The measurement of air supply volumes and velocities in cleanrooms

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Abstract

Air supply volumes and velocities in cleanrooms are monitored by airflow measuring hoods and anemometers but these measuring methods can be inaccurate if used incorrectly. It is demonstrated in this article that measuring hoods are accurate if the air supply passes evenly out of the hood, as occurs when the air volume is measured from a four-way diffuser or no air supply diffuser. However, when a swirl diffuser was investigated, the measuring hood gave readings more than 50% greater than the true volume. The reasons for the inaccuracy, and methods to correct it were established. Vane anemometers give inaccurate readings at the face of high-efficiency air supply filters, and it was found that the most accurate reading was found about 15 cm from the filter face. The number of readings required across the filter face to obtain an accurate average velocity was investigated, as was a scanning method using overlapping passes.

Keywords

Air supply volumes, air velocities, cleanrooms, anemometer, airflow measuring hood

1. Introduction

Cleanrooms minimise the contamination of products made in manufacturing industries, as well as bacterial infection of patients in hospitals. The most common design of cleanrooms is known as 'non-unidirectional airflow'. It has a ventilation system similar to that found in hotels, offices etc. but the final air filters are in the air supply terminals in the ceiling, with a particle-removal efficiency against the most penetrating particles size (usually about 0.3µm) that is usually greater than 99.99%. The air supply volume is much higher than normal rooms, and in the range of about 20 to 100 air changes per hour. The second type of cleanroom is known as 'unidirectional airflow', and particle-free air is supplied from a complete filter ceiling and moves through the room in a piston-like manner at a velocity of about 0.45/s, and exits through a perforated floor. A fuller description of cleanrooms, and how they are tested and operated is given by Whyte (2009).

The cleanliness of non-unidirectional airflow cleanrooms is directly related to the air supply volume, and the unidirectional airflow cleanrooms to the air supply velocity (Whyte 2009). The air supply volume and velocities should be monitored throughout the life of the cleanroom, and monitoring intervals are suggested in ISO 14644-2 (2000). The air volume supplied to a cleanroom can be accurately measured in the air supply ducts using a Pitot static tube but this method is normally only used during initial commissioning and retesting, and monitoring is usually carried out using an airflow measuring hood and an anemometer. The reason is the extra time required to obtain Pitot static readings, the poor access to the air ducts associated with the very large air conditioning systems, and the different expertise of the engineers who commission and balance cleanroom ventilation systems, compared to those that routinely test them.

To measure air volumes, an airflow measuring hood of the type shown in Figure 1 is used to gather the air supplied through a ceiling terminal and the volume is measured as it exits the hood.

Figure 1 Airflow measuring hood used on a diffuser in a cleanroom

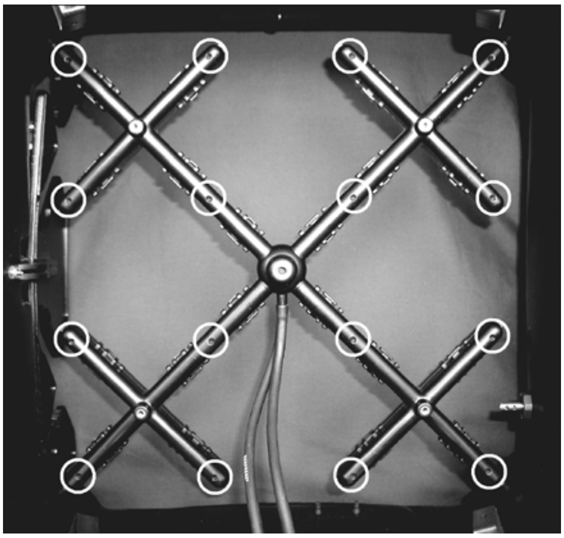

Figure 2 Measuring grid at the exit of an airflow measuring hood with the measuring points circled

There is a measuring grid at the exit of the hood. Typically, sixteen connected holes face the airflow and measure the average total pressure (Figure 2), and sixteen holes at the same position, but facing away from the airflow, measure the average static pressure. The pressure difference between the total and static pressures is the velocity pressure and, with the use of an appropriate correction factor, the velocity is calculated. Knowing the exit area of the hood, the air volume is determined.

