Experimental and CFD Airflow Studies of a Cleanroom with Special Respect to Air Supply Inlets

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Abstract

Investigations were carried out into the air flow in a non-unidirectional airflow cleanroom and its affect on the local airborne particle cleanliness. The main influence was the method of air supply. A supply inlet with no diffuser gave a pronounced downward jet flow and low levels of contamination below it, but poorer than average conditions in much of the rest of the room. A 4-way diffuser gave much better air mixing and a more even airborne particle concentration throughout the cleanroom. Other variables such as air inlet supply velocity, temperature difference between air supply and the room, and the release position of contamination also influenced the local airborne cleanliness.

A CFD analysis of air flow fields in a cleanroom was compared with measured values. It was considered that a turbulent intensity of 6%, and a hydraulic diameter based on the actual size of the air inlet, should be used for the inlet boundary conditions and, when combined with a k–epsilon standard turbulence model, a reasonable prediction of the air flow and airborne particle concentration was obtained.

Key words: Cleanroom, air supply inlets, airborne particles, CFD, turbulent intensity, hydraulic diameter, air turbulence model.

1 Introduction

Cleanrooms are used to minimise the contamination of products made in manufacturing industries, as well as microbial infection in hospitals. There are two basic types of cleanrooms known as ‘unidirectional’ and ‘non-unidirectional’. The unidirectional airflow system works by supplying particle-free air through a ceiling or wall of high efficiency filters and passing it in a piston-like way through the room at a velocity of about 0.45 m/s. The non-unidirectional airflow system uses a conventional ventilation system similar to that found in hotels, offices etc. but with much higher ventilation rates, and final air filters that have very high particle-removal efficiencies and placed in air supply inlets in the ceiling. It is common to find that the air inlets in a non-unidirectional cleanroom are installed without a diffuser, this design probably originating in an attempt to achieve the superior air cleaning of unidirectional air flow, but inlets with diffusers are also used. A fuller description of the design of cleanrooms is given by Whyte (2010).

The cleanliness in a cleanroom is classified by use of ISO 14644-1 (2001), and carried out by ensuring that the concentration of airborne particles does not exceed the limits laid down in the standard. The overall average cleanliness of non-unidirectional airflow cleanrooms is directly related to the air supply filter efficiency, the air supply volume, and the number of particles dispersed within the cleanroom. However, the particulate concentration may vary about the cleanroom and the local particle concentration is dependant on the air flow pattern, and this has been investigated and reported in this article.

2 Description of Cleanroom Investigated and Measuring Methods

The cleanroom studied was an industrial cleanroom, and a plan view of it is shown in Figure 1. Figure 2 shows a section through a minor vertical axis of the room (X-X). The cleanroom had a floor area of 7 m x 4.6 m, with a 2.76 m high ceiling and was therefore 89m^3 in volume. It may be see that the air supply inlets discharged air from one side of the ceiling and the air was extracted through low-level grilles at the opposite side of the room. A stainless steel table, one metre high, was in the middle of the room at a position where products were normally manufactured, and simulated the obstruction to the airflow.

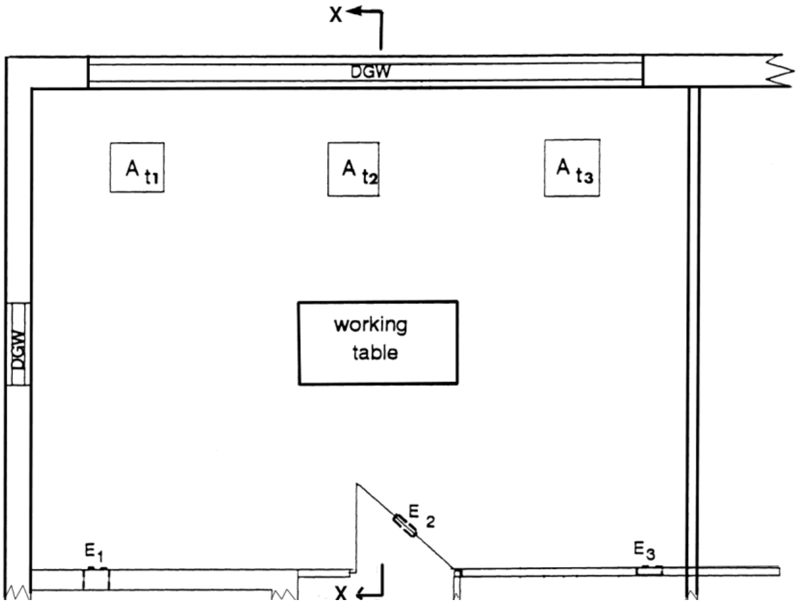

Figure 1 Plan view of cleanroom, A_t = air inlet, E = exhaust grilles, DGW = double glazing window

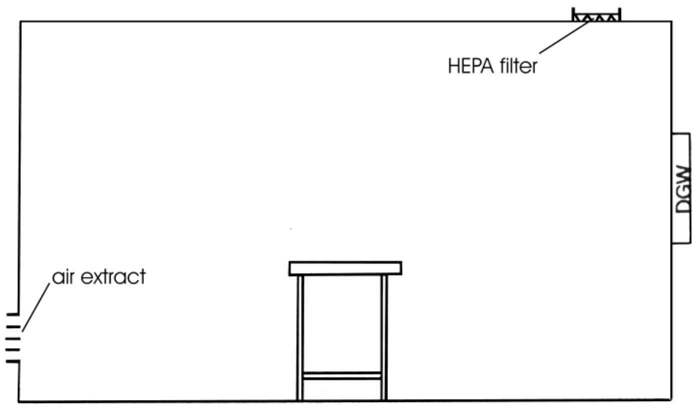

Figure 2 View of cleanroom at section X-X. DGW = double glazed window.

