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A New Experimental Method for Measuring and Visualising Air Flow in Large Food Plants

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ABSTRACT

A new experimental method was developed to visualise air flow patterns and spatial distribution of the mean air velocity in large food plants. The continuous motion of an anemometer measuring air velocity at regular intervals made it possible to rapidly obtain measurements at several thousands of points. The processing of these measurements by Fourier's series and a low-pass filter eliminated the time-fluctuations due to air flow unsteadiness and only the spatial variations of the mean air velocity were kept afterwards. This paper shows how this method was applied in the food industry to a typical meat chiller. The graphical representation of the mean air velocity distribution brought to light dysfunction inside the chiller. The experimental tool presented here can help specialists to carry out diagnoses of the air flow in food plants.

NOMENCLATURE

a_k, b_0, b_k	Coefficients of the development in Fourier's series ($\simeq$m s^{-1})
L	Whole distance covered by the probe (m)
MN	Normed modulus (–)
t	Time (s)
T	Period of the $f(x)$ function (s)
T_p	Temperature of the product (°C)
v	Air velocity (m s^{-1})
v_p	Velocity of the probe (m s^{-1})
X	Width in the plant (cm)
Y	Length in the plant (cm)

Y_i	ith distance from the starting point of the anemometer to the measurement point (m)
Z	Height in the plant (cm)

INTRODUCTION

Whatever the field of activity, controlling the air flow is a critical point in large plants. For instance, in the field of air conditioning, the inside air quality is directly related to the efficiency of the air conditioning system. In the automotive industry, the good functioning of a vehicle depends on its aerodynamics. In the food industry, air flow patterns determine both the efficiency and the homogeneity of the treatments to which products are subjected. Wooton (1986) studied the relative effect of various parameters upon the deep leg temperature in beef carcasses stored in a typical industrial batch chiller equipped with four double-fan refrigerating units suspended from the ceiling. The chilling room was filled with 242 carcasses; the weight, fatness and grade of which varied within the following ranges: $170 < \text{weight} < 420$ kg, $2 < \text{fatness} < 5$ on a 1–5 scale and $2 < \text{grade} < 5$ on a 1–5 scale. The temperature reached 24 h after slaughter varied from 7°C to 18°C. A multivariate analysis showed that carcass location in the room is as important as carcass weight. These two parameters, which explained more than 75% of the variation, are much more important than all the other factors tested (fatness, grade and time of entry into the room). This result clearly illustrates the influence of uneven air distribution on plant efficiency. That is the reason why we developed an experimental method for accurately measuring and visualising air flow in large plants. It is all the more important since, over the last few years, Computational Fluid Dynamics (CFD) techniques which originate from the automotive, aerospace and nuclear industry, have been applied to food processing (Scott & Richardson, 1997). This numerical tool, designed for solving the equations governing the flow of fluids (Navier–Stokes' equations, energy, etc.) within a given flow geometry, makes it possible to obtain, for example, the pressure and velocity distribution within the calculation domain. The experimental method can be used to validate these CFD calculations by comparing the air flow patterns calculated and measured at several thousands of points.

In apparatus used in the food industry, air flows are particularly turbulent and unsteady. Daudin and Kondjoyan (1991) have reported turbulence rates that were equal to 38% on average in an industrial beef carcass chiller and to 25% in a sausage drier, with strong spatial variations. Bouton (1992) has measured air velocities ranging from 0·5 to 15 m s^{-1} at the level of products in an industrial pork offal chiller. The higher velocities coincided with the passage of the measurement system in the vicinity of fans. Due to the strong variability of air velocity in both space and time, it is necessary to measure the mean air velocity at numerous points which is a time-consuming operation. This is incompatible with the accurate representation of air flow patterns.

There is no efficient and rapid method in the literature to describe the air flow in large plants. The Laser-Doppler anemometry technique (Mizier, 1991), which is capable of simultaneously measuring the magnitude and direction of the velocity vector, is very expensive, quite complex and above all unsuitable for large food plants. Standard techniques based on the study of pictures representing the trajectory of particles injected beforehand into the flow, such as smoke, bubbles and metallic dust, provide an approximate representation of the main flow structure (Ruegg *et al.*, 1994; Maghirand & Manbeck, 1993) and using these techniques in a 3D turbulent flow has proved to be tedious. Moreover, the standard average procedure that consists in averaging the air velocity measurements performed at each point of one plant over a period of a few minutes is a very time-consuming process. Using this method, Falconer (1993) studied the air velocity distribution in two cheese stores of identical dimensions but with different air flow rates. He represented the distribution of the mean air velocities measured on 3D graphs (Fig. 1). However, the experimental study included few measurement points, probably because the standard average procedure was tedious. The author could not draw an exact conclusion from his work concerning the whole functioning of the stores studied, except that the velocities were globally and on average higher in the apparatus where the air flow rate was stronger.

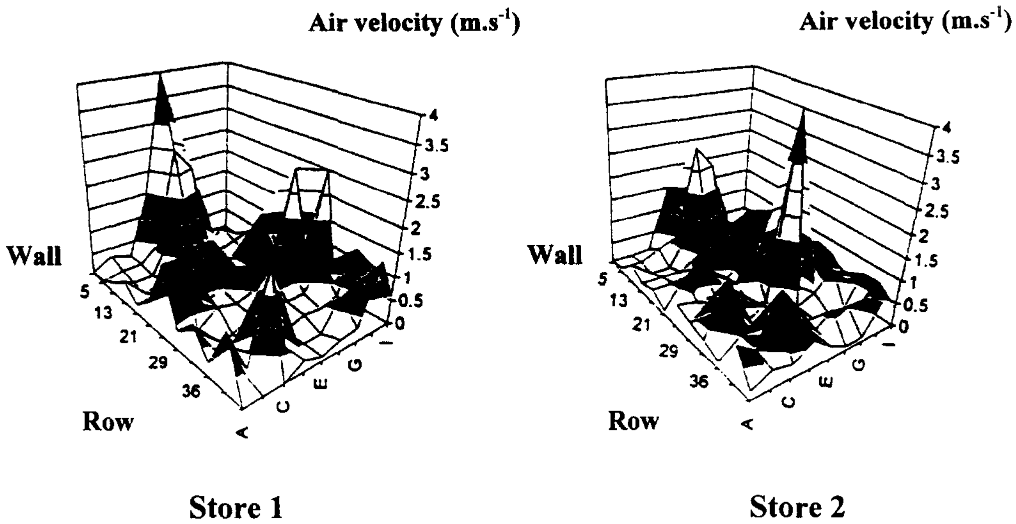


Fig. 1. Graphical representation of the mean air velocities measured in a horizontal section in two cheese stores of similar dimensions, but with different air flow rates: 136000 $m^3 h^{-1}$ in the case of store 1 and 102000 $m^3 h^{-1}$ in the case of store 2 (from Falconer, 1993).

It was therefore necessary to set up a robust, fast, accurate and inexpensive experimental method for visualising air flow patterns and the spatial distribution of the mean air velocity. This paper deals with the principles of this method and shows from two examples that this method makes it possible first to accurately analyze air flow and second to carry out diagnoses of the functioning of the plant studied.

