#12550

Annexe35 HybVent

Hybvent Forum '99

First International One-day Forum on Natural and Hybrid Ventilation

Supplementary Papers

The University of Sydney

Darlington, NSW, Australia

28 September 1999

Hybrid Ventilation in New and Retrofitted Office Buildings

NUMERICAL SIMULATION OF TRANSIENT EFFECTS OF WINDOW OPENINGS

G V Fracastoro & M Perino
Dept. of Energy Technology
Politecnico di Torino
Corso Duca degli Abruzzi, 24 – 10129 Torino, Italy
fax: +39 11 564 4499
e-mail: fracastoro@polito.it, perino@athena.polito.it

ABSTRACT

The simulation of room airing (ventilation by means of door/window opening) by means of CFD techniques requires a specially skilled user, because a number of difficulties arise since the first stage of simulations development, when the user is asked to choose the calculation domain and the time step, and choices which in principle appear correct may frequently lead to meaningless results.
This work is centered on the 2D, transient analysis of a single side enclosure where the ventilation is only due to temperature differences. Wind effect has not been taken into consideration. Different runs have been performed varying: boundary conditions, window sizes and calculation domains. Field model results have been compared to lumped parameter and zone model analyses. A check on conservation principles has shown that CFD results are affected by noticeable inaccuracies for what concerns the prediction of both air temperature and ach's, which may be partially overcome re-scaling the time dependence of the phenomenon.

1 INTRODUCTION

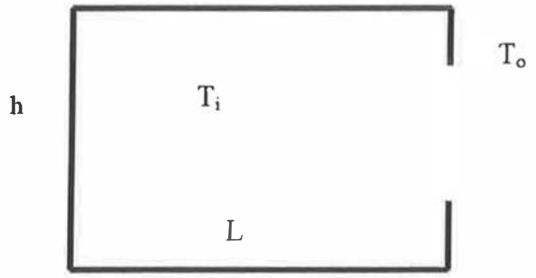
A detailed literature review developed during the fact-finding phase of the Annex 35-HybVent activity has pointed out the difficulties that arise when CFD technique is used to simulate naturally ventilated systems. This specially applies to airing (ventilation by means of door/window opening) (Schaelin et al, 1992, Elsayed, 1998), a simple action which produces qualitatively well known effects.
However, difficulties start since the first stage of simulations development, when the user has to choose the structure of the calculation domain. An apparently reasonable choice may, in fact, lead to surprisingly meaningless results under the physical point of view (e g, cold air entering the room through the upper part of the window and warm air exiting from below), while residuals values would suggest a successful simulation.
Furthermore, the use of simplified models and equations (see for examples Etheridge et al., 1996, Andersen, 1996, ASHRAE Handbook of Fundamentals, 1997, Agnoletto et al. 1981) is usually straightforward, but the user must provide one ore more "empirical" coefficients, whose value is not always known a-priori and may depend on the type of the opening as well as the temperature difference. Moreover, the phenomenon is, by its nature, unsteady and hence the value of temperature difference to be used in these formulas has to be forecasted as an average between initial and final conditions.
In this frame it has been decided to develop a CFD model and investigate the possibility to express the results in concise terms making use of non dimensional quantities such as Grashof number.

2 MODELS FEATURES

2.1 CFD Models

A two dimensional CFD transient analysis has been performed for a natural single side ventilated enclosure using a well known commercial software (FLUENT ®). Only thermal effects have been taken into account and therefore the wind speed has been assumed equal to zero.

In order to avoid difficulties in the description of the domain, the geometry of the room has been assumed very simple (see sketch below).



Two different CFD models have been implemented.

Both have been discretized using a non uniform grid made of 200 x 200 cells, but in the first one only the indoor environment is included (window 1), while in all the other models a strip 2 m wide of outdoor environment has been modelled.

A total of 5 runs have been performed varying window height and temperature differences. Table 1 reports the characteristics of the simulations.

The initial air temperature has been always taken equal to 20° C. The same value has been adopted for the wall temperature, considered constant.

Since the phenomenon is dominated by the buoyancy effect, the Grashof number has been identified as the relevant independent variable. It has been calculated as:

$$Gr = \frac{g \cdot \beta \cdot \Delta T \cdot H^3}{\nu^2}$$

Where ΔT = Temperature difference between wall and outdoors and H = window height.

Table 1 – Model features and boundary conditions.

Model	Outdoor env.	Window height [m]	ΔT [°C]	Grashof Number
Window 1	Not included	1.5	20	$1.17 \cdot 10^{10}$
Window 2	Included	1.5	20	$1.17 \cdot 10^{10}$
Window 3	Included	1.89	10	$1.13 \cdot 10^{10}$
Window 4	Included	1.5	10	$5.64 \cdot 10^{9}$
Window 5	Included	1.5	5	$2.77 \cdot 10^{9}$

The following assumptions were adopted:

- Turbulence model: standard k-ε
- Interpolation scheme: power-law
- Wall functions: standard log-law
- Transient analysis: variable time steps (values from 0.5 s up to 60 s).
- In the first 20-30 s of simulation, time steps larger than 0.5 s lead to numerical instability.
- Number of iterations per time-step: 1000
- Total number of simulated time-steps: about 100 (equivalent to a time span of about 600 s, for each of the simulated configurations for window 2 and 3, and about 150 s for window 4 and 5).
- Computational time: about 3 weeks (for each simulation).
- Hardware: HP Apollo 720 RISC WS (54 Mb RAM memory)

The solution phase is critical due to numerical instability problems and requires particular care. Moreover, as follow from the data listed above, it requires long time and resources.

2.2 Engineering Models

A single-zone model has been developed based on the formula reported by ASHRAE (1997) coupled with the conservation equation for energy:

$$h_p \cdot A_p \cdot (T_p - T_i) = \dot{m} \cdot c_p \cdot (T_i - T_o) + \rho_i \cdot V \cdot c_v \cdot \frac{\partial T_i}{\partial \tau}$$

$$\dot{m} = \rho_o \cdot A \cdot C_d \cdot \sqrt{\frac{g \cdot H \cdot (T_i - T_o)}{T_i}}$$

where the discharge coefficient C_d is given by:

$$C_d = 0.40 + 0.0045 \cdot |T_i - T_o|$$

The two-zone model is described by the following equations

$$h_{p1} \cdot A_{p1} \cdot (T_{p1} - T_1) = \dot{m} \cdot c_p \cdot (T_1 - T_o) + \rho_1 \cdot V_1 \cdot c_v \cdot \frac{\partial T_1}{\partial \tau}$$

$$h_{p2} \cdot A_{p2} \cdot (T_{p2} - T_2) = \dot{m} \cdot c_p \cdot (T_2 - T_1) + \rho_2 \cdot V_2 \cdot c_v \cdot \frac{\partial T_2}{\partial \tau}$$

$$\dot{m} = A \cdot \sqrt{\frac{g \cdot H \cdot \left(\rho_o - \frac{\rho_1 + \rho_2}{2}\right) \rho_o \cdot \rho_2}{\beta_1 \cdot \rho_2 + \beta_2 \cdot \rho_o}}$$

The meaning of the symbols is shown in Figure 1 and in the following list:

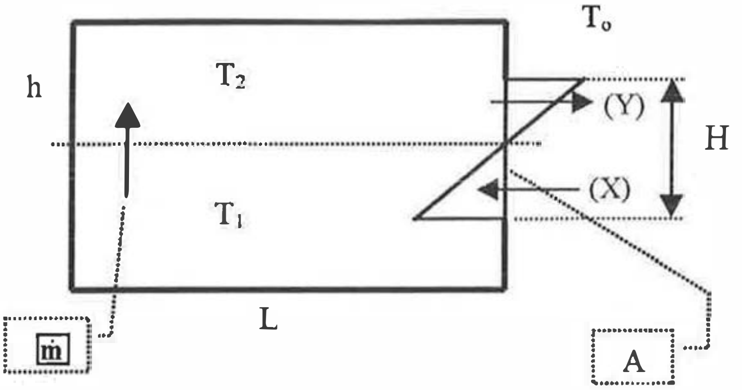


Fig. 1 – Two-zone model – calculation scheme.

A = half window surface
A_p = wall surfaces
h_p = film coefficient
c_v, c_p = heat capacities (constant volume/pressure)
T_p = wall temperature
T = air temperature
V = volume,
β = pressure loss coeff.
Subscripts
o,i,1,2 = outdoor, indoor, zone 1,2

The value of $\beta_1 = \beta_2$ has been determined assuming the initial air flow rate to be equal to the initial flow rate of the single-zone model. The relation for the air mass flow rate has been derived integrating the energy conservation equation along the air flow path from outside (X) to outside (Y). It has been assumed that no air short-cut takes place between the indoor air "exhausted" from the window and the outdoor air entering the room. Furthermore, for the two zone model the air flows only from the lower zone (1) and the upper zone (2), without internal recirculation.

These two models have been solved numerically, discretizing the ODE's and employing an explicit time integration. The model is implemented by means of a spread-sheet software (Excel ®). No particular problems of numerical instabilities have been encountered. However, in order to achieve accurate solutions (with precise energy and mass balances) quite small time steps have to be used, specially in the first part of the simulation (0.1 s for the first 8 seconds, larger, and increasing, time steps for the following time).

Furthermore, in the case of single zone model, Simulink ® (a MatLab toolbox for dynamlc system simulation) was also used for the model simulation. This tool, in fact, might show itself extremely useful in the HybVent system analysis, as it allows an easy coupling of different phenomena calculating quantities such as flow rates, pollutant concentration, temperatures, and introducing also the control strategies. At this stage of development only the flow rate model has been implemented. In all the tested cases, the computational time required for the solution (using a PC Pentium) is less than 1 s.

3 RESULTS

The use of engineering models is simple and straightforward. The only "innovation" introduced by the authors consists in the unsteady state application of formulas expressing the air mass flow rate, $\dot{m}$. The air flow rate, in fact, is evaluated at each integration time step, adopting for the calculation the previous value of air temperature.

The profiles of ach's and air mean temperature versus time (not shown here for brevity) obtained by means of single- and two-zone models[1] are quite similar. The use of other formulas (Etheridge, 1996) for $\dot{m}$ has produced little differences in the final results. In the same way results obtained by means of Excel software are identical to those obtained adopting the Simulink model.

Table 2 resumes the indoor air mean temperature and the ach's values when steady state conditions are practically reached (and the corresponding time required) determined by means of engineering models.

For what concerns the CFD analysis, as mentioned in the introduction, the choice of the geometrical domain has revealed to have a paramount effect in the reliability of the results. Actually, the results of model *window 1*, for which the outdoor environment has been modelled by means of proper boundary conditions (fixed pressure boundaries) are completely meaningless. After a few seconds when the air, as expected, flows from outdoors to indoors in the lower part of the window and vice versa in the upper part (although the neutral level appears to be strikingly low), there is an inversion of the air path and the warm air starts to flow from indoors to outdoors in the lower area of the window, contradicting the common experience. This simple example is a further evidence that CFD analyses choices which appear straightforward (specially for non expert users) may lead to wrong conclusions, even when the numerical indicators (residuals) assume satisfactory results. Being meaningless, the results related to model *window 1* will not be included in the following figures.

Results related to models *window 2* through *5,* are consistent with the expected air flow behaviour. The analysis of air velocity and mass flow rate profiles along the window height shows quite symmetrical trends that develop since the first seconds of simulation. In the first time steps the profile is slightly irregular (and the global mass balance of the rooms is not perfectly satisfied), but after about 2 seconds the curves are smooth and the mass balance is, practically, perfect. Figure 2 shows for the model *window 4*, as an example, the profiles of mass flow rate across the windows at different time steps.

In figure 3 the air changes per hour are plotted versus time for the different models. It must be underlined that the adopted model is 2-D, thus the room depth is assumed to be equal to 1 m. Consequently, the room volume V, adopted for the ach's calculation, is: V = 4.2 m x 2.7 m x 1m = 11.34 m^3.

From a physical point of view the system behaves as if it were an L x h enclosure of infinite width with a continuous "strip" window. The results are strictly applicable only to this type of window, but they may probably be extended to other configurations whether the edge effects of the window sides are negligible (i e, not too high values of ratio height to width). The effect of window to room width ratio is not known a priori and should be investigated.

In order to obtain the actual value of ach related to a particular window and room width, one must multiply the values shown in the following figures by the ratio of window width to room width. Figure 4 shows the air temperature (room mean value) profile versus time. In these charts 8 curves are plotted: four refer to CFD results, four refer to the two-zone model simulations. It is possible to see that there are large discrepancies between the results obtained by means of the two different classes of models. The analysis of the energy balance at each time step has revealed quite large errors in the case of CFD models. This appears unexpected; in fact, during the solution phase of all the models the residuals related to enthalpy were quite low. In figure **5** is possible to see the entity of power imbalance a t various time steps (curves with symbols – refer to the main axis on the left

---

[1] In the two-zone model the room mean temperature is determined as the simple average between zone 1 and 2 air temperatures.

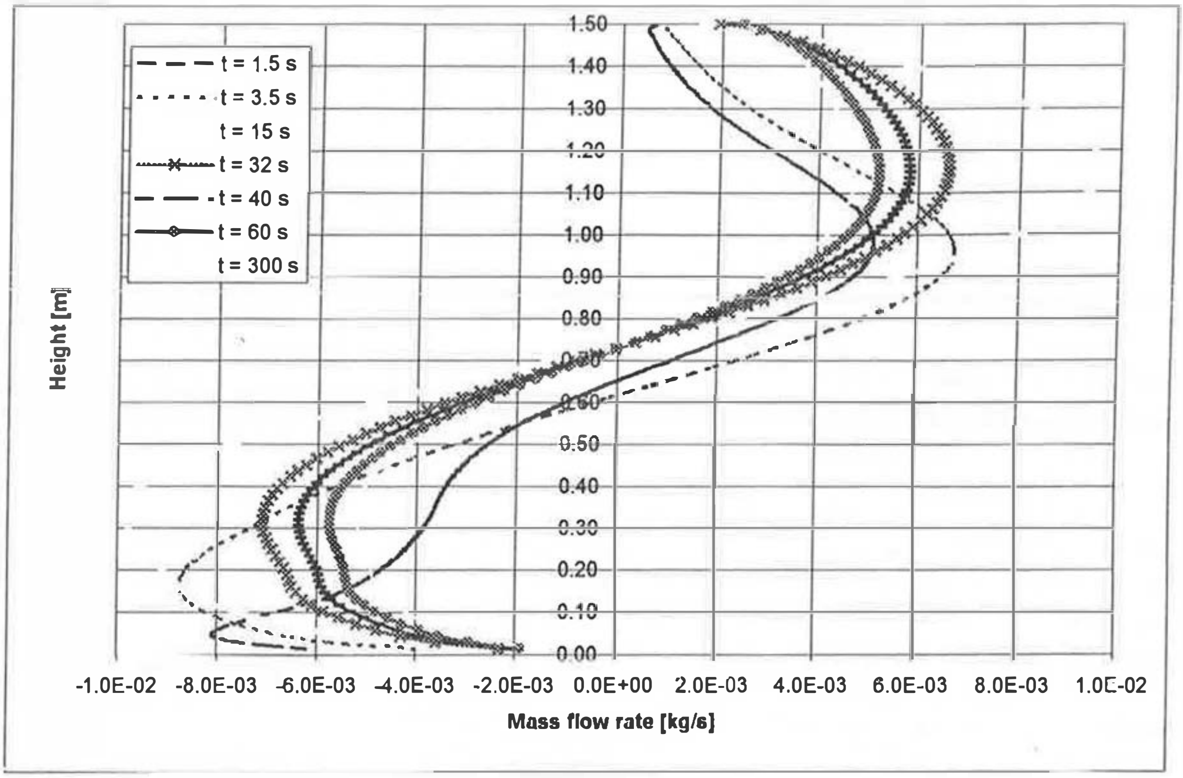


Figure 2 –Temperature profile along the window height for different time steps - CFD results – *Window 4*.