The air change rates of the cleanroom studied were 3, 15 and 20 per hour, these being measured by use of a hood placed over the air supply inlets. The high efficiency air filters were fitted into the supply inlets in the ceiling. Two types of air supply inlets were investigate, these being the type that are most commonly used in cleanrooms i.e. air inlets with 4-way diffusers, and inlets with no diffusers fitted. If no diffusers were fitted, the air exited out of areas of 0.5 m x 0.5 m, and the air supply velocity when the air changes were 3, 15 and 20 per hour, was 0.1 m/s, 0.49 m/s and 0.65 m/s, respectively. If a 4-way diffuser was fitted, the air was directed in four directions at 90° to each other. Each quarter section of the 4-way diffuser had 4 blades that were permanently set so that the air exited at an angle that was determined experimentally to be about 20º to the ceiling. The exit area of a 4-way diffuser was about half the area of that when no diffuser was fitted, and hence the velocity was about double i.e. at 20, 15 and 3 air changes per hour the exit velocity was 0.2 m/s, 1.0 m/s and 1.3 m/s, respectively.

An air change rate of 15 per hour was most thoroughly studied, as it corresponded to an air supply filter face velocity of 0.49 m/s, which was close to the velocity that high efficiency filters are normally designed to be used i.e. 0.45 m/s. The total air supplied from all three inlets was 0.37 m^3/s. Before measuring the room velocities, the air inlet volumes were adjusted to the required volumes and the extract volumes were then adjusted to balance the air supply and give a pressure differential of 10 Pa between the cleanroom and the area outside.

Thermocouples were used to measure the air supply temperature at the face of the HEPA, and the cleanroom temperature was ascertained at the exhaust grilles. The temperature differential between the supply and room air could be set to zero, or the supply air made hotter or colder than the cleanroom air.

A test aerosol was generated by means of a Sinclair-Phoenix particle generator containing a single Laskin nozzle. The concentration of particles ≥ 0.5 μm was measured by an HIAC Royco Model 5130 airborne particle counter.

3 Experiments and Results

3.1 Measurement of Air Velocity and Direction in the Cleanroom

The air velocity and direction was measured using two perpendicular rods placed from floor-to-ceiling on both sides of the room on the plane X-X shown in Figure 1 and 2. String was suspended between the rods and marked at regular intervals to shown where the velocity and direction should be measured. A wisp of smoke was used to ascertain the airflow direction and the velocity was measured in the established direction. Velocities were measured down to about 0.2 m/s by means of a thermal anemometer (Airflow Developments Model TA400), and below 0.2 m/s by using a DISA Type 55 D80 low velocity anemometer. The anemometers were calibrated by means of a low speed wind tunnel (Whyte, Whyte and Green, 2010), where the exact air velocity was obtained by timing smoke over a given distance, and the velocities over the range of 0.1 m/s to 0.6 m/s were shown to be accurate to within +/- 5%.

Two examples of the airflow fields obtained are shown in Figures 3 and 4. These show the airflow from (a) the 4-way diffuser and (b) an air inlet without a diffuser, when the air change was 15 per hour and the temperature of the air supply was the same as the cleanroom air. The actual velocities are given on the figure but the direction arrows are not proportional to velocity, as the velocity range was too large, and at low velocities would give an arrow length that was too small to show the direction.

Figure 3 shows that the air supplied by a 4-way diffuser is ‘thrown’ along the ceiling towards both walls, with a continuation of flow down the nearside wall. It also shows that the diffuser worked as expected by entraining the room air and giving velocity decay away from it that leads to a generally random low-velocity flow in the room with an overall movement to the extract.

Figure 4 shows air flowing as a jet from the air inlet without a diffuser. The air supply flows down to the floor and round the room in a generally clockwise manner and exits out of the extract. The air at the outside wall is seen to rise, possibly to replace the air entrained into the jet, or because the wall was an outside one and warmer than the room.