THE EXPERIMENTAL METHOD

Daudin and Van Gerwen (1996) have summed up the difficulties related to the measurement of air velocity in food plants as follows:

(1) air flow is very turbulent due to the presence of blowing devices and of obstacles. This means that air direction and velocity (Fig. 2) vary rapidly with time and their mean values at one location are thus difficult to measure.
(2) the mean air flow direction varies with space. Consequently, anemometers which are air flow direction-dependent cannot be used.
(3) The mean air velocity varies rapidly in space (Fig. 1). A known value at one or several points has little meaning, particularly if not averaged over a long

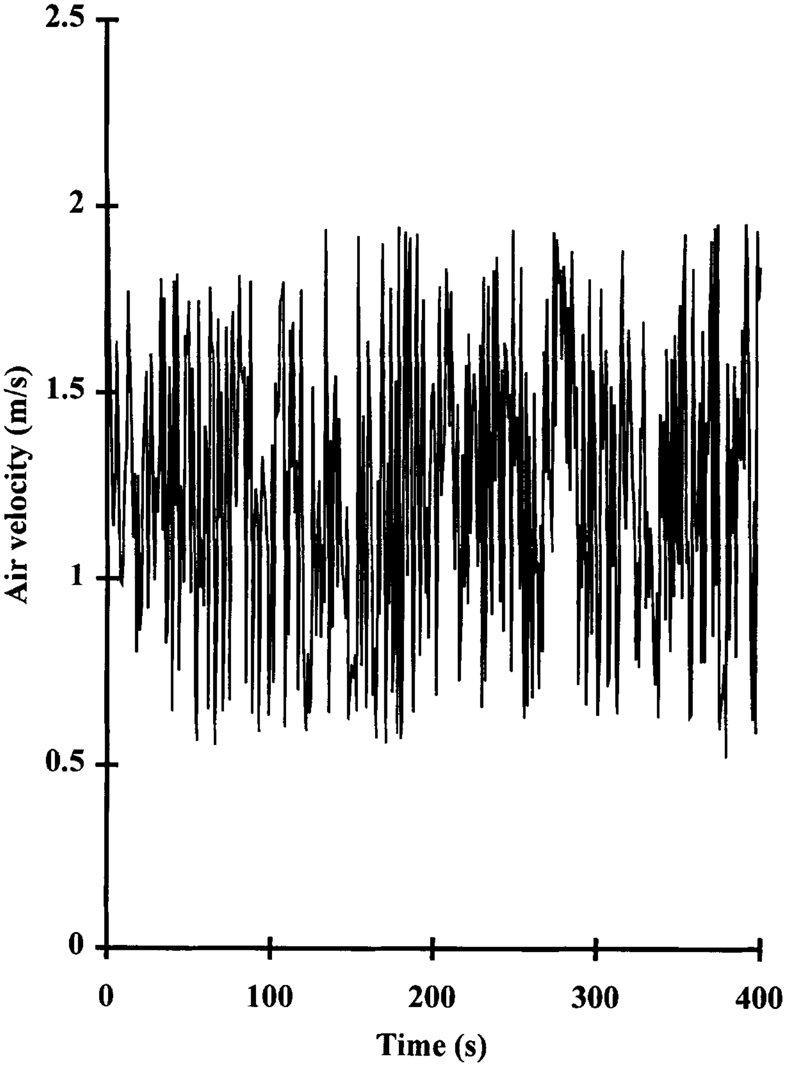


Fig. 2. Variations in the air velocity magnitude due to flow unsteadiness and turbulence at one fixed location in an industrial meat chiller (air velocities were recorded using a hot-film type anemometer and an acquisition frequency of 1 Hz).

period of time. Moreover, air velocity is sometimes very low and below the sensitive threshold of common anemometers.

Given the difficulties mentioned above and because of the time required to use the standard average procedure, an experimental method was developed with the aim of:

(1) recording air velocity at regular intervals using hot-film type anemometers moving slowly and continuously in a plant (these probes are currently used in the industry by engineers on account of their sturdiness),
(2) and then, apply mathematical processing in order to separate velocity variations corresponding to the movement and thus related to space from time-fluctuations due to air flow unsteadiness. Signal processing techniques (Fourier's series and low-pass filter) are suitable in so far as time-fluctuations have a higher frequency in comparison to mean velocity variations in space.

However, although turbulence in the flow has considerable effects on the heat and mass transfers (Kondjoyan *et al.*, 1993), the method presented cannot take into account these effects. A method to improve this limitation could be the use of hot-wire type anemometers, since they are known to be really suited to measure turbulence rates in air flows.

Mathematical basis

The mathematical processing of the values obtained is based on the approximation of a $f(x)$ function by a development in Fourier's series restricted to the first p terms (Angot, 1961).

For a $f(x)$ function whose period is T and represented by a table giving $(n+1)$ ordinate values $y_0, y_1, \ldots, y_n$ for $(n+1)$ equidistant abscissa values $x_0, x_1, \ldots, x_n$, it was demonstrated that the coefficients of the development in Fourier's series can be written as follows (analysis of the system of eqns (1–4) is also known as the Finite Discrete Fourier Transform):

$$a_k = \frac{2}{n+1} \sum_{i=0}^{n} v_i \sin\left(\frac{2\pi}{T} k\, t_i\right) \tag{1}$$

$$b_k = \frac{2}{n+1} \sum_{i=0}^{n} \cos\left(\frac{2\pi}{T} k\, t_i\right) \tag{2}$$

$$b_0 = \frac{1}{n+1} \sum_{i=0}^{n} v_i \tag{3}$$

for a development in Fourier's series applied to the v_i velocity equal to:

$$v_i = b_0 + \sum_{k=1}^{p} a_k \sin\left(\frac{2\pi}{T} k\, t_i\right) + \sum_{k=1}^{p} b_k \cos\left(\frac{2\pi}{T} k\, t_i\right) \tag{4}$$

This requires a measurement of air velocities at regular intervals and a movement of the probe at a constant velocity.

According to eqn (4), it appears that all velocities calculated after processing will be exactly equal to b_0 if no term is kept in the sums, that is the mean air velocity recorded during the anemometer motion. Conversely, if all terms are kept in these sums, the v_i velocity calculated after processing will be equal to the value measured by the anemometer. As a matter of fact, the higher the number p of terms kept in eqn (4), the higher the number of terms of high frequency in the $v_i(t_i)$ function. The anemometer must be moved slowly so that the information contained in the terms of low frequency mainly corresponds to the variation in space of the mean velocity and so that it is possible to separate it from the fluctuations of higher frequency due to air flow unsteadiness.

Moving the anemometers

One of the original features of the method lies in the fact that the anemometer is moved.

In order to facilitate data processing and to determine the point where measurement was performed, it is necessary to measure air velocities at regular intervals and to move the probe at a constant velocity (v_p). Considering these experimental conditions, i.e.:

$$L = v_p T \tag{5}$$

$$Y_i = v_p t_i \tag{6}$$

where L represents the whole distance covered by the probe and Y_i is the ith distance of the measurement point from the starting point of the anemometer (Y_i varies from 0 to L), eqn (4) becomes:

$$v_i = b_0 + \sum_{k=1}^{p} a_k \sin\left(\frac{2\pi}{L} k\, Y_i\right) + \sum_{k=1}^{p} b_k \cos\left(\frac{2\pi}{L} k\, Y_i\right) \tag{7}$$

In the meat industry, carcass conveyors can be used to hang and move the anemometers because of their slow and constant velocity. An example of the application of the experimental method set up in our laboratory to an industrial case where this type of motion was used is given below.