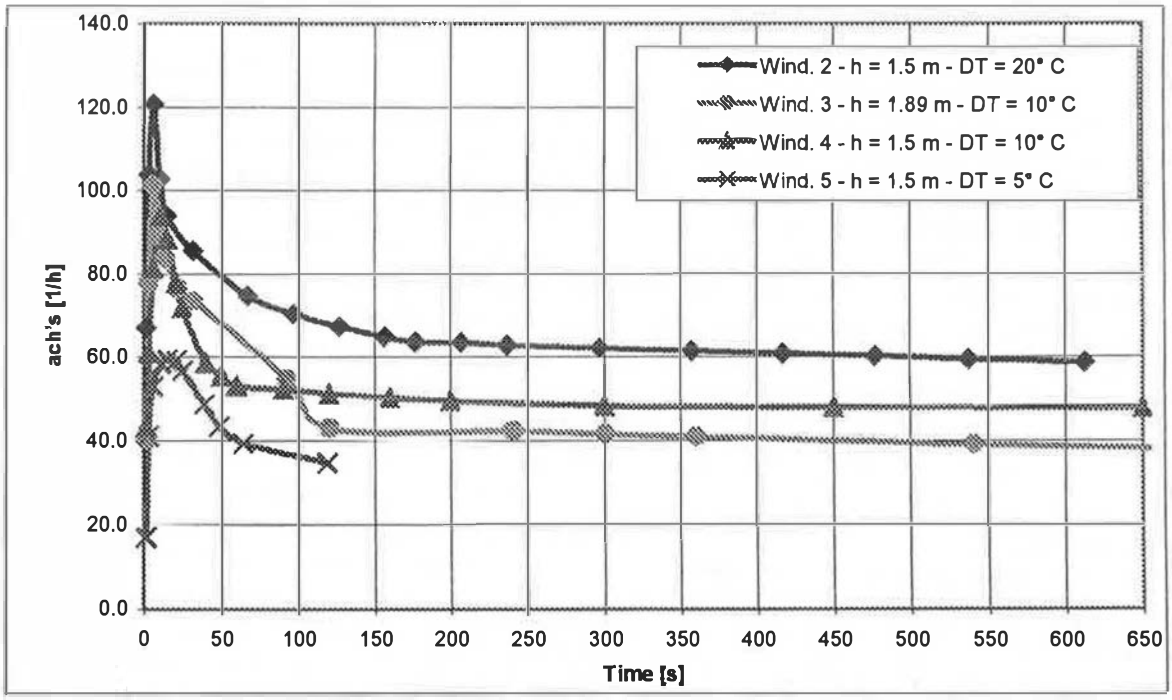
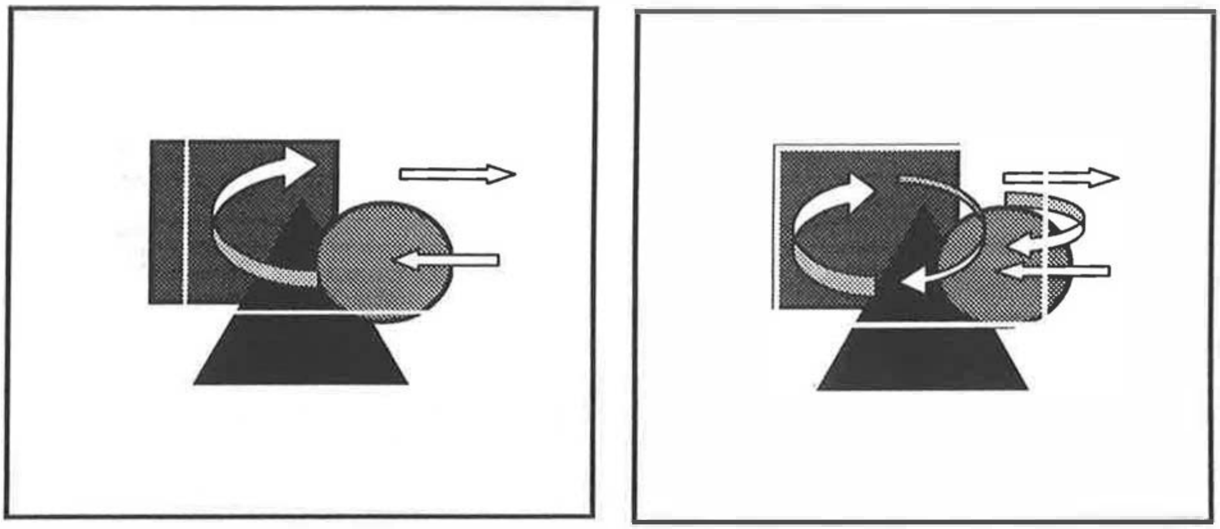


Figure 3 – CFD results – Time history of air change rate.

of the chart) and the relative error defined as: $E = \frac{(\text{Power imbalance}) - \dot{H}}{\dot{H}} \cdot 100$ (symbols - refer to the secondary axis on the right of the chart).

An off-line study of each term of the energy balance equation, performed by means of simplified analytical calculations, has pointed out that the components of the energy balance due to heat fluxes exchanged between walls and air and to enthalpy fluxes are apparently well predicted by the code. Instead, the time variations of internal energy (i e, temperature) seem to be largely underestimated. Everything happens as if the time step intervals adopted for the numerical solution procedure do not coincide with the actual time scale of the phenomenon. Starting from these remarks it has been decided to re-scale all the CFD numerical results on the basis of time steps, $\Delta\tau_r$, obtained imposing the energy balance at each time step: $\Delta\tau_r = \dfrac{\rho \cdot V \cdot c_v \cdot \Delta T}{\dot{Q} - \dot{H}}$.

Figures 6 to 9 show the results of such a procedure in terms of ach's and mean air temperature versus time. It is possible to see a general substantial improvement in the predictions, with a good agreement with profiles calculated by means of zonal models, particularly for what concerns *window 2, 4 and 5* models. In the case of *window 3* model, instead, the performances of the procedure seems to be worst. However, in this last simulation substantial re-circulation of warm exhaust air with fresh entering outdoor air occurs. Due to this mixing, the air actually entering the enclosure has a temperature slightly higher than the outdoor air. This phenomenon is clearly shown by the CFD simulation temperature fields of which figure 10 is an example (fig. 10a – *window 2* model, fig.10b *window 3* model. Time: about 22 s after the window opening).



(a) (b)

Figure 10 – Temperature field after 22 s and main air flow paths – Model window 2 and window 3

The lower temperature differences between indoor and outdoor induce a smaller air flow rate. It follows that the time profile of air mean temperature predicted by CFD calculation differs from that obtained by means of the two-zone model.
From a physical point of view this behavior is probably linked to the fact that in *window 3* model (that from a theoretical point of view should present the same ach's of *window 2*, since both cases have the same Gr number), the air velocities through the larger window are quite low and are influenced by the great clockwise vortex that takes place inside the room due to indoor thermal gradients (while for window 2 the initial vortex is destroyed by the stronger air current that flows in and out the room). Such kind of phenomenon could never be predicted by zonal models, as they assume a priori no re-circulation and mixing.

Table 2 – Steady state values – zone models

Model	n [1/h]	T [°C]	n [1/h]	T [°C]	τ [s]
	Single zone		Two zone		
Window 2	35.4	4.7	37.0	3.7	≈150
Window 3	34.8	12.5	37.1	11.9	≈150
Window 4	27.1	13.0	29.0	12.3	≈220
Window 5	20.8	16.8	22.6	16.4	≈220

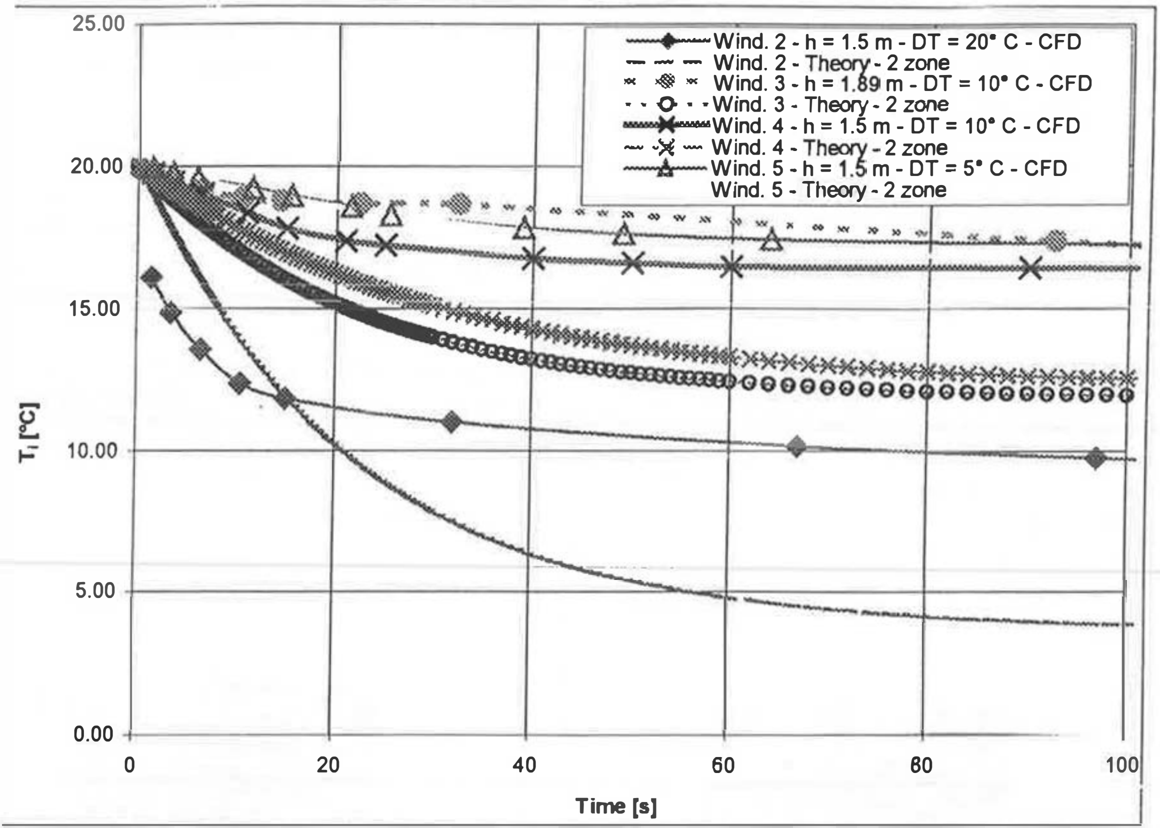


Figure 4 – Air temperature (room mean values) versus time (CFD and 2-zone model).

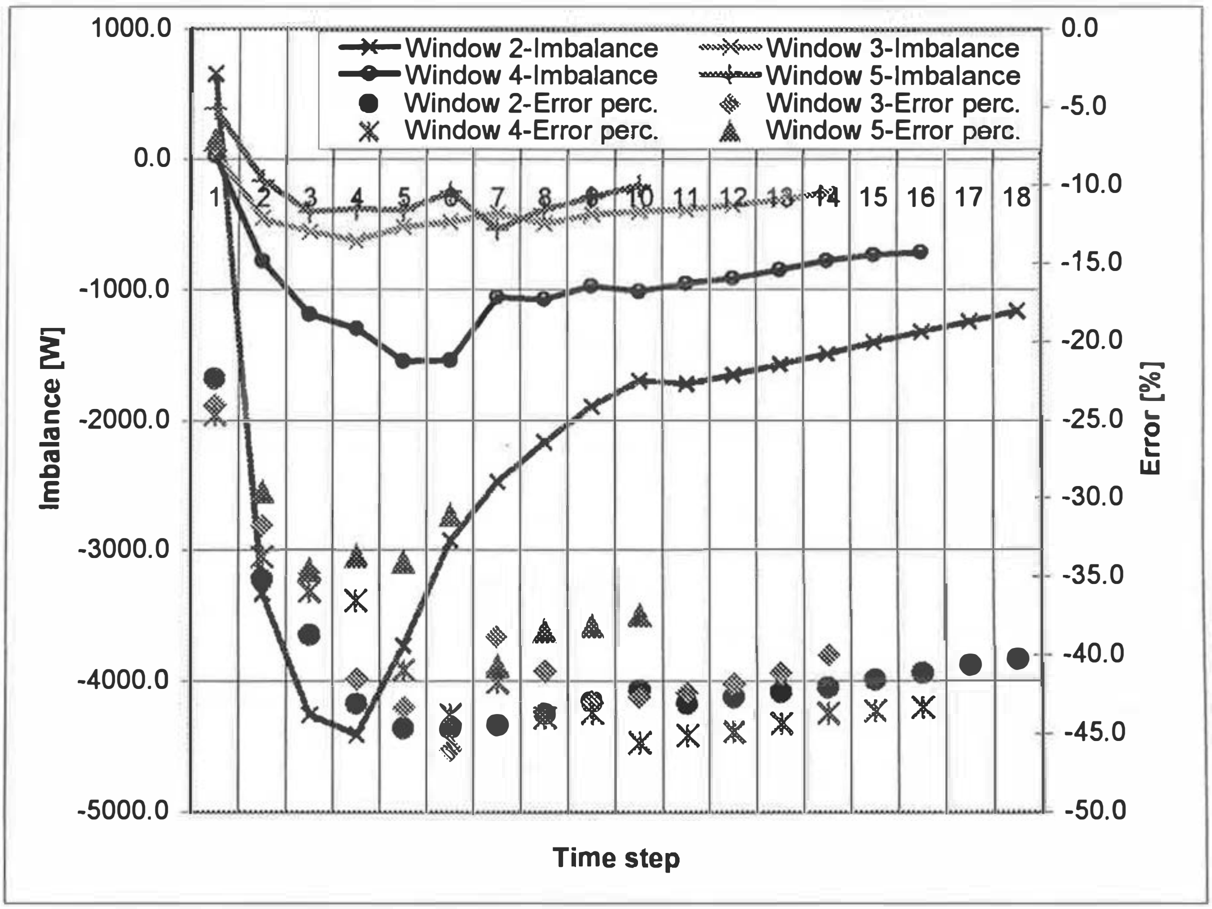


Figure 5 – Power imbalance and relative errors of CFD simulations.

