Original Paper

Indoor Built Environment

Indoor Built Environ 2000;9:127–142

Accepted: May 5, 2000

The Use of Heat Pumps to Induce Airflow on Hot Days in Otherwise Passive Ventilation Systems – A Zonal Modelling Approach

Stephen A. Gage Philippe Ayres James Axon

The Bartlett School of Architecture, University College London, UK

Key Words
Heat pumps · Urban air pollution · Top-down ventilation

Abstract
This paper presents results from a wider study into providing displacement ventilation in urban areas by taking air into buildings from the top without the use of fans. Results from large scale experimental work are given. These results indicate that ventilation airflows can be induced using gravity chillers and heaters in conditions where this type of installation would otherwise fail. The paper also describes initial experiments undertaken to see how far the same equipment can be used for heat recovery. One test installation was modelled using a proprietary zonal model. A further zonal model was created to compare an assisted gravity system with a partially fan-driven system. This system failed on numerous occasions during a hot summer. However, fans are shown to be advantageous in winter heat recovery. A hybrid system is suggested for use under UK conditions.

Introduction

Indoor air quality has been the subject of intensive national and international research efforts because poor air quality is seen to be a health risk, reducing working efficiency and encouraging absenteeism. Much of the research has focused on developing natural ventilation as an alternative to mechanical ventilation and air conditioning. This is generally thought to be healthier than ventilation achieved by forcing air through ducted systems, especially when these systems re-circulate air in a building. Natural ventilation can offer direct local user control. Natural ventilation is quiet if there is no external noise. Natural ventilation can be induced without the need for electrical energy.

Providing natural ventilation in urban areas presents considerable problems. It has been demonstrated that high concentrations of atmospheric pollutants, especially fine particulates, occur at the lowest levels in urban street canyons and in car parking areas [1]. It has also been demonstrated that atmospheric pollution can be drawn into buildings through ventilation openings [2]. The problems are such that prototype special ventilators incorporating filters are being tested for natural ventilation systems [3]. There are other problems related to the nature of urban life. In many urban areas, the level of road noise makes it undesirable to provide natural ventilation by opening windows. Also, opening windows and louvres placed at low level can be an invitation to burglary.

The concentration of urban air pollution is variable in space and the lowest levels are found above the roofs of buildings and in sheltered back-land areas that do not contain pollutant sources. Ventilation engineers are advised to site air intakes to mechanical systems in these

KARGER
Fax + 41 61 306 12 34
E-Mail karger@karger.ch
www.karger.com

1420–326X/00/0094–0127$17.50/0

Accessible online at:
www.karger.com/journals/ibe

Prof. Stephen A. Gage
Faculty of the Built Environment, The Bartlett School of Architecture
University College London, Wates House, 22 Gordon Street
London WC1H 0QB (UK)
Tel. +44 20 7380 7504, Fax +44 20 7504 4831, E-Mail architecture@ucl.ac.uk

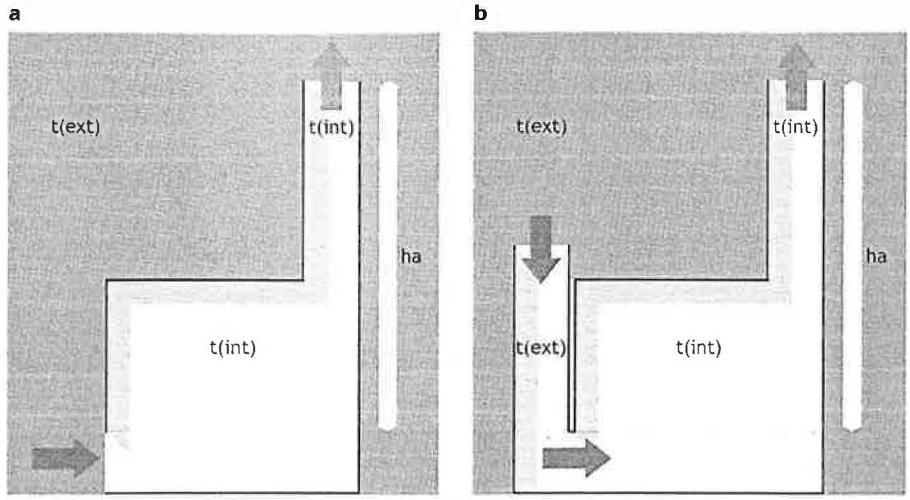

Fig. 1. Gravity displacement ventilation. **a** Conventional gravity displacement. Ventilation occurs when the internal temperature t(int) is greater than the external temperature t(ext). The rate of ventilation is proportional to the temperature difference and the height of the hot column of air, ha. **b** 'Top-down' gravity displacement. The principle is the same as **a**, but the intake duct increases the resistance of the system.

locations, and this advice is likely to become mandatory on an EC wide basis [4]. When buildings under different ownership face common back-land areas it is difficult to ensure that these areas will remain pollution-free; it is consequently more reliable to site air intakes at roof level. However, there is evidence that pollution can also be generated at this level, arising from plants on nearby rooftops and secondary transformation of gases as a result of photo-oxidation. Rooftop air intake is however regarded as best practice in mechanical ventilation systems. It has not been generally used in passive systems in the UK. The only examples of passive roof intake are traditional and modern split duct wind-catcher systems and, in hot dry climates, the use of downdraught evaporative coolers. Both systems have been the subject of recent research [5, 6].

In order to make the most of valuable land, developers and public clients will often seek to create deep plan buildings with rooms that cannot be ventilated or naturally lit from the sides of the buildings. Atria have been created to act as extract ducts but the question of siting intake air remains to be answered. Suggestions have been made, notably by Sir Michael Hopkins & Partners and Ove Arup & Partners in the new Parliament building at Westminster [7] that large-scale wind-driven intakes may be feasible, but this approach only works when the wind is blowing.

'Top-Down' Ventilation

Gravity displacement ventilation has been extensively adopted in passively ventilated and cooled buildings. The addition of an intake duct increases the resistance of the system, but otherwise makes no difference to the principles involved (fig. 1). Cooling can be achieved in a passively ventilated building by passing large quantities of air through the structure during the night to cool it down so that it can absorb occupancy and fabric gains during the day. The aim of night cooling is to achieve conditions where the internal air temperature is less than the external air temperature during the hottest part of the day in summer. In these conditions air will cease to move in the gravity displacement mode shown in figure 1. Therefore, an additional mode in which the airflow is driven to provide fresh air must be provided. If fans are excluded there are two possibilities in windless conditions: (a) Air in the intake duct must be cooled until its temperature is below the internal air. (b) Air in the extract duct must be heated to induce airflow.

These two possibilities can be combined by using a heat pump to cool the air in the intake duct and heat the air in the extract duct (fig. 2). The strategy that has been developed is based on a modified form of gravity chiller. Gravity chillers are used in cold stores to provide a source of cool air and do not employ fans. They consist of coils with fins spaced at 8-mm centres and containing a chilled water/glycol mixture. Rectangular coils are available. These are double-sided devices to be hung vertically from the ceiling with a minimum wall offset of 400 mm. The manufacturer's literature (Asarums Industries AB, Asarum, Sweden) suggests that it is possible to achieve approximately 2.53 $kW \cdot m^{-2}$ of cooling at a 10°C difference between air and coolant temperature using this type of device. This type of heat exchanger is dependent on the airflow which is induced by it and does not restrict airflow while it is in operation.

The amount of cooling required to maintain a minimal airflow rate is small. In theory, approximately 15 $W \cdot m^{-2}$ of cooling will provide intake air 15°C cooler than the external air at 1.0 air changes per hour (ach) in a space 3.0 m high. In effect, the morning condition is extended into the day, gradually eroding as daytime fabric and occupancy gains warm the structure to a point where the internal air temperature is above the external air temperature. If this occurs when the external air temperature is unacceptably high then the air must be cooled further, and its volume flow rate increased so that all occupancy and fabric gains are absorbed into the airflow. Gravity heat exchangers are bulky. It is clearly more economic to provide an installation which maintains ventilation during summer days with an element of cooling than to provide an installation capable of significantly reducing occupancy heat gains. This can only be achieved by placing an increased value on night cooling.

Warm, dry conditions are much more acceptable than warm, humid conditions because perspiration can evaporate from the skin. The approach taken reduces humidity levels in two ways. Water vapour will condense on the gravity cooler at the top of the intake duct, where it can be easily discharged. Air is warmed when it enters occupied spaces, therefore the relative humidity of the internal air before any occupancy gains is lower than the relative humidity of the intake air.

Heat Recovery

Placing the hot and cold coils of a heat pump in the supply and extract ducts of a top-down gravity displacement ventilation system gives the possibility of heat recovery in winter. In this case cooling and heating water flows must be reversed. Convection currents run counter to the general ventilation airflow at intake and extract. A driving pressure difference from the wind is required to overcome these.

Experiments and Results

Research at The Bartlett has concentrated on creating large test installations capable of operating in the open air or in indoor test environments. The question of wind-induced pressure difference has been addressed because in moderate wind conditions these are greater than pressures induced by differences in air temperature. This aspect of the research has been separately reported [8]. The research is summarised in Bartlett Research Paper No. 11 [9].

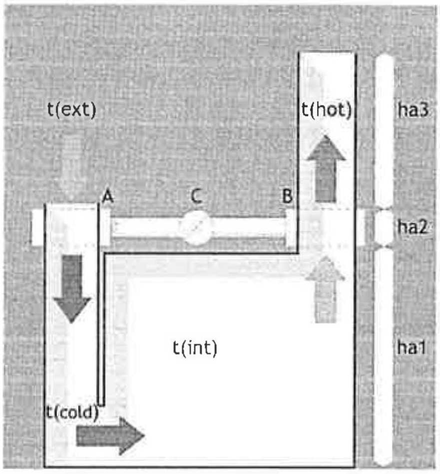

Fig. 2. Using a heat pump to induce ventilation in hot weather. When the external air temperature t(ext) is above the internal air temperature t(int) ventilation can be induced by the heat pump C. The cold coil A reduces the temperature in the intake to below the internal air temperature t(int). Air is then heated by the hot coil B to a temperature above the external air temperature. Airflow is dependent on the temperature difference t(int) – t(cold) and the column height ha1 plus the difference of the mean air temperature at A and B and the column height ha2 plus the temperature difference t(hot) – t(ext) and the column height ha3 and the total resistance of the system.