In the case of static plants without conveyors, a system made up of an independent moving truck and rails of guidance can be used. All technical elements of the specific system designed to test the method are detailed in Mirade's thesis (Mirade, 1996). A truck made up of a metallic plate with four wheels and a rod carrying the anemometers was built. The probes were multidirectional hot-film type anemometers (8450-11M, TSI, USA) whose response time was 0·2 s. They were well suited to measure the velocity magnitude in large food plants (Peyrin *et al.*, 1995).

Tests were carried out in order to find a compromise between a moving velocity high enough to reduce the experimental time and a velocity low enough to be negligible in comparison to air velocity. The number of measurement points had to be large enough to take into account the spatial variation of mean velocity. Finally, a moving velocity of the truck equal to 1·2 cm s^{-1} which was about 90-fold lower than the minimum air velocity measured in the room and an acquisition frequency of 1 Hz were chosen.

Filtering the recorded signal

As mentioned before, and because of the low moving velocity of the anemometers, the spatial variations of mean air velocity correspond to the low frequencies of the signal. Keeping this information amounts to applying a low-pass filter to the recorded signal afterwards.

To determine the cutting frequency of this low-pass filter which must be determined for each plant studied, spectra representing measurements performed by moving the anemometer (the laboratory method) were compared to spectra corresponding to measurements carried out at one fixed location (the standard average procedure); these last measurements were representative of what would be performed anywhere in the room. The spectrum is a representation of the fluctuation amplitude (MN normed modulus) as a function of each frequency of the recorded signal. The MN modulus at a given frequency is calculated from the coefficients of the development in Fourier's series as follows:

$$MN(k) = \frac{\sqrt{a_k^2+b_k^2}}{b_0} \tag{8}$$

As time-fluctuations of the signal can be considered as noise, the comparison was made on an average of 12 spectra. Figures 3a and b show the mean spectra obtained by moving (Fig. 3a) and without moving (Fig. 3b) the anemometer in a small-scale chiller (see part 'Application of the method to 2 cases of the meat industry'), for a measurement line where there were strong variations in air velocity. The difference noted between both mean spectra was attributed to a spatial variation in air velocity recorded when the anemometer was moving. The spectrum in Fig. 3(b) is similar to the spectrum which would have been obtained if the signal plotted in Fig. 2 had been processed. In the present case, the cutting frequency is equal to 20 mHz, i.e. a restriction of the number p of terms to 13 that must be kept in eqn (7). In both spectra, for a frequency higher than the cutting frequency, the MN normed modulus is equal to 0·05, whatever the frequency, which means that time-fluctuations can be considered as noise.

Moreover for the signal considered, as the sampling interval satisfied the conditions of Shannon's theorem, no distortion of the spectrum occurred and so no high frequency was folded onto lower frequency. Consequently, the phenomenon called aliasing could not take place (Max, 1981).

Figure 4 shows the application of the restriction of the number p of terms to 13 in eqn (7): the time-fluctuations of the velocity related to air flow unsteadiness were

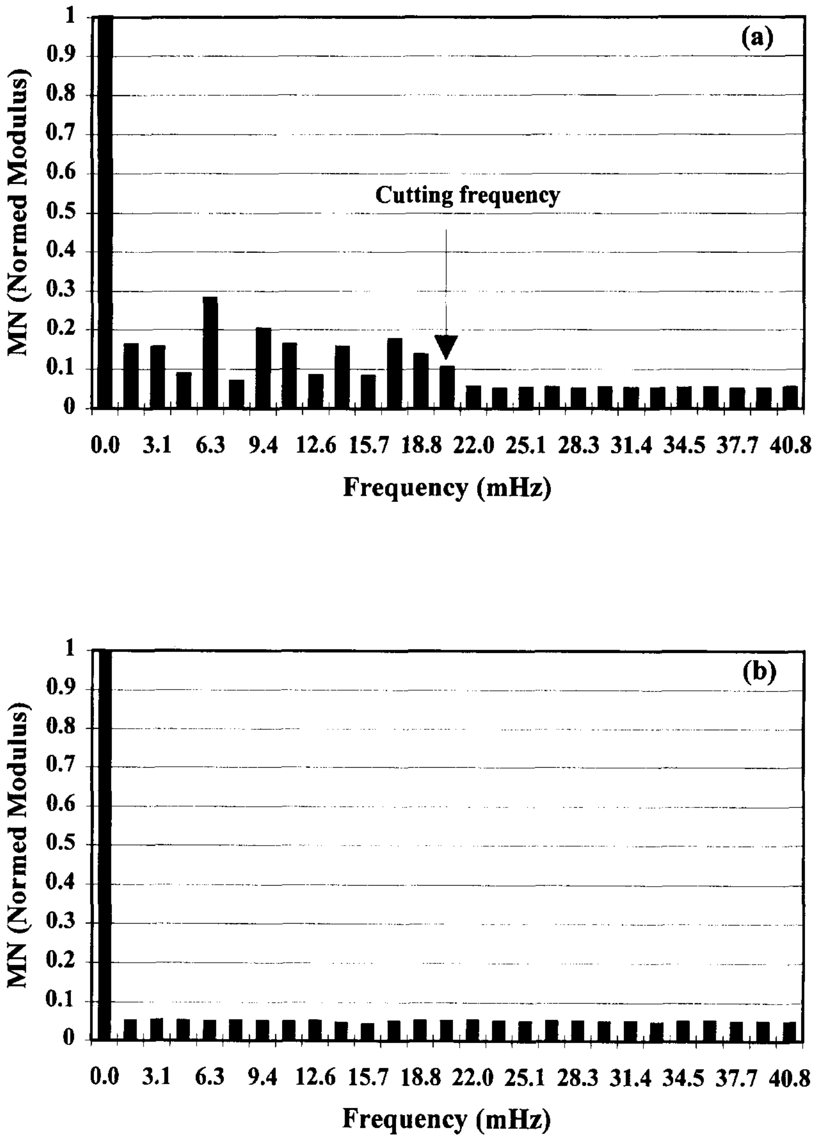


Fig. 3. Graphical representation of the mean spectra representing the air velocity measurements performed (a) by moving and (b) without moving the hot-film type anemometer in a small-scale chiller (the mean spectrum shown in Fig. 3(b) was obtained at one fixed location and it was really representative of what could be performed anywhere in the room).

eliminated from the unprocessed signal afterwards. Only the low frequency variations representing the spatial variation in mean velocity were kept.

Validation and advantages of the method

There is *a priori* no definitive evidence for a pronounced separation between the two parts of the spectrum presented in Fig. 3(a), i.e. between the low and high frequencies; the low-frequency part (below 20 mHz) probably involves undesirable time-fluctuations.