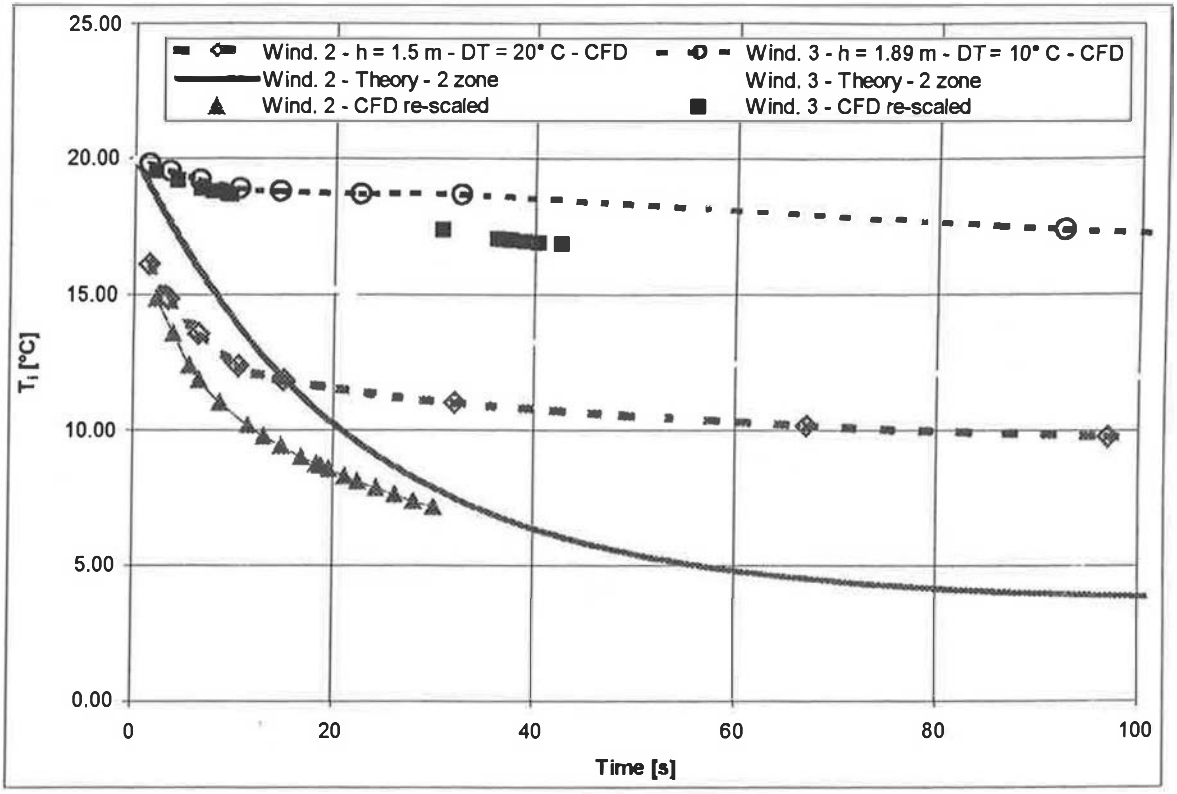


Figure 6 – Air temperature (room mean value) versus time (CFD, 2-zone and re-scaled values) – Models Window 2 and 3.

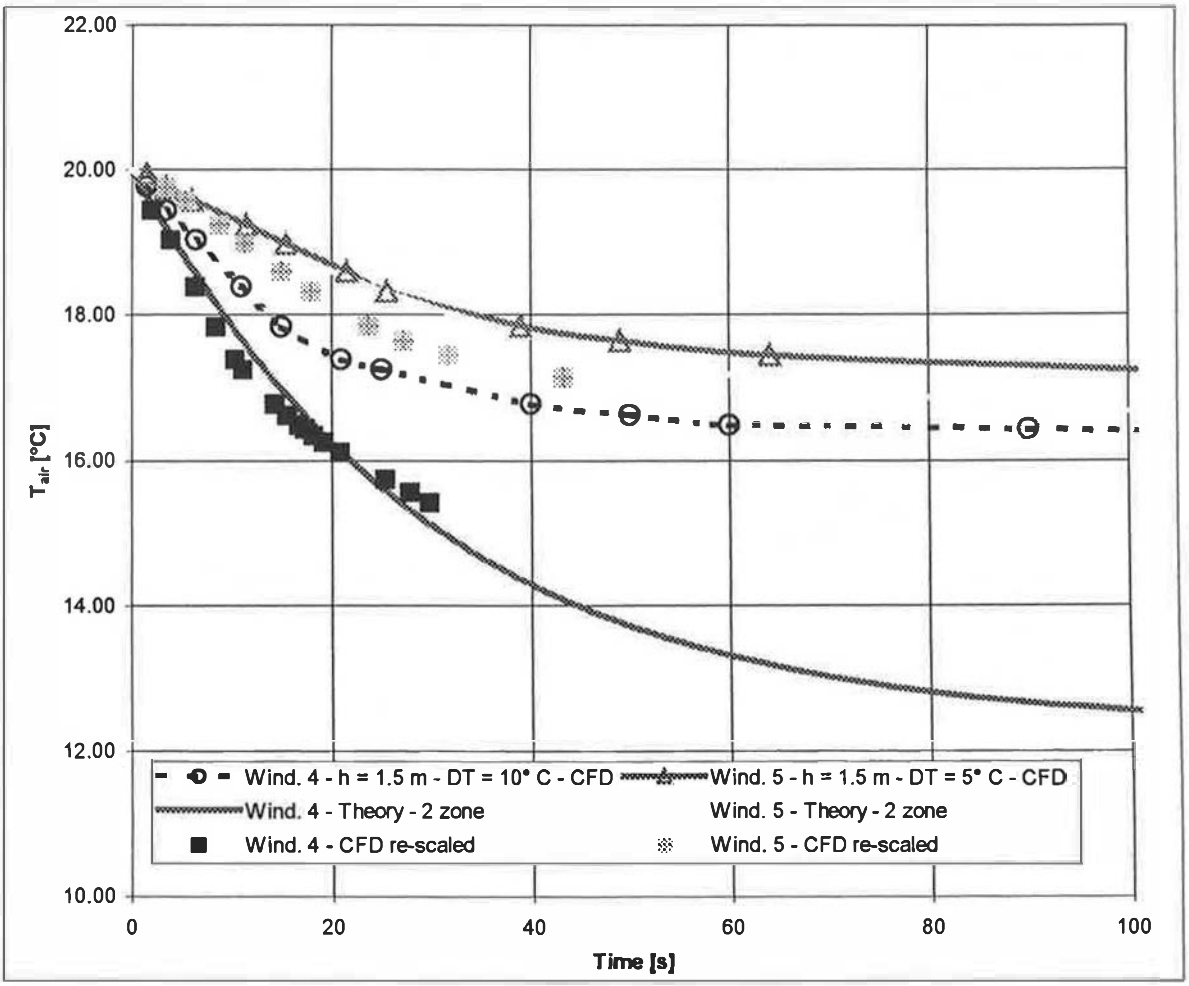


Figure 7 – Air temperature (room mean value) versus Time (CFD, 2-zone and re-scaled values) – Models Window 4 and 5.

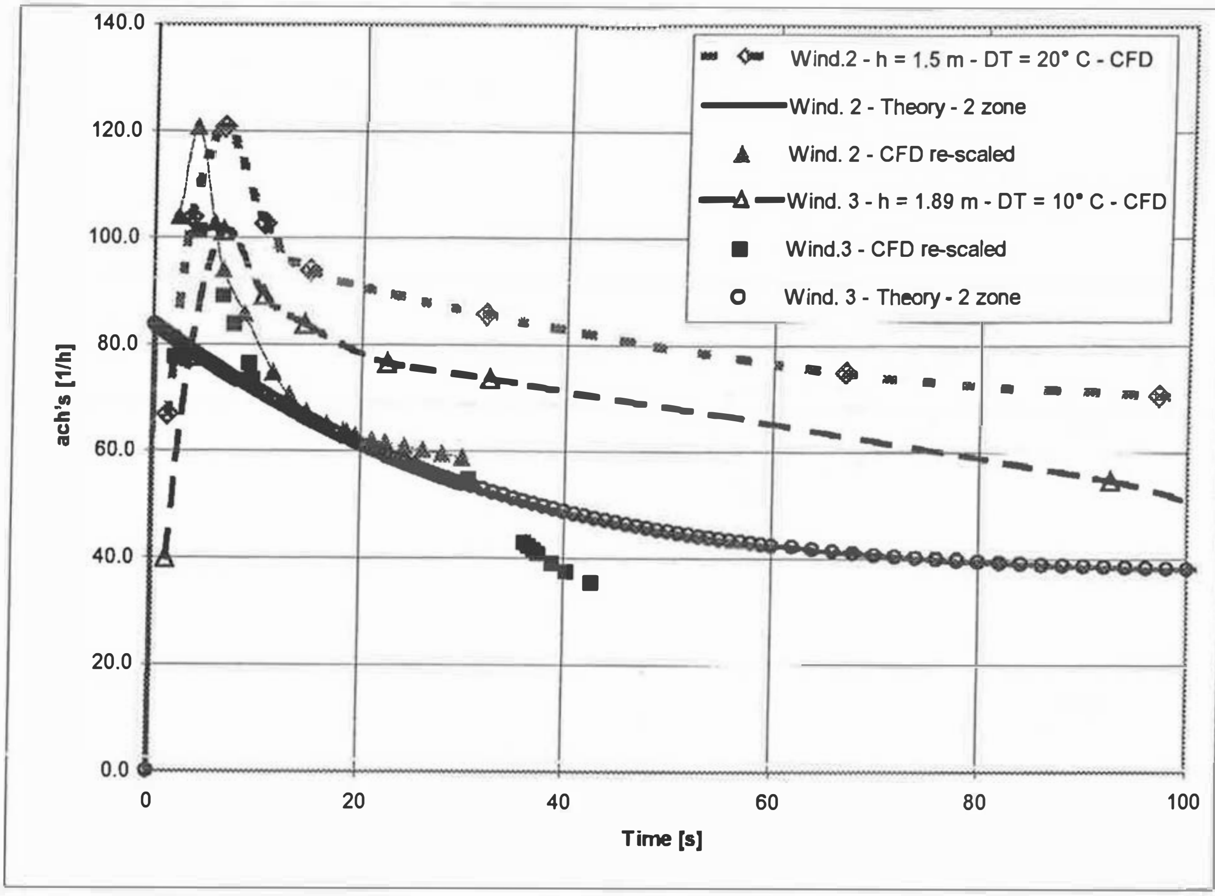


Figure 8 – Air changes per hour versus Time (CFD, 2-zone and re-scaled values) – Models Window 2 and 3.

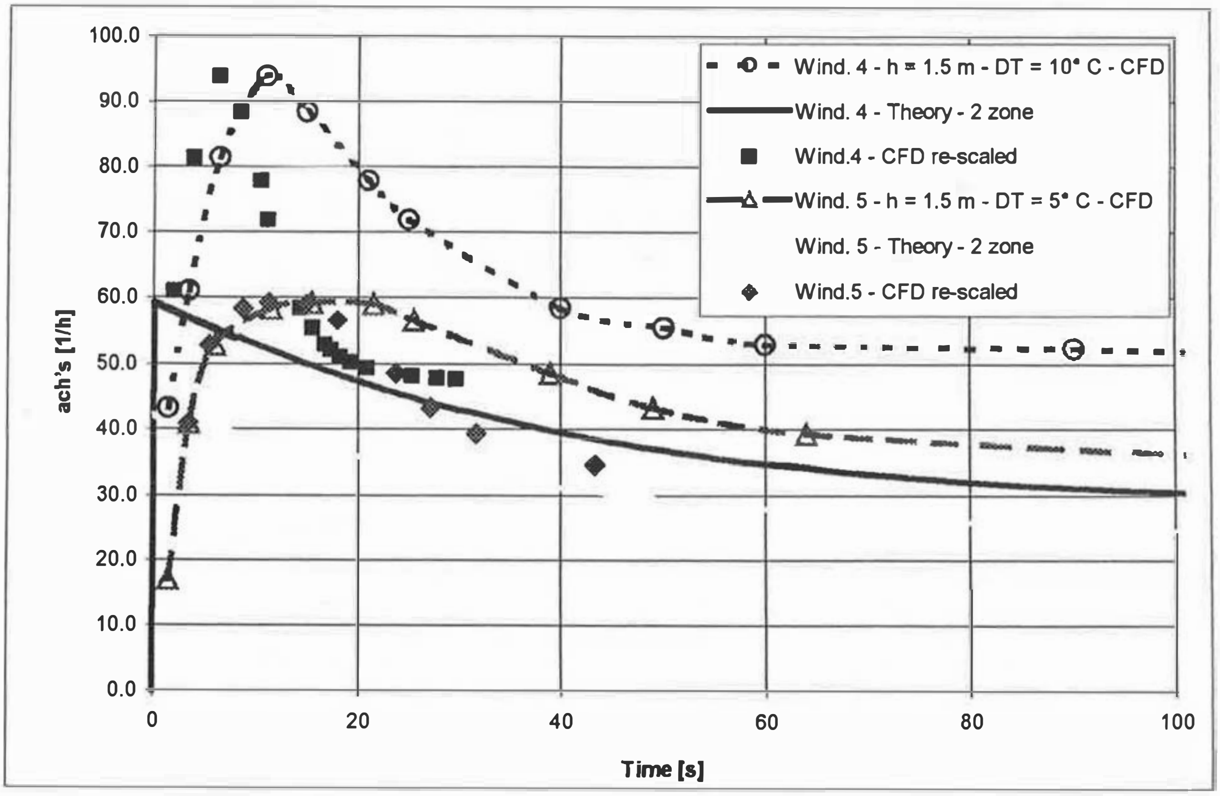


Figure 9 – Air changes per hour versus Time (CFD, 2-zone and re-scaled values) – Models Window 4 and 5.

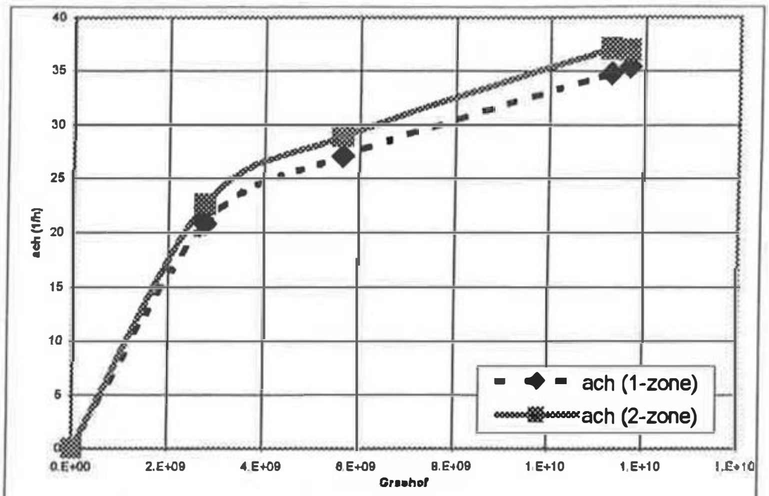


Fig. 11 –Steady-state ach's vs. Grashof number.

4 CONCLUSIONS

The work is, actually, still in progress. The aim of the paper was to verify the applicability of CFD analysis to a simple, yet physically complex phenomenon such as the consequences of opening a window under the mere effect of a temperature difference between indoors and outdoors.
It has been shown that CFD should be used very carefully, with a suitable choice of the calculation domain.
Furthermore, there is an apparent contradiction between the calculation time scale and energy conservation principles. This contradiction has been solved re-scaling the time history by forcing the solution to comply with the energy balance.
Simplified "analytical" models have also been developed, and their results have been compared to the CFD model results. After the time re-scaling, there is a fair agreement between CFD and engineering models, except for *window 3*, for which the CFD analysis revealed a certain degree of outdoor-indoor air recirculation.
A relationship between air changes per hour at steady state conditions and Grashof number has been derived (see figure 11). Both curves show a definite functional dependency upon Grashof number and could be used for first attempt prediction.