Gravity chillers work in free air. In top down gravity displacement ventilation, the temperature difference that they create must be enough to overcome system resistance and drive an airflow that will maximise the cooling effect of the chiller. The installation shown in figure 3 consists of a fixed shade roof with a central shaft. The shaft carries a rotating intake and extract hood with a circular split duct under this. One half of the split duct opens into an intake chamber or plenum that contains a finned chiller coil. The other half of the duct passes through the intake chamber into an extract chamber or plenum containing a finned heating coil. It should be noted that chiller and heater coils are placed around the circumference of the two chambers and do not impede airflow through them. Two separate ducts link the intake chamber to the test cell that then opens into the extract chamber above it. In order to ensure that wind pressures are always positive at the intake position and negative at the extract position the device is turned head to wind using a wind-driven servo linked to a set of chain-driven sun and planet gears. The heat pump used to drive this device is a Cornelius Classic 1000 Remote Cooler (IMI Cornelius Ltd., Alcester, UK). This heat pump has a chilled water reservoir operating around 0°C which is pumped through the chiller coil and a variable temperature hot water output which is pumped through the heater coil. The heat pump operates intermittently, while maintaining a largely constant chilled water supply. In order to avoid peak load overheating in the hot circuit, a fan coil is fitted for experimental purposes. The hot coil temperature can rise to 45°C before the fan coil cuts in.

The hypothesis that gravity chillers and heaters can be incorporated into a gravity ventilation system and induce an airflow was tested in still air. It was found possible to reduce supply air temperatures from 12°C by up to 4°C. Typical test results are shown in figure 4.

The installation shown in figure 5 is equivalent to that illustrated in figure 3 but arranged in three sections. The intake section consists of a rotary intake cowl on a central drive shaft with a fixed shade roof. The intake cowl discharges air into an intake plenum lined with a finned chiller coil. The intake plenum contains a water discharge plate which shields the entry into the duct serving the bot-

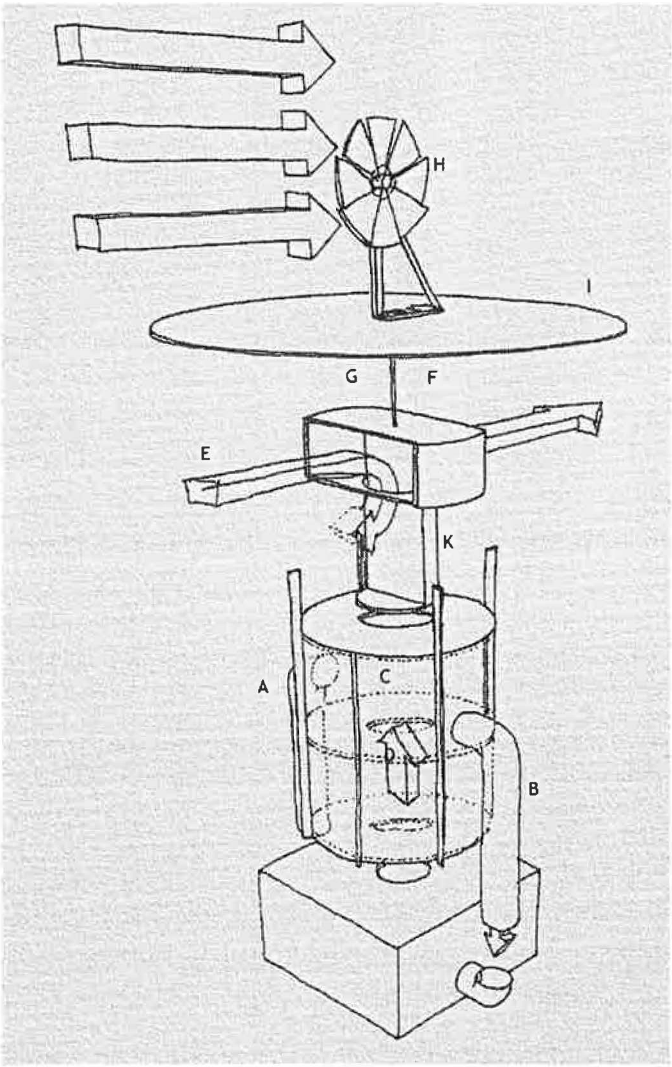

Fig. 3. Rotary shaft duct combined intake and extract using shaft bearings. The wind-driven servo H turns the intake and extract hoods E and F on a central shaft so that they align with the direction of the wind. The central shaft G bears on the fixed shade roof I. Air enters through E into the intake plenum C which contains a finned chiller coil. Water is discharged and the air passes down ducts A and B to the test cell. Air rises from the test cell through duct J into the outlet plenum D which contains a finned heating coil. It then passes through the inlet plenum through pipe K and exits through extract hood F.

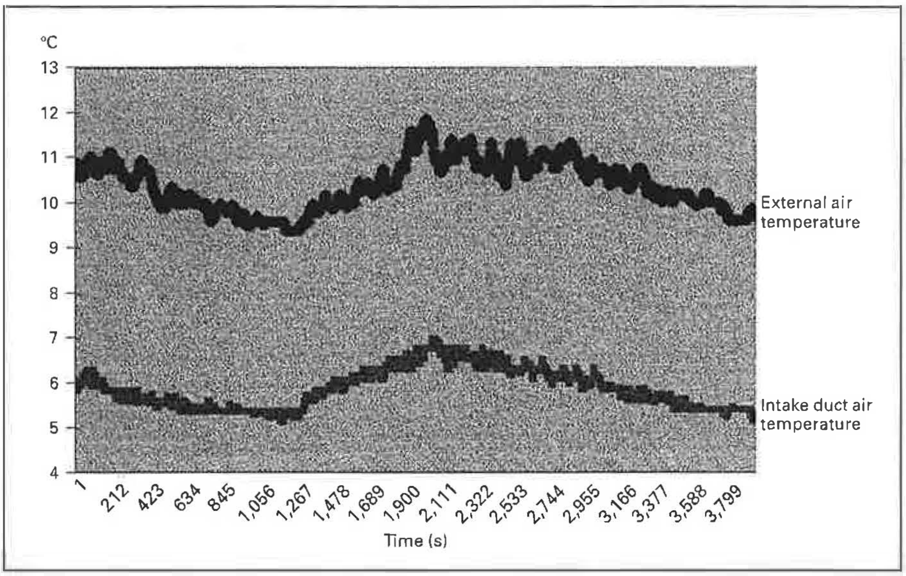

Fig. 4. Graph showing temperature drop between external air temperature and intake air temperature using the equipment shown in figure 3.

tom of a test cell. The central section of the installation contains a variable heater made from nine 40-watt incandescent light bulbs. The outlet section contains an outlet plenum drum lined with a finned heating coil. Above this a vertical extract duct is terminated by a large flat-dished cowl. A variable iris damper is placed at the base of the inlet duct in this system. Through this it is possible to reduce volume airflow through the system. The installation was tested in still air with the orifice diameter reduced from 300 to 200 mm. The same heat pump was used during these experiments but in order to maintain a more constant heat input the heating coil was disconnected and the heater battery was used to supply heat to the system. The gravity chamber will deliver air at 10°C

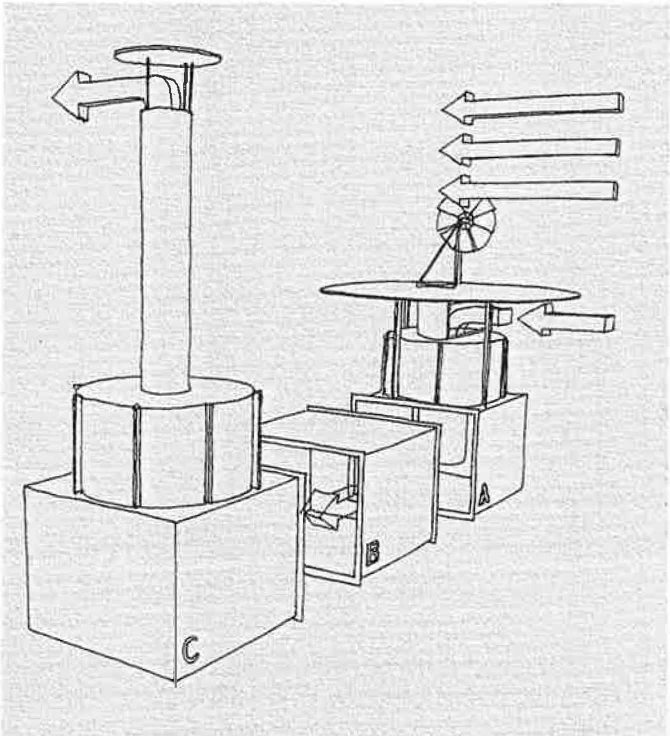

Fig. 5. Separated intake and extract. The components of this system are equivalent to those shown in figure 3 modified as follows: a single intake hood takes air into the inlet plenum which sits on test cell module A. A duct in this test cell module discharges the air at low level. Test cell module B contains a variable heat source. The outlet sits on test cell C and is terminated by a rising duct with a dished rain shield.

