VENTILATION SYSTEM PERFORMANCE

11th AIVC Conference, Belgirate, Italy
18-21 September, 1990

Paper 16

AIR CURTAINS FOR INFILTRATION CONTROL - A COMPUTATIONAL FLUID DYNAMICS ANALYSIS

J K-W Lam, K G Ruddick and G E Whittle

Arup Research and Development

13 Fitzroy Street
London W1P 6BQ
UK.

SYNOPSIS

An investigation has been carried out using computational fluid dynamics methods to study the performance of an air curtain at the door of a heated building. A number of operating conditions have been studied and observations are made on the effectiveness of infiltration control and energy use. Comparisons are also made with previously published design data and results from an accepted infiltration analysis.

It is shown that the calculation method generates plausible and very detailed results which conform well to physical interpretation.

It was found that at too low a supply velocity the air curtain was deflected to outdoors and the energy supply was wasted. Similarly, too high a supply temperature also prevented the air curtain operating correctly.

With the air curtain operating successfully the predicted heat transfer characteristics indicated an efficient use of energy and acceptable indoor temperatures, but with air speeds of up to 2.5 m/s at low level within the building.

1. INTRODUCTION

Air curtains are used to reduce unwanted infiltration through open doors whilst maintaining easy access for people and vehicles. They provide the potential for improved comfort, contaminant and moisture control, and energy efficiency. Their performance depends on a wide range of factors which include: the geometry of the door opening and configuration of curtain, the temperature difference across the opening, wind effects, the ventilation provision in the building, and the momentum and temperature of the discharge air.

In most cases design engineers and suppliers specify and install system which operate successfully as measured against subjective criteria. However, the increasing requirements for higher standards of environmental control and quality and a need to minimise energy use demands a greater attention be focussed on understanding the physical mechanisms involved in order that performance can be optimised.

In this paper a computational investigation is carried out using fluid dynamics to study the performance of an air curtain at a number of operating conditions. Observations are made on the effectiveness of infiltration control and energy use, and comparisons are made with previously published design data.

2. BACKGROUND AND LITERATURE REVIEW

Mott[1] has discussed the principles behind the operation of air curtains and has outlined the basic information required for design. The equations used are those derived from momentum conservation. The curtain is initially discharged at an angle causing the jet to prescribe a parabolic trajectory when subjected to the pressure gradient generated across the opening. It is the change in horizontal component of momentum which offsets the flow which would otherwise occur through the opening. A typical discharge angle is 15°, though this can increase substantially for certain applications.

For a given angle of discharge optimum performance is obtained by taking the return air from opposite the position of discharge, for example through a floor grille in the case of a downward vertical air curtain. However, it is pointed out that this optimum configuration is not always the most practical. Where a return duct is not used then the curtain will part into two streams at floor level. It is stated that it is for the designer to decide in what proportion the curtain should be divided.

Wolfberg[2] describes an air curtain installation in a department store in Minnesota, USA. The doorway is 2.3m height and it is claimed that the door remains open in any weather conditions between the design outdoor temperatures of -28°C and +35°C.

In Hayes and Stoecker[3] the heat transfer characteristics of air curtains are identified. An analysis of performance was carried out based on the fundamental momentum and energy conservation equations, and findings were compared to laboratory experiments.

An important factor in the design of the curtain was found to be the level of turbulence at the discharge. Heat transfer across the air curtain is very dependent on turbulent exchange, a relatively high turbulence level (8%) indicated an inferior performance to that of a lower level (2%).

The trajectory of the jet forming the curtain was identified using smoke as a tracer and by measurements using hot-wire anemometers. For isothermal tests good agreement was found between the measurements and the theoretical analysis. Again for isothermal tests, the pressure differences across the curtain were measured for discharge velocities between 3.0m/s and 15.2m/s and for discharge angles of 15° and 30°, and were found to agree with the theory within an 8% tolerance.

For non-isothermal tests the agreement between experiment and theory was slightly less satisfactory showing that jet deflection, for example, was over-predicted. This was believed to be due partly to the variation of temperature difference across the curtain, with height, caused by internal stratification. Errors were broadly in the range 5 to 15%.

Regarding heat transfer, which was expressed in dimensionless form by the ratio of Nusselt no. to the product of Reynolds no. and Prandtl no., the parameter H/b_o was found to be the most influential (where H is the door height and b_o the slot width). The dimensionless heat transfer relates heat transfer across the curtain to the energy supplied by the curtain. At high values of deflection modulus (essentially based on the ratio of jet inertia to buoyancy) both the discharge angle and the deflection modulus were found to have little effect on dimensionless heat transfer. At this condition the heat transfer coefficient is directly proportional to the discharge velocity. As expected, where jet buoyancy became important (low values of deflection modulus) the dimensionless heat transfer increases substantially. This corresponds to increasing deflection of the curtain leading eventually to the curtain breaking down and being swept aside by flow through the open door. In most cases the minimum value of dimensionless heat transfer occurs at a discharge angle of 15° towards the warm side, although the discharge angle may be increased to 30° without significant increase in dimensionless heat transfer.

In a further paper, which addresses design needs, Hayes and Stoecker[4] describe relationships which allow the discharge velocity and the resulting heat transfer coefficient to be identified for the ratio of H/b°, and an equation is presented to correct for door height and temperature difference. Specific design recommendations are also made regarding the need to minimise turbulence at the discharge of the curtain, and that thick curtains reduce heat transfer rate by approximately 10%. Also because of susceptibility to wind pressures, air curtains should not be installed on opposite sides of the same building. The recommendation is that they should serve a space which is otherwise well sealed. In real operation, a change in wind pressure will cause the curtain to deflect allowing air to pass until the pressure gradient across the curtain adjusts. Thus under dynamically-varying wind conditions the heat transfer rate is expected to be greater than that predicted.

The economics of air curtains are considered by Asker[5]. Curtain efficiency is defined as the ratio of the thermal load imposed by an unprotected doorway compared to that of the air curtained doorway. A number of design points are listed which should be considered to ensure satisfactory control:

- air curtained doors should be located in one wall only;
- whilst ensuring safe passage the area of doorway should be minimised;
- air curtains do not perform well in buildings where a substantial excess of supply or extract air is provided;
- air flow in a room protected by a curtain should be directed to be parallel to the wall in which the curtain is located.