An anemometer is used to measure air velocity and there are two types generally used in cleanrooms. The vane anemometer is the most popular type, having a set of vanes of about 10cm diameter that revolve at a speed dependant on air velocity. The second type is a thermal anemometer which measures velocity using the electrical resistance of a heated thermistor (about 2mm in diameter) that is cooled by the air flow. Knowing the air velocity and the area of the air supply filter, the air supply volume can be calculated.

It has been reported that both the measurement of air supply volume and air velocities in a cleanroom may give inaccurate readings (Anonymous, 2006). As these quantities are of prime importance in determining and maintaining the cleanliness of cleanrooms, the reasons were investigated.

2. Apparatus and instruments used

2.1 Low velocity wind tunnel

A simple and fundamental method was used to accurately measure the air velocities and volumes required for the experiments. A low-velocity wind tunnel was built of clear plastic sheet with inside dimensions of 0.55m x 0.55m and a variable speed fan-filter unit at the front, which drew air through an air straightening membrane and into the tunnel. The front of the fan-filter unit contained a high efficiency particulate air (HEPA) filter with an 'active' filter face area of 0.54m x 0.54m i.e. 0.29m^2. The 'active' area of a high efficiency filter is considered to be the area of the filter media where the air passes through, and does not include the metal frame. The air filter was protected by a grille.

A puff of smoke was introduced at the intake to the tunnel and, after one metre, the time it took to move a further 3 metres along the tunnel was timed using a stop watch, and the velocity calculated. The velocity was measured 10 times and a median average obtained. By varying the fan speed, an exact air supply volume, or filter face velocity, could be obtained.

Different types of diffusers could be fitted to the front face of the fan-filter unit and, additionally, an airflow measuring hood mounted to measure the air volume and investigate the air movement within the hood. In addition, the front of the fan filter unit could be extended by using a tunnel one metre, or 15cm, long. This was done when it was necessary to prevent the air moving sideways during investigations of airflow from the filter.

2.2 Air diffusers investigated

The air supply to a non-unidirectional airflow cleanroom can be either supplied directly from a HEPA filter housed in the ceiling i.e. no diffuser is used, or through a diffuser. The following two diffusers are used in cleanrooms and were investigated in these experiments.

1. A 4-way diffuser of the type shown in Figure 3. This was a 600mm square Trox Technik Type FD. It throws the air sideways, in four directions, this air entraining and mixing with the cleanroom air.
2. A swirl diffuser of the type shown in Figure 4. This was a 600mm square Trox Technik Type DLQ. This type of diffuser twists the supply air and mixes it with cleanroom air.

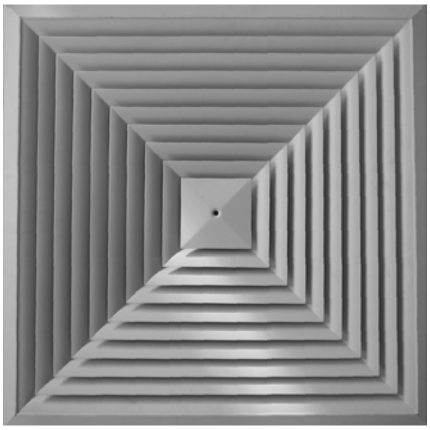

Figure 3 4-way air diffuser

Figure 4 Swirl air diffuser with ribbons to show air exiting

The angle at which the diffuser vanes are set to the horizontal, influences the airflow in the hood. The vanes in the 4-way diffuser appeared to be set at a 40° angle to the ceiling but were curved and exit angle was less than 40°. The actual angle was measured experimentally and found to be 20°. The vanes of the swirl diffuser were straight and the angle was 20°. The surface area through which air passed through the diffuser was less that the total surface area, and when the filter face velocity was 0.45m/s, the velocity through the vanes of the 4-way diffuser was 0.58m/s and through the swirl diffuser it was 3.1m/s.

2.3 Measuring hood and anemometers

The measuring hood used in the experiments was typical of those available and was a TSI Model 8375. The vane anemometer was an Airflow LCA 501 and the thermal anemometer was an Airflow Developments Model TA2. Both anemometers were calibrated using the low-velocity wind tunnel.

