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DISCUSSION PAPERS AND ADDITIONAL PRESENTATIONS

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SECTION 1: Questions and Answers from Papers during Discussion Sessions

SESSION 1: Ventilation and Indoor Air Quality

Title:	**IEA Annex 27: Comparison of Performances of Different Ventilation Systems in Similar Dwellings**
Author:	**W.de Gids, The Netherlands**

Question from: O Jensen
(i) Could you indicate what you mean with the levels of balanced systems, levels 1,2,3, etc?
(ii)How is it possible to have a window open for 2-3 hours when both parts are on work for approx.10 hours!

Answer:
(i) In the paper, and during the presentation, it is mentioned that switch position means a certain air flow rate.

1 means 16 dm^3/s
2 means 26 dm^3/s
3 means 33 dm^3/s

(ii) Although it looks strange, we have registered that over a period of more than 2 years in 80 apartment buildings (see results of "IEA Annex 8, Technical Note 23 of AIVC)

Question from: S Hassid
Are the quantities in Figs.4,5,6,7 & 8 compared for a specific moment or on a seasonal basis?

Answer:
The figures given in the paper concerns measurement of three periods of two weeks during the heating season.

Title:	**Energy Recovery Possibilities in Natural Ventilation of Office Buildings**
Author:	**J Brunsell, Norway**

Question from: A Darmawan
Is the natural ventilation strategy with outdoor air intake at roof and a chimney outlet using the stack effect also applicable for the tropical climate situation? (refer to Figure 7)

Answer:
It could be. However, the report in the paper is for the north European situation. I am not sure what the climate situation in your area is, so I am not sure whether the stack effect will be applicable there.

Title:	**Recommended Ventilation Strategies for New Energy-Efficient Production Homes**
Author:	**J Roberson, USA**

Question from: J Andersson
The figures of the buildings seemed to show flexible air ducts. Is that the case? The key-note speaker of the 16th AIVC Conference, the Energy Commissioner of California, described the US concern of leaking air ducts and the LBL project. Were the ducts tested for air tightness in your presented project?

Answer:
Duct leakage can be a significant issue and should be considered for both energy and ventilation reasons. Current US construction practices use round flexible insulated ducting but measurements show that field installations are generally quite leaky. The designs in the paper considered duct leakage, but not as a rating criterion. So far, no houses have been completed based on the report, so there are no specific measurements. Future work may evaluate these Energy Star Homes and we would hope to include duct leakage on all duct systems. Duct performance is a very active area of research at LBNL.

Question from: D Stevens
Since multipoint supply was the highest rated system in all four climates, would an inline injection fan forcing 50-100 cfm continuously into the return air plenum be a reasonable alternative to give distribution but at a lower cost?

Answer:
This system was not explicitly considered. In principle, having it should be rated as high provided the issue of even distribution in the large duct system was handled correctly. An additional issue that could affect cost is keeping the flow through the injection fan steady during HVAC operation.

Question from: J R Millet
When comparing the different systems, how do you take into account their ability to extract directly the pollutants from wet rooms (exhaust systems seem to be more efficient for this purpose than supply ones).

Answer:
While extraction was considered as one of the potential advantages, it was not explicitly a rating criterion. When rating of two systems are close factors such this one may convince the designer to go with the lower rated system. It should also be remembered that most new U.S. houses have mechanical exhaust systems in wet rooms. Whole house exhaust systems may improve the situation then only slightly or in those rooms not required to have mechanical extract systems.

SESSION 3: Simulation and Design Tools

Title:	**A Design Tool for Natural Ventilation**
Author:	**C Svensson, Sweden**

Question from: D Azzi
Are you thinking of automating the parameter search process for a given desired output using (e.g. a search algorithm)?

Answer:
No.

Question from: S Gage
Why does the system not deal with Spring and Autumn?

Answer:
Contained in annual audit, but otherwise too difficult to do.

Title:	**Heat-Pipe Heat Recovery for Passive Stack Ventilation**
Author:	**S Riffat, UK**

Question from: J. Andersson
All four arrangements of heat pipes show horizontal pipes. Would the effectiveness have been increased if a vertical arrangement had been used adding drive force to the refrigerant in the pipes?

Answer:
In an application like this the position of the heat pipe plays no role.

Question from: J Kronvall
I have some general concerns regarding heat recovery in natural stack ventilation concepts. Once you introduce heat recovery in the stacks, the driving forces will decrease. Have you investigated that influence?

Answer:
Yes, we have investigated this. The driving force due to buoyancy is reduced by heat recovery and this needs to be taken account of in the design. Alternatively one can reduce the risk of low flow rates by employing solar heating of the stack and/or wind assistance. Both of these options are discussed in the companion paper in Session 4 (Page 323 in volume 1 of the Proceedings).

Title:	**Predicting Envelope Air Leakage in Large Commercial Buildings Before Construction.**
Author:	**E Perera, UK**

Question from: J Andersson
With your tool you predict the tightness of the new buildings. Are you also able to predict the long-time effects of the building with increasing age when it starts to crack and when the sealants are drying? For how long a time would the predicted tightness be valid considering a lifetime of perhaps 50-80 years for the building?

Answer:
Future predictions can be made by using the tool to call up component data leakage values for which ageing effects are known. At the moment, the tool calls on average leakage value of components (whose summing up provides the overall leakage) embedded with the data-engine of the design tool. These values can be changed (with caution) to provide an indicator of future performance.

Question from: S Hassid
(a) Is cost effectiveness based on heating or heating and airing calculations?
(b) Are such behavioural factors as opening windows and doors taken into account?

Answer:
(a) Cost effectiveness is based on neither. Cost effectiveness here is the actual implementing monetary cost required to minimise the leakage by a certain amount of percentage points.
(b) User occupant behaviours is not taken into account since the envelope leakage is only a measure of the construction tightness of the envelope.

Question from: W de Gids
Can you tell us more about the accuracy?

Answer:
Accuracy at the moment is about ±20%

Question from: D Etheridge
In my experience, quality of construction is a very important factor in determining air leakage. Does your method take account of this?

Answer:
Agreed. The design tool addresses this implicitly by bringing the designers mind to bear on the effective sealing of the joints. We also recommend that the construction team is briefed prior to construction to ensure that the airtightness measures are implemented as required.

Comment from: P. Op'tVeld
We have developed a more or less similar tool for predicting airtightness in dwellings. In relation to practice we found an accuracy of +20%. It is, however, absolutely necessary if you use data of e-values (AIVC database or others) to provide guidelines/instructions for the workfloor/building site. The execution of airtightness details are only determining the final airtightness.

Title:	**Office Night Ventilation Pre-Design Tool**
Author:	**M Kolokotroni, UK**

Comment from: F Steimle

- If you compare the night ventilation with A/c system, you must mention that A/c systems dehumidify the air but night ventilation can never do it.

- If the temperature in the room is going up to 30°C you get a decrease of human efficiency from 100% down to 60°C. After 5h per year at this level you lose more money than the annual costs per person of a fully equipped A/c system!

Answer:
- It is true that the comparison presented does not take into account dehumidification and it is restricted to sensible cooling. Your observation will be added to the tool and if possible some RH calculations will be added to indicate whether dehumidification is required and how often.

- High temperature for a few hours may occur in naturally ventilated buildings with consequence effects to productivity. However, there are indications and results from recent surveys amongst office workers that occupants prefer naturally ventilated buildings and the control they have over their environment. this might have an effect on their productivity.

SESSION 4: Ventilation Systems

Title:	**Controlled Air Flow Inlets**
Author:	**W De Gids, The Netherlands**

Question from: M Bassett
What is the consequence (in contaminant concentration terms) of relying on airing only in your n=4 houses.? I have to argue this point in contaminant concentration terms in NZ, rather than qualitatively, and wonder if you have data for the Netherlands?