5 REFERENCES

Agnoletto L., E. Grava, La ventilazione naturale degli ambienti, Condizionamento dell'Aria, ottobre 1981.
Andersen K.T., Design of natural ventilation by thermal buoyancy – theory, possibilities and limitations, 5th Int. Conf. On air distribution in rooms, ROOMVENT '96, July 1996, Yokohama.
ASHRAE, Handbook of Fundamentals, Ch. 25, 1997.
Butera F., G. Cannistraro, M. Yaghoubi, A. Lauritano, Benessere termico e ventilazione naturale negli edifici, HTE Energie Alternative, anno 11, no. 59, maggio-giugno 1989.
Elsayed M., Infiltration Load in Cold Rooms, HVAC&R Research, Vol. 4 No. 2, April 1998.
Etheridge D., M. Sandberg, Building ventilation – Theory and Measurements, John Wiley & sons, Chichester, 1996, pp. 89-95,
Schaelin, A., Van der Maas J., Moser, A., Simulation of air flow through large openings in buildings, ASHRAE Transactions, part 2, 1992.

ACKNOWLEDGMENTS

The research activity presented in this work has been developed in the frame of "*Indoor Environment Engineering* "– a project co-funded by the Italian Ministry of University and Reasearch (MURST). The participation of the authors to IEA Annex 35 is supported by ENEA – Ente Nazionale Energie Alternative – Italy.

Outline of HybVent

Per Heiselberg
Indoor Environmental Engineering
Aalborg University
Sohngaardsholmsvej 57, DK-9000 Aalborg, Denmark
fax: +45 9814 8243
e-mail: ph@civil.auc.dk

1 ANNEX DESCRIPTION

The international project Annex 35 “Hybrid ventilation in New and Retrofitted Office Buildings was accepted by the IEA at the Energy Conservation in Buildings & Community Systems Executive Committee Meeting in Washington June 1997. The first year, starting August 1 1997, was a preparatory year and the four year working phase started August 1998. The Annex have participants from 15 countries: Australia, Belgium, Canada, Denmark, Finland, France, Germany, Greece, Italy, Japan, Norway, Sweden, The Netherlands, United Kingdom and USA.

1.1 Background

Soon after the energy crisis in 1973 everybody focused their attention on thermal insulation, airtightness of buildings and heat recovery to decrease energy consumption for heating (and cooling) of buildings. Buildings were designed to be isolated from the outdoor environment with an indoor environment controlled by artificial lighting, mechanical ventilation and heating and cooling systems.

Today, in the design of new buildings and retrofit of old buildings, the attention has been turned towards a more integral energy design with focus not only on thermal insulation, airtightness and heat recovery but also on optimal use of sustainable technologies as passive solar gains, daylight and natural ventilation. The buildings are designed in an interplay with the outdoor environment and are utilizing it to create an acceptable indoor environment whenever it is beneficial.

The extent to which sustainable technologies can be utilized depends on outdoor climate, building use and building location and design. Under optimum conditions sustainable technologies will be able to fulfil the demands for heat, lighting and fresh air, while in some cases supplementary mechanical systems will be needed and in other cases it will not be possible to use sustainable technologies at all.

In well thermally insulated office buildings, which are more and more frequent in IEA countries, ventilation (and cooling) account for more than 50% of the energy requirement, and a well-controlled and energy-efficient ventilation system is a prerequisite to low energy consumption. Natural ventilation and passive cooling are sustainable, energy-efficient and clean technologies as far as they can be controlled, (that is if well modelled and understood). They are well accepted by occupants and should therefore be encouraged wherever possible.

Unfortunately, the design of energy-efficient ventilation systems in office buildings is often turned into a question of using either natural ventilation and passive cooling or mechanical ventilation and cooling. This prevents a widespread use of sustainable technologies because a certain performance cannot be guaranteed under all conditions. In fact in the large majority of the cases a combination of systems, hybrid ventilation, would be beneficial depending on outdoor climate, building design, building use, and the main purpose of the ventilation system.

The number of office buildings to be retrofitted in most IEA countries is now much larger than the potential for new buildings. In many cases there is a large potential for use of sustainable technologies either as a supplement to the existing mechanical systems or as the main part of solutions in cases where classic ventilation systems are impossible to install in an existing building. Innovative hybrid ventilation systems should be developed or improved for that purpose.

1.2 Definitions

Hybrid ventilation systems can be described as systems providing a comfortable internal environment using both natural ventilation and mechanical systems, but using different features of the systems at different times of the day or season of the year. It is a ventilation system where mechanical and natural forces are combined in a two mode system.

The basic philosophy is to maintain a satisfactory internal environment by alternating between these two modes to avoid the cost, energy penalty and consequential environmental effects of full year round air conditioning. The operating mode varies according to the season and within individual days, thus the current mode reflects the external environment and takes maximum advantage of ambient conditions at any point in time. The main difference between conventional ventilation systems and hybrid systems is the fact that the latter are intelligent with control systems that automatically can switch between natural and mechanical mode in order to minimize the energy consumption.

Hybrid ventilation should dependent on building design, internal loads, natural driving forces, outdoor conditions and season fulfil the immediate demands to the indoor environment in the most energy-efficient manner. The control strategies for hybrid ventilation systems in office buildings should maximize the use of ambient energy with an effective balance between the use of advanced automatic control of passive devices and the opportunity for users of the building to exercise direct control of their environment. The control strategies should also establish the desired air low rates and air flow patterns at the lowest energy consumption possible.

Figure 1 shows the definition of hybrid ventilation as agreed on in Annex 35.

Definition of Hybrid Ventilation

Hybrid Ventilation is a two mode system which is controlled to minimize the energy consumption while maintaining acceptable indoor air quality and thermal comfort. The two modes refer to natural and mechanical driving forces.

Purpose of Ventilation

All hybrid systems have to provide air for indoor air quality purposes, but some in addition to that also provide air for thermal conditioning and thermal comfort during working hours.

Purpose of Control System

The purpose of the control system is to establish the desired air flow rate and air flow pattern at the lowest energy consumption possible.

Figure 1. Definition of hybrid ventilation in Annex 35.

1.3 Objectives

The Annex 35 research project is aiming at a better knowledge of hybrid systems and focusing on development of control strategies and performance prediction methods for hybrid ventilation in new and retrofitted office buildings. Its main objectives are:

- to develop control strategies for hybrid ventilation systems in new build and retrofit of office and educational buildings
- to develop methods to predict ventilation performance in hybrid ventilated buildings
- to promote energy and cost effective hybrid ventilation systems in office and educational buildings
- to select suitable measurement techniques for diagnostic purposes to be used in buildings with hybrid ventilation systems

2 STRATEGY AND APPROACH

To fulfil the objectives the work is divided in the following tasks:

Subtask A: Development of control strategies for hybrid ventilation
Subtask B: Theoretical and experimental studies of performance of hybrid ventilation. Development of analysis methods for hybrid ventilation
Subtask C: Pilot studies of hybrid ventilation

An overview of the approach can be seen on figure 2, which is showing a matrix of annex tasks and the research methods used.

Task	Research Method		
	State-of-the-art Review	Theoretical and Experimental Studies	Implementation and Demonstration
Subtask A Development of Control Strategies	• Survey of existing strategies • Local versus central control	• Definition of requirements and evaluation criteria for control strategies • Development of strategies for switching between and combining ventilation modes • Development of strategies for combination of automatic and manual individual control • Control system design	• Demonstration and evaluation of control strategies
Subtask B Development of Analysis Methods	• Survey of available analysis methods	• Achieve better understanding of the physics of hybrid ventilation (air flow control) • Integration of air flow and thermal simulation models • Development of probabilistic analysis method	• Application and evaluation of analysis methods
Subtask C Pilot Studies	• Survey of existing systems and solutions to specific problems • Market survey of components • Survey of building codes	• Analysis of hybrid ventilation components and systems • Analysis of barriers for hybrid ventilation application • Cost-benefit analysis of hybrid ventilation	• Demonstration of hybrid ventilation performance • Technology transfer

Figure 2. Approach of Annex 35 divided into different tasks and research methods.

2.1 Development of Control Strategies

A hybrid ventilation system, which is integrating both natural and mechanical driving forces in the same ventilation system, requires development of new control strategies. These strategies should ensure at any time and for a certain combination of internal loads, outdoor conditions and comfort requirements that the immediate demands to the indoor environment are fulfilled in the most energy efficient manner. As the function of hybrid ventilation is closely related to the use and function of the building a thorough control of hybrid ventilation requires a completely integrated approach where building design, its technical systems (lighting, heating), occupant behavior, surroundings, climatic and meteorological conditions etc., are taken into consideration.

The participants will as a starting point take a typical case in their own country and climate and by theoretical studies, laboratory experiments and field studies of the performance of different control strategies in a hybrid ventilated building develop the most suitable strategies. The main focus will be on development of strategies for switching between ventilation modes and for combining central automatic and individual manual control.

In the development of new control strategies the focus will be on different issues in the participating countries and will cover a range of hybrid ventilation system and building designs. One of the major tasks will be on development of optimum fuzzy controllers that will enable the implementation of real multicriteria control strategies incorporating expert knowledge and on the development and comparison of smart setting and tuning techniques for these controllers. This will enable a rational operation and improved performance of the fuzzy controllers and is a necessary condition for implementing complex control techniques.

2.2 Theoretical and Experimental Studies of Hybrid Ventilation Performance

Thorough understanding of the hybrid ventilation process is a prerequisite for a successful application of hybrid ventilation, for development of optimum control strategies and for development of analysis methods for hybrid ventilation design. The annex will therefore by theoretical and experimental studies investigate the different elements of the air flow process in hybrid ventilation from air flow around buildings, air flow through openings, air flow in rooms to air flow between rooms in a building. The hybrid ventilation process is very dependent on the outdoor climate as well as the thermal behavior of the building and therefore, it is essential to take all these factors into consideration as well as the air flow process of whole systems.

2.3 Development of Analysis Methods

Suitable analysis methods as we know them for mechanical systems are not available for hybrid ventilation systems. Valid methods would give architects and engineers the necessary confidence in system performance, which in many cases, is the decisive factor for choice of system design.

As the hybrid ventilation process and the thermal behavior of the building are linked the development of analysis methods for hybrid ventilation must take both aspects into consideration at the same time and include efficient iteration schemes. This is the case for all types of analysis methods from simple analytical methods, zonal and multizone methods to detailed CFD analysis methods. The subtask will deal with methods on different levels, but a major focus will be on combining thermal simulation models with existing multizone air flow models. In this way the thermal dynamics of the building can be taken into account and this will improve the prediction of the performance of hybrid ventilation considerably. The combined model will be the most important design tool for hybrid ventilation.

The second major development is a new probabilistic analysis method that makes it possible to evaluate indoor climate, energy consumption and certainty of design solution based on the whole operation period. The method should be able to predict the probability that demands of energy consumption, indoor climate and air flow rates are met in hybrid ventilated buildings. The method will be developed, by combining available physical models of the phenomena involved with stochastic models and will be useful in the early design phase.

2.4 Pilot Studies of Hybrid Ventilation

Pilot studies in different countries are used to implement hybrid ventilation systems and demonstrate their performance. The pilot studies are monitored to collect data on performance (IAQ, thermal comfort and energy consumption) and to evaluate corresponding control strategies and analysis methods. The pilot studies include both retrofitted and new build designs and highlight similarities and differences in climatic issues (including seasonal differences), institutional and cultural issues (developers and occupants), and technology transfer issues. The pilot studies concentrate on success stories of hybrid ventilation but also critically highlight problematic cases.

Buildings with hybrid ventilation often include other sustainable technologies like daylightning, passive cooling, passive solar gains etc, and an integrated approach is used in the design of the building and its technical systems.

The pilot studies have hybrid ventilation systems according to the definition applied in this annex. The performance (IAQ, thermal comfort, energy consumption, etc.) of the hybrid ventilation system with the corresponding control strategies are monitored during a one year period. Measurement data are also provided for evaluation of analysis methods and/or control strategies. An analysis of barriers for hybrid ventilation application and a cost-benefit analysis of the alternatives to ventilation system design are provided as well.

Table 1 shows the pilot studies in Annex 35

Table 1. Annex 35 Pilot Studies

Country	Building name	Location	Year of completion	Building type
Belgium	PROBE	Limelette	1975/1997	Office
Denmark	B&O HQ	Struer	1998	Office
Italy	Palzzina I Guzzini	Recanati	1997	Office
Japan	The Liberty Tower	Meiji	1998	Educational
Japan	Tokyo Gas Earth Port	Tokyo	1996	Office
Japan	Fujita Technical Center		1999	Office
Norway	Grong Primary School	Grong	1997	Educational
Norway	School	Oslo	1999	Educational
Norway	Lavollen	Trondheim	?	Office
The Netherlands	Waterland	Leidschenveen	2000	Educational

2.5 Annex 35 Workgroups

The work in subtasks A and B is organized in a number of workgroups, which deals with some of the most important problems in hybrid ventilation design. Table 2 shows the 12 workgroups functioning at the moment and the titles of their work. The workgroups will be closed, when they have finished their work, and new ones may be formed during the remaining working period of the Annex.

Table 2. Annex 35 Working Groups

Workgroup	*Title*	*Leading Country*	*Countries participating*
WG-A1	**Characterization of Ventilation and Control Strategies**	DK	B, DK, F, FIN, N, NL
WG-A2	**Equivalent Energy Performance Targets in Standards and Regulations**	B	B, DK, F, FIN, NL
WG-A3	**Comfort Requirements and Energy Targets**	NL	AU, DK, F, N, NL
WG-A4	**Application of Analysis Methods in the Hybrid Ventilation Design Process**	N	AU, DK, F, N, NL
WG-B1	**Incorporate Thermal Stratification Effects in Network Modeling**	AU	AU, CAN, F, I, N
WG-B2	**Methods for Vent Sizing**	NL	B, CAN, DK, I, NL, S
WG-B3	**Input Data Bank**	UK	G, S, UK
WG-B4	**Develop Probabilistic Methods**	DK	AU, DK, I
WG-B5	**Wind Flows through Large Openings**	S	AU, CAN, DK, F, I, NL, S
WG-B6	**Evaluation of Analysis Tools – Specification of Data Requirement**	AU	AU, FIN, D, I, N
WG-B7	**Integrate or Implement Control Strategies into Models**	AU	AU, F, FIN, G, S, NL, USA
WG-B8	**Climate Data**	AU	AU, B,G, I

3 RESULTS AND END PRODUCTS

The results of Annex 35 will be summarized in two final reports and specific results of individual subtasks will be reported in technical reports and papers.

3.1 State-of-the-art Review

This report will describe the state-of-the-art of hybrid ventilation technologies, of control strategies and algorithms and of analysis methods. The report will provide examples of existing systems. It will show solutions to specific problems (fresh air supply, excess heat removal, etc.) in particular office buildings located in different outdoor climates and using different commercially available hybrid ventilation components.

The report will focus on the impact of differences in climate (including seasonal differences as winter heating and summer cooling), building design, building use and internal loads on energy performance, indoor air quality and thermal comfort.

3.2 Principles of Hybrid Ventilation

This report will describe the principles of hybrid ventilation, including solutions for energy-efficient, comfortable and cost-effective hybrid ventilation and recommendations on control strategies and analysis methods. The report will be written on the basis of experience gained in annex subtasks as well as achievements from previous research (state-of-the-art review).