2.4 Computational fluid dynamics

The air flowing from a air supply filter and diffuser and passing through the measuring hood, was investigated by means of Computational Fluid Dynamics (CFD) using Fluent software (version 6.3, obtained from ANSYS, Inc.). This was carried out to confirm the experimental results and to obtain good visual representation of the airflow within the hood. To assist in the achievement of the latter requirement, Techplot 360 software from Techplot, Inc. was additionally used. To improve the efficiency of the computation, diffuser blades were not included but an area, velocity and exit angle was used that gave an equivalent airflow. The airflow was turbulent and modelled using the basic k-epsilon turbulence model.

Shown in Figure 5 are air streams coming from a HEPA air filter with no diffuser fitted and passing through a measuring hood. The velocity is shown as a magnitude. It may be seen that the air flows evenly from the filter to the exit of the hood.

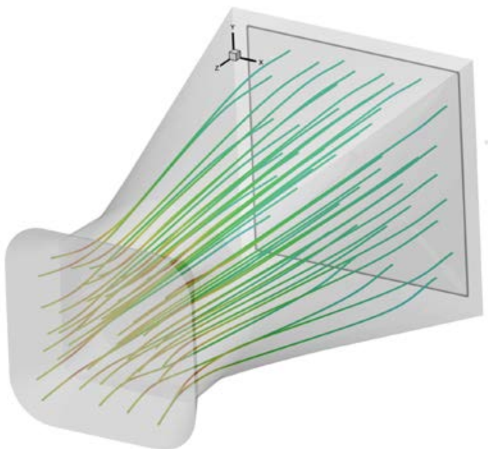

Figure 5 Air flow through a hood coming from a HEPA filter with no diffuser

The airflow obtained from a 4-way diffuser is shown in Figure 6. The air exits from the diffuser and flows to the outside of the hood where there is some vortexing. There was also some vortexing in front of the solid square surface at the centre of the diffuser. It had been anticipated that the air velocity at the exit of the hood would have been uneven but, as seen in Figure 6, this is not so.

Shown in Figure 7 is the airflow in a measuring hood when the air is supplied through a swirl diffuser. It can be seen that the air 'swirls' round the hood. A front view of the exit of the hood is given in Figure 8. As the hood's measuring grid measures velocity in the Z-axis, the magnitude of the velocities in Figure 8 is given in the direction of the Z-axis. The darker the colour, and the longer the lines, the greater is the velocity, and a high velocity is shown round the outside.

A comparison was made between the velocities at the exit of the swirl diffuser, as found from experimental measurements and calculated by CFD modelling. The experimental methods are discussed later in this research paper, where it is reported that the average velocity from the outer 12 measuring points was 2.69m/s and the inner 4 points was minus 0.29m/s. CFD modelling of the flow in the hood, when a swirl diffuser was used, gave an average velocity of 2.44m/s, and minus 0.6m/s for the outer 12 and inner 4 points, respectively. These CFD modelling results, although not exactly the same as the experimental results, are consider close enough to suggest that the airflow shown in Figures 5 to 8 is likely to give a good representation of the true airflow.