Answer:
To be honest, I don't have data for your specific problem. Although we have four studies for the Dutch Ministry of Housing in which we did a year around computer run with our multizone ventilation model to calculate the cumulative exposure to pollutants. The report is only available in English.

SESSION 5: Ventilation Systems

Title:	**Increased Ventilation Airflow Rate: Night and Day Cooling of an Office Building.**
Author:	**C. Martin, France**

Question from: S H Liem

(i) What is the ventilation rate when it is allowed to ventilate without time limit?

(ii) What is the air speed during the day when ventilation is used without limit?

Answer:
(i) The ventilation rate allowed without time limit is the same as during the night: 6 to 10 vol/h.

(ii) We don't know at this time, that is why we need to do further work to assess the comfort parameters in the offices. That is also why it would be better to use a supply system, it is easier to manage the supply air and to avoid high air velocity in the occupied zone.

Title:	**Guidance Tools for Night and Evaporative Cooling in Office Buildings**
Author:	**J R Millet, France**

Question from: S Aggerholm
What are the draw-backs of not including external air conditioners in the control strategy?

Answer:
The outdoor conditions are taken in to account in the additional controls (see par.2.1.3, p.449 of proceedings) both to control the indirect humidification and the heat exchanger runnings.

Title:	**Reducing Cooling Loads with Under Roof Air Cavities**
Author:	**M. Perino, Italy**

Question from: E Perera
Can this simple model be applied to vertical ventilated facade cavities - considering that this would take it out of your measured tilt angle?

Answer:
We have not experienced configurations with tilt angles over 60°, but from a theoretical point of view the model may be applied also for vertical air cavities.

Question from: J R Millet
Do you take into account the wind effect for the calculation of the cavity air flow? Taking it into account could improve the efficiency from 50% to 20% according to experimental and simulation studies done in CSTB.

Answer:
No, the ventilated air cavity model does not take into account the effects of wind on air flow rate value. It takes only into account the changes of the outdoor heat transfer (film) coefficient. However, it should be remembered that this calculation tool has been thought as a design method and not as a simulation procedure. For design purposes the heaviest conditions (i.e. no wind) are usually assumed for calculation.

Title:	**A Method for the Economical Optimisation of the Design Temperatures and the Connecting Flows of a Cooling System**
Author:	**P Sarkomaa, Finland**

Question from:E Perera
This is a very interesting approach. Would you please say whether this technique has been applied to real-life design? If so, how did the buildings perform with respect to the design? Secondly, what explicit value of s would you consider represents an average effectiveness and what value would represent a very good heat exchanger?*

Answer:
Method is useful to define design values of complicated or unusual cooling system. It is general method and it can be applied:

- for individual building
- for district cooling systems
- for applications of industry.

It has been used for these purposes.

Parameters s^* is varying depending on special targets or needs of (investor) customer. Lets say interest rate can vary very much:

- it is in industry 10%....50%
- for commercial building (3%)...6%.....20%
- for normal building (for living) 3% 10%

Also operating time is different for different systems:

- industry - 1a.....25a
- commercial buildings - 15a.....25a

It means that factor of present value a has different values a=2....15.

Also, the price of electricity is varying. It is for industry in Finland e=30...80 US$/MWh and for other about e=60...80 US$/MWh. Also marginal cost heat transfer surface area of very good heat exchangers is varying depending on type-e of heat exchangers

- type of condenser
- type of vaporiser
- type of fan coil units
- type of other heat-exchangers.

It is affected also about situation of market.
There are many values of s^* for good heat exchangers, equations (13) and (14).

It is dependent on:

- Economical parameters

  a = f(interest rate, operating time)
  e = price of electricity, US$/kWh
  h = marginal costs of heat transfer surface areas (US$/$m^2$) with installation costs.
  r = ratio of annual maintenance cost and investment costs. You have to find out statistic to evaluate it. Usually r=0.02 - 0.05

- System parameters
  heat capacity flows ($c=q_m c_p$)
  COP or efficiencies $\eta=\eta_m\eta_{mk}\eta_{cd}\eta_i$
  temperatures of the cooled stream (in room) T_{r1}, T_{r2}, T_{a1}, T_{a2}

- Operating parameters like annual peak load power time t.

All systems have their own s^*_v and s^*_c

Question from: F Steimle
Thank you for the interesting approach to optimization of cooling systems with water loops on both sides. Please give me the definition of your effectiveness for the condenser and the evaporator.

Answer:
Please see pages 443 and 444 Fig 2 in the proceedings.

Title:	**Macroscopic Formulation and Solution of Ventilation Design Problems**
Author:	**J. Axley, USA**

Question from: W de Gids
Have you done some work on the combination of wind and stack with your approach?

Answer:
No, I have not. While the resulting system equations become quite a bit more complex, system behaviour may be determined in terms of design parameter using one of the available macroscopic analysis programs that solve the coupled thermal airflow problem (e.g. ESPair or the next generation of CONTAM currently under development at NIST).

Question from: E Perera
Can one carry out an order of magnitude assessment of the non-linear equations to reduce it to a more tractable linearized form - which would then allow simpler constant analysis?

Answer:
Possibly, but it may prove easier to simply approach the problem of describing system response in terms of the design parameters numerically. Using one of the available multizone analysis tools one may compute response for a range of design parameters - A brute force but, perhaps, not unreasonable strategy.

Question from: A H C Van Paassen
Why is the heat accumulation of the building not implemented in the equations? It may lead to the wrong design for tight tight buildings. In that case natural ventilation will lead to too high temperatures in summer.

Answer:
The simple cases discussed were limited to steady state idealisations for simplicity. As noted in the presentation of the general case, thermal capacity may be directly accounted for. The actual evaluation of feasible combinations of design parameters becomes more complex.

One may, for example, determine feasible solutions as they evolve in time (e.g. schedules for window opening to achieve thermal comfort over a 24 hour period) or, as a second example, one may determine envelopes of feasible solutions (e.g. maximum combinations of window openings to address extreme cases).

SESSION 7: Innovative Cooling

Title:	**Characteristics Values of Natural Ventilation and Air Conditioning**
Author:	**A. Van Paassen, The Netherlands**

Question from J. Andersson
It was an interesting comparison you presented, but I missed a sixth factor to be considered and that is the location of the building and the outdoor environment. In noisy environments and high outside emissions, it might be less advantageous to use open windows for ventilation and cooling.

Answer:
I agree with your proposal to add a sixth factor "the outdoor environment factor". This factor will work out as an advantageous point for air conditioning and mechanical ventilation. Although a hybrid system that uses natural ventilation for cooling during the night and mechanical systems during the day. In the natural ventilation program the problem of air pollution and noise will be studied.

Comment from: E Perera
This comment is in response to the question by Johnny Andersson (Scandiaconsult) regarding natural ventilation in urban areas and city centres. In the EC NatVent project, this whole area is being considered. Technical and strategic design solutions are now being prepared and will provide a means to overcome the technical barrier of noise and air pollution in these situations

Question from: A Thanassios Argiriou
How are the outdoor climatic conditions taken into account in the methodology?

Answer:
Results given are based on northern European climate. The methodology can be extrapolated in Southern climates after appropriate simulations.

Comment from: O Jensen, Denmark
Your presentation is very interesting. It is an attempt to quantify our philosophy. Maybe at the next AIVC conference you can present results based on measurements.

SECTION 2: Papers Not Included in Main Proceedings

ENERGY CONSERVATION IN THE BUILDING SECTOR IN CREECE

by Dimitrios Nomidis
Greek Representative in ExCo of ECBCS Program

Ladies and Gentlemen,

First of all, I would like to welcome you all in Greece.