It is claimed that the use of air curtains to prevent access to food and pharmaceutical plants by most flying insects can be very successful provided the velocity is at least 4.0m/s.

The US National Sanitation Foundation Standard No. 37[6] specifies requirements for air curtains for entranceways in food establishments. Curtain thicknesses and velocities are specified for different types of openings. For example, at a service entrance the minimum curtain discharge thickness should be at least 75mm and the velocity should not be less than 8m/s at a height of 0.9m above the floor.

Learmonth[7] discusses a number of installations and states that for comfort the jetted velocity should not exceed about 6.0m/s, although in industrial applications this could be higher. A sketch is shown claiming to show a representation of the earliest patent application (dated 1904) related to an air curtain.

Some design guidance on the installation of both vertical and horizontal air curtains is provided by Creek[8]. Operating benefits are claimed by allowing on-site flexibility:

- in setting the flow rate and discharge angle of the curtain;
- by providing automatic control of supply temperature;
- by providing on-off fan operation in conjunction with an automatic door.

Baturin[9] describes a practical and theoretical analysis of the air curtain. The effectiveness of the curtain depends on the ratio of the momentum of the curtain discharge and that of the potential air flow through the door. A substantial amount of practical information is given supplemented by examples of the design process. Results of measurements on floor- and side-discharge air curtains including one at a coach depot on the Moscow underground are shown. The door heights are 2.2m to 4.0m.

The flow of outside air through the unprotected doorway, which was at velocities of 2.6 to 3.9m/s, was approximately halved by the curtains. The curtains used typically 11% of the flow rate which would otherwise pass through the unprotected door. Examples are also shown of temperatures within a workshop both with and without an air curtain operating. Substantial improvements in temperatures are indicated by the operation of the curtain.

Developments in air curtains in the USSR and in Sweden are described where the emphasis is on utilising unheated (outside or inside) air[10]. An example of air speed and temperature profiles in the area of a door is shown, indicating the benefits of air curtains.

Simper[11] discusses novel uses for air curtains for, for example, control of dust in quarrying and foundry applications and deflection of raindrops from open-roofed areas. For the latter, power ratings of 0.8 to 1.2kW/m^2 of roof opening are indicated.

Air curtains are claimed to have an energy efficiency of 40 to 95% compared to a closed door[12]. An example is shown of measured temperatures 4m and 20m inside a building in which a 4m x 4m door is protected by an air curtain. With an outside temperature of 0°C and 4Pa under-pressure in the building the temperature at both locations were 6 to 11°C without the curtain operating and 15 to 16°C with the curtain.

Ligtenberg[13,14] reports a combined laboratory and computational approach to investigating air curtain performance, with the emphasis on product development. Wind tunnel tests and computational fluid dynamics analyses were undertaken. A main finding, confirming previous work, was the importance of minimising turbulence intensity at the discharge. A halving of energy losses by operation of a well-engineered air curtain is claimed.

3. ANALYSIS - THE COMPUTATIONAL METHOD

The computational fluid dynamics code, AIRFLO, has been used for these simulations. The code is an implicit-integration, finite-volume Navier-Stokes solver, which has been developed within Arup R&D. It is configured particularly for application to natural and forced convection air flow problems in buildings[15].

The code uses a pressure-coupled formulation involving the simultaneous solution of the momentum equations for velocity components, and the mass continuity equation for pressure. The equation for pressure is derived by substituting the momentum equations (for velocity components) into the statement of mass continuity[16]. A re-arrangement of terms results in a diagonally-dominant equation for pressure. The mass continuity equation is, therefore, exact thus obviating the need for a pressure-correction algorithm such as SIMPLE or its derivatives[17].

The velocity components are calculated at cell faces using a staggered grid. A standard linearization of the equations is used, where the linearized coefficients comprise the convection and diffusion fluxes over the staggered momentum cells and scalar temperature cells, as appropriate.

In the work reported here the energy equation is solved in a segregated form where the act of linearization de-couples it from the momentum and mass continuity equations. Although very simple to implement, this can result in the need for substantial under-relaxation to retain stability, and hence the well-known difficulties of delayed convergence in high Rayleigh number buoyant flows. Although not implemented here, a Newton-Raphson linearization of the energy equation would be expected to significantly enhance the convergence rate for buoyant flow[18]. Buoyancy force is modelled using the Boussinesq approximation.

The 'hybrid' differencing scheme[17] is used, giving central differences at low cell Peclet number and upwind differences, with diffusion suppressed, for high Peclet number. A point Gauss-Seidel solver with multi-directional sweep is implemented for all equations. Appendix A shows the finite-volume form of the equations. The code allows the equations to be solved in either steady-state or transient form, where in the latter a simple Euler integration with forward-differencing in time is used.

4. DESCRIPTION OF SIMULATIONS

The test problem was chosen to represent a shopping mall in winter with an opening to outside protected by a warm air curtain projected downwards from above the opening - a typical application of air curtains encountered by design engineers.

The mall is 15m long, 4m high and 4m wide with an opening of height 2.5m extending across the full width of the mall (see Fig. 1). There is assumed to be negligible leakage of air from the mall except through the door opening. The case of zero wind pressure was chosen. Strong or gusting winds or high leakage would be expected to degrade the performance of the air curtain.

The air curtain unit draws air in at the top, which is then heated and projected downwards and towards the warm mall at an angle of 15° to the vertical. The discharge slot width is 100mm.