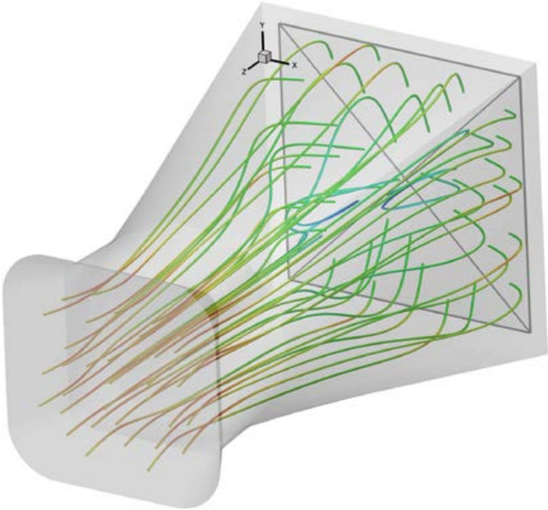

Figure 6   Airflow through a hood coming from a 4-way diffuser

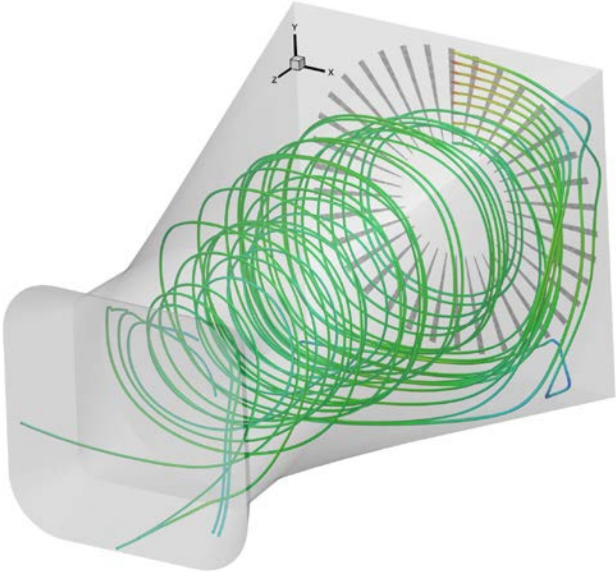

Figure 7   Airflow through a hood coming from a swirl diffuser

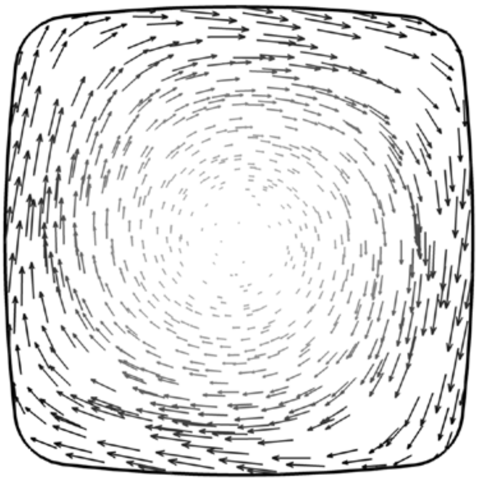

Figure 8 Airflow exiting a hood supplied by a swirl diffuser

3. Experimental measurement of airflow in the measuring hood

3.1 Experimental accuracy

There were two possible inaccuracies that could be introduced into the experimental measurements of the airflow measurement hood. These were, firstly, when the velocity was measured by means of smoke in the low velocity wind tunnel and, secondly, when the volume was measured by the hood. To measure these inaccuracies, the tunnel was set up with a nominal velocity of 0.45 m/s and the velocity measured by the use of smoke ten times. This experiment was repeated 5 times and the variation of the velocity measurements in the tunnel was found to be ± 3.9%. By similar means, the variation in the hood measurement was found to be ±1.2%.

3.2 Calibration of the measuring hood with respect to air supply volume

The air velocity at the filter face of the fan/filter unit, when no diffuser was fitted, was set at nominal velocities of 0.25m/s, 0.45m/s and 0.65m/s by means of smoke measurement. The true air supply volumes coming from the filter face were compared with those measured by the hood. The differences between the two volumes were no greater than 5%.

3.3 Investigation of hood measurements with respect to diffuser design

The air supply volumes measured by the hood when using different diffusers were investigated. The velocity of the supply filter face was adjusted to a nominal velocity of 0.45m/s by measuring smoke in the tunnel, and the true air volume passing through it was therefore 474m^3/hour. The reading of volume from the hood was obtained, and the percentage difference between that reading and the true values are given in Table 1. It may be seen that the hood readings and true readings from no-diffuser and a 4-way diffuser differed by -3.9% and -1.0%, respectively. However, the hood readings obtained from the swirl diffuser were 56% higher than the true air volume.

Table 1 Difference between true and measure volumes with respect to diffuser type

Diffuser type	Difference between true and measured hood volume (%)
None	-3.9
4-way	-1.0
Swirl	+56

3.4 Effect of diffusers on velocity distribution at the hood exit

The placement of the 16 measuring points at the exit of the hood is shown in Figure 2. A preliminary experiment was carried out in which the velocity was measured at the hood exit using a thermal anemometer and it was found that when no-diffuser, or a 4-way diffuser was used, the velocity was fairly uniform across the exit. However, when a swirl diffuser was used, much of the air exited round the outside and at the centre it moved backwards. This flow was the same as that predicted by CFD modelling.