It is a great honor for us to have this Conference of the International Energy Agency in our country.

I wish you best success to your works at this Conference and I carry to you the same wishes and regards of the Secretary General of our Ministry of Development, who, unfortunately, was not able to be here with us today, although he strongly wanted it, due to some urgent and unpostponable obligations of his.

I would like very strongly to take advantage of this annual Conference of the AIVC to officially announce that the participation of Greece to AIVC has been approved (after long efforts of our Ministry for its funding to the Ministry of Finance, I should admit) and our country will participate from now on as a full member of the Air Infiltration and Ventilation Center.

On the occasion of this Conference, I would like to give you a rough picture regarding the Energy Situation of Buildings in Greece and the Policy Measures related to this matter.

Energy Situation of Buildings in Greece

We all are well aware of the fact that the Building sector is one of the most energy spending sectors in the World Economy.

Worldwide, the primary energy consumption in buildings is close to 17 million barrels of oil per day and corresponds almost to the entire daily production of OPEC.

In the European Union, buildings are the major energy consuming sector. At the same time, the European building industry is one of the biggest economic sectors representing a yearly turn-over of the order of 400 billion ECU.

The final energy consumption of the building sector in the European Union is close to 300 MTOE per year, while in Greece close to 4,6 MTOE per year.

The primary energy consumption of the building sector in the European Union, including the contribution of solar energy (direct through solar collectors and indiirect through passive heating and natural lighting) is close to 740 MTOE per year.

In the European Union countries, the primary energy consumption in buildings averages 40 percent of the total energy consumption , ranging from 20 percent for Portugal to 45 percent for Ireland. In Greece, buildings sector accounds for about 30% of the total primary energy consumption of the country.

The individual energy consumption per year and inhabitant for the building sector in the European Union is close to 1 TOE, and presents a slight incresing trend, during the last years, of 0,7 % annually, while for Greece this factor is 0,55 TOE per year and inhabitant, i.e about half the corresponding factor for Europe (apparently due to the much milder climate of Greece compared to the other European countries and the smaller number of air-conditioned buildings) and it presents a clear increasing trend of about 1,8% annually (apparently, due to the constantly accelerating number of air-conditioned buildings in Greece).

The specific energy consumption per year and per square meter for the different types of buildings (residential, commercial, offices, schools, hospitals, hotels) as well as the breakdown of this specific energy consumption according to the energy use (heating, cooling, lighting, equipment), is shown in the following table.

SPECIFIC ENERGY CONSUMPTION OF BUILDINGS IN GREECE

	Energy use				
	Heating	Cooling	Lighting	Equipment	Total
Building type	kWh/m^2.year	kWh/m^2.year	kWh/m^2.year	kWh/m^2.year	kWh/m^2.year
Residential average	95	15	15	60	185
" air conditioned					
Commercial average	74	18	19	41	152
" air conditioned	74	49	17	30	170
Offices average	95	20	24	48	187
" air conditioned	99	36	25	66	226
Schools average	66	2	16	8	92
" air conditioned	99	42	30	9	180
Hospitals average	299	3	52	53	407
" air conditioned					
Clinics average	179.5	48.4	25.8	21.3	275
" air conditioned					310
Hotels average	198	11	24	40	273
" air conditioned		26			285

It should be mentioned at this point, that energy consumption for heating and cooling keeps constantly increasing in Greece, as higher family incomes have enabled the installation of heating and air-conditioning systems in a continiously growing number of new and old buildings. The use of conventional A/C systems, as a matter of fact, has become very popular in the last years. Sales of packaged air-conditioning units have jumped from about 2000 in 1986 to over 100.000 units in 1988. In the tertiary sector too, an ever growing number of buildings proceed to the installation of air-conditioning systems (hotels,offices, commercial buildings, hospitals, schools, etc.).

The impact from the increased use of air conditioners on electricity demand begins to become a serious problem, especially for the greek islands (like Krete), which have an autonomous and with a limited capacity electricity generation and in which the energy cousumption during the summer period, increases dramatically for touristic reasons .

Another factor which raises seriously the energy consumption level during the summer period is the urban heat island effect which is observed in urban areas with high population, like Athens, Pireaus, Thessaloniki and Patras, in Greece. Measurements in the city of Athens have shown that the much higher summer air temperatures which prevail in the central region of the city compared to those of the suburbian areas (with temperature differences ranging from 5-12 °C during the day and 2-5 °C during night) result to doubling the cooling loads of the central areas in relation to those of the suburbian areas. This effect, in combination with the outdoor air pollution and the increased needs for indoor air quality, ventilation and air cleaning, raises dramatically the energy consumption in urban areas.

Energy Conservation Measures in Greece

What Greece has been done for the Energy Conservation in the building sector?

There are a series of legislative and other measures in Greece for energy conservation in buildings and in the broader tertiary sector.

1. First of all, there is a Law for the Insulation in buildings, the Presidential Decree of 4th July 1979 which directly affects the energy consumption in buildings, as it is obligatory and sets minimum insulation standards which result in a specific heating load ranging from 0,6 $W/m^2.°C$ to 1,5 $W/m^2.°C$, depending on the ratio between the external surface area and the volume of the building as well as on the climatic zone of the country (Greece is divided according to this law in three climatic zones: the mild, medium and severe zone).

2. Another law is now under way, prepared by the Ministry of Environment in collaboration with our Ministry of Development / Energy Section, which will provide for:

   a) the submission of an integrated energy study for each building, which will include the potential use of renewable energy sources (usually active or passive solar energy) and bioclimatic design. This study will conclude with the specific energy consumption of the building, which should be lower than some limits, according to the type of the building and the climatic zone. An annex of the law will give all the specifications and instructions for the design and calculations and the overall methodology for the realization of this integrated energy study.

   b) energy certification of new and existing buildings, according to which buildings will be classified in energy efficiency classes in relation to their type. The energy efficiency class of each building will be recorded to all legal papers related to the ownership state of the building, which are necessary for its sale, rent, etc.

   c) energy auditing of buildings, according to a Regulation recently issued by the Ministry of Development / Energy Section, for energy auditing in the Industrial and the Tertiary sector.

   d) improvement of the existing Law for buildings insulation by setting more strict insulation standards and by enforcing the actual implementation of the law.

   e) finally, the law provides incentives for the realization of energy conservation in existing buildings. Those incentives are administrative and financial, mainly tax-reduction incentives.

3. The Law for Economic Development N.1892/90, which gives incentives for investments in general. This law provides, especially for energy conservation investments in the industrial and tertiary sector, high rates of subsidies in the range of 40% - 55% according to the geographical area.

   The energy criteria for the financial support of such investments are:

   - The promotion of R.E.S.

   - Substitution of electricity or fuel oil for fuel gases, treated waste products, R.E.S. or waste heat recovery

- Energy savings higher than 10% per unit produced, under the prerequisite that the investment concerns the energy installation and equipment and not the productive ones.

4. The Operational Energy Programme, which is financed by 75% by European Community Funds and by 25% by the Greek government. This programme includes a Subprogramme for Energy Conservation investments with a total budget of 375 million ECU and a Subprogramme for R.E.S. investments with a total budget of 190 million ECU. This Energy Programme will be running until the year 1999, and gives financial support for the realization of investments from the industrial and tertiary sector, including buildings, as well as from Small and Medium Size Enterprises, in the field of Energy Conservation through the improvement of energy efficiency and the promotion of R.E.S.

   This financial support is given in the form of subsidies in a wide range from 30% up to 55%, for a variety of applications inciuding:

   - Energy conservation in existing enterprises.
   - Cogeneration of heat and power.
   - Substitution of electricity or conventional fuels for natural gas and LPG.
   - Solar systems (active or passive)
   - Wind energy systems for electricity generation or sea water desalination
   - Geothermal energy systems
   - Small hydropower systems
   - Biomass exploitation
   - Photovoltaic systems
   - Passsive systems for heating, cooling and lighting and Bioclimatic systems.

