COMPARISON OF ROOM AIRFLOW CHARACTERISTICS IN A FULL SCALE ENVIRONMENT CHAMBER WITH COLD AIR DISTRIBUTION

S. C. Hu[1], J. M. Barber[1] and H Chiang[2]
[1]School of Architecture and Building Engineering, University of Liverpool, Liverpool, UK
[2]Energy and Resources Laboratory, Industrial Technology Research Institute Hinchu, Taiwan R.O.C

ABSTRACT

This paper discusses the experimental study of direct delivery of cold air into a full scale environmental chamber using different diffusers, i.e. a multi-cone circular ceiling diffuser, a vortex diffuser and a nozzle type diffuser. Comparisons have been made of the following: mean flow patterns, temperature distribution and condensation risk. The vortex diffuser exhibits a higher induction effect than that of the nozzle type diffuser. However, the air speed generated by the vortex diffuser is generally lower than that of nozzle type diffuser. Both the vortex diffuser and the nozzle type diffuser can be considered to have condensation-free characteristics, although some vapor film was observed in the early stage of a "hard start up" test for the nozzle type diffuser.

INTRODUCTION

Cold air distribution has been defined as the supply of cold air directly to rooms. The cold air has a temperature range of 4 °C to 10 °C, being 5 °C to 10 °C lower than the conventional chilled air supply system. The cold air distribution system has become more popular of late because of the increased popularity of ice thermal storage systems which make cold air generation very easy. However, the cold air dumping in the occupied zone and the condensation on diffuser sufarcees are the two concerns in pratical application. One solution to this problem is by the use of Fan-Powered Mixing Boxes (FPMB). The FPMB mix room air with cold air before delivering it to the room. However, there are several disadvantages with FPMB including: (1) the use of small, low efficiency fans and motors in the FPMB offset the fan energy savings arising from the use of cold air (20% to 40% lower than in conventional systems (Elleson1993)), (2) the installation and maintenance of the FPMB is an additional cost to the whole system, (3) the potential noise problem arising from the fans of FPMB. Mixing boxes do not need to be used if the cold air can be adequately mixed with the room air before entering the occupied zone. One solution is the use of a diffuser which induces a large amount of room air into the jet. These high-induction diffusers are designed to mix primary air with a high volume of room air at the diffuser outlet. Thus the effective supply air temperature rapidly increases to the conventional supply temperature in a short distance from the diffuser. There are currently two types of diffuser designed specifically for high-induction performance. One of these consists essentially of a terminal box with a ring of approximately 16 mm diameter holes round its outer edge. This type of diffuser is known as nozzle type diffuser. Detailed dimensions of this type of diffuser are shown in Figure 1a. Although currently on the market, the writers have been unable to obtain detailed performance data for them, especially for non-isothermal flow conditions. The other type is the vortex diffuser (or swirl diffuser) which is claimed to be best suited to installations with relatively high ceilings. This diffuser is provided with stationary radial vanes which aim to cause increased turbulence and mixing by imparting a spiral twist or swirl to the supply air, as indicated in Figure 1b. Similarly, no sufficiently detailed performance data for these, when handling cold air, has been located. As well as the nozzle type diffuser and vortex diffuser, a conventional multi-cone circular diffuser has also been tested for comparison. Figure 1c shows the configuration of the circular diffuser. The angle, ϕ, between the jet outlet and horizontal for the three cones are 40^{o}, 30^{o} and 45^{o}, respectively. There is an extensive body of literature concerning chilled air distribution, but there is very limited literature available on indoor airflow and temperature measurements for cold air distribution systems. Knebel and John (1993) conducted several field tests and showed that cold air can be successfully supplied to a room to produce a comfortable indoor environment under most operating conditions by using the nozzle type diffusers. However, the performance of the associated jet and the resulting field velocities and temperatures are not given. Kondon et al. (1995) performed a laboratory study for cold air distribution systems with a multi-cone circular diffuser. They concluded that there are only minor differences in velocity and temperature distributions between a case of 8 ^{o}C cold air system and a 13 ^{o}C conventional system with the same room temperature. However, substantial condensation and dumping were observed for 8 ^{o}C cases. Shakerin and Miller (1995) tested two types of vortex diffusers under isothermal conditions. Somewhat more induction was observed compared with that of a multi-cone circular diffuser in their study.