It was difficult to accurately measure velocity and direction using an anemometer at the hood measuring points, so an alternative and better experimental method was devised. The volume of air passing through the hood was adjusted to about 474m^3/hr, which was equivalent to a filter face velocity of 0.45m/s. The average air volume and velocity from the measuring hood was then found in three different situations. Firstly, measurements were taken of the normal condition i.e. using all 16 holes. Secondly, the twelve measuring holes in the outer area of the measuring matrix were closed off, and only the measurements from the four centre points were used. Finally, the four centre points were closed and measurements obtained from the outer 12 points. Ten readings were obtained for each condition, and averaged. This was repeated 5 times and the averages given in Table 2.

Table 2 Volume and velocity measurements at the exit of the hood

	Normal 16 points		**Outside 12 points.**		**Inside 4 points**	
	Vol (m^3/hr)	Vel (m/s)	Vol (m^3/hr)	Vel (m/s)	Vol (m^3/hr)	Vel (m/s)
No diffuser	476	1.51	476	1.53	504	1.58
4-way diffuser	478	1.53	491	1.55	486	1.55
Swirl diffuser	738	2.23	838	2.69	-118	-0.29

The air volumes measured by the hood, when supplied from the swirl diffuser and using the 16 normal measuring points, gave readings almost identical to the experimental results discussed in the previous section, and were 55% higher than the readings obtained from both the no-diffuser and a 4-way diffuser. The reason for this increased reading can be explained by the results shown in Table 2 where it can be seen that the outer 12 points of the swirl diffuser gave a substantially higher velocity, which when averaged with the negative results from the inner 4 points, would give a greater reading of volume than the 4-way or no-diffuser.

3.5 Effect of correction mechanisms applied to hood

It was clear from the results discussed in the previous sections of this research paper that when the air velocity at the hood exit was relatively even across the measuring grid, the air volumes were likely to be correct. This was the situation with no-diffuser and a 4-way diffuser, but not with the swirl diffuser. However, an uneven flow caused by the swirl diffuser, and other types of diffusers that cause uneven airflow, might be made even by using the following two possibilities:

1. The airflow can be straightened. Lengths of tubes and honeycombing are often used for this purpose (ISO 7194: 2008). However, the most practical method seemed to be either one partition across the hood, or two partitions across the hood at ninety degrees to each other, as shown in Figure 9. These partitions were placed along the full length of the fabric in the hood. They were constructed from cardboard but, for permanent use, the same impervious fabric as used in the hood could be sown into it to allow easy packing and transportation.

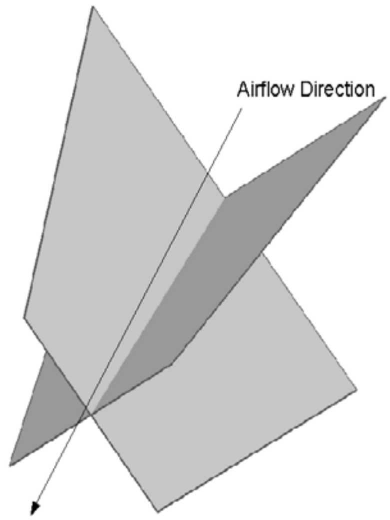

Figure 9 Straightening mechanism

2. A permeable membrane across the intake plane of the hood that would give sufficient pressure drop to even the airflow but not cause a reduction in air flow. Two materials were investigated. These were both light-weight net curtain material, similar to mosquito netting, and used to give sight reduction through a window but allow good light penetration. Two types were used: one was known as the 'fine' mesh and had a thread count of about 12/cm x 32/cm and a pore diameter between the threads of about 800µm x 300µm. The other was call the 'finest' mesh and had a thread count of about 27/cm x 27/cm and a pore diameter of 220/cm x 220/cm.

The filter face velocity was set at 0.45m/s and a comparison made between the volume measured by the hood, and the actual volume passing through the tunnel. Different straightners and permeable membranes were investigated with the swirl diffuser and Table 3 gives the hood measurement values as a percentage of the true values. It may be seen that both straightening mechanisms gave a noticeable benefit, with the partitions giving more benefit.