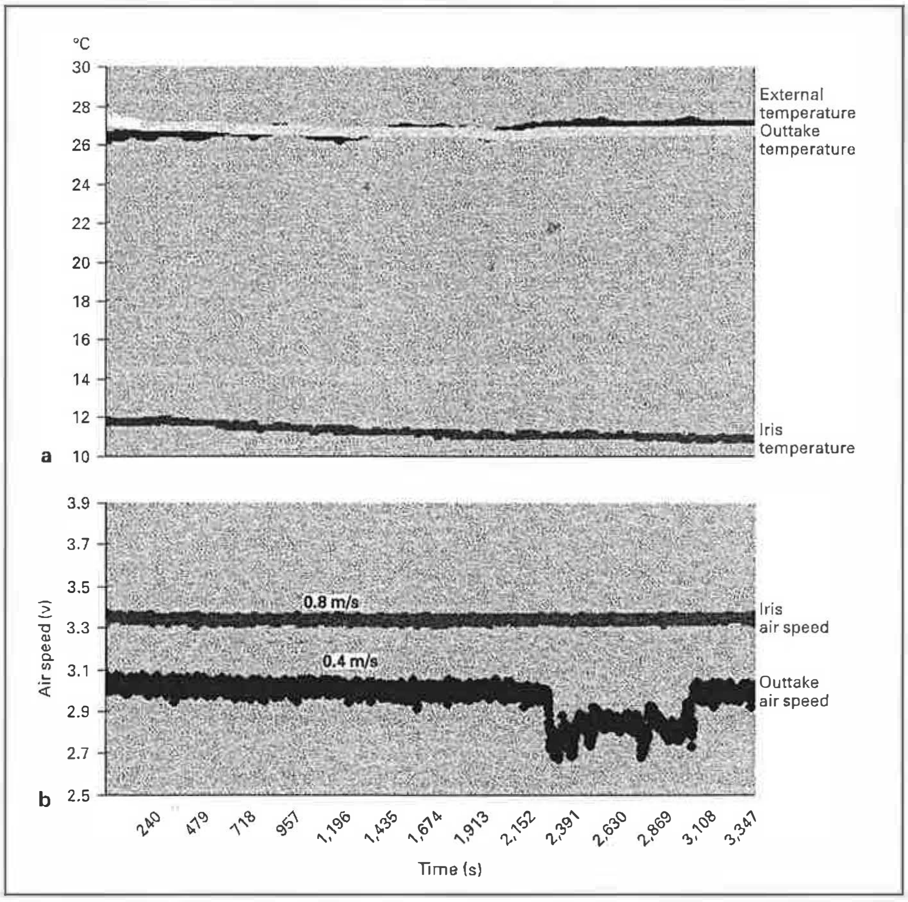

Fig. 6. Graphs showing air temperature differences (**a**) and air speeds (**b**) measured in the equipment illustrated in figure 5. The readings were taken with the heater battery and the chiller on and the iris damper closed to a diameter of 200 mm.

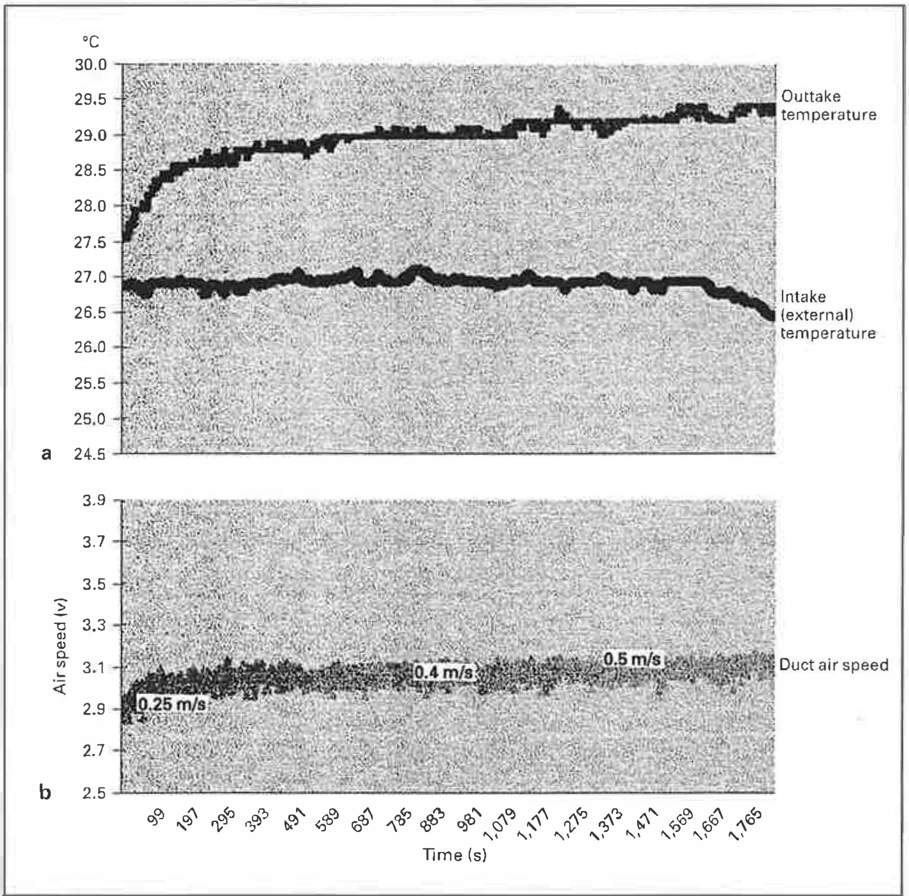

Fig. 7. Graphs showing air temperature differences (**a**) and air speeds (**b**) measured in the equipment illustrated in figure 5. The readings were taken with the heater battery on, the chiller off and the iris damper fully opened to a diameter of 300 mm.

when the external air temperature is 26 °C at a rate of 0.8 m · s^{-1} through the damper. This is equivalent to 0.4 m · s^{-1} through the main 300-mm inlet duct. This temperature drop is achieved in a chamber which is part of a system that presents very little resistance to displacement airflow in other conditions. Typical test results from this system are shown in figures 6 and 7.

Heat Recovery Using the Wind

A preliminary experiment was undertaken using the equipment shown in figure 3 by reversing a wind-driven airflow so that air entered across the hot coil and left across the cold coil. The performance of the equipment was erratic at all wind speeds and especially erratic at wind speeds below 1 m · s^{-1}. At 1 m · s^{-1} average duct speeds were 0.4 m · s^{-1}. Intake air was heated by 9 °C when the hot coil was at 43.5 °C and the cold coil was 2.5 °C. The external air temperature averaged 5 °C. Typical test results from this experiment are shown in figure 8. It may, however, be possible to take heat out of the extract coil and utilise it elsewhere.

Zonal Models

A simplified zonal model of the installation shown in figure 5 was created using Tas software supplied by Environmental Design Solutions Limited (EDSL Ltd., Milton Keynes, UK). Ducts and plena are shown square, changes in airflow direction at intake hood and rainshield plates are omitted, and no attempt has been made to model wind effects. The model is shown in wire frame in figure 9. The chiller coil in the intake plenum was modelled by including a negative equipment gain of 475 W. This was based on a calculation of the heat transfer from the intake to the coil derived from the volume airflow and temperature drop shown in figure 6. Heat gains in the test cell were taken from the heater battery which was running at 360 W.

The Tas model was put into in a virtual climate chamber with a temperature of 26 °C and run for 24 h. The results are shown in figure 10. These results demonstrate that the outtake duct temperature closely corresponds to the test cell temperature indicating that the overall heat flows in the virtual model correspond to reality. The

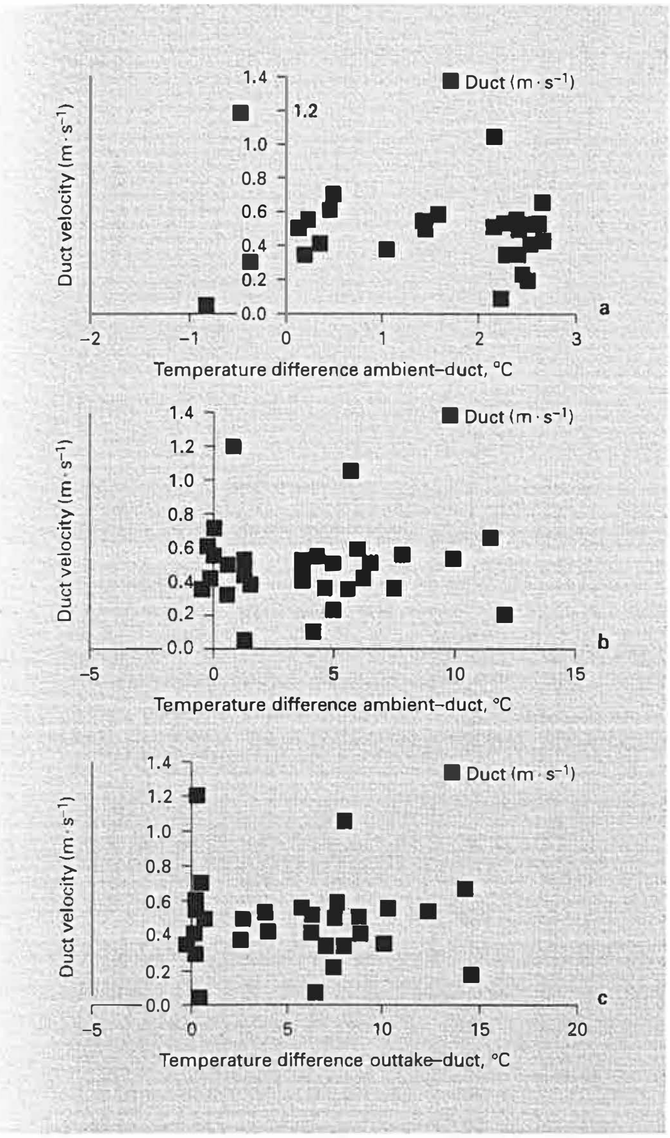

Fig. 8. Scatter graphs showing the effect on air temperature of reversing the heat pump water flows in the equipment shown in figure 3, when exposed to a wind speed of 1 m $\cdot$ s^{-1}. This experiment was performed outdoors and the wind speed reading was that measured at the time of the experiment. **a** The rise in temperature in the intake shaft. **b** The temperature difference between the extract duct and the external temperature. **c** The temperature difference between the intake and extract ducts.

intake duct temperature is substantially below the real world measurement of 7.9 °C and the velocity through the outlet duct is shown at 0.2 m $\cdot$ s^{-1}. This is one half of the observed duct velocity. We can assume that the Tas simulation errs on the side of caution when representing slow moving gravity-driven airflows in vertical ducts.