5. I should mention also the Law 2244/94 which allows for the electricity generation by the private sector through Cogeneration or R.E.S., breaking this way the monopoly of the Public Power Corporation (PPC) for electricity production.

   This law, accompanied by 3 ministerial decisions that provide for some specifications of the law, distinguishes two types of electricity producers, the selfproducers, who mainly cover their own needs in electricity, and the independent producers, who sell all the electricity produced to PPC, and defines the prices of the electricity energy sold to PPC as a percentage of the current PPC tarrifs (that is, from 60% up to 90% according to the type of electricity production and the type of producers).

6. Finally, there is a series of European energy efficiency legislation, in the form of directives, coming from the European Community Programme SAVE, which addresses the building sector and which have been transferred and harmonized in the Greek legislation.

   We distinguish two different types of this legislation:

   - The European directives setting minimum energy efficiency standards. Such directives have been issued for:

     - Non industrial boilers, and

     - Refrigerators and freezers

   - The European directives for energy labelling of the domestic appliances (that is, signing of the domestic appliances with standard labels giving exact information on the energy consumption and the energy efficiency class of each appliance). Such directives have been issued for:

     - Refrigerators and freezers

     - Washing machines

     - Driers

     - Combined washer-driers, and

     - Dishwashers

     Under preparation are labelling directives for:
     packaged air-conditioners, ovens, water-heaters, lamps, etc.

We believe that with the above mentioned legislation framework and measures, which are to be continued and improved as well as furthermore supplemented and extended with new ones, we will manage to cover the largest portion of the energy conservation potential in the building sector in Greece, which is estimated in the rate of 18%.

And with those words, I would like to close my presentation and to wish you all the best and success in your works during the Conference.

Dehumidification by alternative cooling systems
-Sorption-supported dehumidification with
different liquid salt solutions-

Jürgen Röben

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Sorption-supported dehumidification with different liquid salt solutions

Jürgen Röben

1. Introduction

The traditional way to dehumidify the outdoor air in a heating, ventilating and air conditioning (HVAC) system is by cooling the air temperature down below the dew point. For this process a refrigeration system is necessary to realise these low temperatures. Nowadays the disadvantages of refrigeration systems are widely known. An alternative method to dehumidify the air is by separating the process of dehumidification and cooling. There are different ways to get low supply air temperatures for cooling the indoor spaces. It is possible for example to use well water, an evaporative cooling system or, of course, a refrigeration system with relatively high evaporation temperature. This cooling components are well known and already in practice so that this paper focuses on the dehumidification process.

The paper gives a general view of the adsorptive and absorptive dehumidification components. Then a new dehumidifier which uses a liquid desiccant will be described. A small prototype was tested in an experimental plant in the laboratory of the Institute of Applied Thermodynamics and Air Conditioning in Essen. The design of the dehumidifier and the first results of the measurements will be presented.

2. Methods of dehumidification

In the HVAC technology the following dehumidification systems are commonly used: ❶ Condensation on cold surfaces of chillers or water droplets. ❷ Desiccating through the contact with hygroscopic materials.

To ❶: This kind of dehumidification is the frequent applied technology. To get condensate, temperatures below the dew point of the dehumidifying air are necessary. These low temperatures can be realised by an evaporating refrigerant (direct evaporator) or by cold water (water cooled chiller). The cold water is made by a refrigeration plant and its usual supply temperature is approximately 6°C. The refrigeration systems can be basically subdivided into the two following groups: Compression-refrigeration and Absorption-refrigeration-system.

The disadvantages of this systems are:

- Poor controllability due to constant water temperature in the cold water chiller.
- High energy demand for the refrigeration process.
  - ⇨ For the air conditioning it is not always necessary to have supply water temperatures of 6°C.
  - ⇨ Energy demand for the compressor.
- Reheating of the dehumidified air is often necessary.
- Only limited usability of low temperature heat by the refrigeration systems.

By increasing the supply water temperature it is possible to reduce the energy demand for preparing the cooling water with refrigeration systems. By increasing the water temperature of about 1 K the improvement of the coefficient of performance (COP) is approximately 3 %.

To ❷: Desiccating by contact with hygroscopic materials is distinguished by the kind of the used materials: Solid hygroscopic and liquid hygroscopic-materials.

There is a great variety of solid materials which can be used for air dehumidification. Active carbon, active aluminium, silicagel, zeolithes as well as hygroscopic salts are mostly used for technical drying. Silicagel and hygroscopic salts are preferred for the air dehumidifier. Continuous working rotary wheels or discontinuous working packed beds are equipped with such solid desiccants. In this paper the solid desiccants and the design of belonging plants is not described because the main emphasis lies on the use of liquid desiccants.

One of the first liquid desiccants which was used for the dehumidification of air was triethylenglycol. Due to the high vapour pressure the application of this desiccant in open cycle systems is unfit because there are high losses of desiccants which have a negative influence to the environment. However open desiccant cycles working with solutions of lithium chloride, respectively, calcium chloride are partly successful in practice or investigated in promising research projects.

3. Liquid desiccants

The dehumidification of air is possible by using different solid or liquid hygroscopic materials. In this work we only used liquid desiccants like Calciumchloride-, Lithiumchloride- and Klimat 3930S-solutions. The Klimat 3930S-solution is made by the company Solvay Deutschland and consists of 39 Ma-% $CaCl_2$, 18 Ma-% $Ca(NO_3)_2 \cdot 4H_2O$, 5 Ma-% emulsifier S and 38 Ma-% water.

3.1 Requirements on liquid desiccants

In general the desiccants should be non toxic and environmental-compatible, because they are used in open cycle systems at ambient pressure. Furthermore, volatile contents are not allowed in the hygroscopic material, except water, and the desiccant should be non-flammable and non- explosive.

To guarantee a continuous dehumidification process the desiccant must possess a long-term stability. Another advantage of air dehumidification with hygroscopic materials is the possibility to use low temperature heat for the regeneration process. The material costs should not be high and the steady quality of the material is to be guaranteed.

To evaluate the hygroscopic materials, the substance data for the considered temperature range is taken from literature but also by experiments in the laboratory. With this data it was possible to create equations to calculate the following material characteristics: vapour pressure, dew-point temperature, density, specific heat capacity, dynamic viscosity and the surface tension.

4. Testing plant to dehumidify air

The centre of the testing plant is a new dehumidifier which uses liquid desiccants to dry the air. The saline solution ($CaCl_2$- respectively Klimat 3930S-solution) is running in direct contact with the air and both in counterflow. Figure 1 shows the testing plant with the dehumidifier and the several components to create steady conditions.