Table 3 Effect of correction mechanisms on the accuracy of the hood measurements when a swirl diffuser was used

Correction mechanism	Difference between measured and true values (%)
None	56
Fine membrane	16.1
Finest membrane	6.8
Single partition	5.0
Double partition	-3.7
Double partition plus finest membrane	2.4

3.6 Pressure drops caused by correction mechanisms

The mechanisms used to correct erroneous measurements from hoods would only be of practical use if they do not cause a back pressure that reduced the air volume passing through the hood. The pressure drop caused by the use of the fine, and finest, membrane material was measured in the low-speed wind tunnel by means of a static probe. The material was placed across the entrance to the tunnel, which was almost the same area as the hood intake. The filter face velocity at the exit was adjusted to 0.45m/s and the pressure differential measured across the material. The pressure drop was 1.1Pa for the fine material, and 2.2Pa for the finest material. This is a very small percentage of the total losses in an air conditioning plant and associated ductwork used in cleanrooms, where a pressure drop of several hundred Pascals is common, and the HEPA filters alone would contribute between 100 to 150Pa.

No measurable pressure drop was found when using single and double partitions, and a drop in air volume was therefore measured. The volume passing through the tunnel was set at 474m^3/hr, the hood attached to the exit, and the air volume measured by the smoke method. The correction method was then added to the hood and the air volume again measured. None of the correction mechanisms showed a drop in air volume greater than 5%.

4. Measurement of air velocities and air supply volumes by anemometers

When the average air velocity of the air coming from a filter is multiplied by the area of the filter face to calculate the air supply volume, the result can be incorrect (Anonymous, 2006). As the measurement of the area of the filter face is unlikely to be incorrect, it must be assumed that the velocity measurements are the problem. The possible reasons for the problem were investigated by carrying out the following experiments.

4.1 Air velocities with respect to distance from filter face

Air velocities were measured with a vane anemometer at various distances from the filter face. To ensure that a drop in velocity was not caused by the airflow from the filter flowing sideways, a one metre long tunnel was added to the filter face of the fan/filter unit to constrain the airflow. The air velocity at the filter face was set at 0.45m/s and measured at zero distance from the filter's protective grille and at 1 mm distances up to 20cm, and then at 10cm intervals. Measurements were carried out at four points evenly distributed over the filter face and the overall average velocity calculated. This was done for each distance away from the filter face, and the results up to 50cm are shown in Figure 10. After 50cm, the velocity remained relatively constant, with some small variations in magnitude of the type shown in Figure 10 for readings above 15cm. The drop in velocity at 10cm and 15cm from the filter grid, compared to the velocity at the filter face, was found to be 21% and 25%, respectively, and the velocity had levelled out at 15cm.

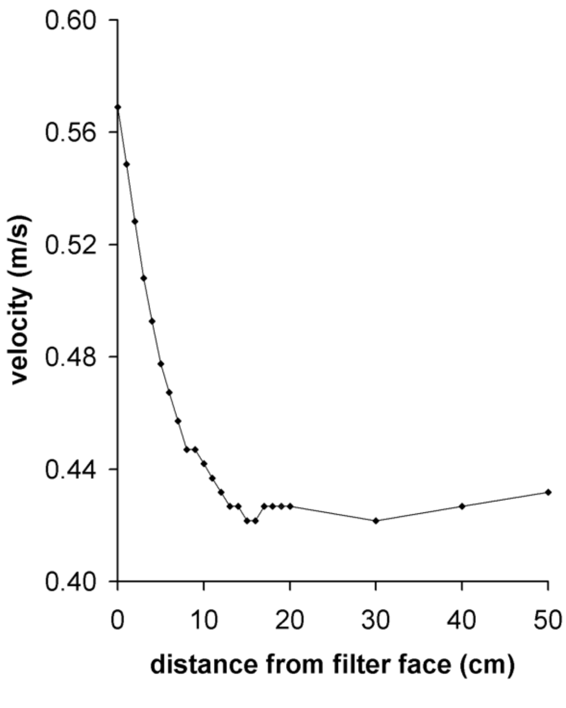

Figure 10 Air velocity with respect to distance from filter face

4.2 Air velocity distribution across a filter face

To obtain information on air velocities across the filter face, a small area was measured using the thermal anemometer. The velocity was measured at 1mm intervals in a horizontal line across the filter, the line passing over areas where the air velocity was likely to be at its highest and lowest because of both the filter and protective grille construction. These velocities are shown in Figure 11. The average velocity was 0.51m/s and the maximum and minimum velocities were 0.86m/s and 0.31m/s, respectively.