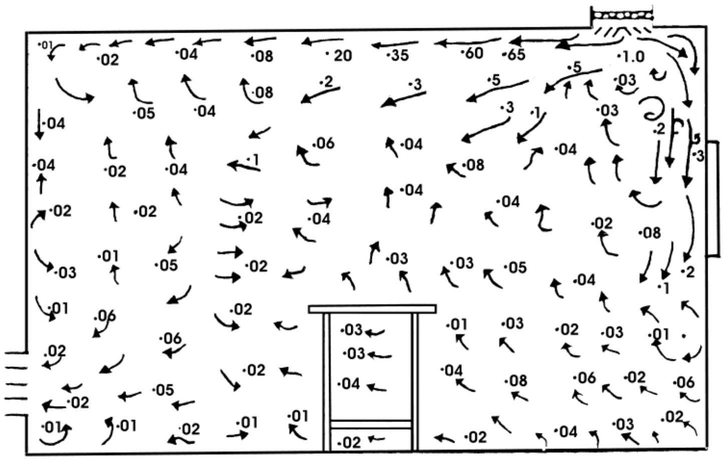

Figure 3 Airflow from a 4-way diffuser

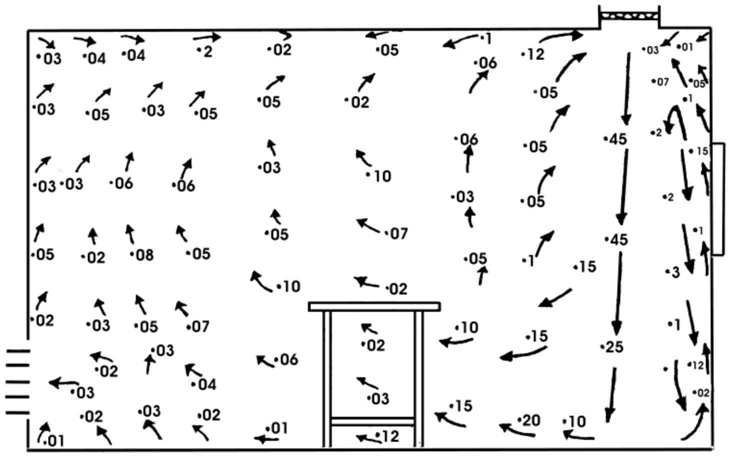

Figure 4 Airflow from an air inlet without a diffuser

3.2 Penetration of particles into the air flowing from an air inlet with no diffuser

The air cleanliness below an air inlet with no diffuser was investigated. Particles generated by a Laskin nozzle were continually released at four points around an inlet with no diffuser, and the concentrations of particles ≥ 0.5 µm measure outside and within the air jet flowing from the inlet. The room air changes were set at 20 per hour, which gave the filter a face velocity of 0.65 m/s, and the temperature difference between the air supply and the cleanroom air was set to zero. The particle concentrations within the jet were calculated as a percentage of the concentration outside the supply air jet, and isopleths plotted as shown in Figure 5.

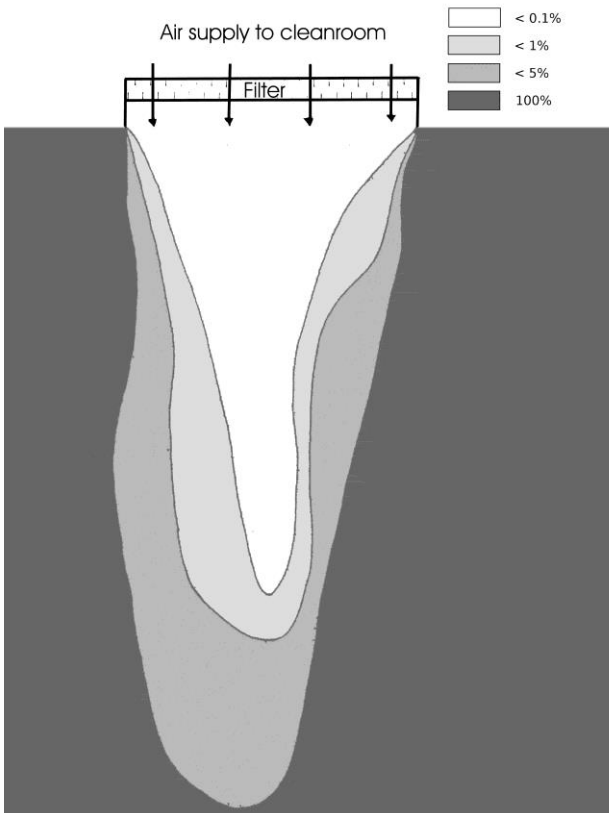

Figure 5 Penetration of particles from the cleanroom into a air jet from an inlet with no diffuser

3.3 Ventilation Effectiveness as Determined by a Constant Release and Transfer of Particles during Steady-State Conditions

The air flow and particle concentration in a non-unidirectional cleanroom was investigated with respect to (a) the type of air inlet (b) the air inlet supply velocity, (c) the temperature difference between the supply and room air, and (d) the point where contamination was released. This was carried out by use of a Performance Index (PI), using an approach similar to that described by Lidwell (1960) and Lidwell et al (1967).

Particles were released at the constant rate at one of two release points using a five-way connector at the end of a tube attached to the particle generator. The airborne concentration of particles ≥0.5µm was measured and allowed to come up to a steady state. The release rate of test particles was then determined from the sum of the products of the concentration of particles and the air volume passing out of the three room exhausts. Assuming that there was perfect air mixing in the room, the steady-state average concentration of the particles in the room air was determined by Equation 1:

$$\text{Steady-state average concentration of particles /m}^3 = \frac{particle\ \ release\ \ rate\,(no/s)}{air\ \ \sup ply\,rate\,(m^3/s)} \qquad (1)$$

When a steady-state condition had been reached, the particle concentrations were measured at sampling points that were one metre from the floor and at the following positions:

Sampling position 1 (SP1): below the middle air supply inlet;

Sampling position 2 (SP2): top of table in middle of cleanroom;

Sampling position 3 (SP3): between the table and the middle exhaust.

To avoid sampling unrepresentative concentrations of particles that would be found close to the release points, the sampling points used were SP2 and SP3 when particles were released from RP1, and SP1 and SP2 when particles were released from RP2.

The PI, which is the ratio of the quantity of the particles measured at the sampling point to that quantity that would have been measured if there was perfect mixing of the airborne test particles, was then calculated for each of the three sampling points using the following equation:

$$\text{Performance Index} = \frac{steady\ state\ average\ concentration\ in\ room\ air/m^3}{particle\ concentration\ at\ sampling\ point/m^3} \qquad (2)$$

Note: the numerator and denominator were mistakenly reversed in the original published equation

The following ventilation variables were investigated:

1. Air inlets fitted with either no diffuser or a 4-way diffuser.
2. Air change rates of 3, 15 and 20 per hour. These were produced from a 4-way diffuser with air inlet velocities of 0.2, 1.0 and 1.3 m/s, respectively, or in the case of an inlet with no diffusers, 0.1, 0.49 and 0.65 m/s, respectively.
3. Temperature difference (ΔT). The supply temperature could be set so there was either no temperature difference between the supply and the room air, or the supply air was either 2°C higher or 2°C lower than the room air. It was difficult to exactly set these differential temperatures but an inaccuracy of not greater than +/- 0.2°C was achieved.
4. Release point of particles. Both were 1 metre from the floor and situated where personnel would stand i.e. either between the middle air supply inlet and the table (RP1), or between the table and the middle exhaust (RP2).