Project Profile Annex 35: HybVent

Title:	"Hybrid Ventilation in New and Retrofitted Office Buildings, Annex 35, a task-sharing Annex to the IEA Implementing Agreement on Energy Conservation in Buildings and Community Systems.
Objective:	• to develop control strategies for hybrid ventilation systems in new build and retrofit of office and educational buildings • to develop methods to predict hybrid ventilation performance in hybrid ventilated buildings • to promote energy and cost-effective hybrid ventilation systems in office and educational buildings
Products:	The results of Annex 35 will be summarised in two final reports • State-of-the-art Review of Hybrid Ventilation • Principles of Hybrid Ventilation
Pilot Studies:	8 Pilot Studies serve to demonstrate implementation of hybrid ventilation concepts and hybrid ventilation performance
Subtasks:	**Subtask A:** Development of control strategies for hybrid ventilation **Subtask B:** Theoretical and experimental studies of performance of hybrid ventilation. Development of analysis methods for hybrid ventilation **Subtask C:** Pilot studies of hybrid ventilation
Time Schedule:	1-year Preparation Period: August 1, 1997 - July 31, 1998 4-year Project Period: August 1, 1998 - July 31, 2002
Participants:	Australia, Belgium, Canada, Denmark, Finland, France, Germany, Greece, Italy, Japan, Norway, Sweden, The Netherlands, United Kingdom, U.S.A.
Operating Agent:	Per Heiselberg, Indoor Environmental Engineering, Aalborg University, Aalborg, Denmark
Subtask Leaders:	Subtask A: Gérard Guarracino, ENTPE, Lyon, France Subtask B: Yuguo Li, CSIRO, Highett, Vic., Australia Subtask C: Marco Citterio, ENEA, Rome, Italy
Meetings:	Proj. Def. Workshop: Aalborg, Denmark, October 22-24, 1997 Kick-Off Meeting: Rome, Italy, March 25-27, 1998 1st Expert Meeting: Trondheim, Norway, October 1-4, 1998 2nd Expert Meeting: Lyon, France, April 20-23, 1999 3rd Expert Meeting: Sydney, Australia Sept. 28-Oct. 1, 1999
Information:	**http://hybvent.civil.auc.dk**

CHARATERISTICS OF AIR FLOW THROUGH WINDOWS

Per Heiselberg, Henrik Dam, Lars C. Sørensen, Peter V. Nielsen, Kjeld Svidt
Indoor Environmental Engineering
Aalborg University
Sohngaardsholmsvej 57, DK-9000 Aalborg, Denmark
Fax: +45 9814 8243
E-mail: ph@civil.auc.dk

ABSTRACT

This paper describes the first results of a series of laboratory investigations that is performed to characterise three different window types. The results show the air flow conditions for different ventilation strategies and temperature differences. For one of the windows values of the discharge coefficient are shown for both isothermal and non-isothermal flow conditions and the thermal comfort conditions are evaluated by measurements of velocity and temperature levels in the air flow in the occupied zone.

It is demonstrated that different window types have quite different characteristics. A combination of different window types in the same natural ventilation design can by using their strong sides improve both ventilation capacity, thermal comfort and IAQ.

1 INTRODUCTION

In natural ventilation systems fresh air is often provided through opening of windows. There is a wide range of possibilities with regard to selection of window type, see figure 1, size and location. However, the knowledge of the performance of individual windows is rather limited and is based on theoretical assumptions on the main driving forces, effective areas and air flow within rooms. It is only possible in window design for natural ventilation to give rough estimates of the thermal comfort, the draught risks and the IAQ levels that can be expected. Some window types are regarded as better than others, but this is only based on qualitative measures and the differences and limitations in the application of individual window types cannot be quantified.

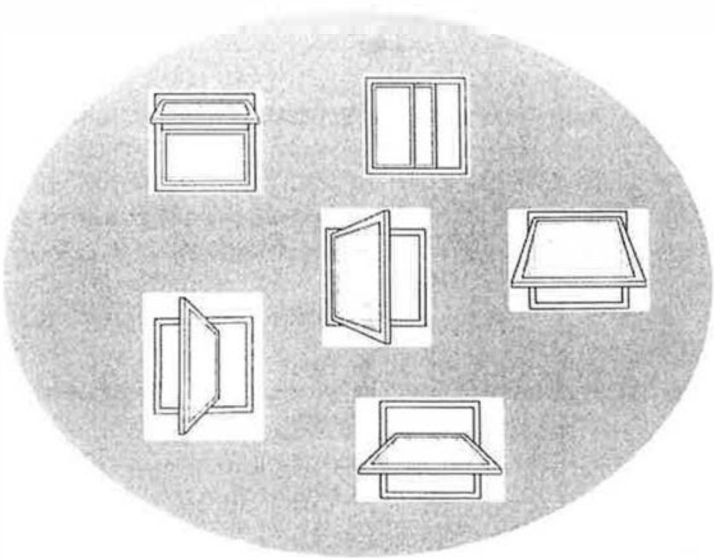


Figure 1. Examples of different window types.

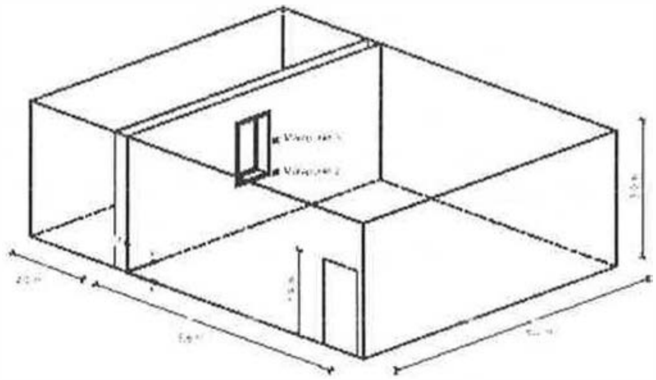
Therefore, there is certainly a need for quantitative information on window performance that can improve the window design methods to a level, where they can match the design methods of air inlets in mechanical ventilation.

This paper describes the first results of a series of laboratory investigations that is performed to characterise three different window types. The results show the air flow conditions for different ventilation strategies and temperature differences. For one of the windows values of the discharge coefficient are shown for both isothermal and non-isothermal flow conditions and the thermal comfort conditions are evaluated by measurements of velocity and temperature levels in the air flow in the occupied zone.

2 DESCRIPTION OF LABORATORY SET-UP

The investigations is performed in a laboratory test room with the size of Length×Width×Height = 8m × 6m × 3m, see figure 2a. The room is divided into two separate rooms by an insulated wall, see figure 2b. The small room can be cooled to a temperature of about 0°C while the large room can be kept at normal room temperature. Three different window types have been placed in the insulated wall, see figure 3a. Window type 1 is a combined side/bottom hung window that is placed close to the occupied zone. Window type 2 is a narrow window that is placed high in the room and has been used both as a top and bottom hung window. Window type 3 is a horizontal pivot window placed close to the occupied zone, see figure 3b-d.

A



B

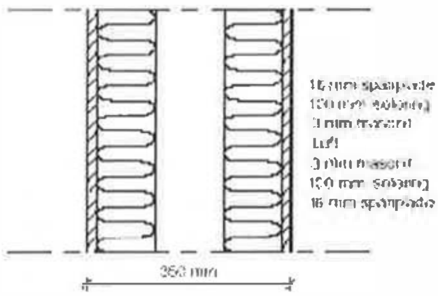
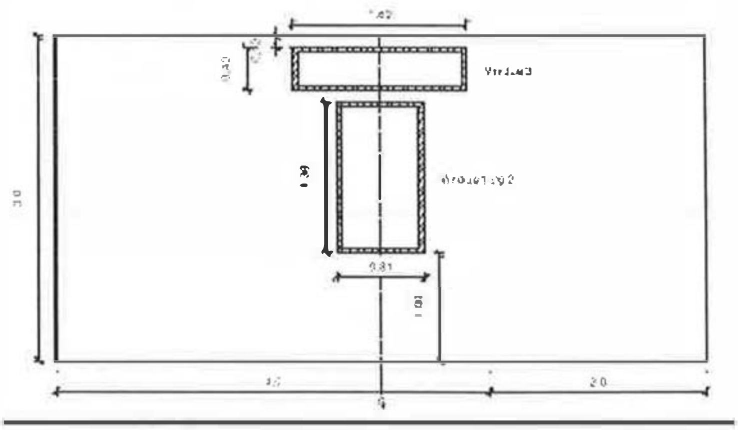
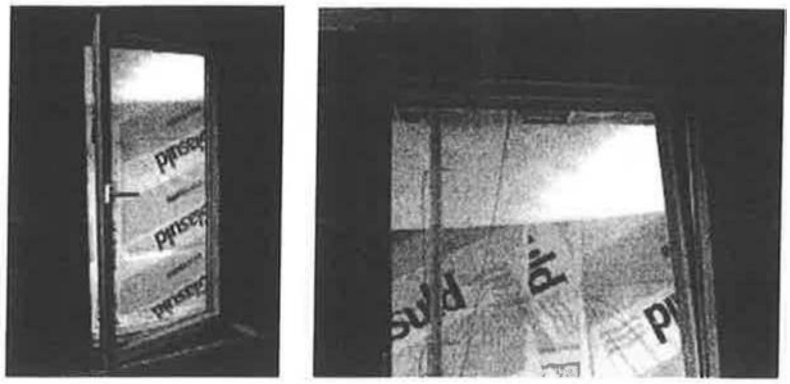


Figure 2. A) Sketch of laboratory test room. B) Sketch of insulated wall construction.

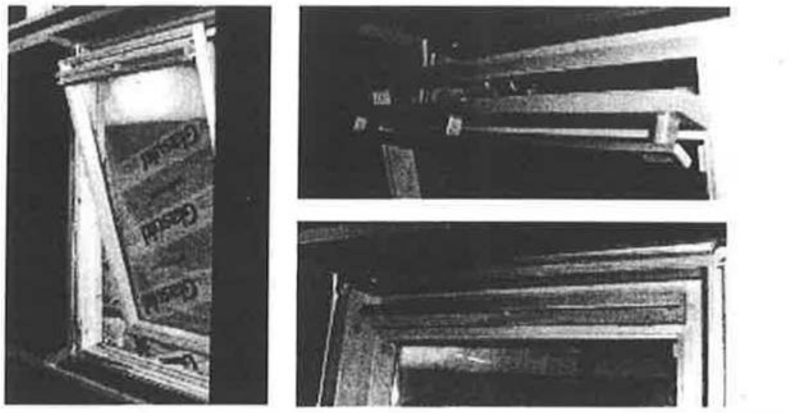
A



B



C



D

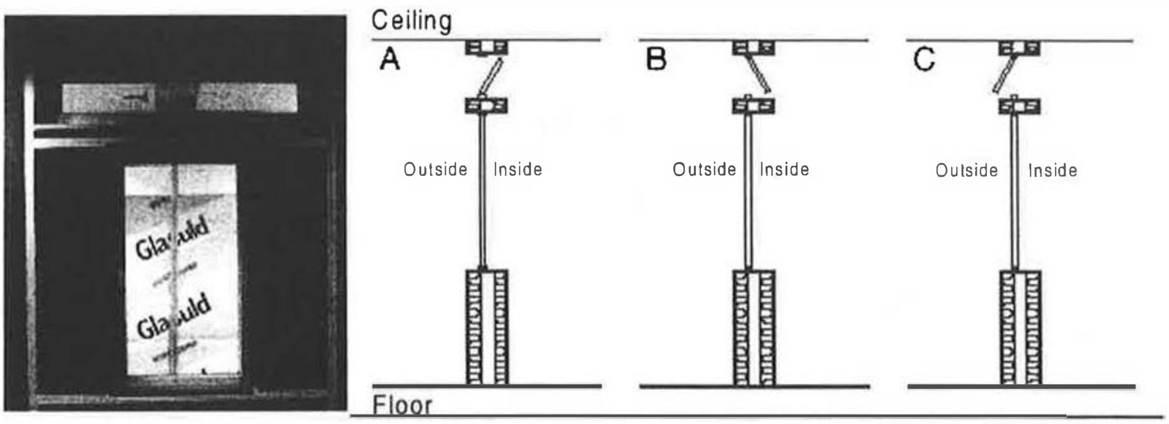


Figure 3. A) Sketch of window location in insulated wall. B) Photo of window type 1, C) Photo of window type 2 and D) Photo of window type 3 (top window) with indication of the three configurations used in the investigation.

3 AIR FLOW THROUGH WINDOWS

The air flow through a window depends on the chosen natural ventilation strategy, see figure 4. Single sided ventilation relies on openings being on only one side of the ventilated enclosure. A close approximation is a cellular building with opening windows on one side and closed internal doors on the other side. With a single opening in the room the main driving force for natural ventilation in winter is the thermal stack effect, where the air will flow into the room in the bottom half of the window and out of the room in top half of the window. The main driving force in summer will be the wind turbulence. Compared with other strategies, lower ventilation rates are generated. Stack induced flows increase with the vertical separation of the openings. Window type 2, with the main opening area divided between the top and the bottom of the window, is therefore more effective than types 1 and 3, where the main opening area is concentrated either in the top or the bottom of the window.

In cross- and stack-ventilation there are ventilation openings on both sides of a space. Air flow from one side of the building to the other and leaves through another window or door. Cross ventilation is usually wind driven while stack ventilation is thermal (and wind) driven. With such ventilation strategies there will only be an inflow of air through the window and the pressure difference will be much higher. The capacity of the opening will not depend on the distribution of the opening area, but only on the total area.

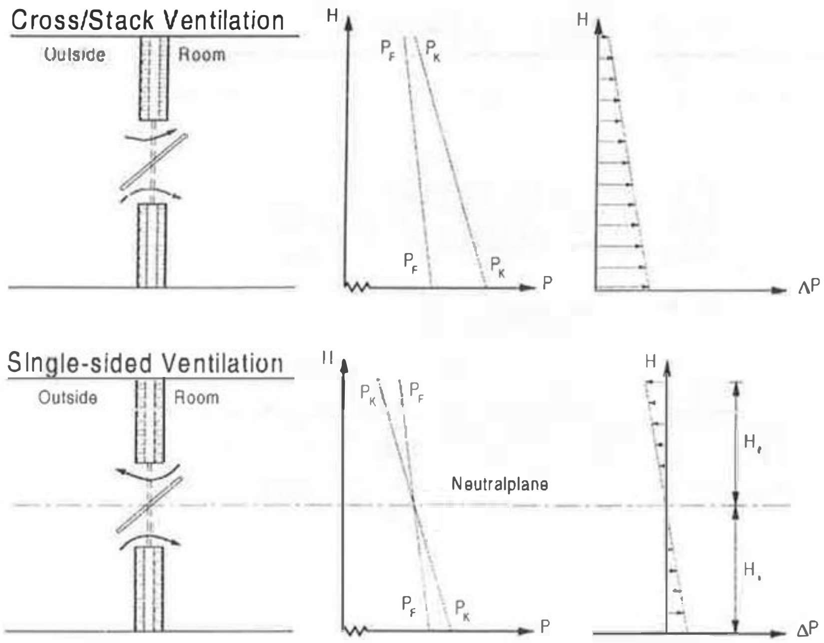


Figure 4. Air flow through a window with a single sided and a cross ventilation strategy, respectively

4 AIR FLOW INSIDE ROOM

The air flow in the room was investigated by smoke tests for both a single-sided and a cross/stack ventilation strategy for all three window types.

For a single sided ventilation strategy air flow through window type 1 and 2 was supplied directly to the occupied zone and dependent on temperature difference and window opening area the air reached the floor from 0,5 – 1.5 m from the window, see figure 5. The air flow along the floor could be characterised as stratified flow. Even very small opening angles resulted in large air flows and high velocity levels in the occupied zone.