Also shown in Figure 11 is the velocity measured along the same line but 15 cm away from the filter face. It can be see that the velocity was less variable with an average, maximum and minimum of 0.45m/s, 0.50m/s, and 0.40m/s, respectively.

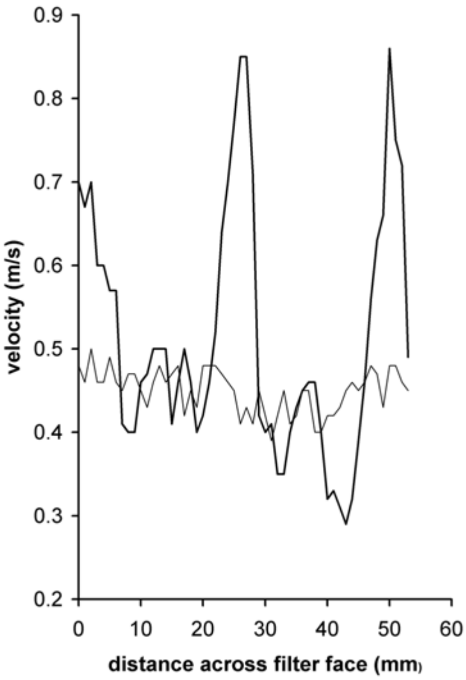

Figure 11 Air velocity with respect to distance across the filter face. Thick line shows velocities at the filter face. Thin line shows velocities 15 cm away from filter face.

4.3 Methods of obtaining the average velocity at the filter face

To ascertain how to correctly obtain the average value of the air velocity coming from a filter, two fan-filter units from different manufacturers were investigated. To constrain the air flowing from the filter, a 15cm tunnel was added to the filter end of the fan/filter unit. The tunnel opening, which was the same area as the filter (540mm x 540mm), was divided into 16 equal areas by means of fine nylon threads strung across the face. The velocities were then measured 15cm from the filter face and in the centre of 16 areas, and then at 8, 4 and 2 equal areas, as well as in the middle of the filter.

The vane anemometer used in these experiments (Airflow LCA 501) took over 40s for the velocity reading to build up from zero to the correct one. Therefore, to obtain an accurate reading, the velocity was allowed to build up to a maximum, and readings that were averaged over 5s were consecutively taken, until a constant reading was obtained.

Table 4 Average velocity from a filter obtained from different numbers of measurements

	Number of measurements				
	16	8	4	2	1
Fan filter unit A					
Average velocities (m/s)	0.48	0.46	0.45	0.45	0.43
% difference	-	-4.2	-6.3	-6.3	-10.4
Fan filter unit B					
Average velocities (m/s)	0.48	0.47	0.47	0.46	0.40
% difference	-	-2.1	-2.1	-4.2	-16.7
Average difference from the two fan/filter units	-	3.2	4.2	5.3	13.6

It is clear that the more velocity measurements taken across the filter face the closer the average result will approach to the true result. Sixteen measurements would be considered an excessive and impracticable number if used in cleanrooms to test a filter with a nominal area of 600mm x 600mm. However, sampling 16 points with a vane anemometer will sample a total area of about 0.13m^2, which is about half of the total filter area. It was therefore assumed that an average of 16 points gave a good estimate of the true average velocity. The average velocity from 8, 4, 2, and 1 point was then calculated as a percentage of the average velocity obtained from 16 points, and shown in Table 4. It may be seen that if only one measurement is taken at the centre of the filter, the average velocity was underestimated in one fan-filter unit by 10% and in the other fan-filter unit by 17%. Inspection of the results in Table 4 shows that two velocity measurements would give a result a little above 5% of the result obtained from 16 measurements, and 4 measurements would give an average velocity a little less than 5% of that obtained from 16 measurements.