The air extracts was also likely to affect the air flow but no modifications could be made to their position. However, it is unusual to find any cleanroom with exhausts sited in any other position other than at low level in the walls.

Seventy experiments were carried out that related the PI to different combinations of the four ventilation variables. The results were analysed by multiple regression analysis available on Minitab Statistical Package (Minitab Inc.). This produced an equation that relates the magnitude of the PI value at a sampling point to the four independent variables in the manner shown in Equation 3:

PI = Constant + k_1 (air change rate) + k_2 (ΔT) + k_3 (type of device: jet = 0 and 4-way= 1) + k_4 (release point: RP1 = 0 and RP2 = 1)

(3)

The actual numerical values of air change rate per hour and temperature difference (°C) were used in the multiple regression analysis. When the temperature was colder than the room air, the numerical value was given as positive, and vice versa. Both the type of air inlet and the number of the release point were assigned the values 0 or 1 as shown in Equation 3.

The PI was considered to be significantly affected by a ventilation variable if the level of statistical probability was less than 0.05. However, the temperature differential was included when the probability was less than 0.1, as it was clear from experiments carried out on the airflow coming from the inlet that the temperature difference between the supply and room air affects the airflow in the cleanroom, and may therefore be expected to have an effect on the PI. Statistically insignificant independent variables were excluded from the equations. The results were analysed to show the effect of the four variables on the PI measured at the three sampling points when the air was supplied by either a 4-way or no diffuser. The resulting equations are as follows:

No diffuser inlet

$PI_{SP1} = 0.714 – 0.032AC + 0.038\Delta T$

$PI_{SP2} = 1.221 + 0.151\ \Delta T – 0.873R_p$

$PI_{SP3} = 0.988 + 0.195\Delta T$

4-Way diffuser

$PI_{SP1} = 1.110 – 0.036AC$

$PI_{SP2} = 2.010 – 1.422R_p$

$PI_{SP3} = 1.65$

Where,
AC = air change rate per hour,
ΔT = the temperature difference (°C)
R_p = release point
Sampling point is shown by a subscript next to the PI

Substituting the values of the independent variables in the equations allows the PIs to be calculated, and the relative importance of the variables to be determined. However, it must be understood that these results do not show the effect of the ventilation variables on the overall concentration of airborne particles in the room, but on localised concentrations at the sampling points.

As the PI is the ratio of the concentration of particles at a sampling point to the average concentration that would have been found had the air been mixed perfectly, the further the PI is away from 1 the less air mixing in the cleanroom is likely to occur. A high PI also indicates a high particle count at the sampling point and vice versa. Using this interpretation, the following conclusions were drawn for each sampling position:

1) The air inlet with no diffuser gave a lower airborne particle concentration under the inlet than the four-way diffuser. The higher the air supply velocities and the lower the supply temperature compared to the cleanroom, the lower the particle concentration under the air inlet. The lowest particle concentration of all the individual experiments was found under the no-diffuser inlet, it being 25 times lower than the average concentration in the room. This agrees well with the results of the investigation of the penetration of contamination into the air jet that are discussed in section 3.2 and shown in Figure 5. The 4-way diffuser gave higher particle counts than those found under the inlet with no diffuser, and the higher the air supply velocity the lower the particle concentration. The air flow under the four-way diffuser was not significantly affected by temperature difference.
2) The airborne particle contamination at the measuring point above the table was found to be mainly dependant on the position of the release point. When particles were released between the table and the exhaust, many fewer particles were transported to the table than when particles were released between the table and the air supply. This was found with both the no-diffuser and 4-way diffuser, and can be explained by the fact that the air flow is generally from the air supply to the extract. When the particles were released between table and air supply inlet and the particles sampled at the table, one individual experiment gave the highest particle concentration of all the individual experiments and was about 5 times greater than the average concentration. The air supply temperature appeared to affect the particle counts at the table when no diffuser was fitted, but not when a 4-way diffuser was fitted. The particle count above the table did not significantly deviate from the average count in the room when the air change rate, and hence air inlet supply velocity, was changed.

3) The contamination at the sampling point between the table and the exhaust was little affected by the four ventilation variables studied, probably because air mixing had occurred over the longer distance the air was required to flow before getting to that sampling point. However, in the case of the jet coming from the inlet with no diffuser, the temperature difference appeared to affect the particle counts.

3.4 Ventilation effectiveness as assessed by lateral dispersion

The effect of the air supply inlet on the mixing of room air was further investigated by releasing particles at one end of the cleanroom and determining how well they distributed throughout the room before being removed by the exhaust. Particles generated by a Laskin nozzle were released at a constant rate into the supply air of inlet 3, which was situated at one end of the cleanroom (see Figure 1) and the concentration of particles ≥ 0.5 μm was allowed to come up to a steady state. The release rate of test particles was determined in the same manner as described in section 3.3 and the steady state concentration in the room air determined by Equation 1. The PI at each of the exhausts was then determined from the concentration of airborne particles at each of the three exhausts and the use of Equation 2. The PIs obtained at the three exhausts when the 4-way diffuser and inlet with no diffuser were used are given in Table 1 and Table 2, respectively.