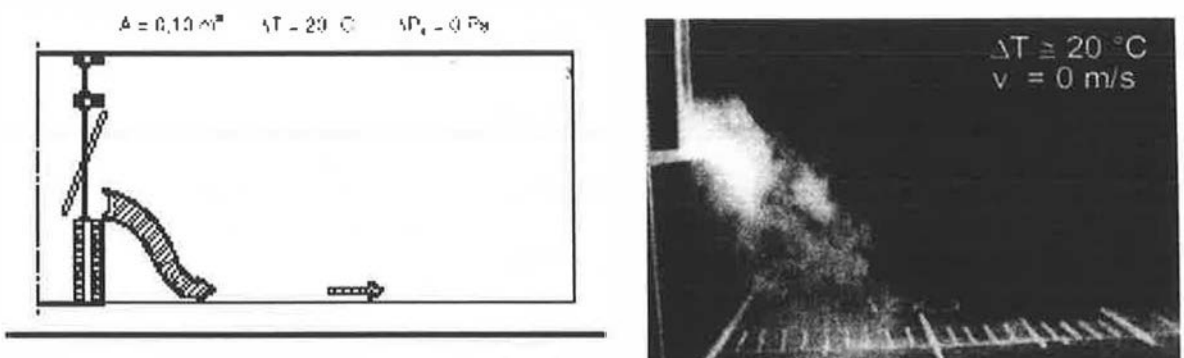


Figure 5. Air flow through window type 2 with single sided ventilation and a temperature difference of 20°C.

For a single sided ventilation strategy the air flow through window type 3 was almost identical for all three configurations on figure 3D. At small opening angles only a very small amount of air entered the room at low velocity. With increasing opening angles the air flow and velocity level increased. In all cases the air flow was downwards along the wall and at large opening angles the air reached the floor and turned into the occupied zone as stratified air flow along the floor, see figure 6.

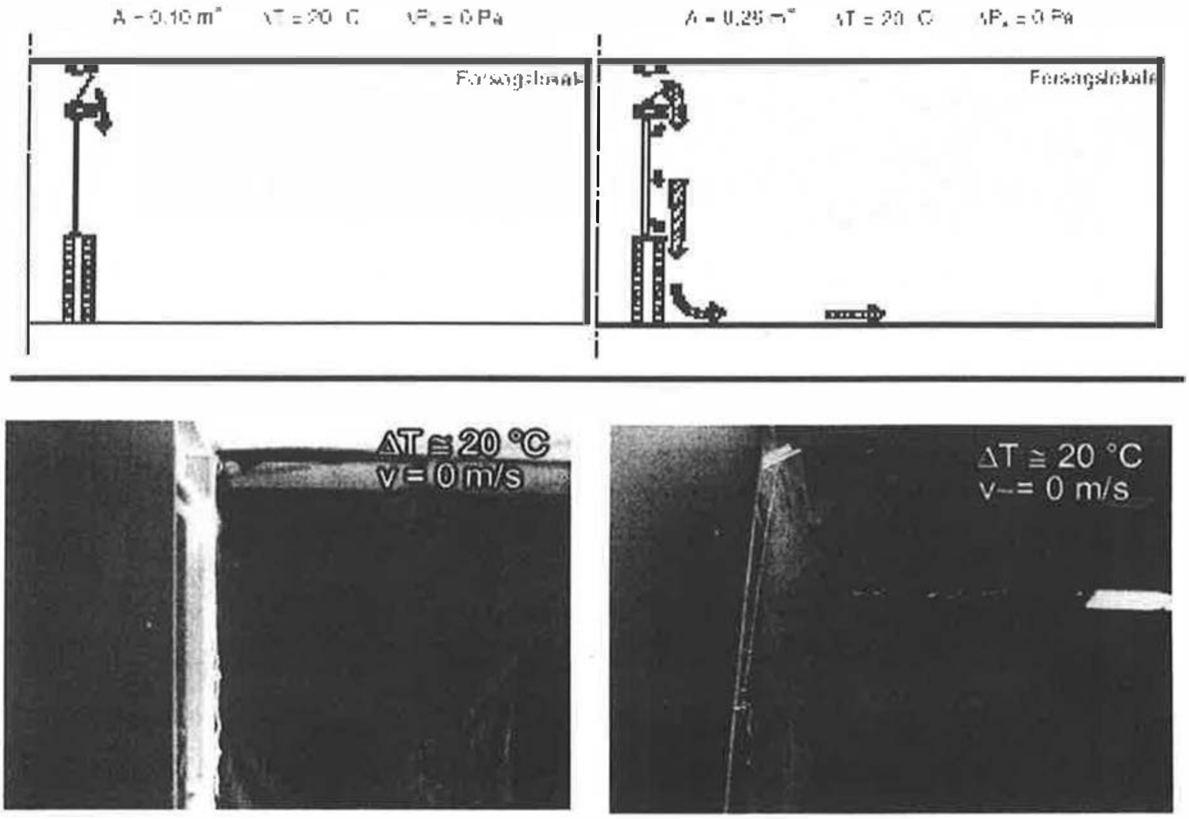


Figure 6. Air flow through window type 3 with single sided ventilation and a temperature difference of 20°C.

For a single sided ventilation strategy window type 3 is the best choice in winter because the air is supplied outside the occupied zone and can be controlled by changing the opening angle. Window type 1 and 2 is not a good choice as the air is supplied directly to the occupied zone and is difficult to control because the amount of air and the velocity levels increase very rapidly with increasing opening angles. In summer with small temperature differences window type 3 will not be able to supply enough air to the room, but will have to be combined with window type 1 or 2.

For a cross- or stack-ventilation strategy the available pressure difference across the openings is generally much higher. For window type 1 and 2 the air flow into the room acted as a thermal jet that reached the floor in a certain distance dependent on temperature difference, pressure difference and opening angle. The problems under winter conditions with high air velocities and with a proper control of the air flow increased. The air flow conditions for window type 3 showed large differences for the three configurations. Generally the air flow acted as a thermal jet. For both a bottom hung window opening in and a top hung window opening out the air flow acted as a thermal wall (ceiling) jet. However, the distance from the wall where the jet separated from the ceiling was larger for the bottom hung window, resulting in lower velocities in the occupied zone. For the top hung window opening in the air flow acted as a free thermal jet and reached quickly the occupied zone resulting in very high air velocities, see figure 7.

A B

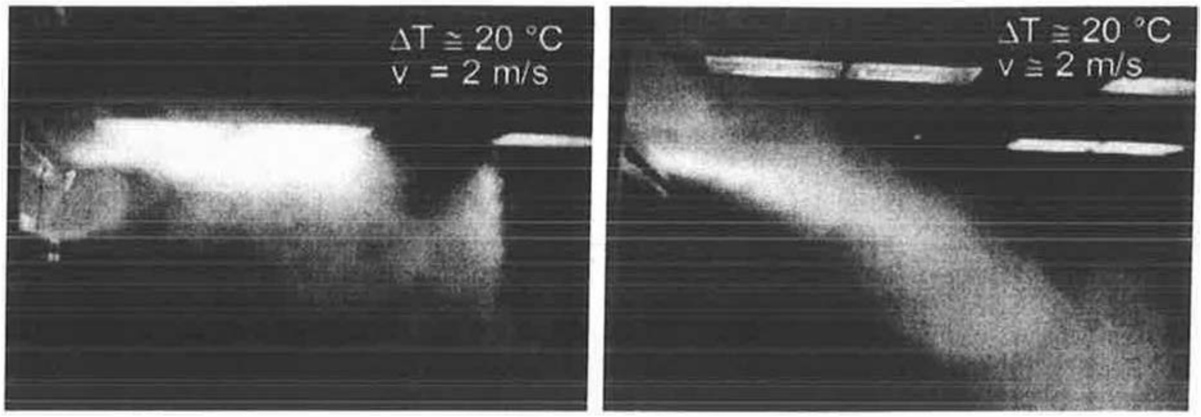


Figure 7. Air flow through window type 3 with cross- or stackventilation and a temperature difference of 20°C. A) Bottom hung, opening in. B) top hung, opening in.

For a cross- or stackventilation strategy window type 3 in a bottom hung configuration is the best choice in winter because the air travels the largest distance before it reaches the occupied zone and the velocity levels therefore will be the lowest. Window type 1 and 2 is not a good choice as the air is supplied directly to the occupied zone at very high velocities and is very difficult to control because the amount of air and the velocity levels increase very rapidly with increasing opening angles.

5 WINDOW AIR FLOW CAPACITY

The air flow through a window can be estimated by equation (1)

$$Q = C_d A \sqrt{\frac{2\Delta P}{\rho}} \tag{1}$$

where Q is volume flow rate (m^3/s)
C_d is discharge coefficient (-)
A is geometrical window opening area
Δp is pressure difference across the window (pa)
ρ is density of air (kg/m^3)

The discharge coefficient is a characteristic parameter for a specific window and takes both the contraction and the friction loss in the window opening into account. The size of the coefficient is only known for very simple opening types. For windows, which have a very complicated geometrical structure, the size of the coefficient is unknown and its dependence on parameters like for example opening area, velocity level (pressure difference) and temperature difference is not known either.

Preliminary measurements on window type 1 with a side hung opening shows some interesting characteristics of the discharge coefficient. The estimation of the geometrical opening area of the window is very difficult because of the complicated geometry and the uncertainty is especially high at small opening angles. The absolute value of the discharge coefficient is therefore uncertain especially at small opening angles and measured values above 1 must primarily be caused by incorrect estimation of the opening area.

Figure 8 shows the discharge coefficient as a function of the pressure difference across the opening for different opening areas, Figure 9 as a function of the opening area of the window and Figure 10 as a function of a reduced Archimedes number ($\Delta T/Q^2$).

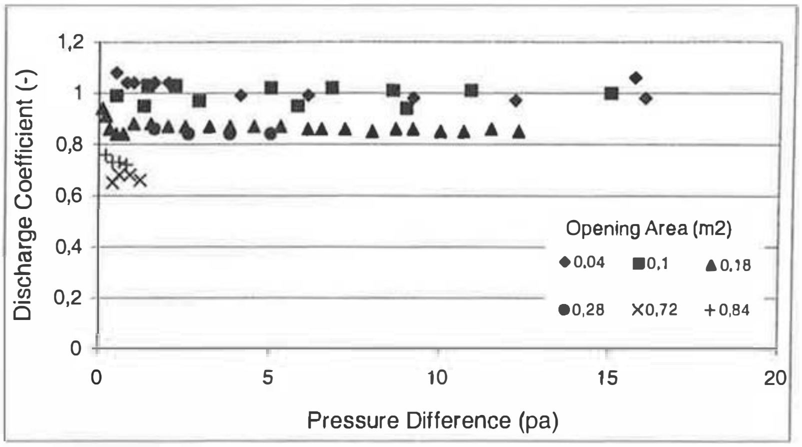


Figure 8. Discharge Coefficient, C_d, for window type 1 (side hung) as a function of pressure and for different opening areas.

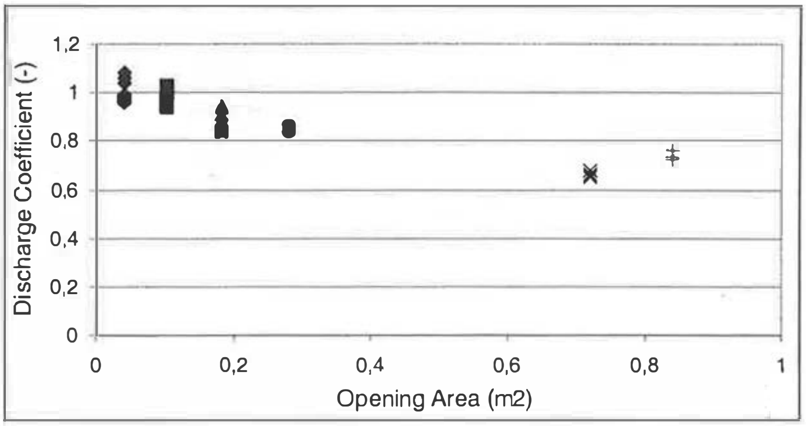


Figure 9. Discharge Coefficient, C_d, for window type 1 (side hung) as a function of the opening area.

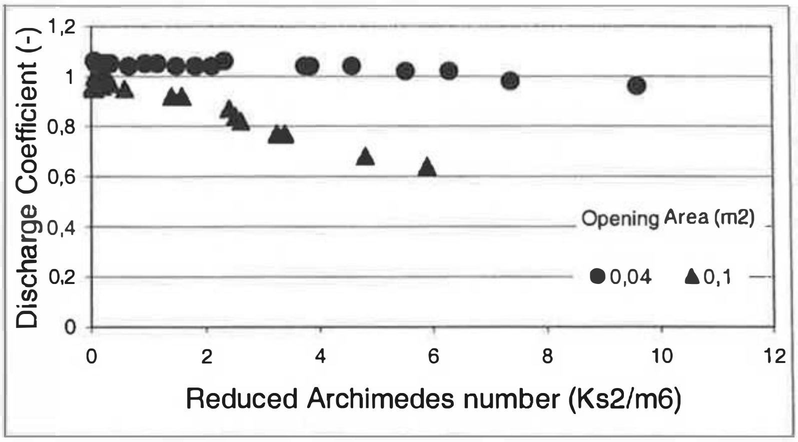


Figure 10. Discharge Coefficient, C_d, for window type 1 (side hung) as a function of a reduced Archimedes number.

The measurement results shows that the in an isothermal case the discharge coefficient is independent of the pressure difference across the window, but dependent on the opening area. In a situation with both a temperature and a pressure difference across the opening the discharge coefficient can be described as a function of the Archimedes number and the opening area. So, the use of a constant value for the discharge coefficient independent of opening area, temperature- and pressure difference can lead to serious errors in the prediction of air flow capacity of window openings.

6 AIR VELOCITIES IN THE OCCUPIED ZONE

The air flow from window type 1 with side hung opening will act as a thermal jet. The distance from the wall, where the air jet will reach the floor will be dependent on the pressure difference (air flow rate) and the temperature difference. Figure 11 shows that the maximum velocity in the air flow along the floor also will be dependent on the air flow rate and temperature difference. The velocity level increases with increasing air flow rate and increasing temperature difference, but decreases with increasing distance to the wall. This is a very typical result for stratified flow conditions.

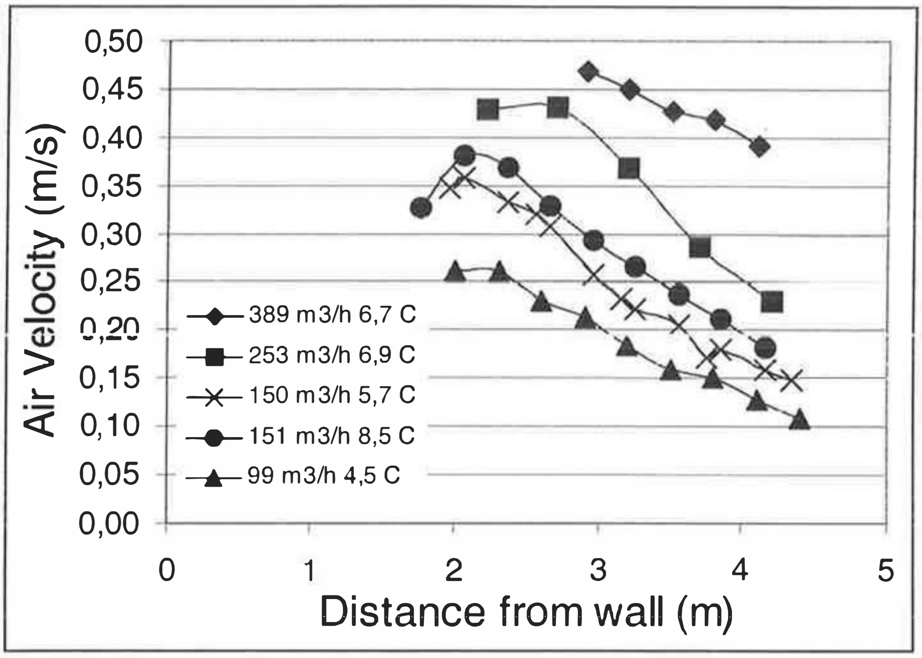


Figure 11. Velocity level in air flow along the floor from window type 1 (side hung) as a function of distance to wall, air flow rate and temperature difference.