The objectives of the study herewith described were therefore (1) to study the flow pattern and temperature distribution of non-isothermal jets issuing from a multi-cone circular ceiling diffuser, a nozzle type diffuser, and a vortex diffuser, (2) to provide some useful suggestions for modifying the currently available multi-cone circular ceiling diffuser to fit the performance requirements of cold air distribution.

EXPERIMENTAL SETUP

A full-scale environmental chamber (length (L) x width (W) x height (H) = 7.2 m x 4.2 m x 2.8 m) was constructed following the ISO-5219 standard.Chilled water (3 ^{o}C) was supplied by a harvester type ice bank. The air volume flow rate was measured by standard nozzles which were located upstream of the diffusers. In order to simulate a uniform floor heat load condition as in office buildings, the floor in the chamber was heated by supplying an appropriate amount of hot-air into the plenum chamber under the floor. The amount of heat added to the plenum was controlled to give the room setpoint temperature (T_r). The room temperature was taken to be the temperature of the air in the chamber's extract duct and consequently, was controlled from a detector mounted against to the common extract point using an electronic PID controller. When steady state conditions had been established, T_r was controlled within ±0.3 °C of the setpoint temperature. The supply air temperature, T_i, was similarly controlled to within ± 0.3 °C of the setpoint temperature, also using a PID controller. The equilibrium was ensured during the experimentation by allowing the same steady state temperature conditions to be established before and during the taking of readings for each case. It could take more than three hours to establish equilibrium. The characteristics of the flow field were measured by a three-dimensional ultrasonic anemometer. The accuracy of the anemometer was ± 2% of the full scale range in use at that moment in time (after zero velocity adjustment).The sampling frequency was 20Hz with a sampling period of 30 seconds. The anemometer was mounted at the extension end of an XYZ mechanical traverse table (with a 0.1mm positioning precision) which was driven by a step motor. T type unshielded thermocouples, with accuracy ± 0.3 °C, were calibrated before they were used. Figure 2 shows the measurement system.

RESULTS AND DISCUSSION

Before taking detail measurements, the temperature and the velocity magnitudes at the center points of the four quarter zones in the chamber (at a level of FL+1.2 m) were measured to check symmetry. The spread of the temperatures was found to be less than 1.0 °C and of the velocity magnitude less than 0.1 m/s. These variations were considered small enough to assume that the flow field was sufficiently symmetrical to the center line of the diffuser. Therefore, only a 1/4 volume of the room was studied. Eight supply flow rate cases were used, as shown in Table 1. Table 1. Cases tested

case	type	Q(l/s)
1	A	47.2 (100cfm)
2	A	70.8 (150cfm)
3	A	94.4 (200cfm)
4	B	47.2 (100cfm)
5	B	70.8 (150cfm)
6	C	23.6 (50 cfm.)
7	C	47.2 (100cfm)
8	C	70.8 (150cfm)

Note: A: multi-cone circular ceiling diffuser, B: vortex diffuser, C:4way nozzle type diffuser

Mean Air Flow Patterns and Temperature Distribution

Figure 3a show the temperature contours of case 2 (supply air flow rate: 70.8 l/s (150cfm)) It was noted that he cold air jets drop directly after discharging from the diffuser. As the flow rate increased to 94.4 l/s (200cfm), a smaller stream formed along the ceiling while the main stream dropped down immediately under the diffuser (see Figure 3b). This appear to result from the configuration of the multi-cone diffuser under test. It was expected that the stream along the ceiling would become larger and extend further if the flow rate had been increased. The angle, ϕ, between the jet outlet and the horizontal, for the diffuser tested was greater than 30° which appears to have produced a downward projection (as indicated by Awbi (1992)). More research is needed to optimize cone angles and the effective areas of the multi-cone circular ceiling diffusers for use with cold air distribution.