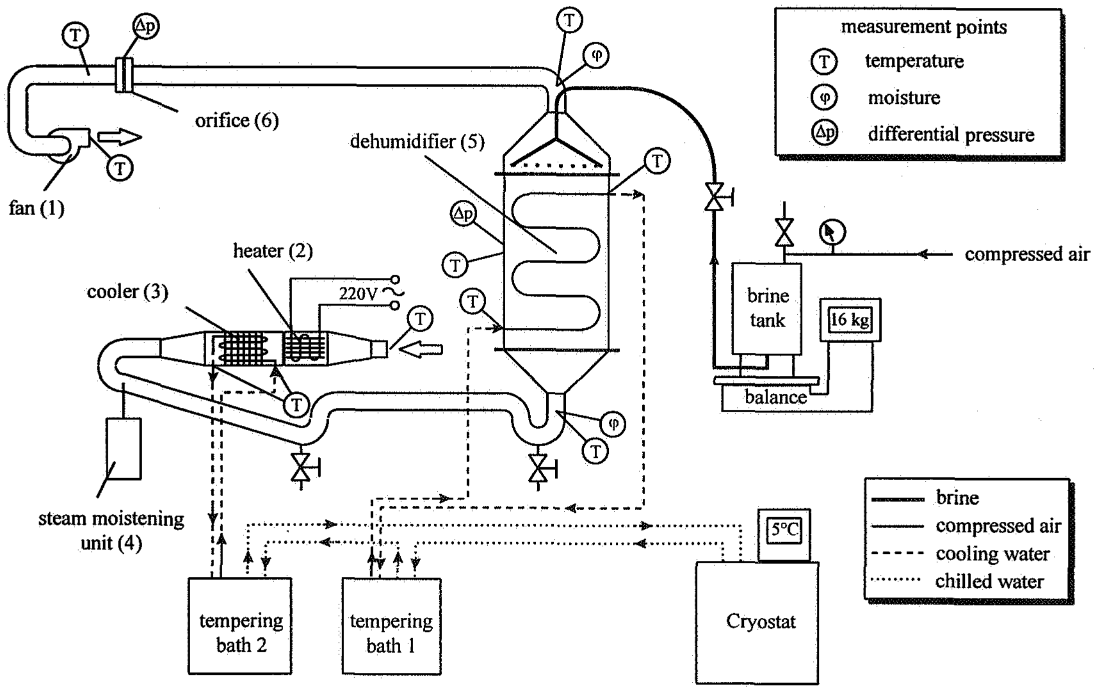

Figure 1: Testing plant with the new liquid dehumidifier.

To carry off the absorption heat it is possible to cool the dehumidifier with chilled water. The water is running through little channels of a double-web-plate in cross-counterflow with the saline solution. The direction of the air, the saline solution and the chilled water are shown in figure 2. The surface for the heat and mass transfer is 4 m² and five double-web-plates are built in the test dehumidifier.

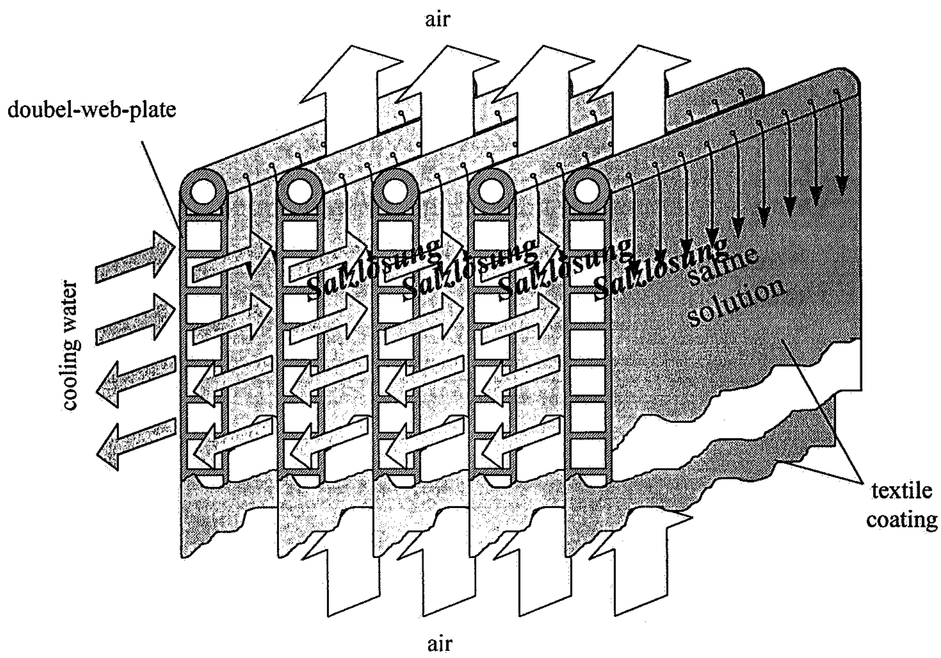

Figure 2: Schematic display of the transfer surfaces

4.1 Measurements

For the investigation of the dehumidifier several measurements were carried through. They can be classified in three groups: The measurements without desiccants, the measurements with the $CaCl_2$-solution and the measurements with the Klimat 3930S-solution. The following measurement program shows the boundary conditions which were changed to describe the behaviour of the dehumidifier:

✹ Without desiccants: Constant air inlet temperature ($t_{G,in}$ = 32,5°C), cooling water inlet temperature ($t_{KW,in}$ = 20°C) and a constant cooling water mass flow ($\dot{M}_{KW}$ = 0,044kg/s). Variable air velocity in the dehumidifier (w_G = 1 m/s to 2,5 m/s).

✹ With desiccants: Constant air inlet temperature ($t_{G,in}$ = 32,5°C) and constant water content of the air ($x_{G,in}$ = 14,5 $g_{water}/kg_{dry\ air}$).
Variable cooling water inlet temperature ($t_{KW,in}$ = 20°C; 25°C), variable air velocity in the dehumidifier (w_G = 1 m/s to 1,8 m/s), variable cooling water mass flow ($\dot{M}_{KW}$ = 0,044kg/s; 0,022 kg/s; 0 kg/s) and a variable mass flow ratio ($\dot{M}_G/\dot{M}_L = \Pi$ = 10 to 110).

4.2 Heat and mass transfer in the dehumidifier

By the measurements it was possible to find equations which describe the heat and mass transfer. To design a dehumidifier it is helpful to use dimensionless numbers. The following dimensionless numbers were used in this research work.

Reynolds-number: $$Re = \frac{w \cdot L^*}{\nu} \quad \text{or} \quad Re = \frac{w \cdot \rho \cdot L^*}{\eta}$$

Nusselt-number: $$\mathrm{Nu} = \frac{\alpha \cdot L^*}{\lambda}$$

Péclet-number: $$\mathrm{Pe} = \frac{w \cdot L^*}{a} \quad \text{or} \quad \mathrm{Pe} = \mathrm{Re} \cdot \mathrm{Pr}$$

Prandtl-number: $$\mathrm{Pr} = \frac{\nu}{a} \quad \text{or} \quad \mathrm{Pr} = \frac{\eta \cdot c_p}{\lambda}$$

Sherwood-number: $$\mathrm{Sh} = \frac{\beta \cdot L^*}{D}$$

Schmidt-number: $$\mathrm{Sc} = \frac{\nu}{D}$$

Where L* is the characteristic length. As in this work exclusively the diameters of passed channels are concerned so that the hydraulic diameter is used:

$$d_h = \frac{4 \cdot A}{U}$$

It was possible to determine the following dimensionless equation to describe the heat transfer in the dehumidifier by the results of the extensive measurements:

$$Nu_G = 2{,}05 \cdot \left(Re_G \cdot Pr_G \cdot \frac{d_h}{L} \right)^{0{,}5}$$

The liquid desiccant and the air in the dehumidifier are in a direct contact so that the velocity of the desiccant has an influence to the air. Therefore, the Reynolds-numbers of the saline solution were calculated with the following equation.

$$Re_L = \frac{\dot{M}_S \cdot \left[1 + \left(\frac{C_{ein} - C_{aus}}{2} \right) \right]}{8 \cdot b \cdot \eta_L}$$