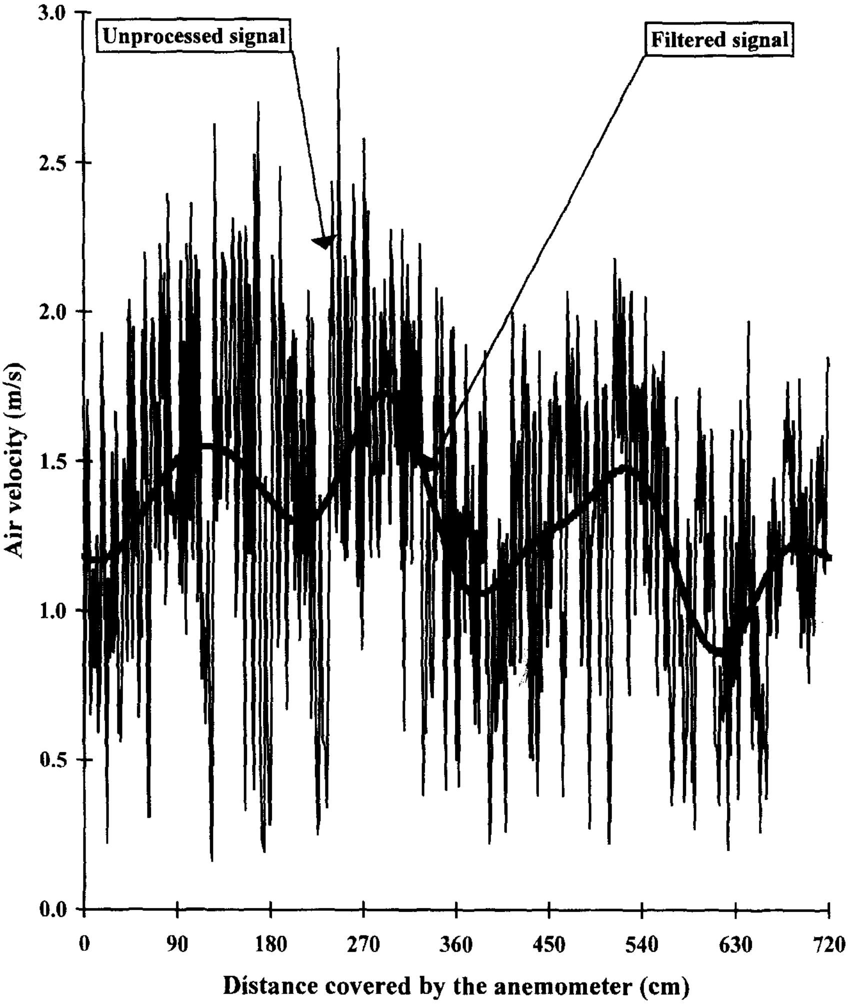


Fig. 4. Example of a filtered signal at 20 mHz corresponding to air velocity measurements carried out in the small-scale chiller by moving the hot-film type anemometer at a velocity of 1·2 cm s^{-1} and using an acquisition frequency of 1 Hz.

In order to rigorously check the validity of data processing, we made a comparison between this new method and the standard average procedure. According to tests concerning the standard average procedure carried out at one fixed location in the small-scale chiller, it appeared that measurements performed at a frequency of 1 Hz must be averaged at least over a 5 min-interval in order to obtain a stabilization of the mean of air velocities (this amounts to apply a kind of filtering to the measurements which is very time-consuming!).

Figure 5 indicates that the differences in mean velocities between both methods are equivalent to the measurement error made when using a hot-film type anemometer, i.e. 0·1 m s^{-1}. In addition to a good accuracy, the new method has the

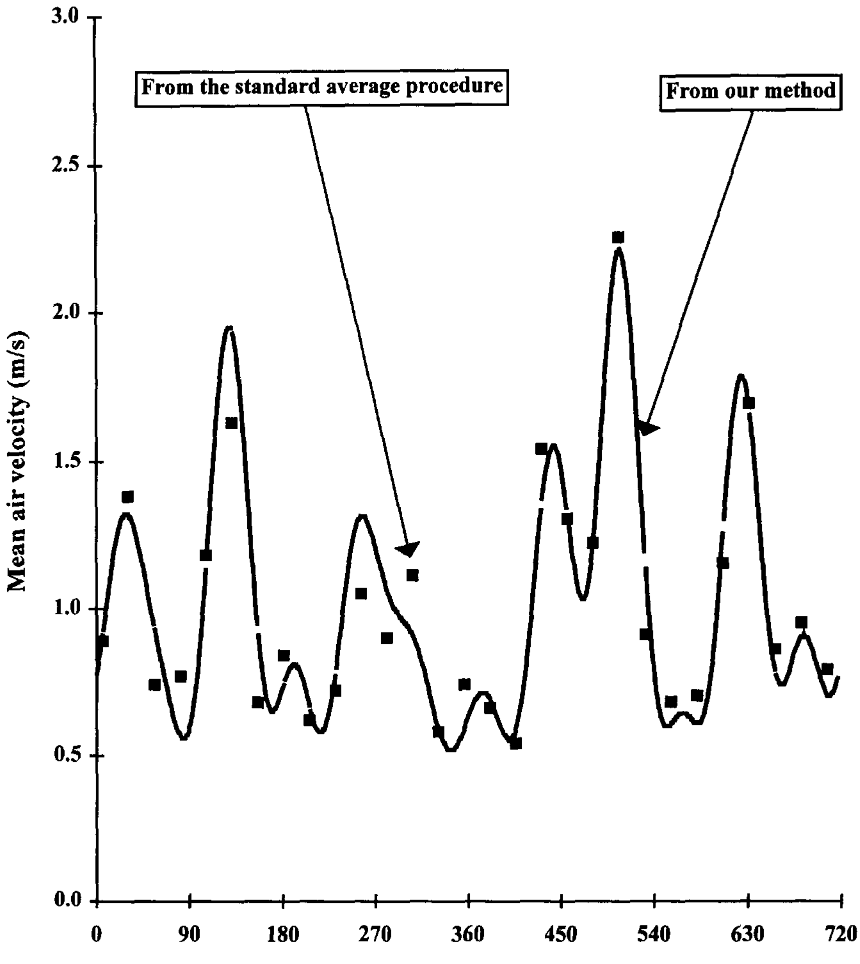


Fig. 5. Validation of the new experimental method set up in our laboratory in comparison to mean air velocities calculated from the standard average procedure.

advantage of being much faster. For instance, only 10 min were required to obtain the continuous curve, that is 600 measurement points (Fig. 5), whereas almost 3 h were necessary to determine the 30 points of mean velocity presented in the same figure. This new method makes it possible to decrease the experimental time by 350 and the number of measurement points can thus be increased.

In addition, keeping only the Fourier's coefficients necessary to calculate the mean velocity instead of the whole unprocessed measurements of velocity reduces by 30 the file size for a given plant.

Lastly, a software was developed for easily and quickly calculating and visualising the air velocity distribution on intensity maps, whatever the section of the plant, i.e. horizontal, longitudinal or vertical.

APPLICATION OF THE METHOD IN THE MEAT INDUSTRY

Taking a small-scale chiller and an industrial pork chiller as examples, this part shows how this experimental method makes it possible to analyze unsteady air flows and to assess the plant operation. The contribution of this method to the study of air flow can only be assessed by means of a computer which makes it possible to quickly examine a series of representations giving useful information on the whole flow patterns. It is more difficult to follow the air flow variations on paper. That is why only some representations chosen for their relevance concerning the assessment of the air flow analysis are given here. All these flow representations stem from experiments carried out in empty plants. This contributes to quickly obtaining a large number of measurement points (about 250000 distributed among 400 measurement lines for each configuration), from which velocity intensity maps with a fine resolution can be plotted. In the figures presented, each line intersection corresponds to a location where the mean air velocity was measured.