Air in the mall is heated by contact with the floor, ceiling, end wall and the shops on either side. This was modelled by assuming a constant surface temperature for these surrounding surfaces (extending from 1m from the opening to the end wall) and a convective heat transfer coefficient of

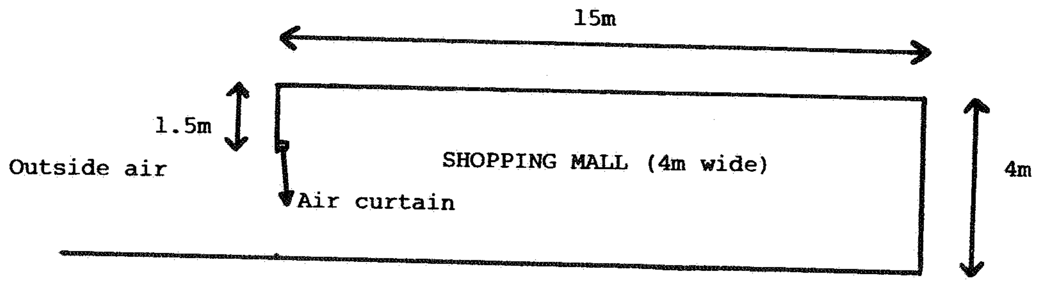

Figure 1. Geometry of test problem (side elevation)

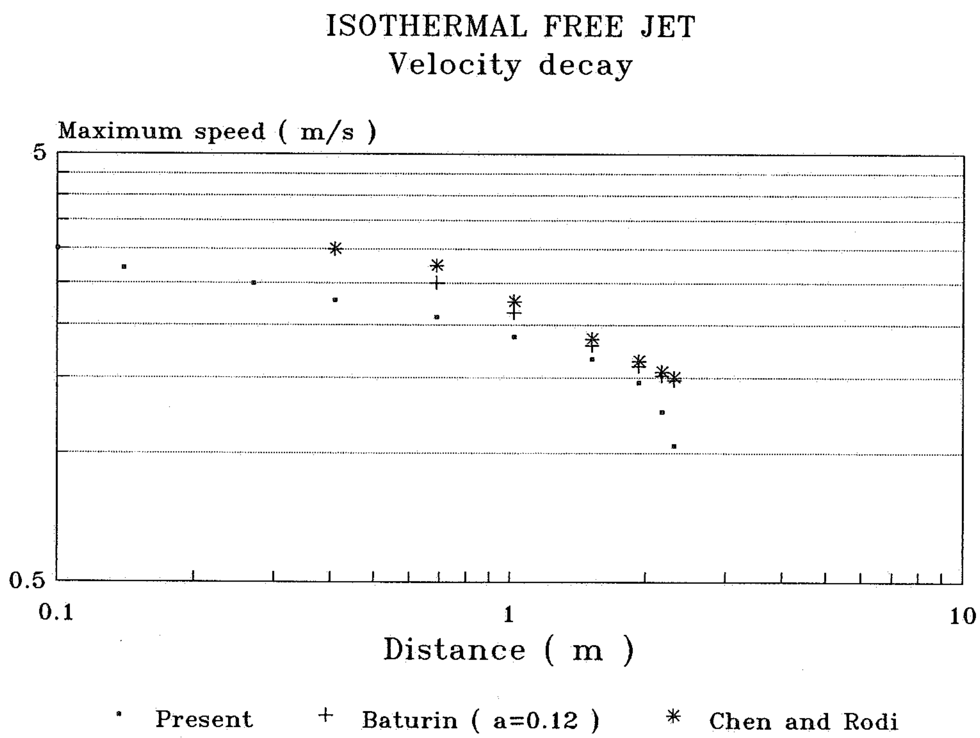

Figure 2. Projection of isothermal jet

$3W/m^2/^{\circ}C$. The outdoor air temperature is $-1^{\circ}C$ and the 'outdoor' boundary is taken 5m from the opening. A fixed external pressure boundary condition is applied here where the mass flow rate across the boundary is based on a constant (equal to the product of density and flow area) multiplied by the linear pressure difference.

The mall was modelled two-dimensionally with attention concentrated on the projection of the air curtain. Three-dimensional edge effects were neglected except for the heat flux from the sides of the mall. The mesh comprised 100 cells horizontally and 56 cells vertically with a maximum cell aspect ratio of twelve at the far end of the mall remote from the door. In the horizontal direction, two cells were used to represent the air curtain discharge. A constant and uniform eddy-viscosity of $1.8x10^{-3}Ns/m^2$ has been used.

From initial conditions of zero velocity and uniform temperature, approximately 1,500 steady-state iterations were necessary to reduce mass and energy flux errors through the doorway down to 2 to 5%. The simulations were carried out on an 8 Mbyte SUN SPARCstation which achieved a throughput of approximately 600 iterations per hour.

5. RESULTS AND DISCUSSION

Six simulations were performed including one (A) to assess the isothermal performance of the air curtain and one (B) to simulate conditions if the opening were not 'protected' by an air curtain. The temperature (Tsupp) and velocity (Vsupp) of the supply air were varied to determine the effect on the performance of the air curtain.

For each simulation the heat lost by convection and diffusion at the opening (Qlost), heat supplied by the air curtain (Qsupp) and heat lost by the surrounding shops (Qload) to the mall by convection were recorded. For an energy balance,

$$Qlost = Qload + Qsupp$$

The thermal comfort conditions for people inside the mall are, for the purpose of these simulations, quantified by the maximum air speed (Umax) and the maximum (Tmax) and minimum (Tmin) temperatures at a vertical plane 5m from the opening up to a height of 1.8m.

The conditions and results of each simulation are shown in Table 1.

	CONDITIONS		HEAT FLUXES			MALL CONDITIONS		
	Vsupp (m/s)	Tsupp (°C)	Qlost (kW/m)	Qload (kW/m)	Qsupp (kW/m)	Umax (m/s)	Tmin (°C)	Tmax (°C)
A	3.0	-	-	-	-	1.8	-	-
B	No supply		1.96	1.94	0.0	0.37	-0.6	2.3
C	3.0	20	7.47	2.15	5.65	0.33	-0.7	2.3
D	3.5	20	8.69	2.37	6.65	0.40	-0.6	2.1
E	4.0	20	2.20	0.50	1.58	2.50	15.6	17.0
F	4.0	30	14.50	2.39	12.60	0.41	-0.6	2.1

Table 1. Summary of test conditions and results

The isothermal simulation was performed to allow comparison with established data for the velocity decay of free plane jets. Figure 2 shows the projection of the isothermal jet for this geometry with discharge angle of 15°. The velocity decay from Baturin[9] is found from:

$$U_x = U_o\, 1.2\, [(2aX/b) + 0.41]^{-0.5}$$

where U_x = velocity at distance X (m/s)
U_o = initial jet velocity (m/s)
a = a constant for the type of outlet (0.12)
b = slot width (m)

and that from Chen and Rodi[19] from:

$$U_x = U_o\, 2.4\, (X/b)^{-0.5}$$

It can be seen that in the proximity of the discharge the predicted velocity is lower than expected. This indicates an enhanced and unrealistic diffusion which is believed to be due to the relatively poor mesh resolution and to associated factors related to turbulence modelling and differencing scheme. At a large distance, velocity is seen to fall off rapidly as the jet becomes influenced by the proximity of the floor such that it can no longer be considered as 'free'.