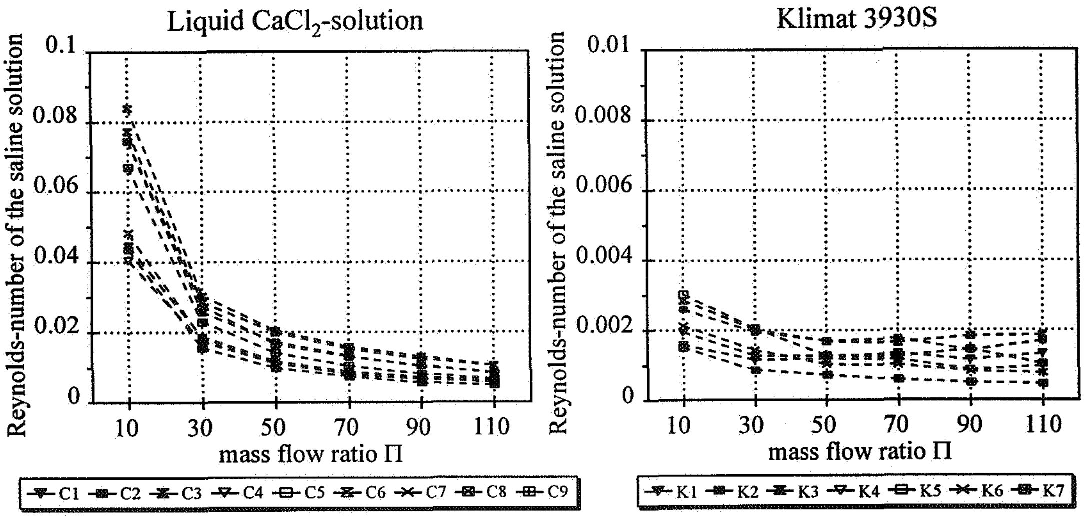

Figure 3: Reynolds-numbers of all series of measurements with liquid desiccants in dependence of the mass flow ratio.

Figure 3 shows the Reynolds-numbers of liquid $CaCl_2$- and Klimat 3930S-solution in dependence of the mass flow ratio Π. The dimensionless equation to describe the mass transfer in the dehumidifier includes therefore the Reynolds-number of the liquid desiccant.

$$Sh_G = 2{,}05 \cdot \left(Re_G \cdot Sc_G \cdot \frac{d_h}{L} \right)^{0{,}5} \cdot \left(1 + K \cdot Re_L^{0{,}33} \right)$$

The equation to describe the mass transfer includes a correction term. In this term K is a specific material constant.

By an empirical way it was possible to find the specific material constants for the investigated liquid desiccants.

Liquid $CaCl_2$-solution	$K = 0{,}21$
Liquid Klimat 3930s-solution	$K = 0{,}60$

The range of dehumidification Δx in dependence of the mass flow ratio of both desiccants is shown in figure 4.

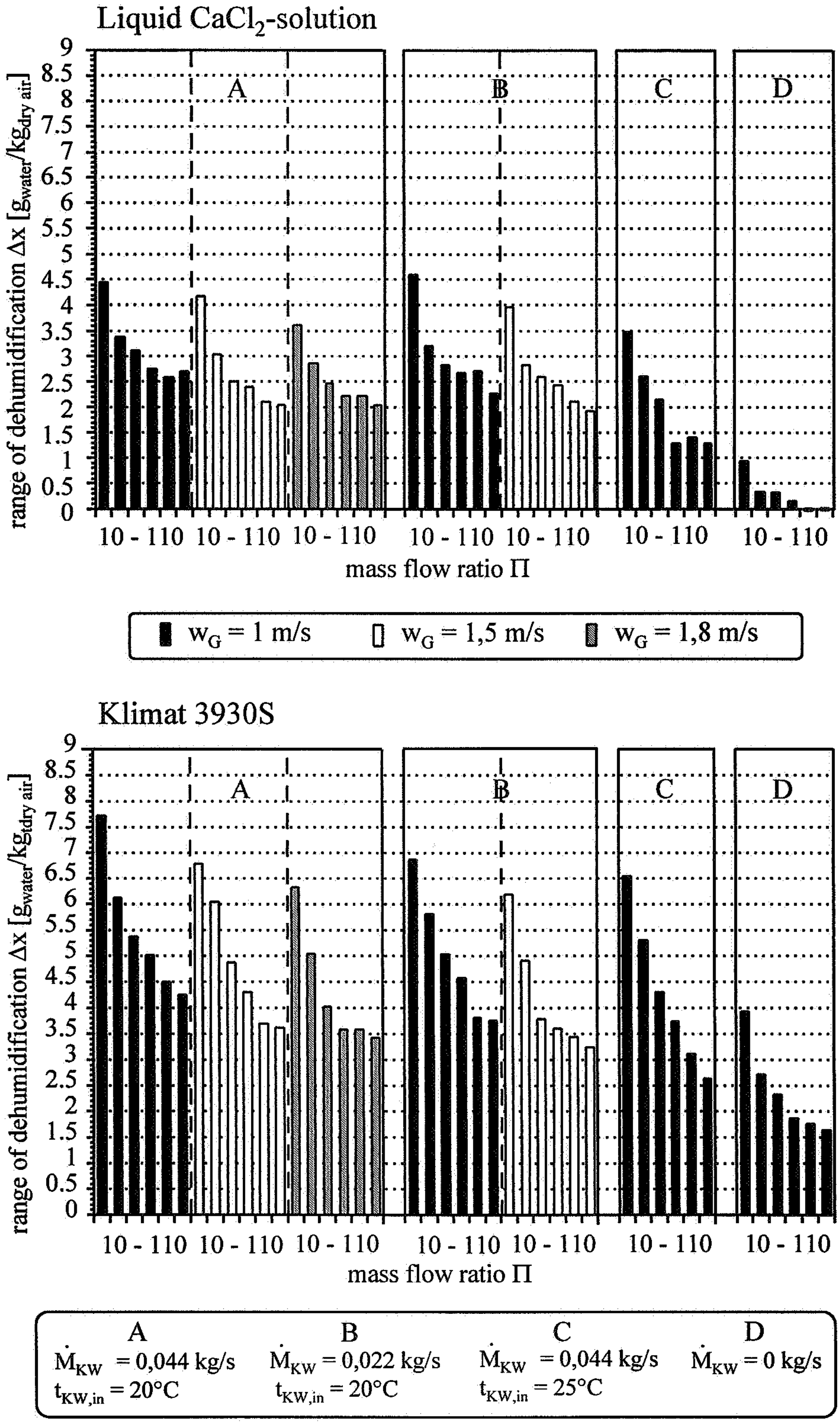

Figure 4: Achieved range of dehumidification Δx in dependence of the mass flow ratio of both desiccants.

It is obvious that the range of dehumidification by using the liquid $CaCl_2$-solution is lower than by using Klimat 3930S. The results of series D are very interesting, because this measurements were done without chilled water for the removal of the absorption heat. The range of dehumidification by using Klimat 3930S is here up to $\Delta x = 4\ g_{water}/kg_{dry\ air}$ and by using $CaCl_2$-solution only $\Delta x = 1\ g_{water}/kg_{dry\ air}$.

5. Conclusions

The results of the theoretical and experimental research work can summarised to the following points:

1. Modelling of equations to describe the desiccant material characteristic for a temperature range from 10°C to 70°C and the considered range of concentration of saline solution ($CaCl_2$-, LiCl-, Klimat 3930S-solution).
2. Modelling of an equation to describe the heat transfer in the new dehumidifier.
3. Modelling of an equation to describe the mass transfer in the new dehumidifier.
4. Comparison of the dehumidification behaviour of different liquid desiccants.

With this results it is possible to design a dehumidifier, which operates in according to the investigated test dehumidifier.