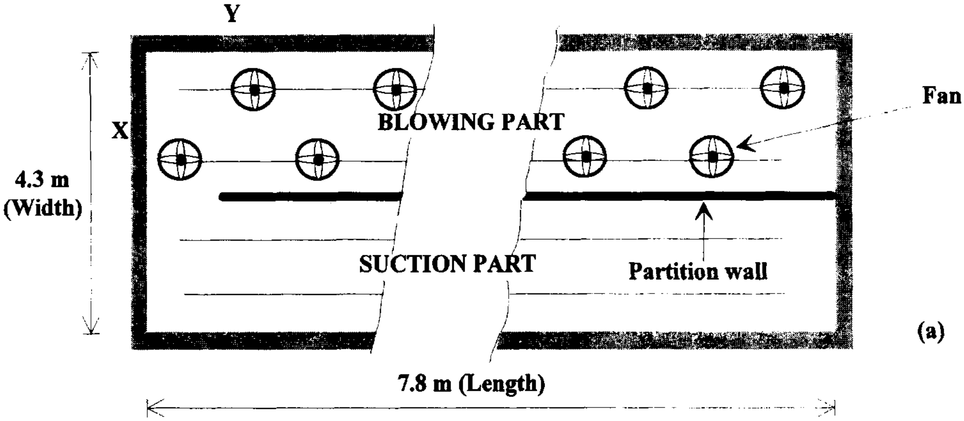
The small-scale chiller

In order to validate the method and to avoid industrial constraints, a small-scale chiller was built ($7{\cdot}8 \times 4{\cdot}3 \times 4$ m).

Figures 6(a) and 6(b) describe the geometry and functioning of this apparatus. The small-scale chiller was divided into two parts by a partition wall whose length was smaller than that of the plant (in practice, this makes it possible to transfer products from one part to another). In the 'blowing part', air was introduced through flexible shafts placed in two staggered rows (Fig. 6a). To simulate the blowing of an industrial helicoid fan, the shafts were fitted just at the output with static vanes which spread and swirled the air flow coming out. The air circulated afterwards below the partition wall and went up in the 'suction part' to be finally sucked up in the ceiling and sent to the air conditioning system (Fig. 6b). The whole air flow rate was steady and equal to 40000 m^3 h^{-1}.

To understand how the air flows in the small-scale chiller, it is advisable to simultaneously examine Figs 7, 8 and 9 which present parts of the empty plant according to the three directions in space.

Figures 7(a) and 7(b) show the large heterogeneity of the flow in the small-scale chiller in so far as air velocities almost range from 0 to more than 3 m s^{-1}. In the horizontal section of the blowing part (Fig. 7a) at $Z = 210$ cm, i.e. 90 cm below the blowing of fans, interactions between air flows originating from the different fans placed in two staggered rows clearly appear with velocities higher than 2·5 m s^{-1}. Moreover, the velocities lower than 1 m s^{-1} indicated in this figure at $X = 74$ cm and $X = 174$ cm are located straight above the fans. Fig. 7(b) accounts for an air flow distribution radically different in the suction part of the chiller with velocities higher than 2 m s^{-1} near the wall placed at $X = 424$ cm and velocities equal to 1 m s^{-1} just behind the partition wall (at $X = 234$ cm). Between these two distinct



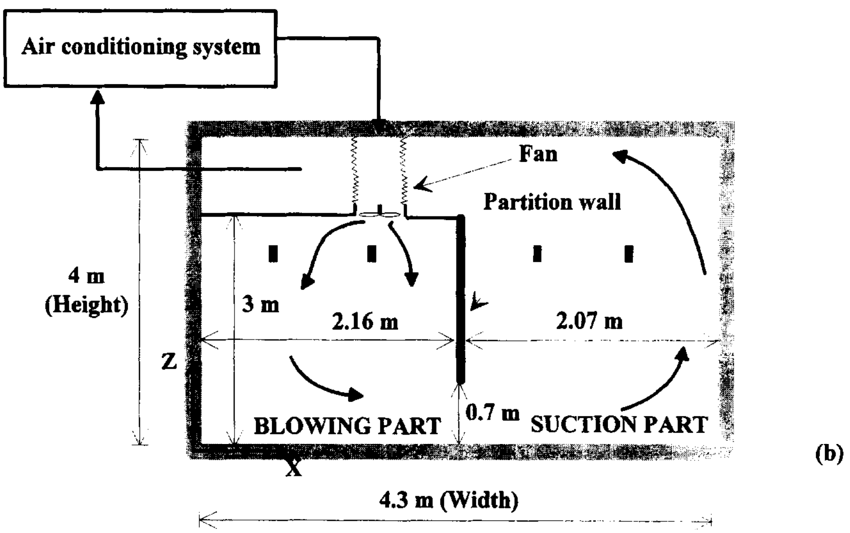


Fig. 6. Description of the geometry of the small-scale chiller: (a) top view (b) vertical section with a schematic description of the air flow.

and much ventilated areas, there is a large area characterised by very low velocities ranging from 0 to 0·5 m s^{-1}. Owing to the fact that the length of the partition wall was smaller than that of the plant, so that a part of the air flow rate was directly blown in the suction part, a 3-dimensional effect is underscored in Fig. 7(b) for lengths of Y ranging from 165 to 305 cm. That the reason why the air velocities at this place were higher than 1 m s^{-1}.

The large heterogeneity of the air flow shown in Figs 7a and 7b is again underlined in Fig. 8 which represents the air velocity distribution on a longitudinal section located at $X = 174$ cm, that is in the axis of the second fan row. Analyzing this view (Fig. 8) shows that the interactions between the blowing of fans lead to a succession of areas where velocities are higher than 2·5 m s^{-1} for a height $Z = 210$ cm and to hollows of ventilation which are located exactly straight above each fan: the velocity magnitude is about 0·5 m s^{-1} for lengths $Y = 150$, 250, 350, 450, 550 and 650 cm.

Besides the acceleration of the air flow along the partition wall for values of X ranging from 164 to 204 cm, Fig. 9(a) accounts for rather low air velocities in the