Figure 4a shows the velocity vectors of case 5. It is observed that the trajectory of the air jet traveled horizontally, attaching to the ceiling and extending to around 2.7m from the diffuser. No dumping was observed in the occupied zone. Comparing the 3D velocity data of the multi-cone circular diffuser and the vortex diffuser tested, it should be noted the jet flow of the circular diffuser is radial while that of vortex diffuser is rotational For the same supply flow rate cases (case 2 and case 5), the air velocity magnitude of case 5 was generally lower than that of case 2. A strong rotation was seen at the diffuser outlet area, but this rotational motion decayed to produce a straight radial flow within about three diameters (1m) of the center of the diffuser. A very uniform temperature distribution through the whole field was observed (see Figure 4b). Even in the outlet area, the temperature increased to a value close to the room temperature which implied a very good mixing of room air with supply cold air adjacent to the outlet of the vortex diffuser. The flow pattern at the region close to the ceiling is similar to a three dimensional wall jet type flow. Incoming cold air attached to the ceiling and traveled downstream along the ceiling due to a strong Coanda effect. For the low flow rate case (case 6), the jet penetration length was approximately 2.3 m from the diffuser. A stagnation area formed at the upper-right region, due to the lack of air circulation. Much induction (mixing) of ambient air with primary air was observed (upward vectors) under the diffuser, although the supply flow rate was only 23.6 l/s (50 cfm). For the medium supply flow rate case with supply flow rate of 47.2 l/s (100 cfm), the jet length extended to approximately the opposite wall. As for the high supply flow rate case, there was no separation point observed , as seen in Figure 5c. In general , most of the excessive penetration jets could decrease the uniformity of airflow with a stream of air at floor level and hence cause a lower temperature zone in that area. However, for the high induction diffuser under test, excessive penetration of the jet did not create a greater temperature non-uniformity at the floor zone. This was because of it's high induction characteristic which entrained a large amount of room air in the jet before it traveled to the floor level. The jet velocity decay coefficient (K) is defined as:

$$K = \frac{V}{V_o} \frac{(x + x_p)}{\sqrt{A_o}} \quad (1)$$

In general, K values for the 4-way nozzle type diffusers under test were approximately 6.0 being higher than the values for currently available diffusers as indicated in ASHRAE Fundamentals (1997). The temperature distributions for case 6 to case 8 were generally similar. The temperature gradient in the occupied zone was very small (less than 1 °C), while the temperature gradient in the jet region was very large. A lower temperature region close to the wall of the case 8 could be observed (see Figure 5d) which may be due to an excessively long jet penetration. However, the temperature difference between the room setpoint temperature and air temperature in this region is less than 1 °C. For the vortex diffuser, the neck area was assumed to equal to the effective area. The effective area of the nozzle type diffuser was taken to be the gross area of the nozzles (9 nozzles for each way). The K value of the nozzle type diffuser tested was found to be 6.0 as mentioned before, whilst the K value of the vortex diffuser in the present study was around 2.2. This latter value was close to that of Shakerin and Miller (1996) and Shiyoa et al. (1994). The smaller K value of the vortex diffuser indicates that more air was entrained into the jet i.e. the induction was higher than that of nozzle type diffuser. The entrainment of the jets from the linear diffusers and from radial diffuser can be evaluated roughly by the following empirical formulas, respectively.

$$\frac{Q_x}{Q_o} = \frac{\sqrt{2}\,(x + x_p)}{KA_o^{1/2}} \qquad (2)$$

$$\frac{Q_x}{Q_o} = \frac{2\,(x + x_p)}{KA_o^{1/2}} \qquad (3)$$

Equations (2) and (3) were derived from equation (1) and the empirical relation of $\frac{Q_x}{Q_o}$ and $\frac{V_x}{V_o}$ (ASHRAE Fundamentals 1997). Assuming the neck area was the same as effective area of the vortex diffuser, the K value of the nozzle type diffuser was around three times higher than that of the vortex diffuser while the vortex diffuser had a fifteen times greater effective area than that of nozzle type diffuser. Consequently, the entrainment of the vortex diffuser was slightly greater than that of the nozzle type diffuscr.In general, it is concluded that the vortex diffuser tested shows a higher entrainment than that of the nozzle type diffuser. However, due to other factors such as room volume, diffuser location, room height etc., a high entrainment vortex is not necessary good for a cold air distribution system. Probably the nozzle type diffuser is more suitable than the vortex diffuser for handling small volume flow rates associated with a large turn-down ratio VAV systems. In general, due to the short throw of the vortex diffuser, a larger number of vortex diffusers than the nozzle type diffusers would seem to be required for treating similar same room areas.