Preliminary analysis have showed that it is possible to develop an equation system that can be used to predict the velocity level in the occupied zone as a function of opening area, pressure difference and temperature difference. This can be used to predict the comfort performance of window openings and estimate the limitations of a specific window type. In this way the design of window openings for natural ventilation becomes not only a question of providing the necessary opening area to ensure satisfactory capacity but also a question of selecting the optimum window type for thermal comfort.

7 CONCLUSIONS

The results have been promising and work will continue by investigation of the performance of other window types, especially those located at high levels in the room.

The results showed that the discharge coefficient for a window opening varies considerably with opening area and temperature difference and that the use of a constant value can lead to serious errors in the prediction of air flow capacity.

It should be possible to develop equations systems to predict thermal comfort in the occupied zone because of air flow from window openings, which is very important for the selection of optimum window types.

A further next step could be to investigate the dynamics of window opening and the performance of different control systems in controlling the air flow through the windows. But as long as we do not know the characteristics of the window, it is very difficult to define the requirements to the control system.

GUIDELINES ON COST EFFECTIVE NATURAL AND LOW ENERGY VENTILATION STRATEGIES FOR RETROFITTING TO UK OFFICES

Paul Robinson, John Palmer and Earle Perera
Building Performance Assessment Centre, Building Research Establishment
Garston, Watford, WD2 7JR, UK
e-mail: palmerj@bre.co.uk

ABSTRACT

Work carried out in the UK, on behalf of, and fully funded by the Department Transport and Regions, has produced guidelines for the cost effective refurbishment of offices for natural and low energy ventilation. The aim of the work is to counteract the current trend towards the installation of air conditioning when other low energy strategies will provide suitable comfort conditions. This paper shows ways in which comfort conditions can be achieved with zero or minimal energy use in a range of office types, using natural ventilation or simple mechanical low energy systems.

1 INTRODUCTION

Many existing offices either overheat in the summer or use excessive amounts of energy to maintain acceptable temperatures. The reasons are increasing internal heat gains from office IT equipment, poor efficiency lighting systems, density of staff and original poor building design causing excessive solar gains. The trend is for problem offices to have air conditioning systems replaced - at the end of the life of the existing services - or installed in previously naturally ventilated offices when refurbishment occurs. Introducing or replacing air conditioning is an energy intensive solution and is unnecessary for most office buildings in the UK climate. The trend needs reversing; natural and low energy ventilation strategies work and can be incorporated during refurbishment.

There is much recent advice on using natural and low energy ventilation in *new* non-domestic buildings [1,2]. Existing buildings being refurbished can also benefit from many of the strategies to displace, avoid or reduce the use of air conditioning (AC) and the associated energy consumption. This UK Government funded study concentrates on the refurbishment case.

The retrofitting of natural and low energy ventilation systems in existing UK office buildings during refurbishment can maintain comfortable indoor conditions and result in significant energy and CO_2 savings compared to the common solution of just installing AC.

Where natural and low energy ventilation strategies do not eliminate the need for air conditioning altogether, they can improve the situation by one or more of the effects below:

(i) reducing the AC load (by reducing internal gains and using the thermal mass of the fabric with night cooling)
(ii) reducing the area of the office where AC is needed (spatial mixed mode)
(iii) reducing the proportion of the year when AC is needed (temporal mixed mode).

2 NATURAL AND LOW ENERGY VENTILATION SYSTEMS

A retrofit natural or low energy ventilation strategy comprises measures which act to reduce the internal gains as far as possible, absorb heat in the fabric of the structure and promote the flow of cooler outside air through the building. The measures are listed in Table 1.

Removal of suspended ceilings acts to expose the thermal mass of the floor slab and increase floor to ceiling heights. Increased height of the office space increases the efficiency of natural ventilation at the occupied level by providing a larger air flow path across the office space and improving thermal comfort by allowing air to stratify. Night cooling by ventilating with cool night air and storing the “coolth” in the exposed heavy floor slab reduces peak temperatures the following day.

The introduction of an atrium in a deep plan building to reduce the depth of the floor plan improves daylight penetration to the centre of building and promotes stack ventilation to draw air through the adjacent office space from windows or vents on the façade.

3 REFURBISHMENT LEVELS

It can be seen from the list of measures in Table 1 which might make up a natural ventilation or low energy ventilation strategy, that they involve varying levels of disruption to the building. BSRIA have put forward four likely levels of office refurbishment, together with the opportunities for natural and low energy ventilation measures to be introduced for each level of refurbishment [3]. This information is incorporated into Table 1.

Level 1 is a *minor* refurbishment and involves the introduction of opening windows, reduced window area, modern internal blinds, low energy IT option on replacement, re-painted interior and re-designed office layout to maximise access to available daylight.

Level 2 is an *intermediate* refurbishment, as level 1 but with mid-pane blinds for solar control, a new, more energy efficient lighting and control system, removal of false ceiling to expose thermal mass and raise ceiling height, providing the possibility of occupant controlled night cooling.

Level 3 is a *major* refurbishment, as level 2 but with external solar control, possible use of stair cores as ventilation stacks, BMS controlled night cooling with motorised window/vent opening.

Level 4 is a *complete* refurbishment, options as level 3 and with radical changes to air flow paths, for example by addition of a central atrium, or use of a double facade to drive stack ventilation.

Table 1 Natural and low energy ventilation measures to be incorporated at the of four levels of refurbishment

Refurbishment measures: options at each level	Refurbishment level 1. Minor	Refurbishment level 2. Intermediate	Refurbishment level 3. Major	Refurbishment level 4. Complete
• Repaint interior with cool, light colours	x	x	x	x
• Layout of work stations and office equipment near extract	x	x	x	x
• Choice of low energy office equipment on replacement	x	x	x	x
• Replacement opening windows with multiple openings	x	x	x	x
• Reduced window area	x	x	x	x
• Good daylighting from positioning of windows	x	x	x	x
• Some solar control by glazing choice and internal blinds	x	x	x	x
• Reduction of unwanted infiltration	x	x	x	x
• Efficient electric lighting systems and controls		x	x	x
• Removal of suspended ceiling		x	x	x
• Night cooling by leaving windows open - manual control		x	x	x
• Added solar control by use of mid-pane blinds		x	x	x
• Controllable windows or vents, perhaps by the BMS			x	x
• Use of stair wells or service shafts for stack ventilation			x	x
• Added solar control by use of external blinds			x	x
• Use of a double facade or solar chimney to act as a ventilation stack				x
• Introduction of an atrium in a deep plan building				x

We have added a level of refurbishment, *AC Upgrade* which involves just the replacement or introduction of a typical fan coil AC system and allows comparison with the low energy ventilation approach. The new or replacement AC system is assumed to be a fan coil AC system, with a COP of 3, set point of 22°C, typical low velocity fans and 100% air re-circulation, giving 5 air changes per hour. It is assumed to operate when the building is occupied.

Simple low energy mechanical ventilation (Hybvent) can be used in conjunction with the measures and strategies described above for each level of refurbishment where the result of their being retrofitted does not quite eliminate the need for AC. A simple low energy mechanical ventilation solution should use high efficiency fans with a minimum of ducting and only be used when and where the natural ventilation driving forces need to be supplemented. The system investigated here, in conjunction with cases where the natural ventilation refurbishment *just* fails to meet the summer comfort criteria, is a simple mechanical extract using best practice low energy fans, operating only in the peak summer months, in the areas of the office building where it is required.

4 ASSESSMENT METHOD

The benefits of retrofitting natural ventilation and low energy ventilation solutions to existing office buildings during refurbishment have been assessed using the BRE design tool NITECOOL. NITECOOL predicts hourly temperatures, number of hours over a certain temperature and cooling season energy use (if mechanical ventilation or AC is specified) for a variety of design options such as glazing area and type, orientation, solar control, air change rate, window or vent opening area, building thermal mass, night ventilation, and internal heat gains.

The program predicts dry resultant temperatures, (DRT - based on a weighting of 50% air temperature and 50% radiant temperature) and so are the most closely linked to a human thermal comfort (no allowance is made for air movement). Therefore, the maximum internal temperature during the occupied hours of the hot period is used to assess the overheating risk and success or failure of the natural ventilation strategies incorporated at each level of refurbishment.

The acceptability criteria used to assess the success of the low energy ventilation refurbishment strategies in this study are:

- an internal maximum dry resultant temperature for the July hot week of 28°C (this is 1°C above the maximum outside air temperature used in the simulation)
- a maximum of 15% of occupied hours over 27°C in the July hot design week (with an overall yearly maximum hours over 27°C of 2% of occupied hours).

4.1 Weather data for modelling

NITECOOL works with a week of CIBSE Kew weather data for each of the cooling season months, May-September. Maximum and minimum air temperature data (correlated with solar radiation and wind speed) is contained in the NITECOOL programme. The programme assumes a sinusoidal profile between daily maxima and minima. The work described here uses two different temperature profiles as used for predicting either comfort or energy use:

(i) **peak temperature assessment**; hot design period data (a warm five days with maximum temperature of 23.3°C and minimum of 15°C, followed by two very hot days with max 27°C and minimum of 16°C), and
(ii) **summertime cooling energy use**; average weather data for May-September.

4.2 Office types

In order to make realistic comparisons in typical office buildings, four common office types [4] have been used in this analysis to assess the benefits of natural ventilation and low energy ventilation for refurbished offices. Each office type has been simulated with NITECOOL for each level of refurbishment. Results from a South facing office (high solar gain), and results averaged for North, East, South and West orientations are presented.

5 SIMULATION RESULTS FOR DIFFERENT OFFICE TYPES

5.1 Type 1 Naturally ventilated, cellular

It is estimated that 68% of England and Wales office stock (71.7km^2 gross floor area) is Type 1.

Characteristics:

- Simple shallow plan buildings
- often small and in converted residential accommodation
- naturally ventilated
- predominantly cellular
- typical building depth 10-20m, typical office

- depth, 4-10m
- individual windows
- lower illuminance levels than other types
- local light switches and heating controls
- occupied 2500h/y (equivalent to 10h/d, 5d/wk, assuming that the office is open for work 50 weeks of the year).

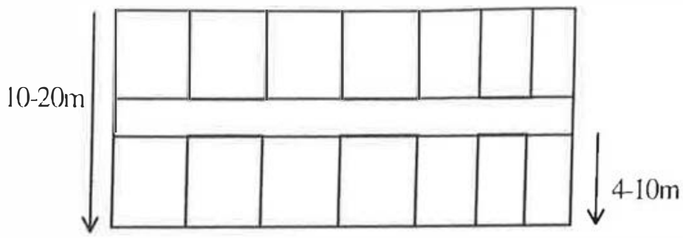


Figure 1 Type 1 typical office plan

Internal gains:
- typical- 30W/m^2
- good practice - 20W/m^2

Table 2. Type 1 office: base case and refurbishment levels - performance of South facing office

Case	Features	Max temp °C	% hours over 27°C in July hot week	Cooling energy/y (average temps) kWh/m^2/y
base case	• internal gains - 30W/m2 • 7 ach single sided ventilation (daytime) • occupation -10h/d (08-18:00) • 60% glazing ratio • single glazed • internal blinds with solar trans=70% • suspended ceiling (indicating lightweight construction with respect to accessible thermal mass).	38.1°C	85%	0
ac upgrade	• base case + fan coil AC system	22.0°C	0%	47.0
Refurb level 1	• base case + 25W/m^2 internal gains, reduced glazing ratio (40%), double glazing, 10ach daytime, new office layout and repainting with light colours	32.6°C	63%	0
Refurb level 2	• level 1 + internal gains 20W/m^2, mid-pane blinds (solar transmission of 40%), removal of suspended ceiling (increased thermal mass), 10ach day and for night cooling	27.0°C	0%	0
Refurb level 3	• as level 2 + external louvres (solar transmission of 20%), use of stair cores as stacks , BMS controlled motorised vents/windows, day time 10ach, night ventilation 13 ach	26.3°C	0%	0
Refurb level 4	• as level 3 + option of using double façade to get around site problems of noisy/polluted façade (same ach)	26.3°C	0%	0

Key: the shaded block indicates where natural or low energy ventilation strategies provide acceptable temperatures

5.2 Type 2: naturally ventilated, open plan

It is estimated that 9% of E&W office stock (9.3 km^2 gfa) is of this type.

Characteristics:

- purpose built, sometimes converted industrial space
- typical size 500m^2 - 4000m^2,
- largely open plan with some cellular offices and special areas
- typical depth 10-15m
- light levels, internal gains and hours of use are usually higher than type 1 cellular,
- occupied 3000h/y (=12h/d, 5d/wk, 50wk/yr)
- Internal gains: typical - 35W/m^2, good practice - 25W/m^2

Figure 2 Type 2 office typical plan

Table 3. Type 2 office: base case and refurbished levels - performance of South facing office

Case	Features	Max temp °C	% hours over 27°C in July hot design week	Cooling energy/y (average temps) kWh/m²/y
Base case	• internal gains - 35W/m2 • 10ach cross ventilation (daytime) • occupation -12h/d (07-19:00) • 60% glazing ratio • single glazed • internal blinds with solar trans=70% • suspended ceiling (indicating lightweight construction with respect to accessible thermal mass).	36.9°C	81%	0
AC Upgrade	• base case + fan coil AC system	22.0°C	0%	54.9
Refurb level 1	• base case + 30W/m² internal gains, reduced glazing ratio (40%), double glazing, 10ach daytime, new office layout and repainting with light colours	33.1°C	61%	0
Refurb level 2	• level 1 + internal gains 25W/m², mid-pane blinds (solar transmission of 40%), removal of suspended ceiling (increased thermal mass), 10ach day and night (night cooling)	27.5°C	8%	0
Refurb level 3	• as level 2 + external louvres (solar transmission of 20%), use of stair cores as stacks , BMS controlled motorised vents/windows, day time 10ach, night ventilation 13 ach	26.7°C	0%	0
Refurbishment level 4	• as level 3 + option of using double façade to get around site problems of noisy/polluted façade (same ach assumed)	26.7°C	0%	0

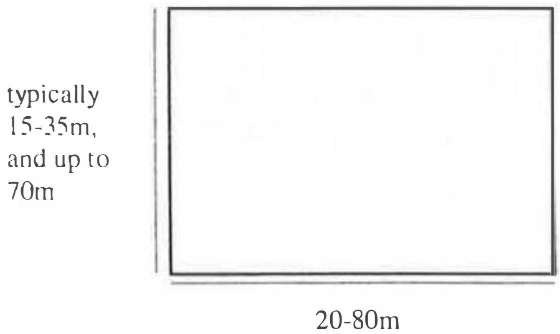
Key: the shaded block indicates where natural ventilation strategies provide acceptable temperatures

5.3 Type 3: air conditioned, standard

It is estimated that 14% of E&W office stock (15.7km² gfa) is of this type.

Characteristics

- Similar in occupancy & planning to Type 2
- usually deeper floor plan (up to 70m, typically 35m),
- tinted or shaded windows,
- occupied 3200h/y (=13h/d, 5d/wk, 50wk/y)



Internal gains:

- typical - 40W/m²
- good practice -25W/m² perimeter zone

Modelled example:

- 35mx35m=1225m²,
- passive zone (6m perimeter band)=696m²
- service zone (central) - 10% of gfa (from Econ 19) = 11.1mx11.1m=123m²
- core zone is 6m deep band within the perimeter band = 406m².