Table 1 Performance indexes at the three exhausts when 4-way diffusers were studied

Air change rate	**ΔT (°C)**	**PI - exhaust 1**	**PI - exhaust 2**	**PI – exhaust 3**
3	-1.05	1.03	N/A	0.97
3	-0.16	1.03	N/A	0.98
15	0	1.11	1.13	0.84
15	0	0.99	0.93	1.03
15	+0.05	0.92	1.05	1.09
20	-0.04	1.12	1.03	0.89
20	0.10	1.11	1.08	0.90

N/A = not applicable as there was insufficient air supply to pressurise the cleanroom unless the exhaust was blanked off

Table 2 Performance indexes at three exhausts when inlets with no diffusers were studied

Air change rate	**ΔT (°C)**	**PI - exhaust 1**	**PI - exhaust 2**	**PI – exhaust 3**
3	-0.13	1.07	N/A	0.93
3	-0.03	1.09	N/A	0.91
15	-0.13	0.83	0.79	1.24
15	-0.02	0.74	0.75	1.36
20	0	0.72	0.56	1.38
20	0	0.70	0.54	1.47

N/A = not applicable as there was insufficient air supply to pressurise the cleanroom unless the exhaust was blanked off

It can be seen when a 4-way diffuser was investigated, the lowest value of PI was 0.84 (16% lower than the average value in the cleanroom), and the highest was 1.12 (12% greater than the average value). When inlets with no diffuser were studied, the lowest air change rate (3/hour) gave good air mixing but 15 and 20 air changes per hour reduced the mixing. The lowest PI was obtained when the air change rate was 20 per hour and this was 0.70 at exhaust 1; the highest was 1.47 at exhaust 3. It can therefore be deduced that 4-way diffusers gave better mixing of air in the cleanroom than inlets without diffusers and, when inlets with no diffuser were used, the air supply tended to flow out the nearest exhaust.

4 Computational Fluid Dynamic (CFD) Simulation of the Air Flow

The air flow in the non-unidirectional flow cleanroom was studied by computational fluid dynamics (CFD). The CFD analysis was used to confirm the experimental results, and to determine how useful CFD would be in predicting air flow and particle contamination in a non-unidirectional cleanroom. Because the actual velocities and direction of the air flow was available for comparison with those determined by CFD, it was also possible to make an investigation of the choice of different turbulence models, and the turbulent intensity and hydraulic diameter that should be used to describe the boundary conditions of the air inlet.

4.1 Modelling and Meshing the Cleanroom

A 3D model of the cleanroom was drawn and meshed using Gambit 2.4.6 (Ansys Inc). Meshing requires the volume of the cleanroom model to be divided by grids into small elements (cells). Tetrahedrals, hexahedrals, and wedges were used, either as a single shape or combination of shapes, and were chosen to minimise distortion of the element shapes at boundaries, and hence minimise inaccuracies and produce good convergence of the CFD solution. Refinement of the grid, by decreasing the global element size, allowed the elements to fit the geometry of the room better and reduced the discretization error. However, this increase in accuracy may be offset by the unacceptably high computational requirements, and thus a balance between the number of elements and computational requirements was sought.

A cleanroom is rectangular in shape and can therefore be successfully meshed by hexahedrals. Hexahedrals are a good choice as, when distorted, they still give a reasonable solution to the CFD analysis. However, the four-way air supply inlets required the use of wedges, as hexahedrals were topologically a poor fit. The meshing also included a boundary layer at the walls, ceiling and floor. To ensure good meshing at the table, the room model was divided into sections that allowed the meshing from each table surface to line up with the outer wall. A grid size dependency study was carried out on element sizes of 0.075m, 0.05m and 0.1m. There was practically no difference in the airflow field between an element size of 0.075m and 0.05m and, although an element size of 0.1m gave a similar overall airflow pattern, the air ‘throw’ from the diffuser was found to diffuse more quickly. An element size of 0.05 m was therefore chosen which gave a mesh of 988,736 cells (990,395 nodes) in the 4-way diffuser analysis, and 952,512 cells (986,040 nodes) in the no-diffuser inlet analysis. The meshed models were then imported into Fluent 6.3 (Ansys Inc) and analysed using three dimensional, double-precision models.

4.2 Inlet Boundary Conditions

Diffusers at the air supply inlet to the cleanroom used vanes to direct air. These vanes are likely to be about 2 mm thick and, assuming that this thickness should contain a minimum of 3 elements to obtain an acceptable mesh, the complete meshing of the diffusers would lead to an unacceptably small element size and excessive computational requirements. In the case of the 4-way diffuser, the air inlet opening was divided by diagonals and the velocity direction from each of the resultant four areas was set at 90° to each other, with a 20 degree angle between the ceiling and the exiting air. The velocity was set at 1.0 m/s. This approach caused the Fluent software to reduce the air supply volume, the reduction varying according to the angle. This was compensated by increasing the inlet area. The inlet with no diffuser gave no such problems and the velocity was set at 0.485 m/s in the vertical direction.

4.3 Choice of Turbulent Intensity, Length Scale and Turbulence Model

CFD analysis has been used to obtain air flow fields in non-unidirectional airflow cleanrooms and other controlled environments with terminal air filters and high air change rates (Bottani et al 2008; Cheong et al 2006; Rouaud et al, 2002). However, the scientific literature on this subject is limited and the best choice of turbulent intensity and hydraulic diameter to describe the boundary condition at the air supply inlet of a non-unidirectional airflow cleanroom is unclear, as is the best choice of turbulence model. These problems were investigated by comparing the experimentally-produced airflow fields with those from CFD simulations.