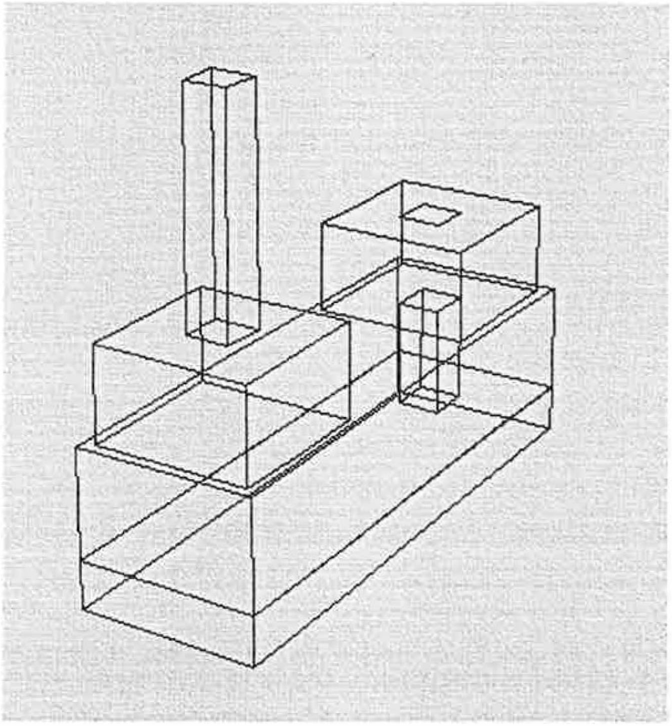

Fig. 9. Wireline of simplified Tas model of figure 5.

Kew Test Cell

A further zonal model was created to compare the performance of a heat pump-assisted ventilation system with a fan-assisted and fan-driven system. The model is of an experimental 6 × 6 × 6 m aluminium cell situated at Kew in West London. The cell is lined with 300 mm of insulation throughout. A larger version of the ventilation system described in figure 5 is part of the system. All ducts and plena are insulated. The upper half of the walls and ceilings are lined with 150 mm of concrete and 50 mm of concrete is placed over the floor insulation.

For this experiment the cell was modelled as above and is shown in wire frame in figure 11. The cube was assumed to be occupied from 06.00 to 18.00 h, with lighting gains of 15 W $\cdot$ m^{-2} taken up in the lowest part of the space. Occupancy gains were assumed to arise in a zone above this at 35 W $\cdot$ m^{-2}. Inlet and outlet apertures were variable. A heat pump was assumed as part of the model which was capable of transferring up to 40 W $\cdot$ m^{-2} from the inlet plenum to the extract plenum under simulated summer conditions. We also assumed that there was a mechanism in place which could transfer up to 30 W $\cdot$ m^{-2} from the extract plenum and supply this heat to the lower zone of the main space.

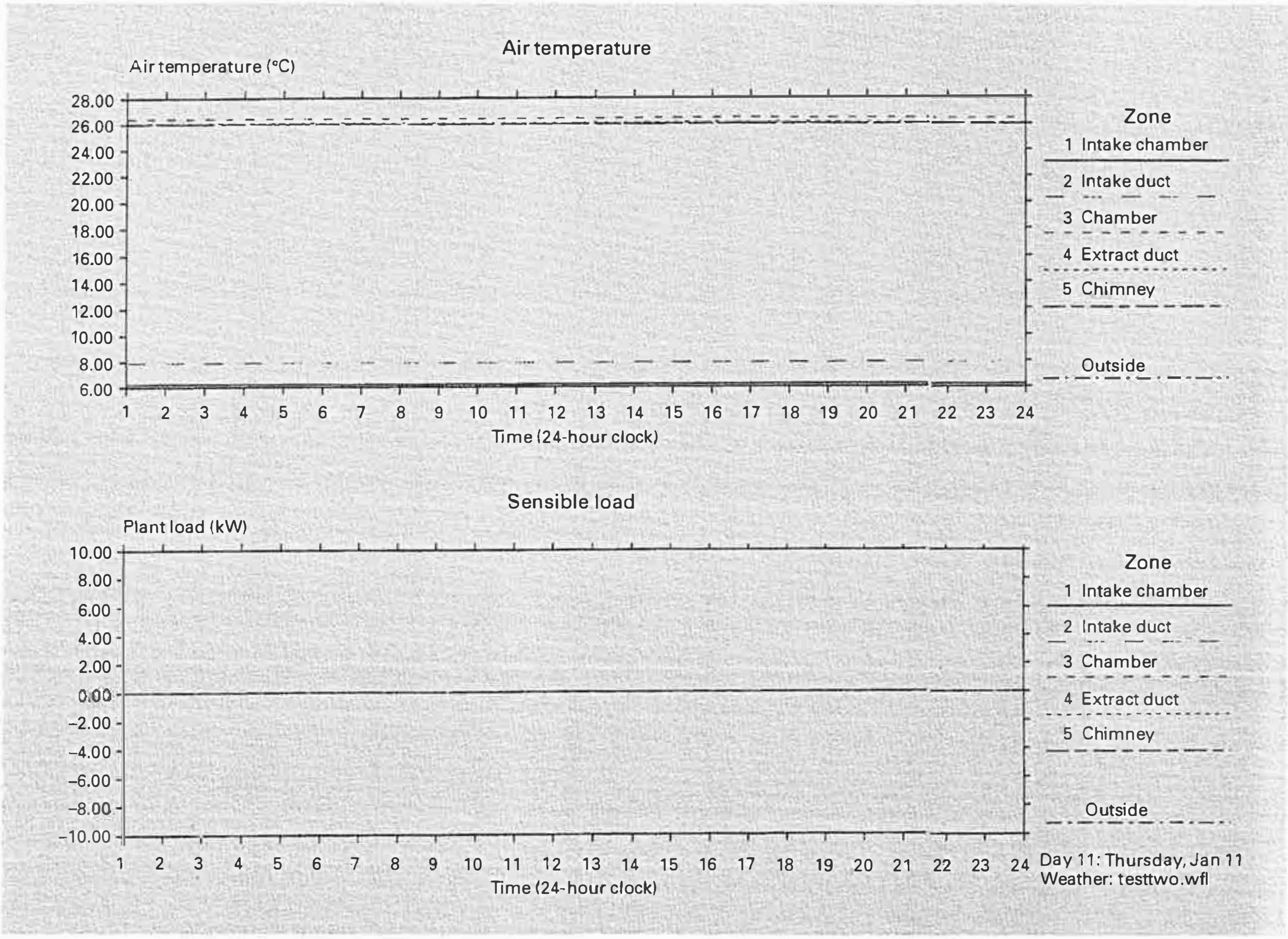

Fig. 10. The model shown in figure 5 running in a virtual climate chamber at 26 °C.

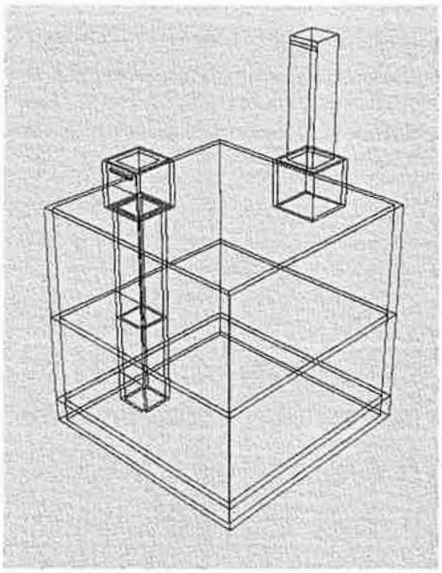

Fig. 11. Wireline-based 6 × 6 × 6 m test cell at Kew.

In addition to the above the cell could be mechanically ventilated with external air to give up to 20 ach in the bottom half of the main space (the inhabited zone). It was assumed that fan energy consumption and resulting air heat gain was $0.3\ W \cdot l^{-1}$.

Target environmental conditions for the period 06.00–18.00 h were set as follows: resultant temperatures 18–25 °C, inhabited zone air change rate 1.5–2.5 ach.