Surface Condensation Tests

Surface condensation tests on the three types of diffusers were performed under an initial room condition of RH=90% and T_r=36 °C. Such a high initial temperature and humidity condition would represent a condition known as "hard start up" and can be seen in many circumstances. An example would be for an emergency weekend executive staff meeting that required the room to be cooled immediately during a weekend set back control sequence. This situation was tested for each diffuser type in turn. At the start of the tests, due to the heat gain from ducts and other equipment, the supply air temperature decreased slowly to 5 °C after about 12 minutes. With the multi-cone circular diffuser, much surface condensation and even some water droplets were observed on the surfaces of inner cone. With the nozzle type diffuser, it was found that in the first 32 minutes in the vicinity of the outlet area, the dew point temperature decreased slower than that of the diffuser surface temperature.(see Figure 6). Some vapor film was observed on the surfaces of the lower part of it after about 5 minutes of starting the test. However, no water-droplets were observed. The vapor film disappeared soon after the surface temperature achieved 20 °C. No condensation was found on the surfaces of the vortex diffuser throughout the period of the test.

NOMENCLATURE

A_n	neck area of the diffuser
A_o	effective area of the diffuser
K	centerline velocity decay coefficient .
Q_o	diffuser flow rate (m^3/s)
T	local mean air temperature (°C).
T_d	dew point temperature
T_{eff}	effective draft temperature (°C)
T_i	jet inlet temperature (°C)
T_r	room temperature (°C)
Ts	diffuser surface temperature (°C)
V	local mean air velocity (m/s)

V_x centerline velocity (m/s)
V_n neck velocity of diffuser (m/s)
V_o effective velocity of diffuser (m/s)
X horizontal distance from center of the diffuser to measuring point (m)
X_p virtual origin of jet (m)
ϕ angle between the jet outlet and the horizontal.

CONCLUSIONS

The results of this study enable the following conclusions to be drawn:
(1) Instead of using a FPMB (Fan Powered Mixing Box), the system tested supplied cold air directly. Both the vortex diffuser and the nozzle type diffuser appear capable of providing high induction room air, maintaining a relatively small temperature gradient in the occupied zone.

(2) The multi-cone circular ceiling diffuser tested suffered an early jet separation resulting cold air dumping in the occupied zone. Much surface condensation was observed at the surface of inner cone of the diffuser. Some modification of the currently available circular ceiling diffuser appears to be required for use in cold air distribution systems. This could involve optimizing the cone angles and outlet effective area.

(3) The vortex diffuser exhibits a higher induction than that of the nozzle type diffuser. However, the mean air speed in the room served by a vortex diffuser system is generally lower than that with nozzle type diffusers.

(4) Both the vortex diffuser and nozzle type diffuser can be considered to have condensation-free characteristics, although some vapor film was observed in the early stage of a "hard start up" test for the nozzle type diffuser.

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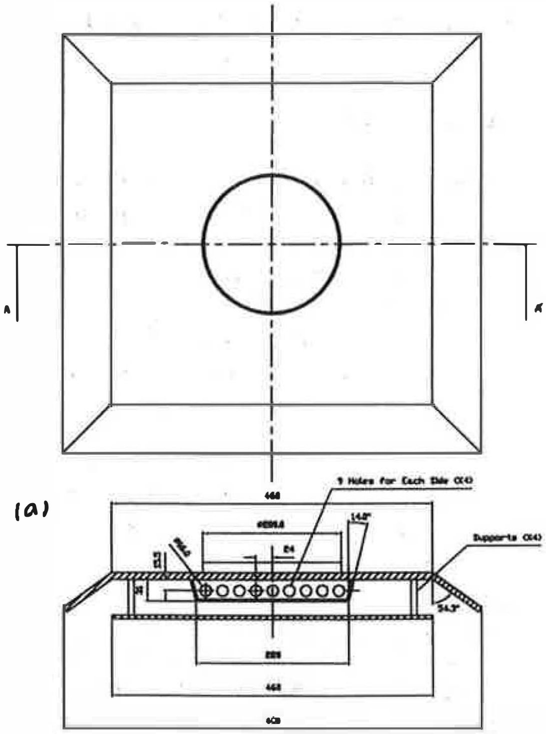