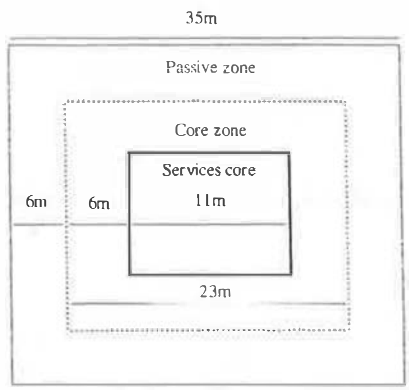


Figure 3 Type 3 office floor plan

Table 4. Type 3 office: base case and refurbishment levels - performance of South facing office

Case	Features	Max temp. °C	% hrs over 27°C in July hot week	Cooling energy/y (average temps) $kWh/m^2/y$
Base case	• internal gains - 40W/m2 • 5ach (daytime) • occupation -13h/d (07-20:00) • 80% glazing ratio • single glazed • internal blinds with solar trans=70% • suspended ceiling (indicating lightweight construction with respect to accessible thermal mass).	45.0°C	96%	91 (type 3 typical use)
AC Upgrade	• base case + fan coil AC system	22.0°C	0%	66.9
Refurb level 1	• base case + $35W/m^2$ internal gains, reduced glazing ratio (40%), double glazing, 10ach daytime, new office layout and repainting with light colours	36.6°C	82%	0
Refurb level 2	(a) natural ventilation: level 1 + internal gains $25W/m^2$, mid-pane blinds (solar transmission of 40%), removal of suspended ceiling (increased thermal mass), 10ach day and night (night cooling)	28.4°C	21%	0
HYBVENT	(b) low energy mechanical ventilation: level 2(a) + simple extract fans to assist natural driving forces, to give 10ach day and night (and 15ach for hottest July periods)	27.9°C	12%	3.1
Refurb level 3	• as level 2 + external louvres (solar transmission of 20%), use of stair cores as stacks, BMS controlled motorised vents/windows giving 7ach day and night ventilation	26.9°C	0%	0
Refurb level 4	• as level 3 + central atrium replacing service core giving 10ach day and night ventilation	26.7°C	0%	0

Key: the shaded block indicates where natural or low energy ventilation strategies provide acceptable temperatures

5.4 Type 4 air conditioned prestige

An estimated 9% of England and Wales office stock ($9.9km^2$ gfa) is of this type.

Characteristics

- usually deep plan (up to 70m, typically 50m)
- purpose built or refurbished to high standard
- occupied 3500h/y (=14h/d).

Internal gains:

- typical - $45W/m^2$,
- good practice - $30W/m^2$ (perimeter zone with good daylighting + controls, $35W/m^2$ core zone)

Modelled example:

- $50m \times 50m = 2500m^2$
- passive zone (6m perimeter band)=$1056m^2$
- service zone (central) - 15% of gfa (from Econ 19) = $19.4m \times 19.4m = 376m^2$
- core zone = 9.3m deep band within the perimeter band = $1063m^2$.

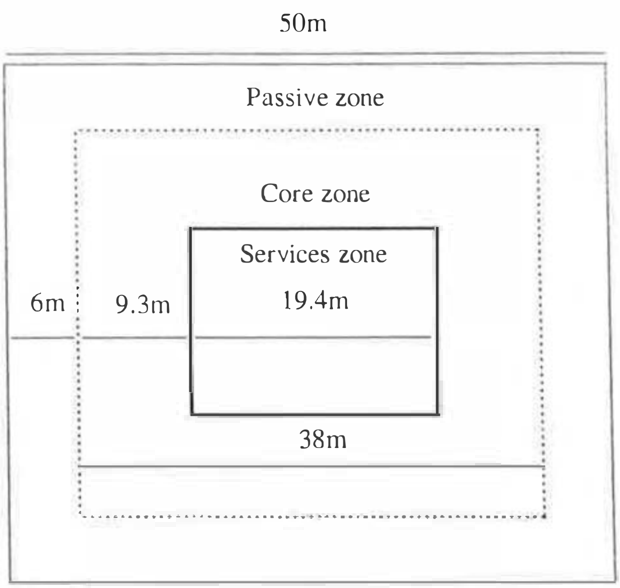


Figure 4 Type 4 office floor plan

Table 5. Type 4 office: base case and refurbishment levels - performance of South facing office

Case	Features	Max temp. °C	% hrs over 27°C in July hot week	Cooling energy/y (average temps) $kWh/m^2/y$
Base case	• internal gains - 45W/m2 • 5ach (daytime) • occupation -14h/d • 80% glazing ratio • single glazed • internal blinds with solar trans=70% • suspended ceiling (indicating lightweight construction with respect to accessible thermal mass).	45.8°C	97%	108 (type 4 typical use)
AC Upgrade	• base case + fan coil AC system	22°C	0%	73.4
Refurb level 1	• base case + $40W/m^2$ internal gains, reduced glazing ratio (40%), double glazing, 10ach daytime, new office layout and repainting with light colours	37.5°C	85%	0
Refurb level 2	(a) natural ventilation: level 1 + internal gains $30W/m^2$, mid-pane blinds (solar transmission of 40%), removal of suspended ceiling (increased thermal mass), 10ach day and night (night cooling)	29.2°C	26%	0
HYBVENT	(b) low energy mechanical ventilation: level 2(a) + simple extract fans to assist natural driving forces, to give 10ach day and night	28.0°C	14%	6.6
Refurb level 3	• as level 2 + external louvres (solar transmission of 20%), use of stair cores as stacks, BMS controlled motorised vents/windows giving 7ach day and night ventilation	27.5°C	7%	0
Refurb level 4	• as level 3 + central atrium replacing service core giving 10ach day and night ventilation	27.2°C	5%	0

Key: the shaded block indicates where natural or low energy ventilation strategies provide acceptable temperatures

6 RESULTS

The shaded areas of Table 6 highlight where a low energy ventilation refurbishment level has succeeded in eliminating the need for AC. All refurbishment levels from 2 upwards, where thermal mass of the floor slab is exposed and night cooling made possible, are successful with all four of the typical offices modelled in this study. For office types 3 and 4, level 2 refurbishment has to be supplemented by a simple low energy mechanical system (Hybvent) to provide acceptable summer temperatures without AC.

Table 6 shows a summary of the modelling results. The mode of servicing required to provide acceptable summer temperatures is recorded for each refurbishment level. The table also shows the summertime energy use (kWh/m^2y) for providing acceptable temperatures in the example office buildings, averaged over East, West, South and North facing offices (core zone not included), for the different levels of refurbishment of the four offices types.

Table 6 Natural and low energy ventilation to provide acceptable internal summer temperatures: mode of servicing and energy use (kWh/m^2y)

Refurbishment level	Type 1	Type 2	Type 3	Type 4
AC Upgrade	**AC** - 43.4	**AC** - 51.7	**AC** - 61.2	**AC** - 69.3
level 1	**AC** - 33.9	**AC** - 41.6	**AC** - 51.0	**AC** - 54.3
level 2 (with night cooling)	**NV** - 0	**NV** - 0	**Hybvent** - 3.4	**Hybvent** - 5.4
level 3 (with night cooling)	**NV** - 0	**NV** - 0	**NV** - 0	**NV** - 0
level 4 (with night cooling)	**NV** - 0	**NV** - 0	**NV** - 0	**NV** - 0

Key: AC= air conditioning required for acceptable temperatures Hybvent = simple low energy mechanical ventilation required for acceptable temperatures **NV** = natural ventilation provides acceptable temperatures

As a check on the methodology used, the NITECOOL predicted energy use for the AC Upgrade of office types 3 and 4 (i.e. installation of new, efficient AC system into an unmodified and poorly specified offices) has been compared with the ECON 19 *typical* practice data (without humidification). The predictions consistent with ECON 19 figures shown in tables 4, 5.

Level 1 refurbishment It can be seen from Table 6 that for all office types, the measures introduced at the level 1 refurbishment - though not enough to allow natural and low energy ventilation to provide summer comfort - result in a reduction in AC energy consumption of about 20% from the *AC Upgrade* option.

Level 2 refurbishment Two approaches to low energy ventilation provision are proposed for this level of refurbishment: natural and low energy mechanical (Hybvent).

Natural ventilation For shallow plan office types 1 and 2, the level 2 refurbishment allows natural ventilation to provide acceptable summer temperatures, with no electrical energy consumption. Level 2 refurbishment measures are as for level 1 but with mid-pane blinds (about 40% solar transmission), new efficient lighting system and controls (reduces heat gains from electric lights) and removal of suspended ceiling. Removing the suspended ceiling has several beneficial effects:

- it increases head height which allows more effective cross ventilation and gives a larger reservoir for warm stale air to accumulate without affecting occupants
- it allows access to thermal mass in the floor slab and, with manually operated night cooling (cool night air coming through open windows and vents) can be used to build up a store of "coolth" to reduce the peak temperatures of the next day.

Low energy mechanical ventilation For the deeper plan office types 3 and 4, the level 2 refurbishment does not satisfy the summer comfort criteria. However, it is only just outside the band of acceptability for a couple of months in the middle of the summer and the effect of introducing simple low energy mechanical ventilation was investigated. A simple extract system with low energy fans was modelled to provide a higher air change rate (10ach) and this was found to bring acceptable summer temperatures to both office types with minimal energy consumption (less than 8% of the AC Upgrade energy consumption).

Level 3 refurbishment provides acceptable summer temperatures for all office types using natural ventilation. For office types 1 and 2 which were bought into the comfort zone by refurbishment level 2, then the extra measures which can be incorporated in level 3 provide more flexibility and ventilation effectiveness where, due to site factors (for example polluted, noisy environment to one facade of a building), the strategies possible at level 2 are not able to be fully implemented.

Level 4 refurbishment is mainly applicable to deeper plan type 3 and 4 offices. As level 3 refurbishments of all office types provide acceptable summer conditions in the July hot design week, the extra measures which can be incorporated in level 4 provide more design options for effective ventilation where site or building restrictions prohibit the use of the strategies possible at level 3.

7 COST ANALYSIS

Plant capital and maintenance costs A typical air conditioning system has been estimated to cost £200/m^2 with maintenance costs of £15/m^2.

A simple low energy mechanical ventilation solution would be an extract system using fans with best practice performance (0.75W/l/s). The cost of installation and maintenance of such a system is estimated to be: £20/m^2 for installation, £0.90/m^2/y for maintenance.

The installation and maintenance costs have been added on to level 2 refurbishment costs for office types 3 and 4.

Low energy ventilation on-cost bands when refurbishing an office anyway
Level 1 on-cost for low energy ventilation refurbishment (if it is assumed that redecoration and minimal repairs would happen at the same time anyway) of £100-£250/m^2.

Level 2 on-cost for low energy ventilation refurbishment (if it is assumed that modernization and improvement of space would happen at the same time anyway) of £50-£300/m^2.

Level 3 on-cost for low energy ventilation refurbishment (if it is assumed that modernization and improvement of space would happen at the same time anyway) of £300-£500/m^2.

For *level 4*, there are two possibilities, for estimating on-costs: relative to a standard refurbishment package of space modernization and improvement, or relative to a major structural alteration (involving changes to cladding, core, all finishes):

> Level 4 on-cost for low energy ventilation refurbishment (if it is assumed that modernization and improvement of space would happen at the same time anyway) of £500-£700/m^2.
>
> Level 4 on-cost for low energy ventilation refurbishment (if it is assumed that a major structural alteration - involving changes to cladding, core, all finishes - would happen at the same time anyway) of £0-£200/m^2.

Maintenance costs for natural ventilation are shown with those for air conditioning for comparison in Table 7.

Refurbishment level 3 and 4 maintenance costs, though still those associated with natural ventilation are much higher than those for levels 1and 2 as it is assumed that the higher levels will include BMS control of automated windows and vents.

Table 7 Maintenance costs, £/m² y

Office type	1	2	3	4
Refurbishment level 0, with fan coil AC	15	15	15	15
Refurbishment level 1	1.1	1.1	1.1	1.1
Refurbishment level 2	1.1	1.1	1.1	1.1
Refurbishment level 3	4	4	4	4
Refurbishment level 4	N/A	N/A	4	4

NB. Maintenance costs for refurbishment level 1-4 assume AC is avoided, else level 0 costs apply

The above estimated costs – together with other estimates of typical office refurbishment costs [3,4] - have been used to assess the simple payback figures and lifecycle costs. This information is shown in Tables 8 and 9.

Table 8 Simple Payback Periods

Simple payback, years	Type 1	Type 2	Type 3	Type 4
level 0 (full ac)				
level 1				
level 2 (with night cooling)	0-5.2y	0-5.1y	0-6.1y	0-6.4y
level 3 (with night cooling)	6.1-18.4	6.1-18.2y	5.9-17.8y	5.9-17.8y
level 4 (with night cooling) – relative to space modernization /improvement refurb (£300/m^2)	n/a	n/a	17.8-29.6y	17.8-29.7y
level 4 (with night cooling) – relative to structural upgrade refurb (£800/m^2)	n/a	n/a	0y	0y

The light shaded area indicates where the refurbishment level avoids the need for installation or replacement of AC.
The dark shaded area indicates where the refurbishment level *with simple fan assistance* (Hybvent) avoids the need for installation or replacement of AC.

Table 9 Lifecycle costs relative to air conditioning over 25 years

lifecycle costs relative to level 0: -ve is a saving, +ve is an extra cost (£/m^2 over 25y)	BRECSU type 1	BRECSU type 2	BRECSU type 3	BRECSU type 4
level 0 (full ac)	0	0	0	0
level 1				
level 2 (with night cooling)	-£381 to -£631	-£385 to -£635	-£308 to -£608	-£345 to -£595
level 3 (with night cooling)	-£109 to -£309	-£113 to -£313	-£123 to -£323	-£122 to -£322
level 4 (with night cooling) – relative to space modernization /improvement refurb (£300/m^2)	n/a	n/a	+£123 to -£77	+£122 to -£78
level 4 (with night cooling) – relative to structural upgrade refurb (£800/m^2)	n/a	n/a	- £27 to - £227	-£28 to -£228

The light shaded area indicates where the refurbishment level avoids the need for installation or replacement of AC.
The dark shaded area indicates where the refurbishment level *with simple fan assistance* (Hybvent) avoids the need for installation or replacement of AC.

These financial parameters are based on modelling estimates of savings in cooling, ventilation, lighting and heating energy resulting from the refurbishment measures which result in the avoidance of air conditioning.

8 CONCLUSIONS

All office types, on refurbishment involving retrofit of low energy ventilation strategies, show significant energy savings from the avoidance and reduction of air conditioning, whilst providing comfortable internal temperatures.

For deeper plan types 3 and 4, more minor levels of refurbishment with natural ventilation strategies were not sufficient to provide acceptable internal temperatures. However, when the ventilation was supplemented by low energy mechanical systems, they were found to provide the required comfort levels with minimal energy use.

ACKNOWLEDGEMENTS

The authors gratefully acknowledge the UK Government funding of this project through the DETR.

REFERENCES

1. CIBSE, "Natural ventilation in non-domestic buildings", Applications Manual AM10, CIBSE 1997, ISBN 0 900953 77 2.

2. Building Research Establishment, "Natural ventilation in non-domestic buildings", BRE Digest 399.

3. Kendrick C, Martin A, Booth W, "Refurbishment of air-conditioned buildings for natural ventilation", BSRIA Technical Note TN 8/98, August 1998, ISBN 0 86022 498 8.

4. Energy Consumption Guide 19; Energy Consumption of Office Buildings. BRECSU