Using the Tas simulation program empirical conclusions have been drawn about the performance of the

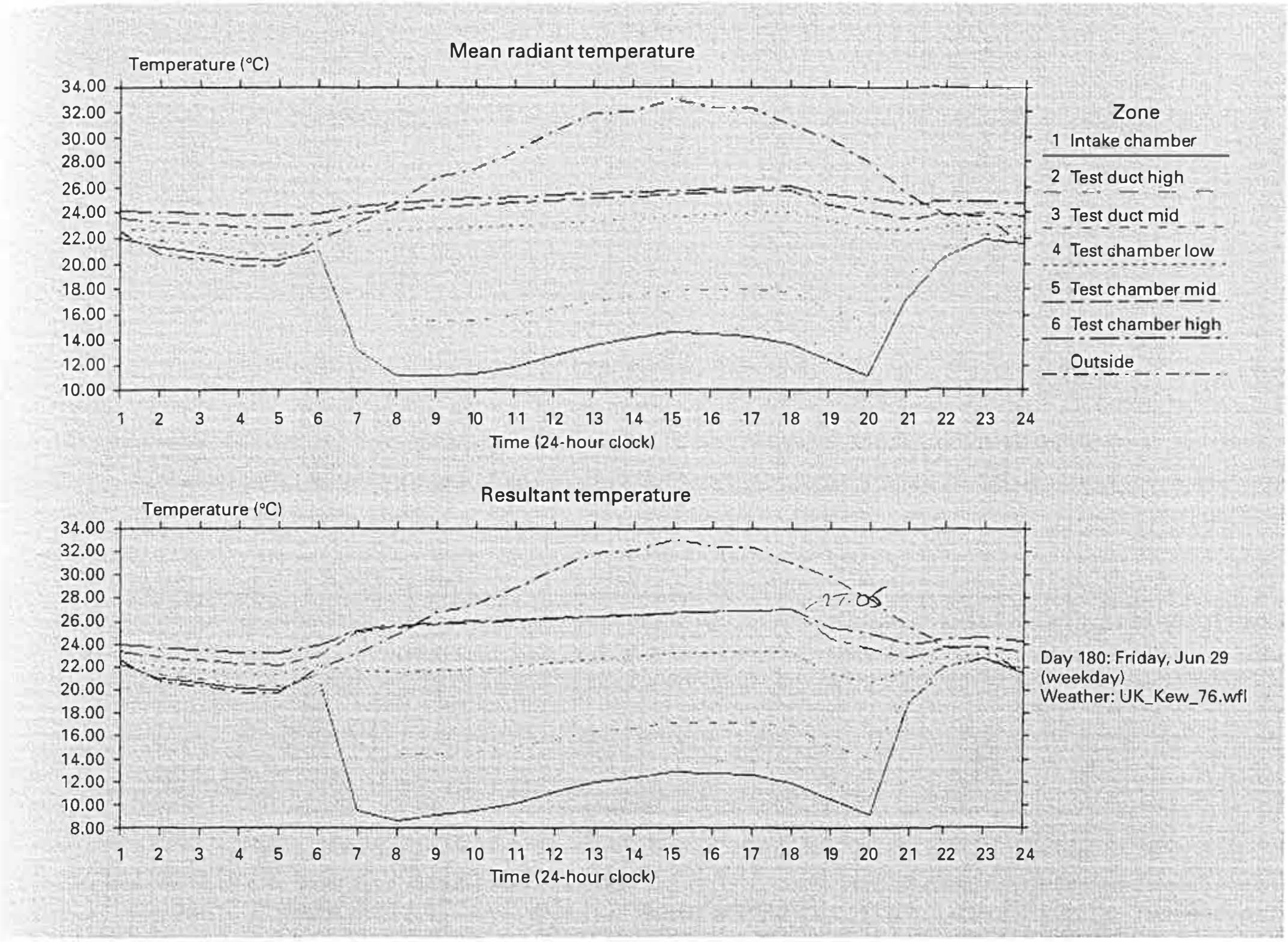

Fig. 12. Simulation of a hot day after a hot night. Results from the test cell running with approximately 38 $W \cdot m^{-2}$ gravity downdraught cooling from 06.00 to 21.00 h.

hypothetical test cell in a variety of conditions. However, these conclusions are limited by the nature of the science that is embedded in the Tas simulation and by the nature of the virtual Tas cell that has been described.

A further limitation to the conclusions that can be drawn is the climatic database that is used. Most of our investigations use data from 1976, a year when a very hot summer followed and was followed by a cold winter. Some other investigations use data from Kew 'try', which is an average database. This database contains data from a much cooler summer and much warmer winter than 1976. The Tas simulation is time-dependent and gives results which assume that the system has been running for 10 days previously.

Discussion

Hot Day after Hot Night

The first question asked was: what is the least energy-intensive method of achieving target conditions during a hot day after a hot night? The data given in figure 12 for such a situation shows a simulation of what happened on day 180 Kew 1976 with a heat pump in operation transferring 37.8 $W \cdot m^{-2}$ (heat transfer figures are given in $W \cdot m^{-2}$ of the main test cell floor area) from intake to extract air. This is equivalent to an electrical energy consumption of 12.6 $W \cdot m^{-2}$ over a 15-hour day. In this model the cell purges itself by gravity airflow from 21.00 to 06.00 h.

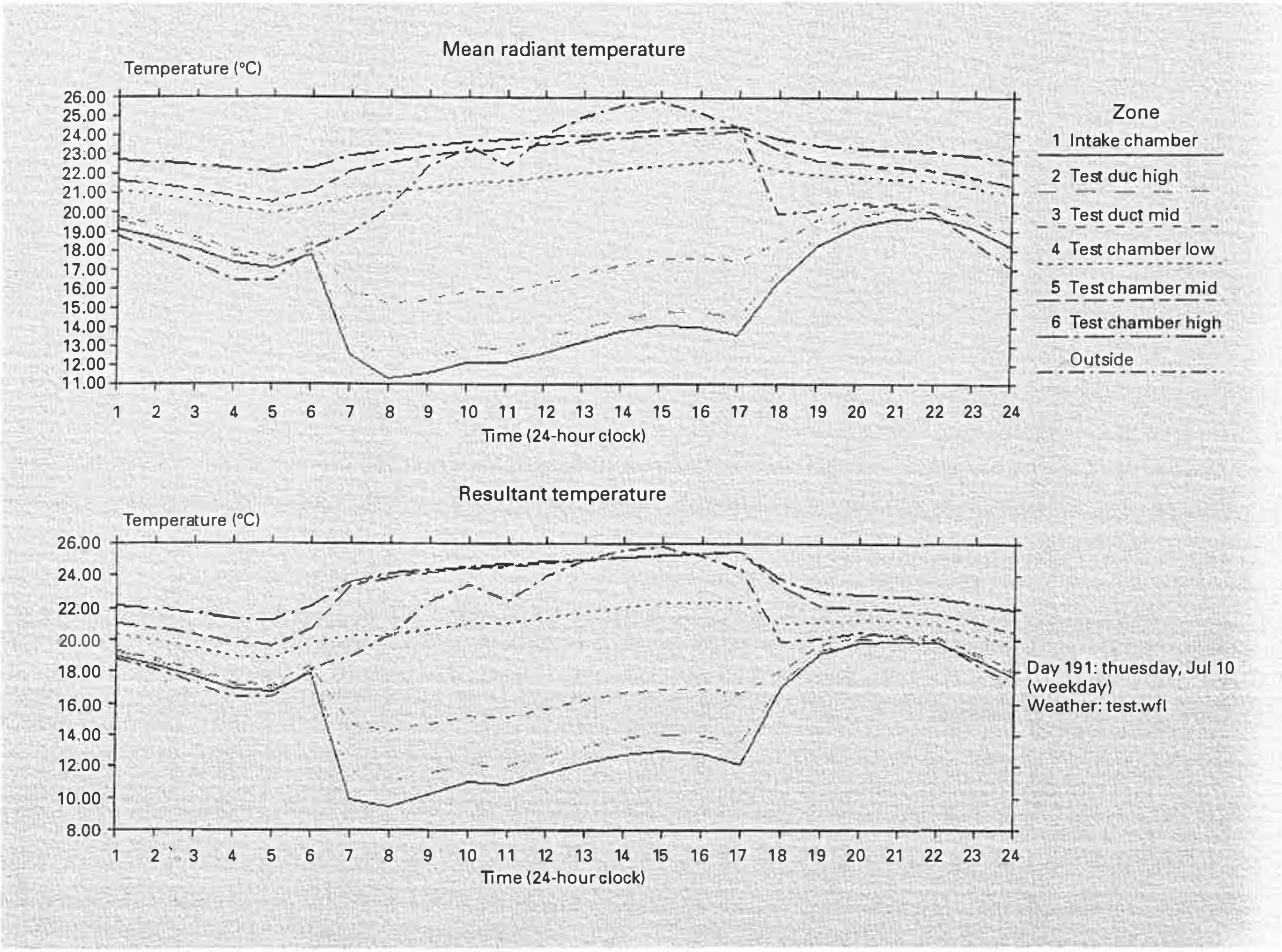

Fig. 13. Simulation of a hot day after a warm night. Results from the test cell running with 25 W · m^{-2} gravity downdraught cooling from 06.00 to 18.00 h.

It was not possible to achieve the target conditions with mechanical ventilation during the day after gravity purging at night for a maximum day temperature of 32°C, or with mechanical ventilation during the day after mechanical purging at night for a maximum day temperature of 33°C.

Further investigation indicated that the following rule held true for the model. If the mean external night temperature exceeded 16°C and the peak external day temperature exceeded 25°C then acceptable temperature conditions could not be realised using mechanical ventilation. However, acceptable conditions could be achieved using a heat pump after night purging by gravity air circulation. The ratio of the ventilation openings during the day compared to the night was 0.06:1.00.

A limiting example of this particular situation (using a manufactured weather file based on day 191 Kew 1976 is shown in figure 13). The heat pump is transferring 25 W · m^{-2} from the intake to the extract air. This is equivalent to 8.33 W · m^{-2} of electrical energy over a 12-hour day.