For an unprotected opening (B) cold air penetrates

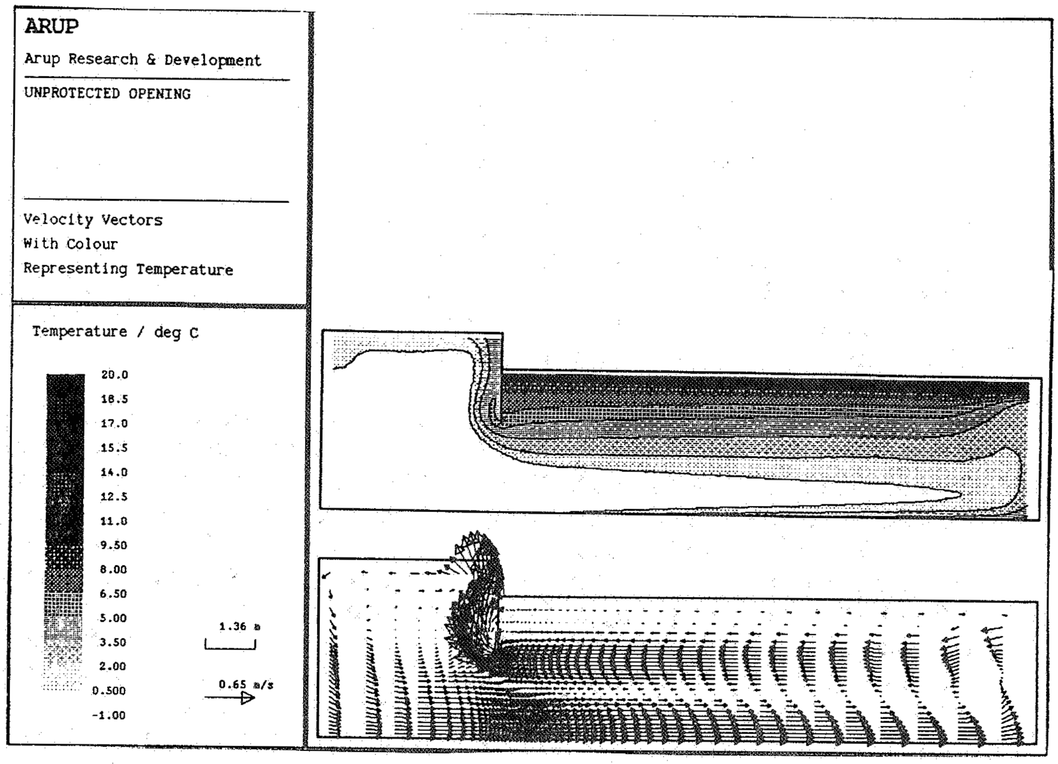

Figure 3. Unprotected opening (simulation B)

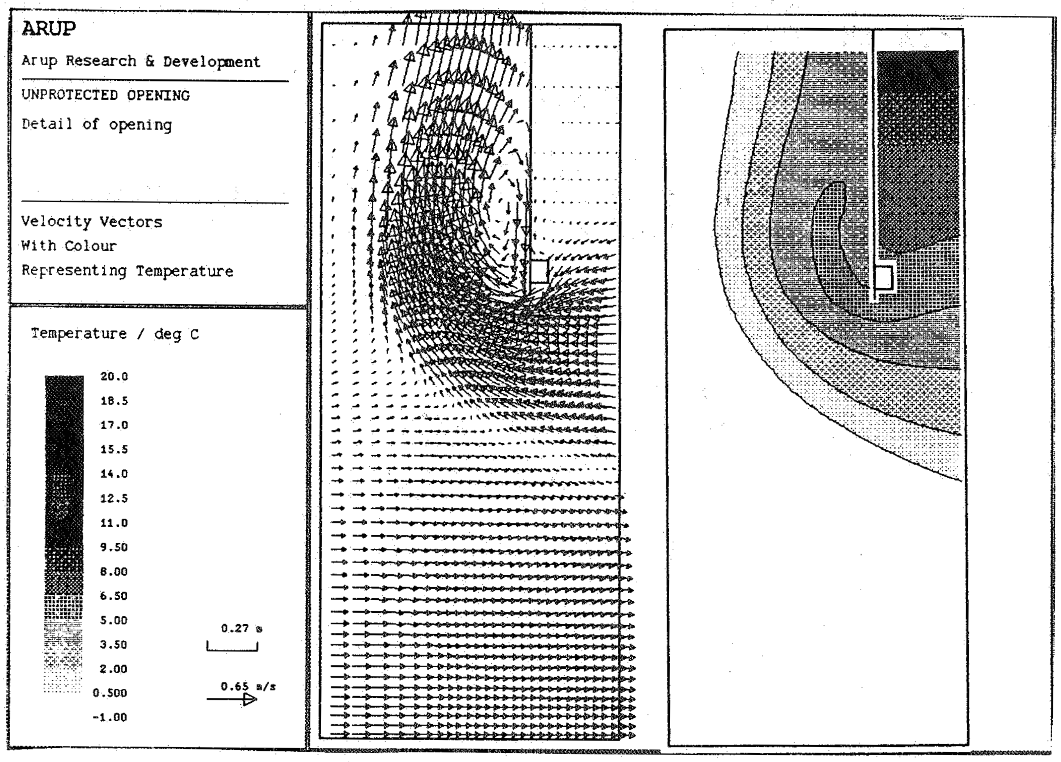

Figure 4. Unprotected opening (simulation B) - detail

at low level far into the mall causing unacceptable conditions for occupants. An equal quantity of air warmed by the surroundings of the mall escapes from the upper part of the opening. Figures 3. and 4. show the velocity and temperature fields (with only one-in-four velocity vectors plotted in Figure 3. to simplify presentation). There is a tendency for stratification of warmer air above the level of the opening and it can be seen that in the plane of the door the neutral-plane is approximately two-thirds the way up the height of the door.