An alternative method of obtaining the average velocity of the air coming from a filter face is to scan the whole area of the filter face over a given time interval. This method is not used when testing a cleanroom but in view of the difficulty demonstrated in this article to obtain an average velocity, it seems a possible alternative. The vane anemometer used in these experiments had a 'time constant' facility that allowed it to average the readings over a selected time period of 5s, 10s, 20s or 30s. To obtain an average velocity, the anemometer was placed 15cm from the filter face and the velocity allowed to rise to a constant value; this could take 40 to 50 seconds. The filter face was then scanned backwards and forwards in slow parallel sweeps in a manner that would sample the complete filter face as adequately as could be done within the time available. 5s, 10s, 20s, 30s and 60s time intervals were investigated and it was found that a time interval below 30 seconds gave results that were inconsistent and greater than +/- 5% of the true result, and a 30s scan was required to give repeatable and accurate readings (<5% of the average value) from a filter of a nominal size of 600mm x 600mm.

5. Discussion and Conclusions

The routine measurement of the air supply volume and velocities in a cleanroom is usually carried out using airflow measuring hoods and anemometers. However, it has been reported that these measurements may be unreliable, the reasons being uncertain (Anonymous, 2006).

A typical airflow measuring hood was studied and was found to accurately measure the air supply volume from an air terminal without a diffuser, and a 4-way diffuser. However, when used to measure the air supplied from a swirl diffuser the measurement was 55% greater than the true volume. This inaccuracy was caused by the air swirling round the hood and, at the exit where the measuring points were placed, the velocity round the outside was very much greater than the centre area. Twelve of the sixteen measuring points were positioned round this outside area, and hence the volume measured was much higher than the actual volume. This problem was corrected by use of partitions in the hood to prevent swirling, or a permeable membrane at the entrance to the hood to even out the airflow. The back pressure caused by such correction mechanisms was shown to be small and unlikely to influence the air volume measurements. It is suggested that such correction mechanisms should be used in airflow measuring hoods to prevent inaccurate readings caused by swirling or uneven flow from diffusers.

The air supply volume from a ceiling diffuser can be measured by removing the diffuser and obtaining the average velocity of the supply filter, and multiplying that result with the 'active' area of the filter face. The velocity of the air coming from the filter face of unidirectional airflow benches or enclosures, isolators, and mini-environments is also routinely measured in a similar manner to ensure that they supply the correct air velocity. The accuracy of measurement of the average velocity of the air coming from a high efficiency filter in such situations was investigated by consideration of (1) uneven velocity distribution across the filter face (2) type of anemometer used and (3) measuring distance from the filter face.

Within the filter manufacturing industry, a filter is considered acceptable for use in a cleanroom if the velocity distribution across a filter face is less than +/- 20% of the average velocity. This variation is likely to cause difficulties in obtaining an accurate average velocity if insufficient velocity readings are taken across the filter face, but it is unclear how many measurements are needed. ISO 14644-3 (2005) suggests a single reading in the middle of the filter. This will be sufficient if the airflow is even across the filter face, but in two fan-filter units investigated the average value was underestimated by 10% and 17% compared to that obtained from 16 measurements distributed evenly across the filter face. Our experiments suggest that to obtain a result within about 5% of the correct result, for a filter of a nominal size of 600mm x 600mm, the filter face should be divided into at least two equal areas, and preferably four, and velocity readings taken at the centre of these areas, and averaged. A 1200mm x 600mm filter would need twice the number of readings. Care has to be taken to ensure the anemometer velocity has reached a maximum before velocity readings are taken and the readings should be taken 15cm from filter face. An alternative method is to place the vane anemometer 15cm from the filter face, wait until the velocity reading has reached a constant value and then scan over the whole filter face in slow parallel passes. A minimum time of 30s should be used for scanning a 600mm x 600mm filter, and 60s for a 1200mm x 600mm filter.

A single reading of velocity obtained by a thermal anemometer samples about 5mm^2 of the filter face compared to about 8000mm^2 when a vane anemometer is used. As the air velocity across a high efficiency filter face is uneven, a vane anemometer should be a better choice than a thermal anemometer for ascertaining the correct average velocity of air coming from a filter.

When a vane anemometer was placed up against the grill on the filter face, the velocity reading was found to be about 25% greater than the true velocity, and the velocity dropped off as the distance away from the filter increased, until a constant and correct velocity was reached. It is assumed that this is caused by the greater air turbulence at the filter face. It has been industrial cleanroom practice for many years to obtain the correct velocity by measuring 10cm from the filter face. ISO 14644-3 (2005) recommends a distance of 15cm to 30cm from the filter face. Our experiments suggest that 15cm is the best choice.

Acknowledgements

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