Hot Day after Cool Night

The next set of conditions tested were those where the mean external night temperature falls below 16°C and the peak external day temperature exceeds 25°C. Day 182 Kew ('try') meets these criteria. The results presented in figure 14 show how it was possible to achieve target conditions using mechanical ventilation during the day after night purging by gravity air circulation. The fan en-

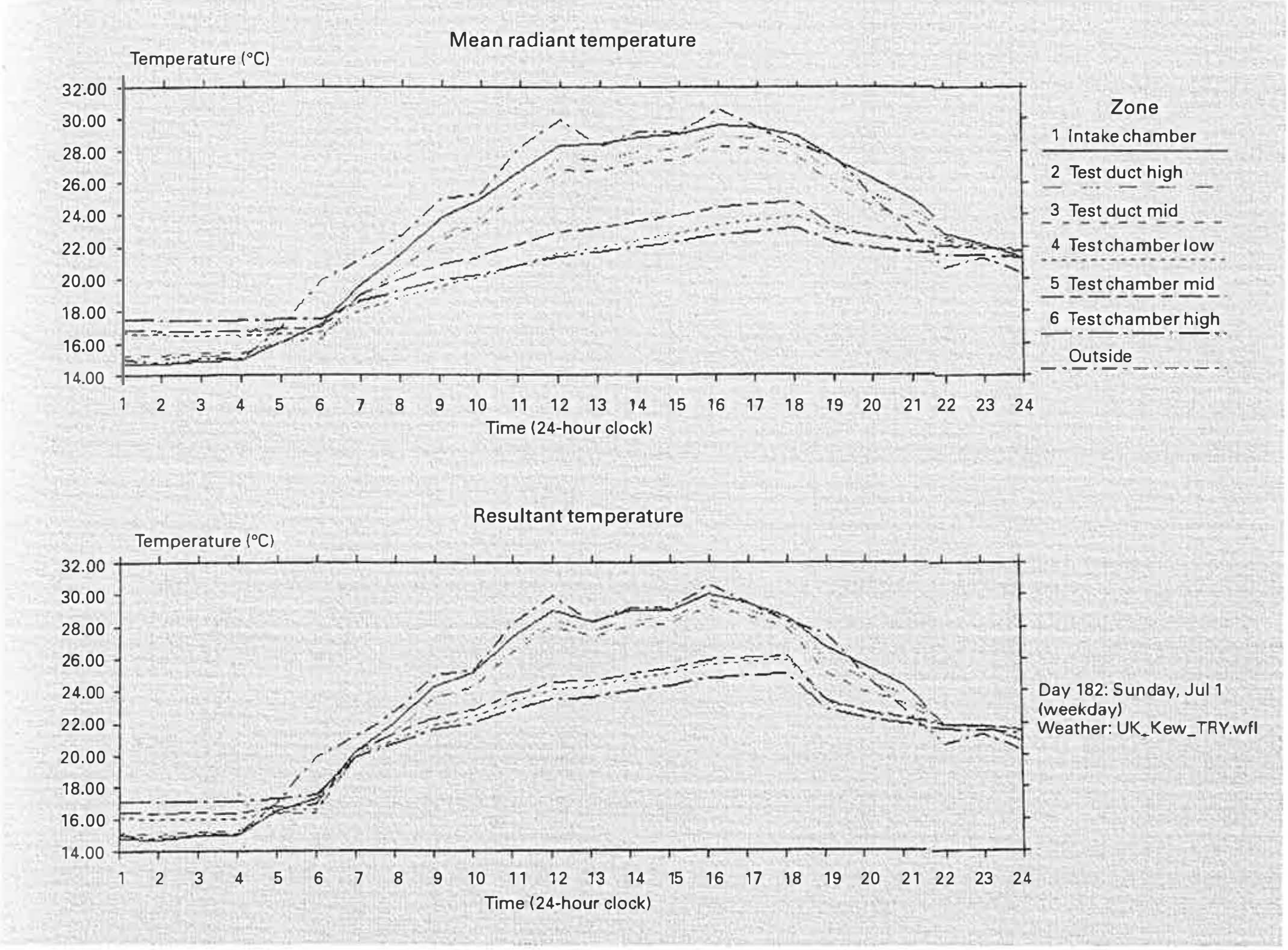

Fig. 14. Simulation of a hot day after a cool night. Results from the test cell running without gravity downdraught cooling.

ergy consumption to induce daytime ventilation was 0.5 $W \cdot m^{-2}$ over a 12-hour day.

This consumption is significantly less than that required when a heat pump was used to induce airflow in the same conditions. An investigation showed that the heat pump must move 9.4 $W \cdot m^{-2}$ equivalent to 3.1 $W \cdot m^{-2}$ electrical energy to achieve the same effects as the fan outlined above. Further investigations indicated that the following rule holds true for the model. When external mean night temperatures fall below 16°C and peak day temperatures exceed 25°C, target conditions can be met if mechanical ventilation is used to provide daytime ventilation after night purging by gravity air circulation. It should be noted that the highest day peak temperature from the data set that occurred after a night with mean temperature below 16°C is that given in the example above.

Autumn Conditions

As both mean night and peak day temperatures fall, a point is reached where mean external day temperatures fall below 14°C. The following rule then appears to hold true for the model. When peak external day temperatures fall below 14°C ducts must be closed at night to retain daytime gains. The building will ventilate and maintain target conditions during the day without any additional energy requirement. A limiting example of this situation (day 282 Kew 1976) is shown in figure 15.

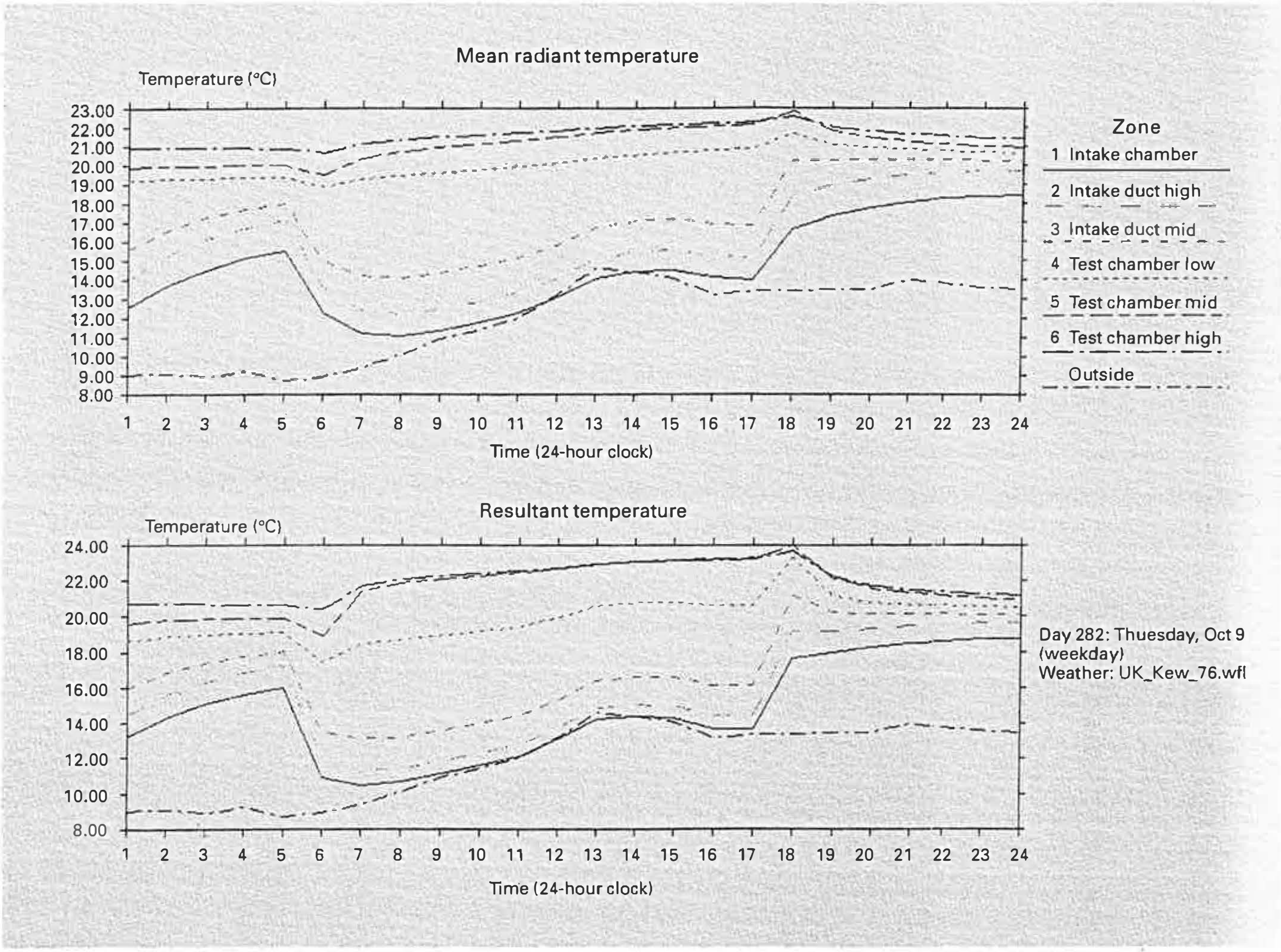

Fig. 15. Autumn conditions with vents closed from 18.00 to 16.00 h.

Winter Conditions

In the winter the mean external day temperature falls to a point where ventilation and fabric losses are such that the target minimum temperature cannot be maintained. At this point heat has to be taken out of the extracted air and returned to the building. The following rule then holds good for the model. When the mean external daytime temperatures fall below 4 °C, either the incoming air or the building fabric must be heated. A limiting example of this situation (day 350 Kew 1976) is shown in figure 16. This example also demonstrates that it is possible to take heat out of the extract plenum and still maintain a gravity-driven ventilation airflow. The results show that initially relatively little heat transfer is required at 10 $W \cdot m^{-2}$ during a 12-hour day.

When mean external day temperatures fall to –1 °C a much greater amount of heat must be transferred. This is illustrated in figure 17 showing day 31 Kew 1976, when approximately 30 $W \cdot m^{-2}$ had to be transferred.