6. Acknowledgement

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7. Symbols and Indices

Symbols:

a	thermal diffusivity
A	transfer area
b	width
C	water content (solution)
d	diameter
D	diffusion coefficient
K	specific material constant
L	length
L*	characteristic length
$\dot{M}$	mass flow
t	temperature
U	circumference
w	velocity
x	water content (air)

Indices:

G	gas (air)
h	hydraulic
in	intake
KW	chilled water
L	saline solution
S	salt

Greek Symbols:

α	heat transfer coefficient
β	mass transfer coefficient
Δ	delta
η	dynamic viscosity
φ	density
λ	thermal conductivity
ν	cinematic viscosity

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Title

Thermal analysis of rooms with diurnal periodic heat gain, ThermSim Part 1 : Derivation

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Thermal analysis of rooms with diurnal periodic heat gain, ThermSim
Part 1 : Derivation

Synopsis

Temperature and cooling demand in a room summertime is influenced by numerous factors like : internal gains, ventilation, solar gain, behaviour of occupants, thermal inertia of the room, and outdoor conditions (climate).

The thermal environment and cooling demand summertime is often analysed using advanced computer programs. These programs require detailed input describing every feature of the room. Often the overview, transparency and some of the physical insight is lost using these advanced computer programs.

In a predesign phase of a project it is preferable to do simple calculations of the thermal behaviour of a room. These simple calculations often gives more physical insight and overview than using computer programs. Simple calculations also gives a quality assurance of later computer analysis of the room.

In this paper a simplified thermal analysis of a room is presented, called *ThermSim*, which can be used as a hand calculation method in the predesign phase of a project.

In rooms with significant solar gain, the total heat gain to the room at any time of day, can be approximated with a simple cosine function. This assumption together with a thermal one-mass-model, and a frequency analysis model often used in electric circuits analysis, forms the basis of the thermal room model. The solution of this model gives a simple equation which can predict the temperature in the room, exposed for a heat wave midsummer.

The method shows in a transparent way the time-lag between maximum heat gain and maximum occurring room temperature. I addition the "thermal build up" in a heavy room from day to day during a heat wave is easily predicted. ThermSim is compared (comparison found in part 2) with more advanced computer analysis and shows good agreement when the model assumptions is fulfilled.

List of symbols

Symbol	**Description**	**Unit**
A_{fac}	Facade area	m^2
A_{win}	Area for whole window construction (including frame)	m^2
C_{air}	Heat capacity of air (can be set to 0.34 Wh/kgK)	Wh/kgK
L	Mechanical or natural air flow rate	m^3/h
n	Air infiltration in ACH	1/h
$\bar{q}$	Daily mean heat gain	W
$\hat{q}$	Daily amplitude heat gain	W
t	Time	h

t_{max}	Time for maximum heat gain and external temperature to occur	h
T_i	"Effective" room temperature	°C
$\bar{T}_e$	Mean daily external temperature	°C
$\hat{T}_e$	Daily amplitude external temperature	°C
U_{fac}	U-value facade construction	W/m²K
U_{win}	U-value window construction	W/m²K
V	Room air volume	

1 Introduction

Thermal design of rooms are often done using advanced computer simulation tools. These tools are often cumbersome to use and give little insight in the physical process which is simulated. In the early stage of a design process it can be beneficial to use simple hand calculation for a rough predesign of a room. This gives a much better physical insight to the thermal process in the room. In addition it can be a valuable quality assurance for later computer simulations.

This paper describes a simplified method for simulation of temperatures and cooling load, called *ThermSim*, which can be used for hand calculation or it can easily be implemented in a spreadsheet. This method can be used on most rooms provided they have a daily variation in the heat gain that can be approximated with a sinusoidal variation in the gain. This is often the case with rooms exposed to solar radiation.

The method shows in a transparent way how the temperature evolves from day to day during a heat wave. In addition it gives the daily variation in room temperatur, with maximum and minimum room temperature. Or, it can be used to estimate necessary cooling load for keeping the temperature and daily temperature variation at an acceptable level.

This paper, part 1, derives the method and interpret the different terms in the model. Part 2, which is given in a accompanying paper, contains tables which simplifies the use of the method, along with examples and comparison to advanced computer simulations.

The model is based on a five main assumptions :

1. Daily variation in heat gain and external temperature is approximated by a sinusoidal function
2. Room air temperature, surface temperatures and "building structure" temperature is "merged" into one mean effective room temperature
3. The effective heat capacity of the room is limited to a finite thickness of the building constructions
4. Every input to the model is either constant or approximated with a diurnal sinusoidal variation
5. Heat loss to adjacent rooms are negligible

2 Formulation of model

2.1 Physical and mathematical derivation

In summertime rooms with a external windows are exposed to a diurnal variation in heat gain. This variation can often, with good approximation, be estimated with a sinusoidal function on the form :

$$q(t) = \overline{q} + \hat{q}\cos\left(\frac{2\pi(t - t_{\max})}{24}\right) \qquad \text{(W)} \qquad (1)$$

q(t) is heat gain in Watt as a function of time (t), $\overline{q}$ is the daily mean heat gain (W), $\hat{q}$ is the heat gain amplitude[1] (W), t is time (hours) and t_{max} is the time when maximum heat gain occur. The period is of course 24 hours.

If there is a ventilation system (natural or mechanical) supplying the room with the air flow rate L_{vent} the cooling effect (heat loss) is :

$$q_{vent} = C_{air}L_{vent}(T_i - T_e) = H_{vent}(T_i - T_e) \qquad \text{(W)} \qquad (2)$$

C_{air} is the volumetric heat capacity of air which can be set to 0.34 Wh/m³K, L is the air flow in m³/h , T_i is the room air temperature (°C) and T_e is the external temperature (see below). Air flow is assumed constant, and mechanical cooling (cooling coil) is not considered here (treated separately in section 2.3). Heat gain from fans in mechanical ventilation has to be added to the other heat gains in equation (1).

Heat loss to the external that can be written :

$$q_{ext} = \left[C_{air}nV + \sum U_{win}A_{win} + \sum U_{fac}A_{fac}\right](T_i - T_e) = H_{ext}(T_i - T_e) \quad \text{(W)} \quad (3)$$

where U_{win} is the U-value of the window and A_{win} is the window area including the frame(m²), n is the infiltration rate in ACH and V is the room air volume (m³), U_{fac} is the U-value for the facade construction and A_{fac} is the facade area.

The external temperature (T_e) in equation (2) and (3) , varies during the day, and this variation can be estimated with a sinusoidal function in the same form as (1) :

$$T_e(t) = \overline{T}_e + \hat{T}_e\cos\left(\frac{2\pi(t - t_{\max})}{24}\right) \qquad \text{(°C)} \qquad (4)$$

[1] The amplitude can be taken as the difference between the maximum heat gain q_{max} and the minimum heat gain q_{min} divided by two :

$$\hat{q} = \frac{q_{\max} - q_{\min}}{2}$$

If the room temperature fluctuates there will be heat accumulation in the building structure, and to some extent in the room air. If the roomtemperature rises dT_i during a small timespan dt, the heat accumulation is :

$$q_{acc} = \left(\sum A_{sur} C_a'' + C_{air} V\right) \frac{dT_i}{dt} = C_a \frac{dT_i}{dt} \qquad \text{(W)} \qquad (5)$$

Where A_{sur} is the area of all surfaces in the room having significant heat capacity, C_a'' is the effective heat capacity pr. square meter for the surface (the specific heat capacity of the accumulating layer in the construction), dT_i is the infinitesimal temperature rise during the infinitesimal timestep dt.