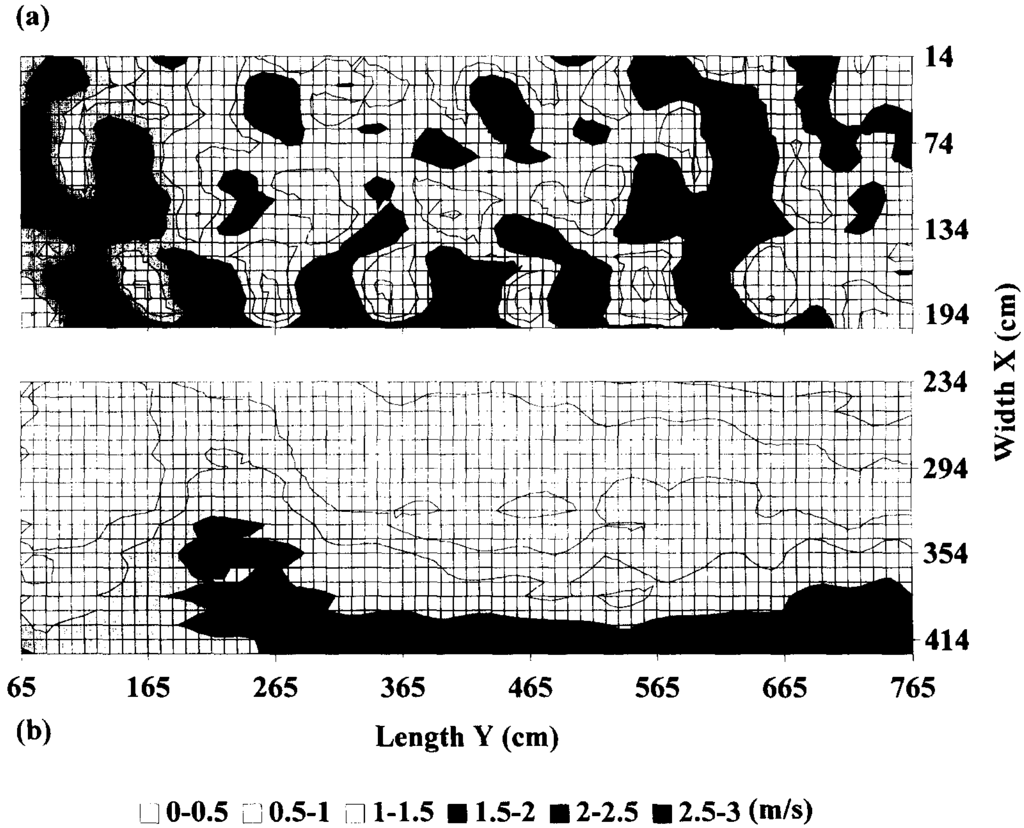


Fig. 7. Distribution of the measured air velocities in the small-scale chiller in a horizontal section located (a) in the blowing part at 90 cm below the blowing of fans and (b) in the suction part at 2·1 m above the ground.

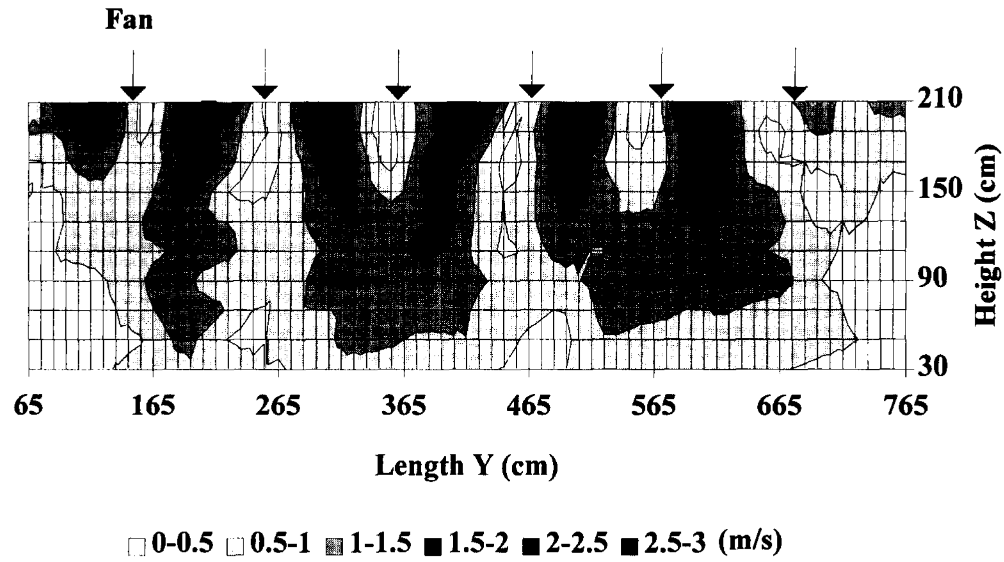


Fig. 8. Distribution of the measured air velocities in the small-scale chiller in a longitudinal section located at a width X of 174 cm, i.e. in the axis of the second fan row.

blowing part, below 1 m s^{-1} for heights Z ranging from 30 to 150 cm. This can probably explain why, in practice, carcass forequarters sometimes get cold less rapidly than thicker rear quarters. The vertical section of the suction part of the chiller confirms the existence of a poorly ventilated area (Fig. 9b); consequently, a large part of the air flows above the ground and along the wall opposite to the partition wall instead of circulating in the middle of the part where products are usually placed. This lack of ventilation is obviously harmful to the chilling of the products present at this place because it strongly reduces heat and mass transfers. According to Brown and James (1992), the cooling time of pork carcasses is increased by 10% when air velocity varies from 1 to 0·5 m s^{-1}. In fact, the existence of an area where air velocities are close to 0 m s^{-1} is much more harmful to carcass chilling than the air flow heterogeneity in so far as, in view of this large size, it is liable to strongly affect the chilling kinetics for a long time. This increases the cooling time of carcasses and lowers the efficiency of the process.

Measurements carried out by our laboratory in several industrial chillers with an identical geometry have led to similar findings. Bouton (1992) experimentally showed in an industrial pork offal chiller that the air velocities were very high below the partition wall (ranging from 1 to 4·3 m s^{-1}) and that the major part of the air flowed below the products at about 40 cm above the ground. He also measured air velocities equal to 0·3 m s^{-1} around the offals in the middle of the suction part. Rudelle (1993) showed in an industrial bovine carcass chiller that air velocities were on average twice higher at the level of carcasses in the blowing part (1·3 m s^{-1}) than in the suction part (0·6 m s^{-1}). She also noticed a lack of ventilation around carcasses in the suction part and very high air velocities below the partition wall at

40 cm and 70 cm above the ground, as well as near the wall opposite to the partition wall.

The lack of ventilation underscored in the small-scale chiller confirms the conclusions resulting from the experiments carried out in similar industrial chillers.

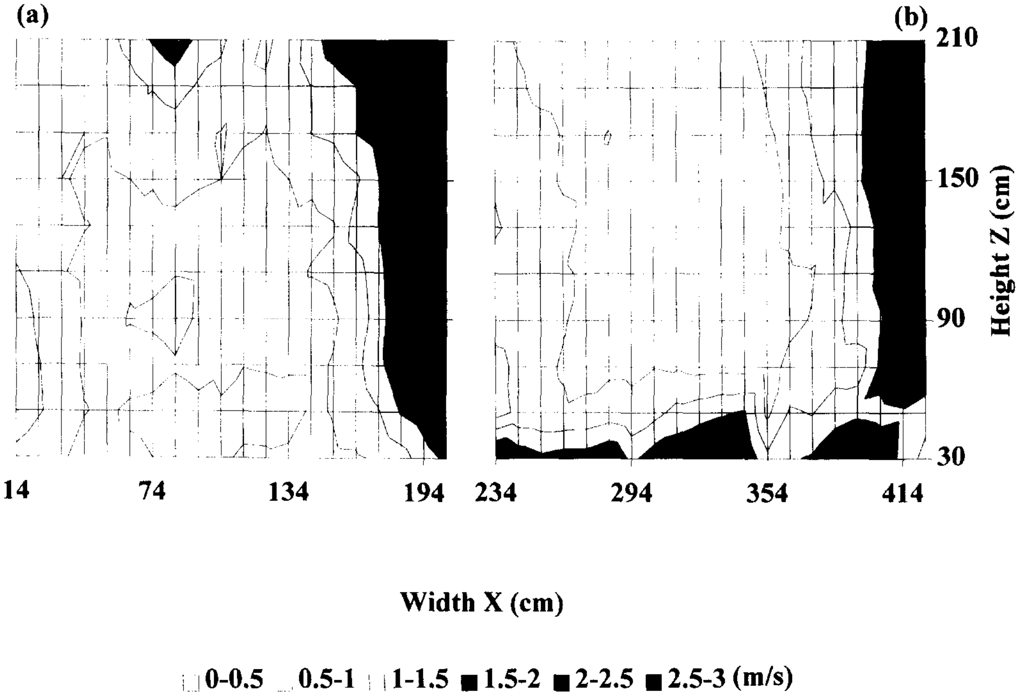


Fig. 9. Distribution of the measured air velocities in the small-scale chiller in a vertical section located at a length Y of 405 cm (a) in the blowing part (b) in the suction part.