The maximum velocities through the open door are found to be 0.3 m/s (at -1°C) into the mall at low level and 0.7 m/s (at 5 to 6°C) to outdoors at high level. These figures compare to a maximum velocity of 0.38 m/s indicated by a numerical infiltration calculation which assumes an external temperature of -1°C and a <u>uniform</u> internal temperature of 5°C. Using the latter model, the assumption of uniform internal temperature locates the neutral plane at the mid-height of the door hence generating a symmetric inflow / outflow velocity profile. The total flow rate and energy flux through the door indicated by the CFD code was 0.35 m^3/sm and 1.96 kW/m, and by the infiltration model was 0.32 m^3/sm and 2.31 kW/m, respectively. A simpler calculation from the CIBSE Guide[21] indicates a flow rate of 0.43 m^3/sm. The comparison is encouraging and gives confidence in the modelling approach.

The infiltration model, which uses ten vertical segments through which the flow is calculated, is based on that described by Liddament[20], where the flow equation is:

$$Q = 1.29\ C_D\ A\ \Delta P^{0.6}$$

where Q = volume flow rate (m^3/s)
C_D = discharge coefficient (0.6)
A = area (m^2)
ΔP = pressure drop (N/m^2)

Figures 5. and 6. show the velocity and temperature fields for simulation C with Vsupp= 3m/s, which corresponds to the initial design estimate for the outlet velocity. It can be seen though that the air curtain is swept aside by the pressure gradient generated by 'stack' effect. Cold air penetrates the mall virtually unimpeded and conditions are similar to those for an unprotected opening. The energy provided to heat air and drive the fan in the air curtain unit is entirely wasted as the warm

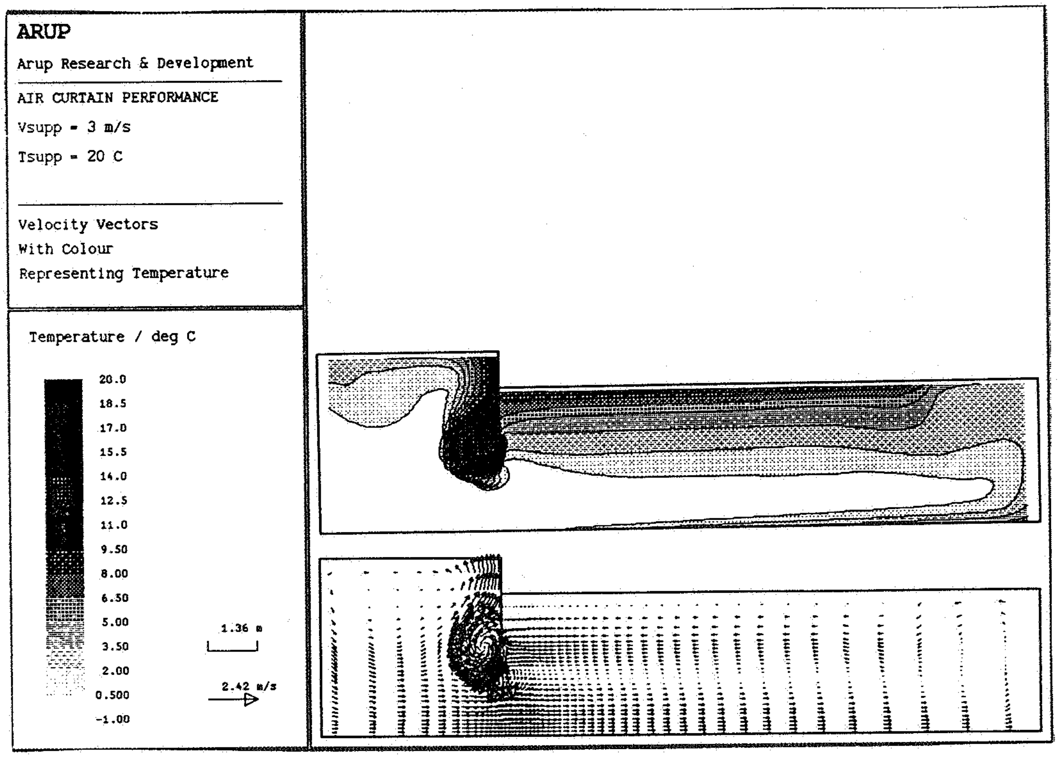

Figure 5. Air curtain with Vsupp=3m/s, Tsupp=20°C (simulation C)