Turbulent Intensity

The most appropriate choice of the degree of turbulent intensity was investigated by comparing the ‘throw’ of air from both the jet and 4-way diffuser as calculated by means of CFD simulation, and as measured. The velocities of the air ‘throw’ from the diffuser were chosen for study, as this was an important influence of the airflow in the cleanroom. This study was carried out using a cleanroom with 15 air changes per hour, when the air supply came from either a 4-way diffuser with an air supply velocity of 1.0 m/s, or an inlet without a diffuser and a velocity of 0.485 m/s. The turbulent intensities that were initially studied were 20% and 1%, when the hydraulic diameter was 0.5 m, the grid element size was 0.05 m, and the standard k-epsilon turbulence model was used. In the case of the jet, the velocities were ascertained in a line from the centre of the diffuser down to the floor. In the case of the 4-way diffuser, the velocity was measured in a line 10 cm below the ceiling and between the edge of the diffuser and the wall. These velocities are shown in Figures 5 and Figure 6, along with the experimental measurements.

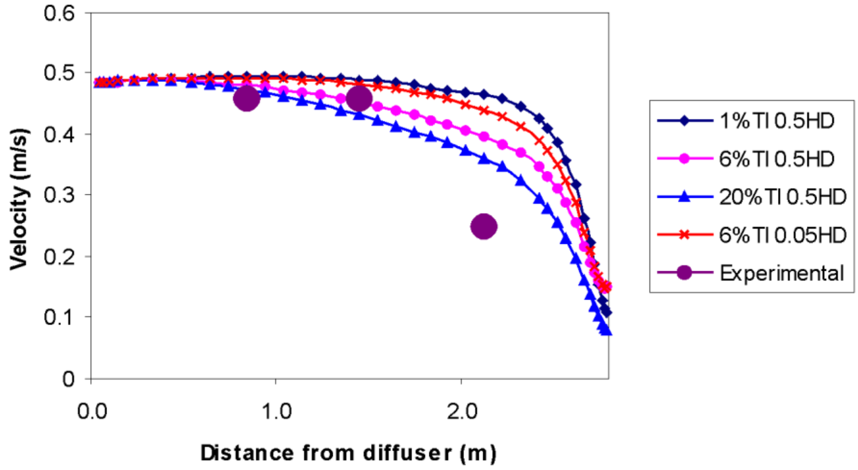

Figure 6 Effect of turbulent intensity and hydraulic diameter on the air throw from an inlet without a diffuser

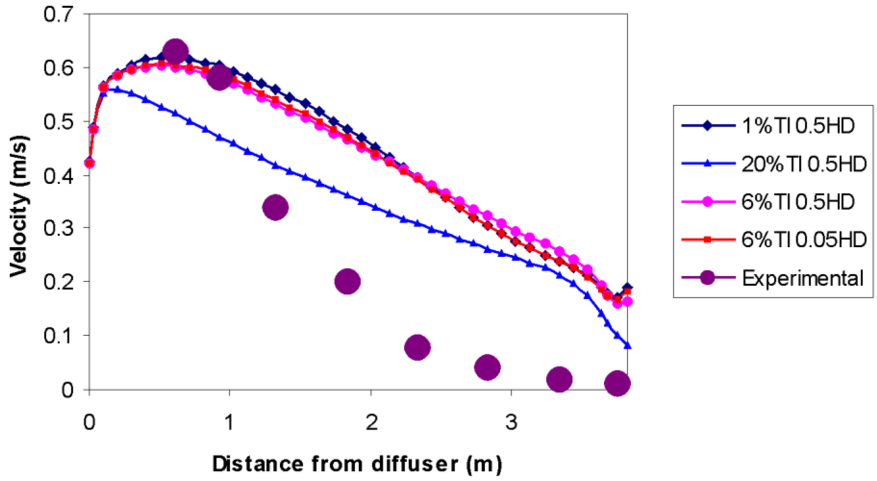

Figure 7 Effect of turbulent intensity and hydraulic diameter on the air throw from a 4-way diffuser

Angellotti et al (2003) have investigated the turbulent intensity found a short distance away from a HEPA filter face and, although they did not report the average turbulent intensity, inspection of their diagrams shows that it was about 2% to 3%, with a maximum not greater that 5%. We also measured the turbulent intensity at a HEPA filter face. This was carried out when the average filter face velocity was 0.45 m/s and using an Airflow Developments UA6 ultrasonic anemometer, but was measured against the filter face of a HEPA filter. It was found that the average turbulent intensity was in the region of about 3%, with the highest value of 8%. It can be seen in Figure 5 and 6 that as the turbulent intensity increases, the throw of the air from the jet air supply decreases and approaches closer to the experimental results. However, it was thought that a turbulent intensity should not be used in the CFD analysis that was greater than found in the practical situation, and a turbulent intensity of 6% was therefore adopted throughout the CFD analysis. The affect of a turbulence intensity of 6% on the throw of air from the inlets is shown in Figure 5 and Figure 6.