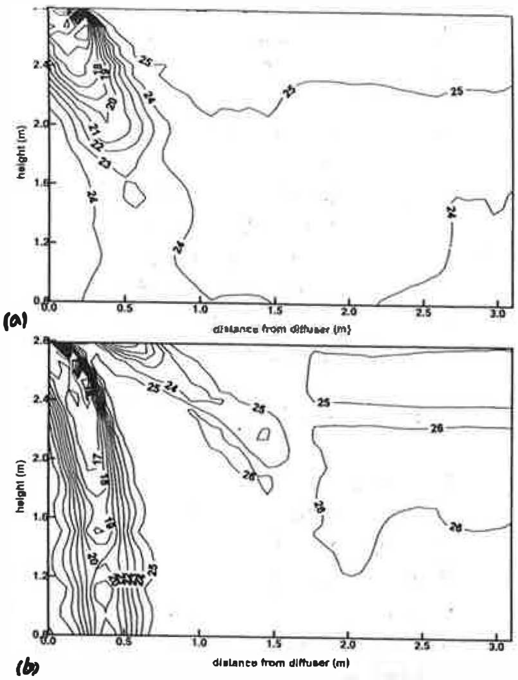

Figure 3.Measurement results of multi-cone circular diffuser, (a) case 2,(b) case 3

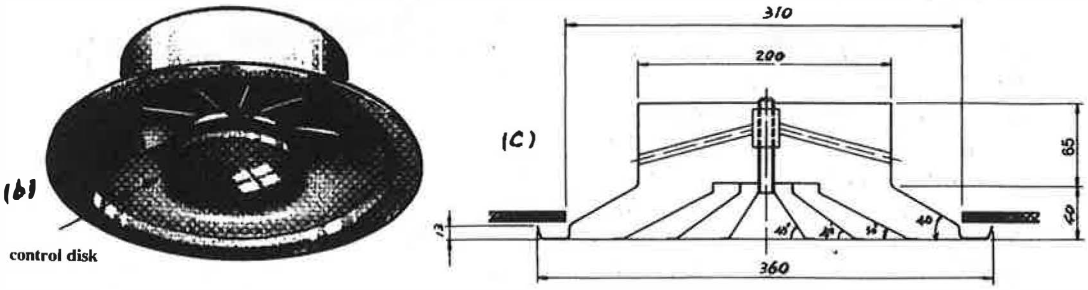

Figure 1. (a) The four-way nozzle type diffuser, (b) The vortex Diffuser, (c) The multi-Cone Circular Diffuser

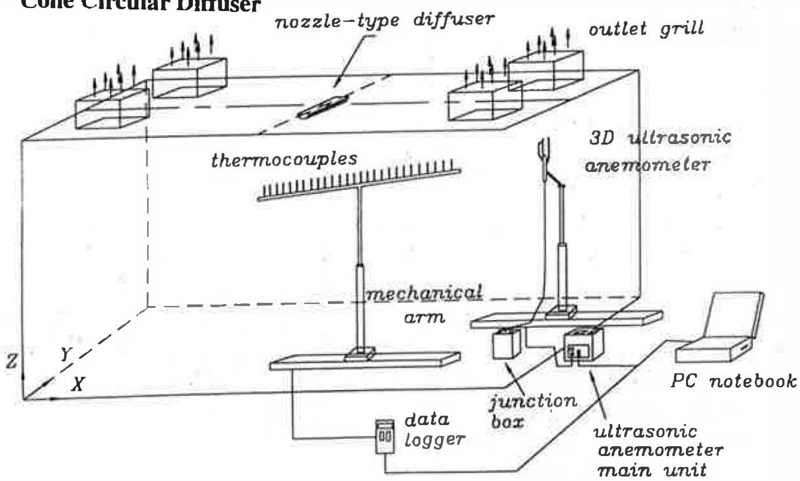

Figure 2.Schematic of measurement system.

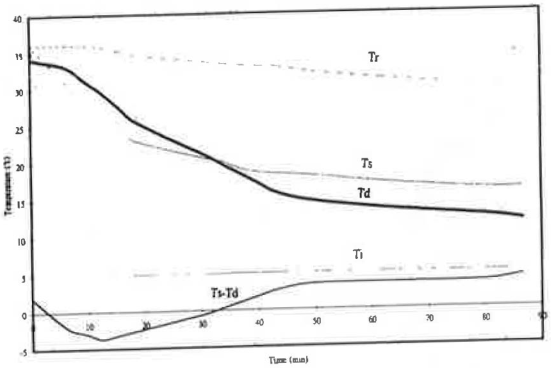

Figure 6.Condensation test for nozzle type diffuser (T_s = diffuser surface temperature, T_i = supply air temperature, T_r = room temperature, T_d = dew point temperature).