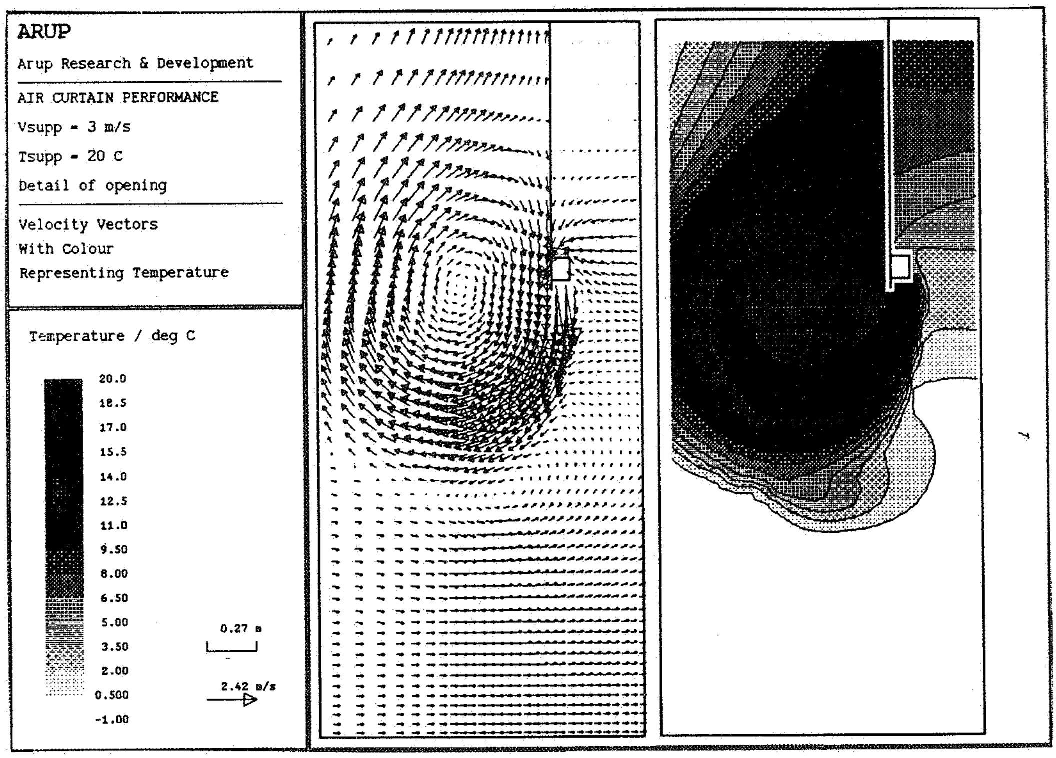

Figure 6. Air curtain with Vsupp=3m/s, Tsupp=20°C (simulation C) - detail

air jet is deflected and lost outside the mall.

Simulation D with a slightly higher supply velocity of 3.5m/s gives a similarly poor behaviour to simulation C.

Figures 7. and 8. show the velocity and temperature fields for simulation E with a higher supply velocity of 4m/s. Here the air curtain is seen to behave as desired. The curtain extends to the floor preventing the large infiltration found in simulations B and C. Incoming air is warmed by mixing with the curtain to give an acceptable and fairly uniform mall temperature of 16°C. The air curtain does, however, produce high air speeds (2.5m/s) within the mall which may cause some discomfort. The efficiency is good since the energy expended in heating the supply air is not wasted but serves to keep air within the mall at a reasonable temperature.

Simulations D and E indicate that the critical supply velocity for correct operation of the air curtain is somewhere between 3.5m/s and 4m/s and that heat transfer across the curtain for a supply velocity of 4 m/s is about 2.2 kW/m. This compares with a critical supply velocity of 2.1 m/s from Hayes and Stoecker[4]. For the higher supply velocity of 4.0 m/s, Hayes and Stoecker indicate a heat transfer rate of 3.6 kW/m (see Appendix B). Not surprisingly the agreement is not perfect, but it is encouraging considering the complexity of the flow and the difficulty of the theoretical analysis on which these predictions of critical supply velocity and heat transfer are based.

Simulation F was performed with a supply velocity of 4m/s, but with a raised supply temperature of 30°C. The additional buoyancy forces caused the jet to break away from the floor and to be swept aside and exhibit similar performance to simulation C.

Clearly care should be taken when sizing and selecting air curtains since the implications, on both performance and energy use, of incorrect selection are dramatic. However, further work is needed to identify the extent to which the present results may be contaminated with numerical discretisation errors.

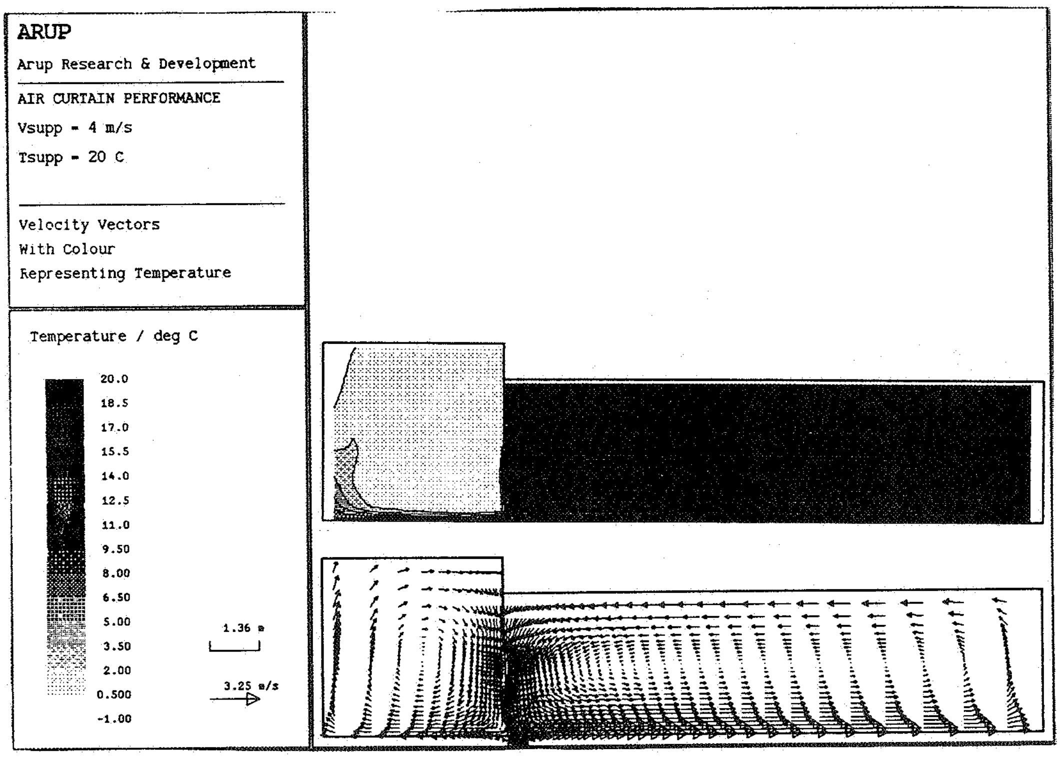

Figure 7. Air curtain with Vsupp=4m/s, Tsupp=20°C (simulation E)