Hydraulic Diameter

It was not clear from the scientific literature if the hydraulic diameter of the air inlet in a non-unidirectional airflow cleanroom should be based on the full size of the air inlet, the distance between the diffuser vanes, or the size between the folds of the media in the HEPA filter. The effect of a hydraulic diameter of 0.5m and 0.05m was therefore studied in both the no-diffuser and 4-way diffuser inlets when the turbulent intensity was set at 6%, the grid element size at 0.05 m, and using the same cleanroom conditions as described in the previous section. It can be seen from Figure 6 that in the case of the inlet with no diffuser, a smaller hydraulic diameter (0.05 m) gave a greater air ‘throw’ from the diffuser than a hydraulic diameter of 0.5 m, and a hydraulic diameter of 0.5 m approached closer to the experimental measurements. As shown in Figure

7, there was no noticeable difference in the case of the 4-way diffuser. As the air inlet was 0.5 m x 0.5 m, it did not seem scientifically correct to increase the hydraulic diameter above this size to get closer to the experimental results, and so a hydraulic diameter of 0.5 m was adopted and used throughout the CFD analysis.

Turbulence Model

A large number of models that simulate air turbulence are available in Fluent. The most appropriate ones, which were within the available computational resources, were the standard k–epsilon model and its two variants i.e. realisable k–epsilon model and the RNG k–epsilon model. The k-omega turbulence model was also available but did not appear to be an appropriate choice because of the special attention that has to be given to the boundary layer when using this model. The appropriateness of the k-omega turbulence model was checked by simulating the airflow field produced by the 4-way diffuser and this showed that the air exiting from the diffuser passed along the ceiling at a much reduced velocity. No further investigations were therefore carried out using the k-omega turbulence model.

The turbulence models were initially run in the 'steady-state' condition, which is the normal approach to many CFD problems. However, it was found that the majority of the simulations failed to produce a convergence of the solution much below about 1×10^{-4}. A common problem that prevents an acceptable convergence is poor meshing which causes distortion of the meshing elements, but an investigation of this possibility suggested that the models studied were well meshed. The solution to the problem was indicated by the oscillations in the steady-state analysis, which can be indicative of an unsteady state problem, and this possibility was confirmed by mapping the areas of the cleanroom where residual problems were found. The problem areas connected with the inlet without a diffuser was found to be at the outside of the jet at the near side wall. In the case of the 4-way diffuser, the problem areas were where the air stream from one diffuser met another, as well as the top corner of the ceiling furthest away from the diffuser. Good convergence was obtained by changing over from 'steady-state' to 'unsteady state' analysis, and convergence criteria of less than 1×10^{-6} were then met.

The airflow fields calculated by CFD when using the three k–epsilon models were compared with each other and the experimental results. The cleanroom studied was supplied with 15 air changes per hour, and there was no temperature difference between the supply and room air. The turbulent intensity was set at 6%, the hydraulic diameter of the air inlet was 0.5 m, with the grid element size of 0.05 m. It was found in the case of the 4-way diffuser that, for all practical purposes, the air flow fields were identical in the standard, realisable, and RNG models, and close to the actual measurements. In the case of the jet of air from the inlet without a diffuser, the airflow field obtained from the standard k–epsilon model and realisable k–epsilon model gave practically identical results that were close to the experimental results. However, when the RNG model was used, a slightly different airflow field was obtained that did not match the actual measurements and the results from the other two CFD models. Because of these results, the standard k–epsilon model was selected for use in the CFD analysis, using a turbulent intensity of 6% and a hydraulic diameter of 0.5 m for the inlet boundary conditions.

5 Comparison of CFD and Experimental Results

The airflow fields obtained from CFD analysis and the experimental results were compared. This was carried out using a non-unidirectional airflow cleanroom with the two types of air inlets, an air change of 15 per hour and no temperature difference between the air supply and the room. It should be noted that the extracts are off-set from the plane investigated (see Figure 1). The airflow field found by experimental measurement is shown in Figure 3 and should be compared with the CFD simulation shown in Figure 8, and Figure 4 with Figure 9. It can be seen that the airflow obtained by the CFD simulation predicted the actual airflow reasonably well, although the throw of air from the inlets was greater than the actual results. It has been reported (Rouaud et al 2002) that the standard k–epsilon model predicts more diffusion of the air stream than measured experimentally and this is contrary to that found in our study. However, a grid dependency study of our model showed that the element size used (0.1 m) predicted less diffusion that the larger grid sizes associated with studies such as Rouaud et al 2002 and this may explain at least part of the difference.

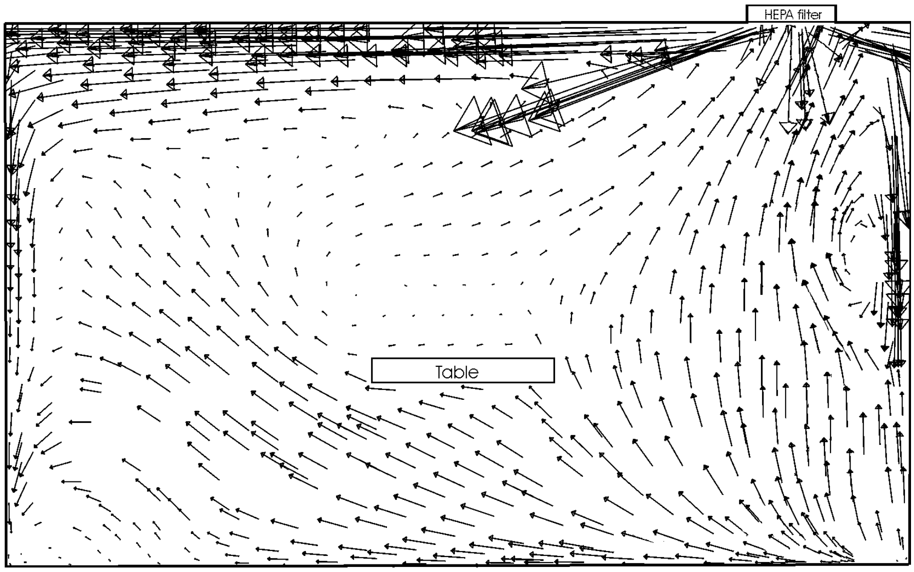

Figure 8 CFD simulation of the air flow field in a cleanroom with 15 air changes per hour and a 4-way diffuser