An industrial pork offal chiller

Applying the method to air velocity measurements in an industrial chiller filled with pork offals contributed to assessment of its capacity.

The functioning and geometry of the industrial chiller were similar to those of the small-scale chiller previously described (Fig. 6a and 6b). The free space below the partition wall was equal to 0·7 m and the two parts of the chiller were of equal width, i.e. 2·3 m. Only the plant length was very different: 20 m instead of 7.8 m. The pork offal conveyor was used to move a hot-film type anemometer with a velocity of about 3 cm s^{-1}. The acquisition frequency was equal to 125 mHz, that is one measurement of air velocity every 8 s. The mean air velocity distribution was calculated from the average of measurements recorded on each plant length with the measurement system hung among offals. This method showed that, apart from walls, the mean air velocity varied very little in the blowing part and that the higher velocities were measured in the suction part near the wall opposite to the partition wall and not in the middle where offals moved on.

The thermal consequences of this uneven air distribution are significant. The drop in temperatures (T_p) measured in a pork liver of 1·5 kg and presented in Fig. 10 illustrates these effects (Daudin *et al.*, 1992). In theory, when a product is cooled with air at a constant air velocity and temperature, the slope of logarithm of the dimensionless temperature at one location, defined by $((T_p - T_{air})/(T_{p\ initial} - T_{air}))$, is constant after a short while (Cleland, 1990). In Fig. 10, an obvious change in slope, particularly marked below the surface, occurs after 60 min, corresponding to the liver entry into the poorly ventilated area of the suction part. Another change in slope occurs afterwards as a result of the liver circulation along the wall opposite to the partition wall where air velocities increase.

According to the example displayed in Fig. 11, using the new experimental method at an industrial scale provides a description of the air flow quite satisfactory and above all more precise. However, the main difficulty was to determine a cutting frequency in so far as the separation between the low frequency information representing the spatial variation of the mean velocity and the time-fluctuations was not as clear as in the mean spectra performed in the small-scale chiller (Fig. 3a). A cutting frequency of 18 mHz was therefore chosen. In spite of a small number of mean velocity points, Fig. 11 indicates the blowing interactions created by the different fans, gives useful information about the respective sizes of the largely and poorly ventilated areas and makes it possible to clarify the thermal consequences experimentally measured (Fig. 10).

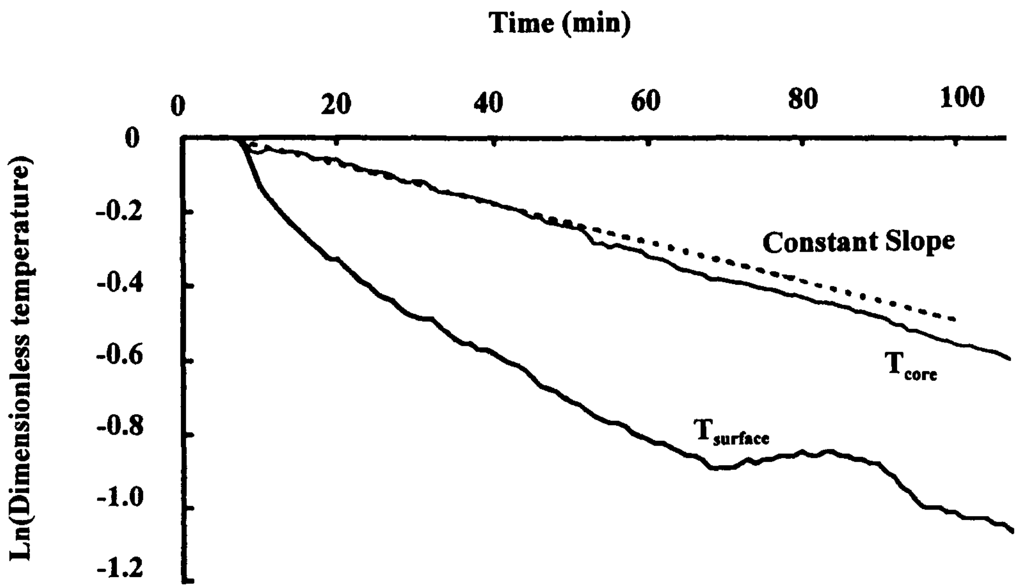


Fig. 10. Variation of the logarithm of the dimensionless temperatures measured in a pork liver of 1·5 kg moving in an industrial chiller (from Daudin *et al.*, 1992).