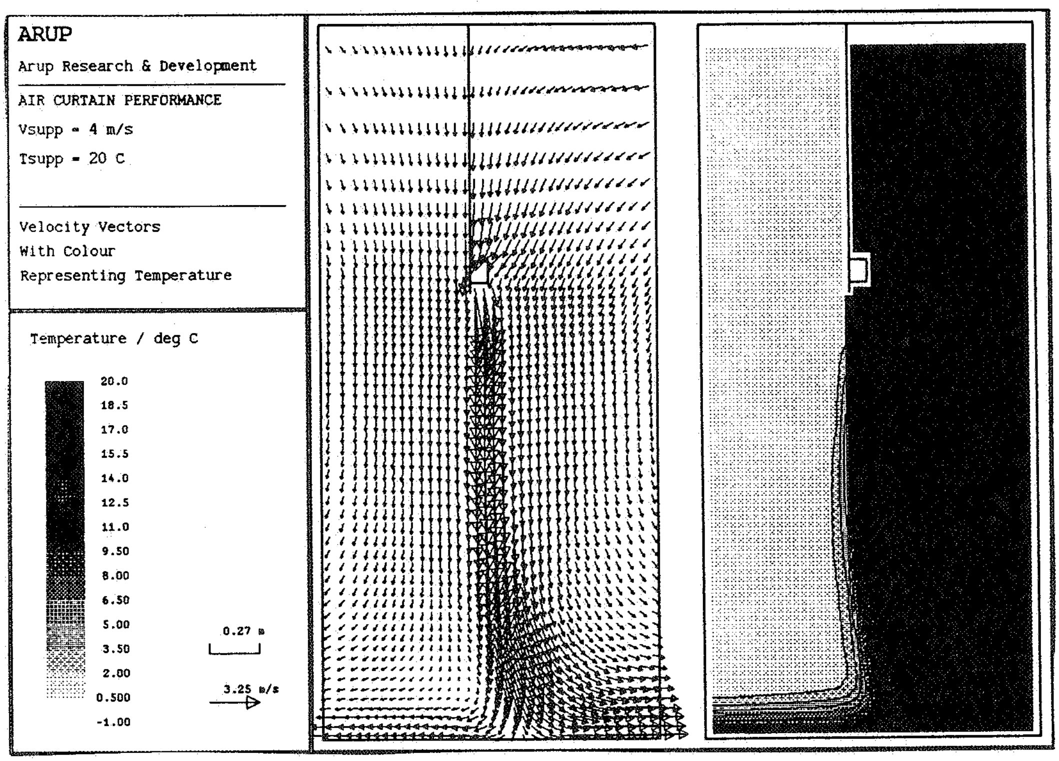

Figure 8. Air curtain with Vsupp=4m/s, Tsupp=20°C (simulation E) - detail

6. CONCLUSIONS

An investigation has been undertaken of air curtain performance using computational fluid dynamics methods, and findings have been compared to previously published design data.

Initial predictions of flow through an open door and the comparison with an accepted infiltration analysis show the calculation method generates plausible and very detailed results which conform well to physical interpretation.

For an external temperature of -1°C the heat loss through the unprotected open door was found to be 1.96 kW/m giving a mean internal temperature in the occupied zone 5 m distant from the door of 0.9°C. With the air curtain operating, but not successfully (since the air curtain was deflected to outdoors), the heat loss increased to between 7 and 9 kW/m without any improvement in internal conditions. With the momentum in the discharge from the unit increased such that the air curtain operated correctly the heat loss was 2.20 kW/m and an internal temperature of 16.3°C was achieved.

The corresponding heat transfer coefficients are 0.85 $kW/m^2 {}^oC$ without a curtain, 3.6 $kW/m^2 {}^oC$ with the curtain operating incorrectly, and 0.051 $kW/m^2 {}^oC$ with the curtain operating successfully. Clearly, when the air curtain is operating incorrectly the energy supplied via the curtain is totally wasted.

With the air curtain operating successfully, an increase in the temperature of the supply air from 20°C to 30°C caused the curtain to be deflected to outdoors resulting in a very high energy use and no improvement of temperature conditions beyond those generated without a curtain present. This behaviour was due to the increased buoyancy force.

It should be noted that when the air curtain was operating successfully high air speeds (2.5 m/s) were generated at low level within the building.

It was found that for the configuration studied, the critical momentum flow at the discharge of the air curtain should be significantly higher than that indicated by the design method, although further work is needed to establish mesh independence and freedom from contamination by numerical discretisation errors.

7. APPENDIX A - Discretized CFD equations

7.1 Discretized momentum equation in X direction

$$a_P U_P = a_N U_N + a_S U_S + a_E U_E + a_W U_W + a_H U_H + a_L U_L + b_U$$

where a = convection / diffusion fluxes
- a function of (C, D)

C = convection (mass) flux
D = diffusion flux
$a_P = a_N + a_S + a_E + a_W + a_H + a_L + V\,\rho_P / dt$
b_U = momentum source + $V\,\rho_P^{\circ}\,U_P^{\circ} / dt$
V = cell volume
ρ = density
U = computed velocity
U° = velocity at previous time step
dt = time step

Subscript P refers to the in-cell variable value, and subscripts N, S, E, W, H and L refer to surrounding north, south, east, west, high and low variable values. Superscript 'o' refers to the variable value at the previous time step.

The form of the function defining a_N, etc, follows from the selection of the discretization scheme, eg. upwind, central, hybrid differencing.

7.2 Discretized mass continuity equation

$$(\rho_P - \rho_P^{\circ})\,V / dt + C_N + C_S + C_E + C_W + C_H + C_L + b_c = 0$$

where b_c = mass source

7.3 Discretized energy equation

$$a_P h_P = a_N h_N + a_S h_S + a_E h_E + a_W h_W + a_H h_H + a_L h_L + b_h$$

where h = enthalpy (Cp T)
Cp = specific heat
T = temperature
b_h = energy source + $V\,\rho_P^{\circ}\,h_P^{\circ} / dt$

8. APPENDIX B - Air curtain design procedure

When selecting an air curtain the design engineer must consider a number of parameters affecting performance:

- discharge configuration - overhead, side or

floor;
- warm- and cold-side temperatures;
- slot width, supply velocity and, hence, flow rate;
- supply temperature;
- discharge angle.