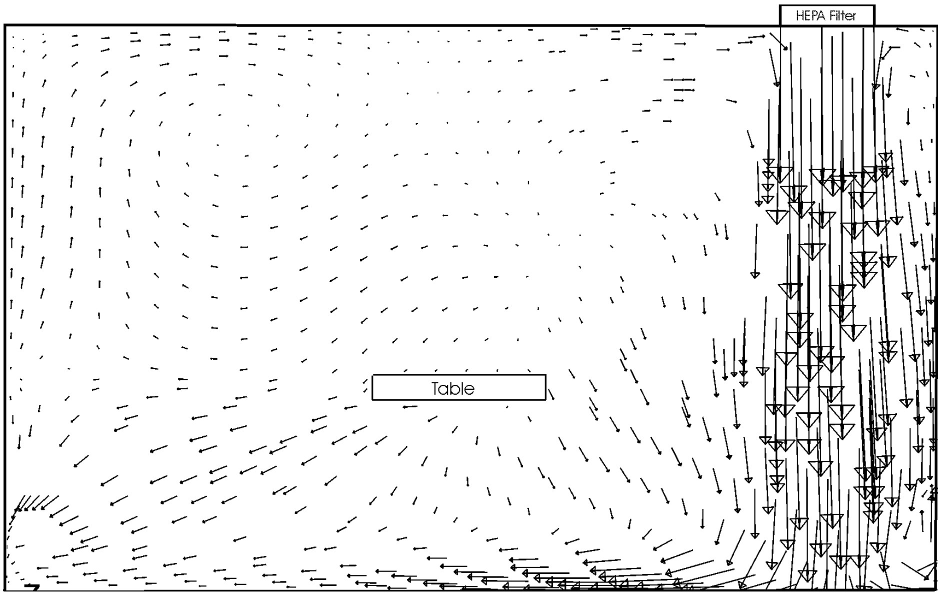

Figure 9 CFD simulation of the air flow in a cleanroom with 15 air changes per hour and an air jet from supply inlet

6 Discussion and Conclusions

The overall average airborne cleanliness of a non-unidirectional cleanroom is determined by the volume of air supplied to it, but the local cleanliness is determined by the air flow within the room. Experimental investigations were carried out into the influence of the following variables on particle concentrations: type of air supply inlets, air inlet velocity, temperature difference between the air supply and the room, and the position where airborne contamination is released. In addition, the actual airflow field within the cleanroom was measured experimentally, as well as determined by CFD analysis. Most of the studies were carried out on a cleanroom with an air supply of 15 air changes per hour that had a supply filter face velocity of 0.485 m/s. This air supply condition was chosen as HEPA filters installed in cleanrooms are designed to supply air at a velocity of 0.45 m/s, and therefore the airflow pattern in the cleanroom that was studied is likely to be close to that found in many cleanrooms.

It was found that the main variable that influenced air flow and local particle concentrations within a non-unidirectional cleanroom is the method of air supply. When the inlet had no diffuser, the air supply gave a pronounced downward jet air flow to the floor, which reduced the penetration of particles into the air stream, and the cleanest conditions in the cleanroom were found below this inlet, but poorer conditions in much of the rest of the cleanroom. A 4-way diffuser gave much better mixing of the air throughout the cleanroom and hence a more even concentration of particle contamination. Of lesser influence on the air flow was the air velocity at the air inlet, the temperature difference between the supply and room air, and the place where contamination is released.

The overall conclusions deduced from this study were that the air inlet to a non-unidirectional cleanroom should not contain a diffuser if a clean zone is required under the air inlet. If mixing of the air and similar particle concentrations throughout the cleanroom is more desirable, then a 4-way diffuser (or diffuser with similar air mixing properties) should be used. If no diffuser is used, then the increase in cleanliness below the diffuser will be associated with a decrease in the cleanliness in the rest of the room. When a cleanroom is tested according to ISO 14644-1 and particle concentrations are measured at positions spread about the room, poor air mixing could lead to airborne concentrations at one or more of the test positions exceeding the limits set. If production equipment is not placed under the air inlet, or moved away from its original position, then production may end up being carried out in a dirtier than average area. It is suggested that a 4-way diffuser (or diffuser with a similar air mixing efficiency) is fitted to an air supply inlet and if lower levels of airborne particles, or microbe-carrying particles, are required at critical areas, then an enhanced air device, such as a unidirectional airflow workstation, should be used.

The airflow in the cleanroom was measured and compared with that obtained by CFD analysis. This allowed a study of the best choice of turbulent intensity and hydraulic diameter for specifying the boundary conditions of the air inlet, as well as the choice of turbulence model. It was concluded that a turbulent intensity of 6% and a hydraulic diameter derived from the full size of the air inlet gave the closest results to the actual airflow field, while still remaining within a practical range of values. The k–epsilon standard turbulence model was considered to be a good choice in our application. The CFD analysis gave a good prediction of the airflow field, although the ‘throw’ of the air from both designs of air inlets was greater than observed. It was thought that the CFD analysis could be used with reasonable confidence to predict the airflow in a non-unidirectional cleanroom and to ascertain the likely concentrations of airborne particles and microbe-carrying particles throughout a cleanroom.

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