiments it became clear that this technique did not yield any useful results. This was caused by flow reversals and the unstable nature of the flow. Therefore the multi position technique was abandoned and the experiments were restricted to measuring velocity magnitudes. The following procedure was used:

On a grid of a number of radial and axial positions, velocity distributions were obtained from measurements during 10 minutes at 200 Hz. The probe was positioned in such a way that the wire and the prongs were parallel to the roof. Examples of a number of characteristic velocity distributions are depicted in Figure 8. The distributions were divided into four categories:

- Normal distributions that do not lie against the frequency axis (e.g. distribution 30/9.7 in Figure 8). These distributions are at locations where the flow is apparently stable and where there are no flow reversals. Hence the mean velocity can be calculated from the first moment of the velocity distribution (equals the arithmetic mean of the velocities).
- Narrow distributions that lie against the frequency axis (e.g. distribution 140/57 in Figure 8). These distributions are at locations where the flow is apparently stable but where flow reversals occur. Calculating the mean velocity magnitude by calculating the first moment of the velocity distribution would result in a very large overestimation of the mean because flow reversals are neglected. Therefore the distribution was fitted with equation (15). Initially, this fit was carried out with a Marquardt-Levenberg algorithm, but the results proved to be very dependent on the guess values. Therefore the fit was done with a rigorous scan of the (μ, σ) -domain.
- Skewed distributions that lie against the frequency axis (e.g. distribution 140/9.7 in Figure 8). These distributions consist of a large number of normal distributions. Since the velocity distributions lie against the frequency axis, it is *impossible* to calculate a sensible mean. The only way of evaluating the velocity distribution quantitatively is to assign a velocity interval by eye.
- Wide distributions that do not lie against the frequency axis (e.g. distribution 60/9.7 in Figure 8). It is possible to calculate a mean of this distribution by calculating the first

moment. However, this mean is not very reliable. Therefore a velocity interval has also been indicated for this category of distributions.

The calculated means and estimated intervals are shown in Figure 9.

In Figure 10, partial time records of the velocity distributions in Figure 8 are given. At locations 60/9.7 and 140/57, a slow periodic fluctuation can be seen. In the autocorrelation function of these time records (see Figure 11), it can be seen that the characteristic time of

Figure 9. Predicted (curved line) and measured velocities (dots: time averaged values; vertical lines: estimated interval) at a measurement grid. For all axial positions except at $x = 260$ cm, the radial positions are: (left side) 93.6, 75.8, 57.3, 28.9, 19.3, 13.5, 9.7, 3.9, (axis) 0.0, (right side) 3.9, 9.7, 13.5, 19.3, 28.9, 38.5, 47.9, 57.3, 75.8, 93.6 cm. For $x = 260$ cm the radial positions are: 47.9, 38.5, 28.9, 19.3, 9.7, 0.0, 9.7, 19.3, 28.9, 38.5, 47.9 cm.

Figure 10. Time records at four locations.

this fluctuation is about 12 seconds. No distinct periodicity can be discerned in the autocorrelation function of the air inlet velocity signal.

DISCUSSION

At various locations in the drying chamber it is impossible to calculate a sensible mean velocity due to flow reversals and large variations in the velocity. Therefore ranges have been indicated within which the velocity magnitude may vary. The following can be noted about the fluctuations:

- The time records and velocity distributions show that the flow pattern changes continuously, but the flow pattern does not alternate between two meta-stable patterns; this would yield bi-modal velocity distributions and these were not observed.
- A distinct slow periodicity can be discerned in the velocity signal (see Figure 10 and Figure 11) at several locations in the drying chamber. This periodicity is not caused by variations in the air feed rate, since no obvious periodicity was observed in the time history record of the inlet air velocity.

The exact causes of the large fluctuations or instabilities is not known.

The broad velocity ranges as estimated with the single hot wire technique raises the question whether the velocities could have been measured more accurately by using more sophisticated measurement techniques. One could consider:

- Triple hot wire anemometers—with a triple hot-wire one can measure a more accurate instantaneous velocity magnitude. Flow directions can not be measured if the flow is unstable or if flow reversals are present.
- 3D laser Doppler anemometers—with this type of equipment one can measure a velocity magnitude as well as a velocity direction, even when flow reversals are present. With this technique, it is possible to determine a more accurate time average for the local velocity.

Both techniques yield that the increase in accuracy in the average local velocity has little significance because of the large variations in local velocity.

A technique that produces much more useful information is one that measures the velocity at a large number of locations at the same time. In this way the velocities at various locations can be correlated. For velocity fields that

Figure 11. Autocorrelation functions at four locations.

are purely two dimensional, laser image velocimetry can be used¹¹. Three dimensional applications are under development and appear to be extremely promising in spray drying research.

CONCLUSION

For the interpretation of the hot-wire anemometer measurement data, it proved to be essential to consider velocity distributions instead of arithmetic means as they reveal the unstable nature of the flow pattern and produce more accurate time-averaged velocities in the case of flow reversals.

Considering the unstable nature of the measured flow pattern, reasonable agreement was still found between the measurements and CFD results:

- The CFD model correctly predicts a fast downward flowing core and low velocities in the outer region.
- The predicted shape of the velocity profile of the core near the inlet is consistent with measurement data.
- The widening of the core toward the air outlet as predicted was also found in measurements.

NOMENCLATURE

a	fitting parameter, s^{-2}
c_u	constant in k - ε equation, 0.09
d	characteristic length, m
D	hot-wire calibration constant, 7.0225
E	bridge voltage, V
F	hot-wire calibration constant, 4.562
h^2	pitch factor of hot-wire, 1.0
I	turbulence intensity, —
k	kinetic energy of turbulence, $m^2 s^{-2}$
k^2	yaw factor of hot-wire, 0.062
n	hot-wire calibration constant, 2.027
$P(U)$	probability function
U	velocity, especially in x -direction, $m s^{-1}$
$\underline{U}_e$	effective cooling velocity, $m s^{-1}$
$\underline{u}_e^2$	standard deviation of the effective cooling velocity, $m^2 s^{-2}$
V	velocity in radial direction, $m s^{-1}$
W	velocity in tangential direction, $m s^{-1}$
x	axial direction
y	radial direction
z	tangential direction

Greek letters

ε	turbulent energy dissipation, $m^2 s^{-3}$
λ	Taylor micro scale, m
μ	mean velocity, $m s^{-1}$
ν	kinematic viscosity, $m^2 s^{-1}$
ρ	autocorrelation function
σ^2	variance, $m s^{-1}$
τ	lag time, s

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ACKNOWLEDGEMENTS

This research is part of the project Modelling Quality in Spray Drying and was sponsored by the Unilever Research Laboratory Vlaardingen.

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The manuscript was received 5 March 1996 and accepted for publication after revision 31 October 1996.