Overhead discharge is usually preferred over floor discharge for reasons of comfort and for both ease of installation and contamination control.

In the case of overhead discharge, Hayes and Stoecker[4] suggest a procedure for estimating the supply momentum required for a given slot width, b_o, door height, H, and design inside, Tw, and outside, Tc, temperatures. From their theoretical and experimental work the critical supply velocity, U_o'', to ensure that the air curtain reaches the floor is estimated as:

$$U_o'' = 3.4 \text{ m/s}$$

and the coefficient for the rate of transfer of sensible heat across the curtain, h_o'', as:

$$h_o'' = 0.071 \text{ kW/m}^{2\circ}\text{C}$$

under the conditions H= 2.13m, H/b_o= 25, Tw= 23.9°C, Tc= -23.3°C, with discharge downwards and towards the warm mall at an angle of 15° to the vertical.

Applying a correction factor, K_B, to extrapolate to an air curtain designed for H= 2.5m, b_o= 0.1m, Tw= 16°C, Tc= -1°C, where,

$$K_B^2 = (H/2.13m)*(Tw_{(abs)}/Tc_{(abs)} - 1) / (297K/250K -1)$$

$$Tw_{(abs)} = Tw + 273$$

$$Tc_{(abs)} = Tc + 273, \text{ and}$$

$$U_o' = K_B * U_o'',$$

gives K_B = 0.62, and

$$U_o' = 2.1 \text{ m/s}$$

To ensure that the air curtain establishes itself properly after any disruption and to allow a margin for error a "velocity factor", F= $2^{0.5}$ = 1.4, is employed, giving,

$U_o = F * U_o' = 1.4 * 2.1$ m/s

$= 2.9$ m/s

The rate of sensible heat transfer across the curtain, Q, is then :

$Q_{2.9m/s} = (F * K_B * h_o'') * H * (Tw - Tc)$

$= 1.4 * 0.62 * 0.071$ kW/m^{2o}C $* 2.5$ m $* (16^oC + 1^oC)$

$= 2.6$ kW/m

For the higher supply velocity of 4m/s,

$Q_{4m/s} = 2.6$ kW/m $* (4$m/s $/ 2.9$m/s $)$

$= 3.6$ kW/m

9. REFERENCES

1. MOTT, L.F.
"Design of the air curtain. Refrigeration, Air Conditioning and Heating", pp.1-23, September 1961.

2. WOLFBERG, H.A.
"Get 1350 tons of cooling from 875hp in shopping-parking facility". Heating, Piping and Air Conditioning, May 1964.

3. HAYES, F.C. AND STOECKER, W.F.
"Heat transfer characteristics of the air curtain". ASHRAE Trans. vol.75, pt2, pp.153-167, 1969.

4. HAYES, F.C. AND STOECKER, W.F.
"Design data for air curtains". ASHRAE Trans. vol.75, pt2, pp.168-180, 1969.

5. ASKER, G.C.F.
"What, where and how of air curtains". Heating, Piping and Air Conditioning, June 1970.

6. NATIONAL SANITATION FOUNDATION, Standard No. 37, "Air curtains for entranceways in food establishments", 1970, 1980, 1985.

7. LEARMONTH, R.A.

"The use of air curtains". H & V E, September 1970

8. CREEK, H.P.
"Warm air curtains". H & V E, October 1971.

9. BATURIN, V.V.
"Fundamentals of industrial ventilation", Chapter 15 - Air curtains, Pergamon Press, Oxford, 1972.

10. "Doorway protection with an unheated air curtain", Building Services Engineer, vol.43, April 1975.

11. SIMPER, J.I.
"New uses for air curtains". Building Services Engineer, vol.43, November 1975.

12. "Air curtains - Energy savings for industry". HAC December 1986 / January 1987.

13. LIGTENBERG, P.J.J.H.
"Economical air curtains". Biddle Air Curtains Ltd, Enfield, Middlesex, UK.

14. LIGTENBERG, P.J.J.H.
"It's curtains for cold air". HAC pp.30-32, October 1989.

15. HOLMES, M.J., LAM, J.K-W., RUDDICK, K.G. AND WHITTLE, G.E.
"Computation of convection, conduction and radiation in the perimeter zone of an office space". Proc. Int. Conf. Roomvent 90, Oslo, June 1990.

16. ZEDAN, M., AND SCHNEIDER, G.E.
"A coupled strongly implicit procedure for velocity and pressure computation in fluid flow problems". Numerical Heat Transfer, vol.8, pp.537-557, 1985.

17. PATANKAR, S.V.
"Numerical heat transfer and fluid flow". Hemisphere Publishing Corporation, New York, 1980.

18. GALPIN, P.F., AND RAITHBY, G.D.
"Treatment of non-linearities in the numerical solution of the incompressible Navier-Stokes equations". International

Journal for Numerical Methods in Fluids, vol.6, pp.409-426, 1986.

19. CHEN, C.J. and RODI, W. "Vertical turbulent buoyant jets - A review of experimental data". Pergamon Press, Oxford, 1980.

20. LIDDAMENT, M.W. "Air infiltration calculation techniques - An application guide". Air Infiltration and Ventilation Centre, International Energy Agency, 1986.

21. CIBSE Guide. Air Infiltration, Table A4.5. London, 1986.

Discussion

Paper 16

C-A Roulet (LESO, Switzerland)

Since water is between 200 and 400 times more efficient than air to transport heat (the figure depending on the considered phenomenon), why insist on using air for that purpose? Water temperature conditioning may solve many problems encountered in air conditioning.

M J Holmes (Arup R & D, UK)

There is no general solution to cooling problems, the structural form may influence the air conditioning system, that is a concrete or steel frame. For example, the type of voids generated could mean a ceiling or floor systems are more appropriate - concrete favours floor supply. An obvious comment on water cooled equipment at desks could cause problems with flexibility.