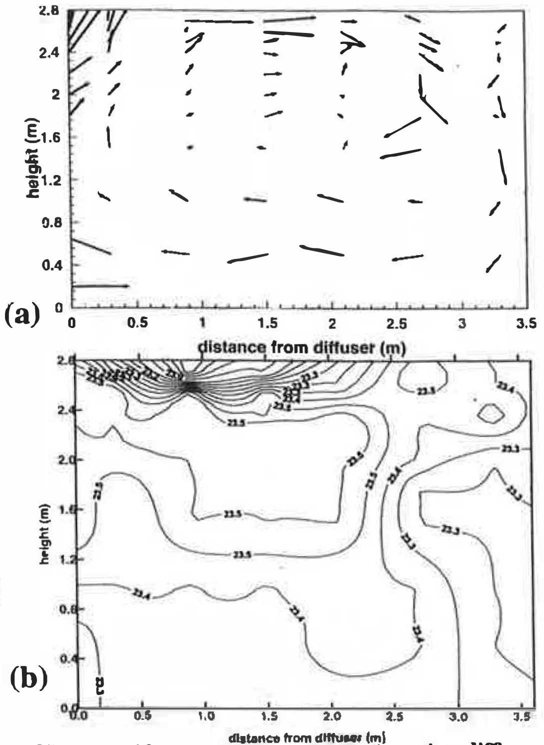

Figure 4.Measurement results of vortex diffuser (a) velocity vectors of case 5 with supply flow rate of 70.8 l/s (150cfm), (b) temperature distribution of case 5

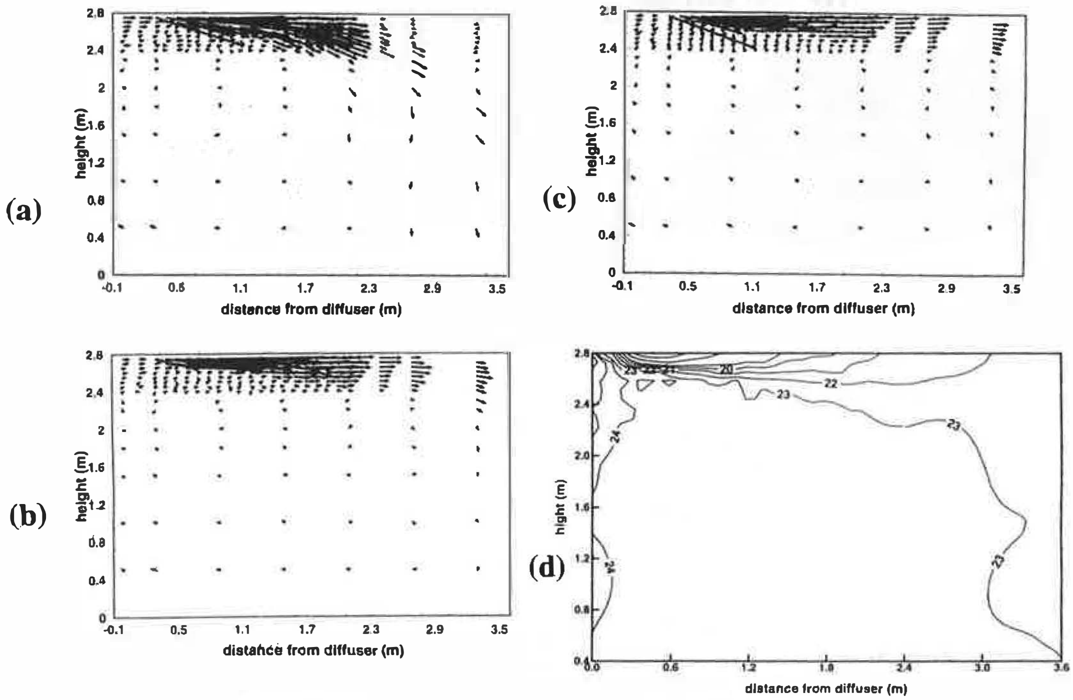

Figure 5.Measurement results of nozzle type diffuser (a) velocity vectors of case 6; (b) velocity vector of case 7 (c) velocity vector of case 8; (d) temperature distribution of case 8.