Winter Conditions/Heat Recovery

Commercial air-to-air heat exchangers operate at approximately 1 $W \cdot l^{-1} \cdot s^{-1}$ of air moved and at 70% efficiency when the intake-to-extract temperature differences approach 20 °C. As a result 1 W of electrical energy will transfer 16 W of heat. Comparative heat pump efficiencies are substantially less; 1 W of electrical power is assumed to transfer 3 W of heat. The case for direct air-to-air, or air-to-water-to-air heat exchange in winter conditions is considerable.

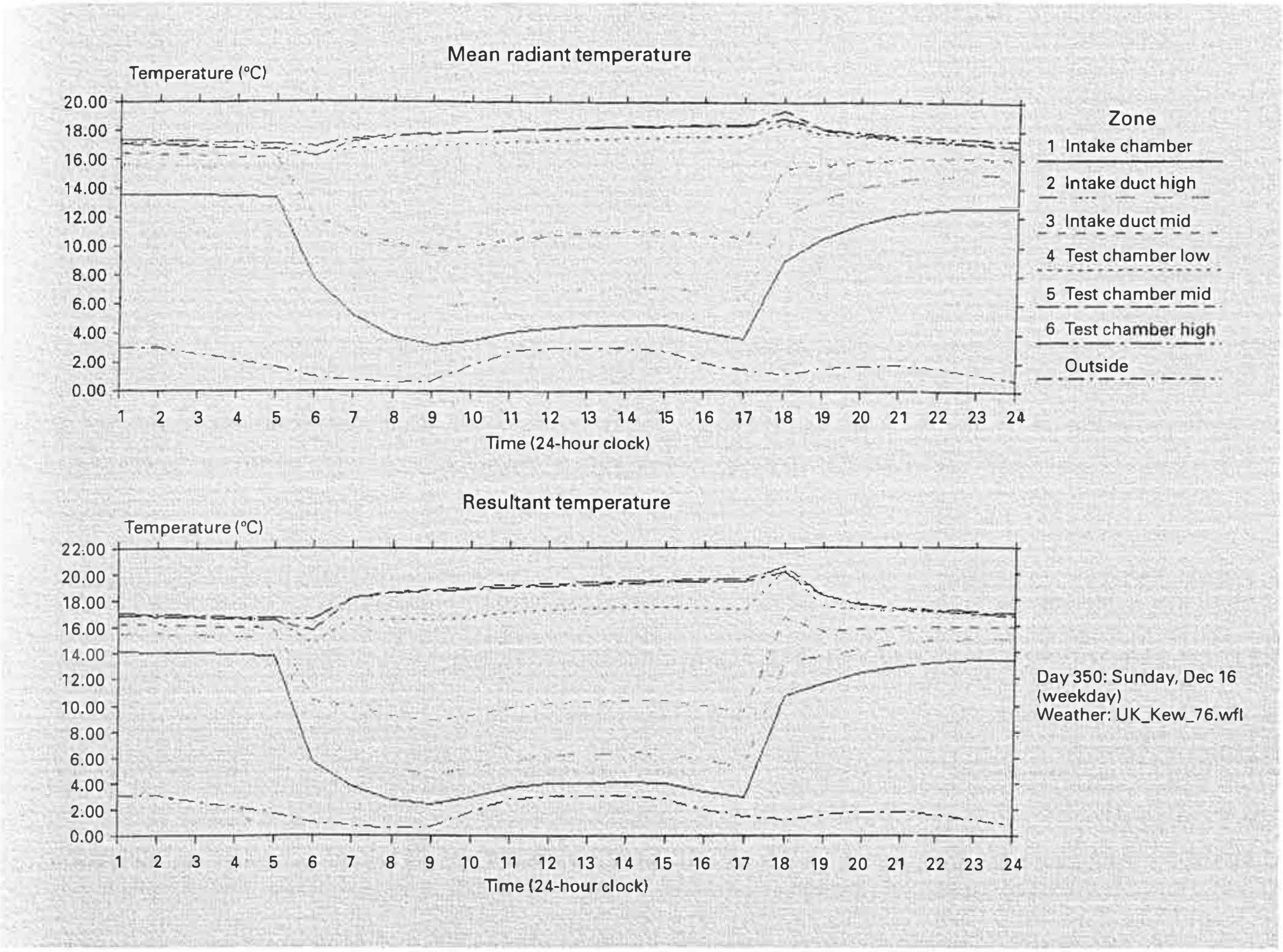

Fig. 16. Winter conditions with 10 W · m^{-2} heat transfer from 06.00 to 18.00 h.

Spring Conditions

Gradually, as the weather becomes milder, the need for heat transfer diminishes, followed by a period when heat carry-over will guarantee acceptable morning start conditions. Nights are usually very cold. Cool mornings and evenings will take out internal heat gains. As day temperatures rise it becomes necessary to recommence cooling the building during the day. The following rule was found to hold for the model that progressively opening ventilation ducts during the day permits purging by gravity air circulation when the peak day temperatures exceed 18 °C. The results using a limiting example (day 126 Kew 1976) are shown in figure 18.

The Lessons of 1976

So far as climate is concerned 1976 was an extreme year. We have undertaken a day-on-day analysis based on the studies above. The results are as follows:

(1) The number of days when no energy input was required to maintain comfort conditions: 270.

(2) The number of summer days when a heat pump had to be used to maintain daytime comfort conditions: 20.

(3) The number of summer days when daytime mechanical ventilation could be used: 28.

(4) Wind speed is another important parameter for those days when daytime mechanical ventilation could be used. It appears that the weather conditions when hot days are followed by cold nights are those where there is a

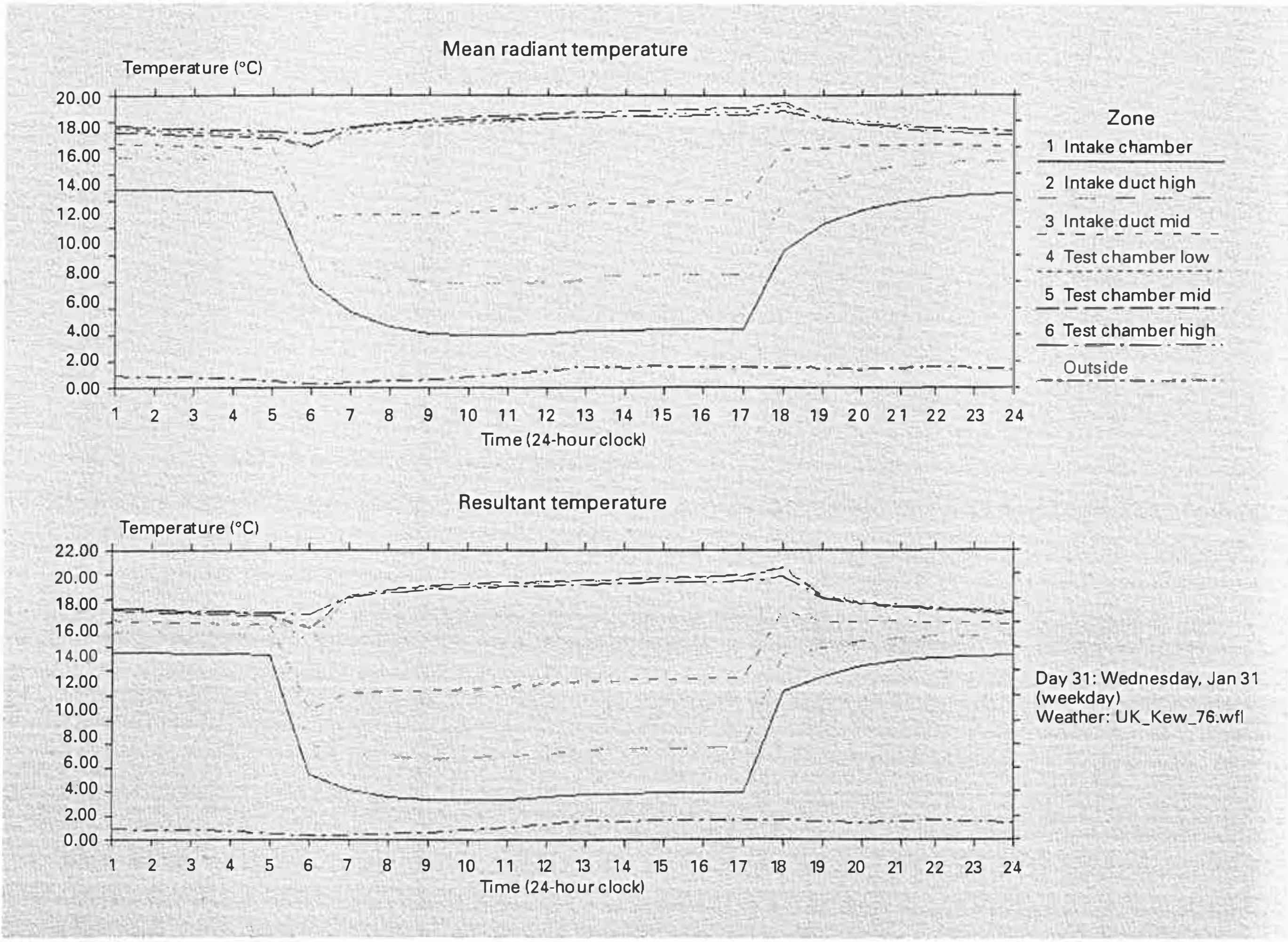

Fig. 17. Winter conditions with 30 $W \cdot m^{-2}$ heat transfer from 06.00 to 18.00 h.

considerable amount of wind. On 25 of the 28 days wind speeds exceeded 2.5 $m \cdot s^{-1}$. These wind speeds would be reduced in sheltered situations but may, of themselves, be sufficient to induce the daytime ventilation rates that were required. A more detailed treatment of this issue is given by Levermore et al. [10] in their report to the EPSRC.

(5) The number of winter days when heat recovery was required during the day: 36.

Several lessons were learnt from the simulations using the climate conditions of 1976:

(1) For 74% of the year, the test cell would maintain target environmental conditions without any energy input.