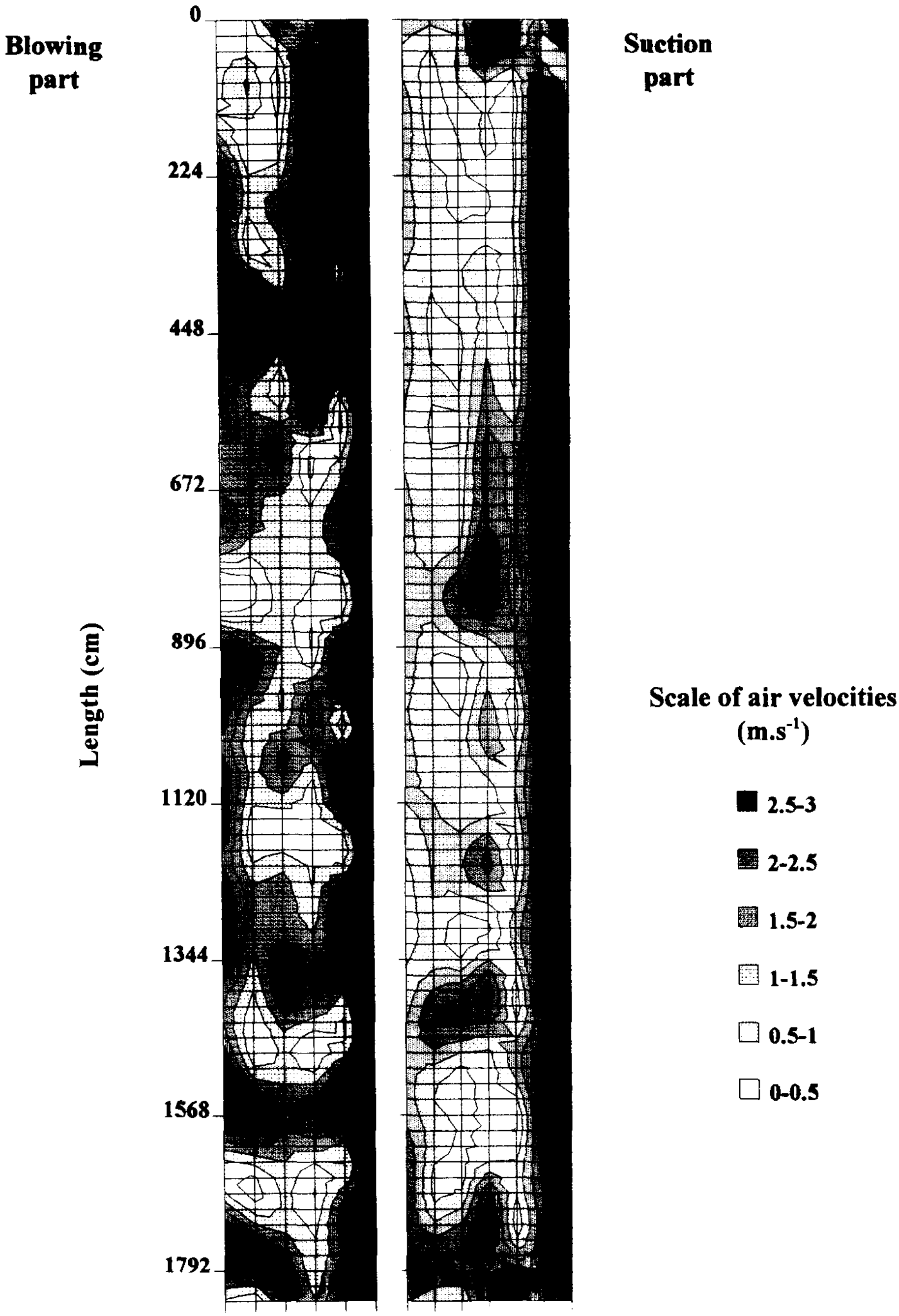


Fig. 11. Distribution of the air velocities in an industrial pork offal chiller in a horizontal section located at 70 cm above the ground (the velocities were calculated after processing the measurements carried out in our laboratory, with the new experimental method).

CONCLUSION

The method developed in our laboratory and designed for measuring air velocity in an unsteady and turbulent flow is rapid and accurate: time saving in comparison to the standard average procedure is equal to 350 and its accuracy is similar to the measurement error, i.e. 0·1 m s^{-1}. In addition, this method is very simple to use and suitable for processing industrial air flows where anemometers can be moved by the product conveyor. It makes it possible to explain how the plant operates and thus to reveal a possible dysfunction. The spatial distribution of measurement points is sufficient to accurately analyze and locate flows. This can help to understand the failure of thermal treatments that was observed for some products. Lastly, increasing the number of measurement points will contribute to fully validate the CFD calculations (Mirade *et al.*, 1995).

In addition, in order to complete this experimental method which cannot be used to assess the air flow direction, our laboratory has recently built an 'electronic weather cock' equipped with an optical coding system that properly measures the air flow direction inside pilot plants for air velocities higher than 0·9 m s^{-1} (Peyrin *et al.*, 1996).

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REFERENCES

Angot, A. (1961). *Compléments de Mathématiques*. Collection technique et scientifique du C.N.E.T., Revue d'Optique.

Bouton, V. (1992). Contribution à l'amélioration du fonctionnement et de la conception des installations frigorifiques. Mémoire d'ingénieur, Ecole Supérieure des Ingénieurs et Techniciens Pour l'Agriculture, Val de Reuil, France.

Brown, T. & James, S. J. (1992). Process design data for pork chilling. *International Journal of Refrigeration*, **15**(5), 281–289.

Cleland, A.C. (1990). *Food Refrigeration Processes: Analysis, Design and Simulation*. Elsevier Applied Science, London.

Daudin, J.D. & Kondjoyan, A. (1991). Influence de l'indice de turbulence de l'écoulement sur les procédés de traitement thermique de solides par l'air. *Récents progrès en génie des procédés, Lavoisier, Paris*, **5**(13), 287–294.

Daudin, J. D., Bouton, V., Prod'Homme, G. & Mourlhon-Dallies, J. L. (1992). Air flow characteristics in a chiller for pork offals and cooling consequences. *38th International Congress of Meat Science and Technology*, Clermont-Ferrand, France, Vol. 6, pp. 1195–1198.

Daudin, J. D. & Van Gerwen, R. J. M. (1996). Air circulation: how to cope with this critical point. New developments in meat refrigeration, EC\CE/AMST/I.I.R. *Chillers* 30–37.

Falconer, R. M. (1993). Cooling air distribution in a cheese store. I.I.F.-I.I.R., Commissions B1, B2, D1, D2/3, Palmerston North, New Zealand, Vol. 3, pp. 406–413.

Kondjoyan, A., Daudin, J. D. & Bimbenet, J. J. (1993). Heat and mass transfer coefficients at the surface of elliptical cylinders placed in a turbulent air flow. *Journal of Food Engineering*, **20**, 339–367.
Maghirand, R. G. & Manbeck, H. B. (1993). Modeling particle transport in slot-inlet ventilated airspaces. *Transactions of the ASAE*, **36**(5), 1449–1459.
Max, J. (1981). *Méthodes et Techniques de Traitement du Signal et Applications aux Mesures Physiques*, Tome 1. Editions Masson, Paris.
Mirade, P. S., Daudin, J. D. & Arnaud, G. (1995). Simulation en deux dimensions de l'aéraulique de deux tunnels de réfrigération des viandes. *Revue Internationale du Froid*, **18**(6), 403–412.
Mirade, P. S. (1996). Contribution à l'amélioration des procédés de traitement des solides par l'air grâce à la description et à la simulation de l'aéraulique—application au refroidissement et au séchage des viandes. Thèse de Doctorat de l'Université de Poitiers, Thermique, France.
Mizier, M. O. (1991). Le profil 3D d'un débit d'air à Mach 4 vu au laser. *Mesures*, **639**, 85–87.
Peyrin, F., Kondjoyan, A. & Daudin, J. D. (1995). Analyse des difficultés de mesure de la vitesse de l'air dans les Industries Agro-Alimentaires. *Industries Alimentaires et Agricoles*, **10**, 728–735.
Peyrin, F., Mirade, P. S. & Daudin, J. D. (1996). Conception et test d'une girouette électronique pour l'étude de l'orientation des écoulements d'air. *Journées mesures I.N.R.A., Cary le Rouet*, **IV**, 66–71.
Rudelle, S. (1993). Contribution à l'amélioration du fonctionnement et de la conception des installations frigorifiques. Mémoire d'ingénieur, Ecole Supérieure des Ingénieurs et Techniciens Pour l'Agriculture, Val de Reuil, France.
Ruegg, T., Stangier, R., Stoeckli, B., Tanner, C., Dorer, V. & Lommel, A. (1994). Air flow and comfort measurements in churches. *ROOMVENT'94, Air distribution in rooms, 4th International Conference*, Krakow, Poland, Vol. 2, pp. 455–468.
Scott, G. & Richardson, P. (1997). The application of computational fluid dynamics in the food industry. *Trends in Food Science and Technology*, **8**, 119–124.
Wooton, A. E. (1986). Factors affecting the chilling rate of beef sides. I.I.F.-I.I.R., Commission C2, Bristol, U.K., Vol. 3, pp. 115–121.