(2) Direct heat recovery is the most energy-efficient winter heating strategy.

(3) Free running passive gravity driven airflows can play a significant role in summer cooling.

(4) For some days in the summer, mechanical or wind-driven ventilation is the most economical way to maintain comfort conditions.

(5) For a similar number of days, active cooling is required. This will induce gravity driven ventilation airflow without the use of fans.

Duct Resistance

It was demonstrated that the physical test cell constructed (fig. 5) had a much higher duct velocity than was shown by its virtual Tas representation. The second Tas virtual test cell designed contained ducts that were 750 × 750 mm^2. These served an internal space 5.4 × 5.4 m on plan. For this cell the virtual duct velocities were also low,

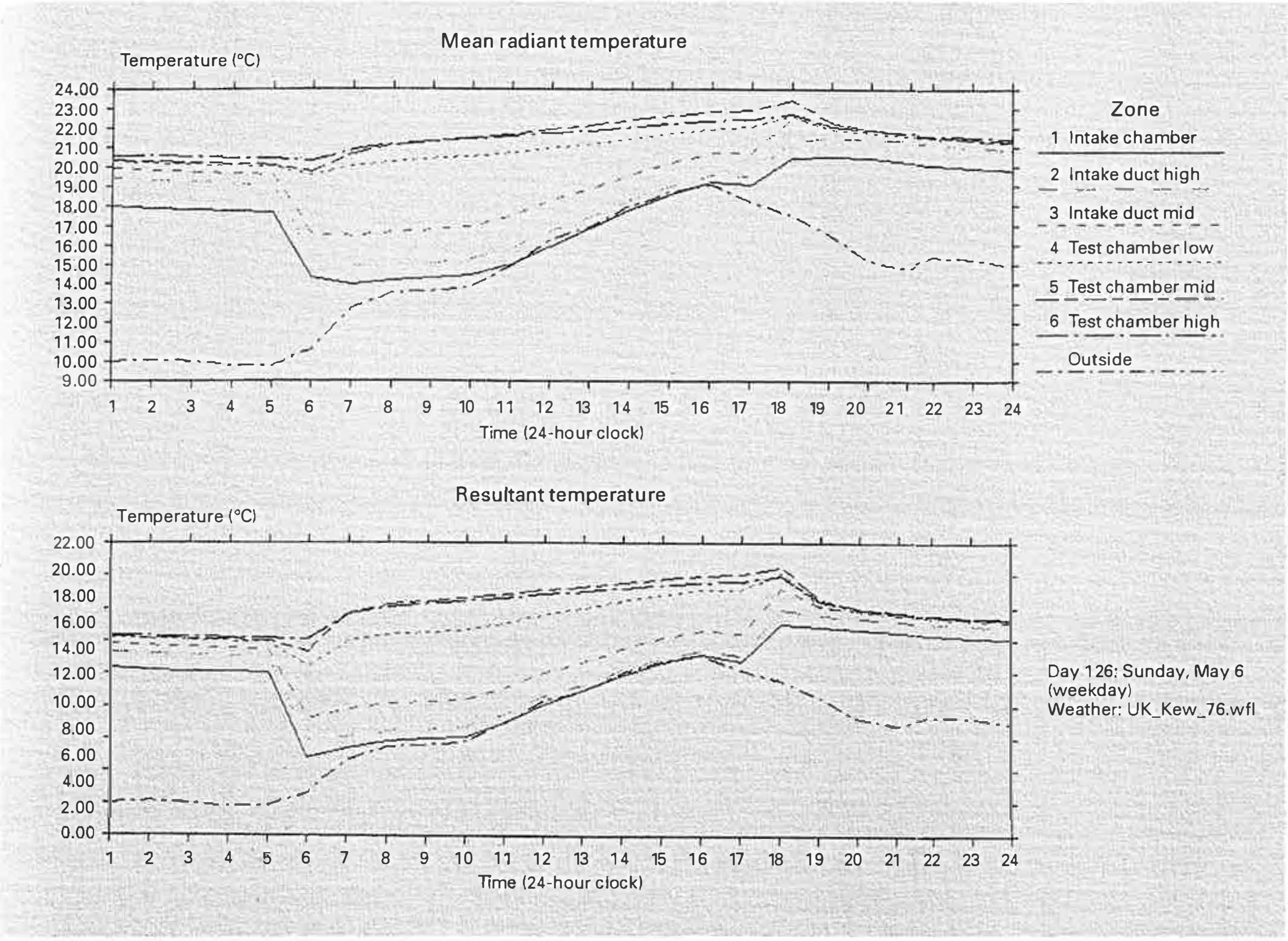

Fig. 18. Spring conditions with vents open from 06.00 to 18.00 h.

typically 0.35 $m \cdot s^{-1}$ during night purging. No additional resistances beyond those derived by the Tas program have been included in the demonstration.

The physical experiments conducted cause us to view existing data on slow speed gravity flows and ducts with caution.

The driving gravity pressure in the system shown in figure 2 is dependent on the temperature differences in the columns of hot, warm and cold air in the system. The driving pressures can be approximately expressed as follows:

$$\Delta p = \frac{\rho g}{300} (ha1(t(int) - t(cold)) + ha2(t(mean\ hot\ chamber) - t\ (mean\ cold\ chamber)) + ha3(t(hot) - t(ext)))$$

This is equivalent to:

$$\Delta p = 0.043\ ((ha1(t(int) - t(cold)) + ha2(t(mean\ hot\ chamber) - t\ (mean\ cold\ chamber)) + ha3(t(hot) - t(ext)))$$

The system as a whole will have a resistance (R) that gives a maximum residual gravity pressure [Δp(RG)]. This is the dynamic velocity pressure at the point of greatest constriction, that is the point where air is moving at its greatest velocity

$$\Delta p(RG = \frac{\Delta p}{R}$$

This relationship can also be expressed as

$$\Delta p(RG) = K(RG)\ \Delta p$$

where K(RG) is the residual gravity velocity pressure factor. It is a measure of the pressure loss of a complete gravity system expressed as the maximum velocity pressure in the system.

Calculations based on the experimental results demonstrate that the test installation shown in figure 3 has a value of K(RG) = 0.3. For the test installation shown in figure 5 K(RG) = 0.4. Calculations based on CIBSE data [11] suggest that the installation shown in figure 3 should have a value of K(RG) = 0.1 and the installation shown in figure 5 should have a value of K(RG) = 0.17, excluding duct resistance. This poses a considerable problem which must be the subject of further research if this type of technology is to be realised. It is clearly unacceptable to initiate a large-scale programme of physical testing or CFD modelling on a built project-by-project basis. There are two possible lines of investigation. Firstly, if the component industry is prepared, with appropriate government assistance, to investigate this issue, it may be possible to fabricate a range of intake and extract assemblies of the type shown in figure 5 which may be empirically tested to establish their performance characteristics. It may then be possible to calculate the performance of the rest of the system using existing techniques. Alternatively, it may be possible to create a generic description of this type of system and modify an existing zonal model so that it can be easily applied to a range of largely similar installations. This latter approach must be validated empirically.

Conclusions

A gravity ventilation system which takes air in from the roof of a building without the use of fans will fail in conditions when the external air temperature is above the internal air temperature, and when there is no wind. It is, however, possible to reduce the temperature of the incoming air and raise the temperature of the outgoing air by using a heat pump. A ventilation airflow can be successfully induced in this way.

The resulting systems are passively operated for most of the year and provide a form of gravity-driven variable air volume air cooling on summer days which can also operate in overcast humid, windless conditions and maintain cooling during periods when temperatures are high at night. It is suggested that the peak daytime cooling load energy demand could be met by utilising solar power. The minor drawback in this approach is that it ignores those summer conditions when mechanical or wind-driven ventilation is all that is necessary to ensure comfort. The major drawback in the approach is that it ignores direct winter heat recovery. Commercial air-to-air heat recovery units contain fans which could provide summer daytime ventilation and which could drive this air through compact chiller units when required. This is not incompatible with an otherwise passive top down gravity system operating for much of the year.

The climate of southern England is erratic and this kind of mixed mode system is probably inevitable. In colder climates, where hot days can be assumed to follow cold nights in summer and where winters are long, it may be more appropriate to consider a purely mechanical system. In hotter climates, where warm nights are the norm in summer and winter heating requirements are negligible it may be more appropriate to consider a purely gravity-driven system fitted with gravity chillers and heaters for use in the summer.

References

1 Ajiboye P, Hesketh M, Willan P: The significance of urban pollution and its dilution with height. Proc 18th AIVC Conf, 1998, pp 258–266.
2 Kukadia V: The effects of external atmospheric pollution on indoor air quality. Proc 17th AIVC Conf, 1997, pp 41–49.
3 Ajiboye P, Hesketh M: Natural ventilation for offices. BRECSU, 1999, pp 18–19.
4 Irving S, Palmer G: Location of ventilation inlets to minimise contamination: State of the art review. BRE/CIBSE NVG Semin Proc 1997.
5 Gage S, Graham M: Static split duct ventilators. Building Research and Information, in press.
6 Cook MJ: Use of CFD for modelling passive downdraught cooling. Proc 18th AIVC Conf, Athens, 1997, 2, pp 603–611.
7 Dunster B, Pringle J: Research into sustainable architecture. Architectural Design 1997;67:27–31.
8 Gage S: Combined wind driven inlet and outlet terminals. EPIC Conf, 1998, pp 191–198.
9 Bartlett Research Paper No 11: Top Down Ventilation and Cooling in Urban Areas. University College London, London 1999.
10 Levermore G: Report on EPSRC research grant GR/C38288, 1999.
11 CIBSE Guide/Chartered Institution of Building Services Engineers, ed 5. London, CIBSE, 1986, vol C, table C4.35.