We are now ready to formulate the heat balance for the room. According to the first law of thermodynamics heat gain minus heat loss will equal heat accumulation :

$$\overbrace{q_{gain}(t)}^{Gain} - \overbrace{\left(q_{vent} + q_{ext}\right)}^{Losses} = \overbrace{q_{acc}}^{Accumulation}$$

or

$$\overbrace{\bar{q}_{gain} + \hat{q}_{gain} \cos\left[\omega\left(t - t_{\max}\right)\right]}^{Gain} - \overbrace{\left(H_{ext} + H_{vent}\right)\left(T_i - \bar{T}_e - \hat{T}_e \cos\left[\omega\left(t - t_{\max}\right)\right]\right)}^{Losses} = \overbrace{C_a \frac{dT_i}{dt}}^{Accumulation}$$

For mathematical convenience it can be more compactly written as :

$$\frac{dT_i}{dt} + \frac{T_i}{\tau} = \frac{T_\infty}{\tau} + \delta \cos\left[\omega\left(t - t_{\max}\right)\right] \qquad \text{(K/h)} \qquad (6)$$

where the new parameters : the frequency ω, the timeconstant τ, the stationary temperature T_∞ and the amplitude coefficient δ have been introduced, which is given by :

$$\omega = \frac{2\pi}{24} \qquad \text{(1/h)} \qquad (7a)$$

$$\tau = \frac{C_a}{H_{vent} + H_{ext}} \qquad \text{(h)} \qquad (7b)$$

$$T_\infty = \frac{\left(H_{ext} + H_{vent}\right)\bar{T}_e + \bar{q}}{H_{vent} + H_{ext}} \qquad (°\text{C}) \qquad (7c)$$

$$\delta = \frac{\left(H_{ext} + H_{vent}\right)\hat{T}_e + \hat{q}}{C_a} \qquad \text{(K/h)} \qquad (7d)$$

2.2 Solution

The solution to equation (6) can be written in the closed form :

$$T_i(t) = \overbrace{\Delta T e^{-t/\tau}}^{Transient} + \overbrace{\overline{T}_\infty}^{Stationary} + \overbrace{\hat{T}\cos\left[\omega\left(t - \tau_{lag} - t_{\max}\right)\right]}^{Periodic} \qquad (°C) \qquad (8)$$

For a more detailed mathematical derivation of (8) see the appendix. The solution is the superposition of a transient temperature, a mean stationary temperature, and a diurnal periodic temperature. T_∞ is the mean stationary value defined above, ΔTa is the transient temperature-difference given by :

$$\Delta T = T(0) - T_\infty - \hat{T}\cos\left[\omega\left(t_{\max} - \tau_{lag}\right)\right] \qquad (K) \qquad (9)$$

T(0) is the intial temperature (00.00 the first day). The temperature amplitude $\hat{T}$ related to the .periodic temperature variation is given by:

$$\hat{T} = \frac{\delta}{\sqrt{\tau^{-2} + \omega^2}} \qquad (K) \qquad (10)$$

For interpretation of the solution it is wise to introduce the parameter time-lag, defined by :

$$\tau_{lag} = \frac{\arctan(\tau\omega)}{\omega} \qquad (h) \qquad (11)$$

2.3 Estimation of cooling load

If the calculated temperature is exceeding accepted limits, we need to estimate necessary cooling capacity to maintain comfortable temperature conditions in the room. By cooling capacity we mean mechanical cooling, which comes in addition to cooling by the external air flow rate (mechanical, natural or infiltration).
Equation (8) can be used to calculate the cooling load if we do the following simplification :

- We assume that diurnal periodic stationary condition has been reached. It implicates that the transient term in equation (8) has become negligible.

The total cooling load can then be estimated by the sum of the to effects :

1. Of removing so much of the mean heat gain that stationary mean temperature is reduced to a desired mean temperature $\overline{T}_{cool}$. This load can be called the mean cooling load, denoted : $\overline{q}_{cool}$

2. And removing so much of the amplitude heat gain that the temperature amplitude is reduced to a desired temperature amplitude $\hat{T}_{cool}$. This cooling load can be called the amplitude cooling load, denoted : $\hat{q}_{cool}$

The mean cooling load can be found by setting the stationary term in (8) equal to the desired mean cooling temperature $\overline{T}_{cool}$, and reducing the heat gain with $\overline{q}_{cool}$:

$$\overline{q}_{cool} = (H_{ext} + H_{vent})\overline{T}_e + \overline{q} - (H_{ext} + H_{vent})\overline{T}_{cool} \qquad \text{(W)} \qquad (12)$$

To reduce the amplitude temperature variation around the stationary mean temperature to a desired level (e.g. 2 °C), we have to remove the heat :

$$\hat{q}_{cool} = (H_{ext} + H_{vent})\hat{T}_e + \hat{q} - \hat{T}_{cool} C_a \sqrt{\tau^{-2} + \omega^2} \qquad \text{(W)} \qquad (13)$$

where $\hat{T}_{cool}$ is the allowed temperature amplitude (variation). The maximum total cooling load is then given as :

$$q_{cool} = \overline{q}_{cool} + \hat{q}_{cool} \qquad \text{(W)} \qquad (14)$$

3 Discussion and interpretation

As seen in equation (8) there are three terms in the solution, that are significant to the resulting temperature. We have three constants with the unit of temperature (°C) : the transient temperature difference ΔT, the stationary temperature $\overline{T}$ and the temperature amplitude $\hat{T}$. In addition there are two constants with unit time (hours), the timeconstant τ and the time-lag τ_{lag}. In the following these constants, and how they influence the solution, will be discussed.

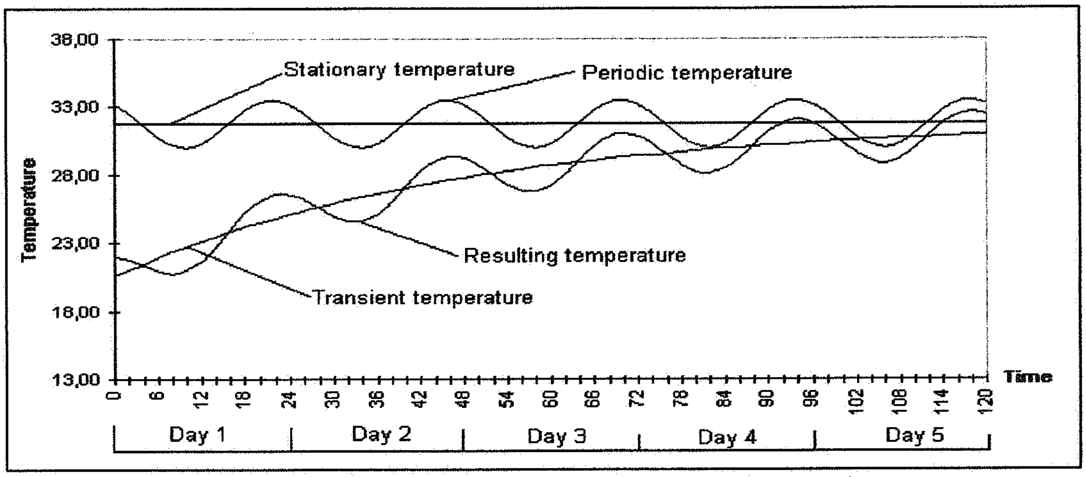

Figure 1 : Graphic illustration of the three terms in eq. (8), and the resulting roomtemperature

3.1 Transient term

The transient term given by :

$$\overbrace{\Delta T e^{-t/\tau}}^{Transient} \qquad (°C) \qquad (12)$$

determine how fast periodic stationary temperatures in the room is reached. ΔTa is the difference between the initial temperature (00.00) before the heatwave began, and the maximal occurring temperature at time 00.00 (after a long time). Since the initial temperature is lower than the stationary temperature, ΔTa will always be negative!
Typical values of ΔTa is between -5 °C and -20 °C.

How fast the transient term vanishes, that is how fast stationary conditions are reached, is entirely determined by the timeconstant τ. The timeconstant can vary from a few hours to more than 100 hours, depending on how large the thermal inertia of the room is. Table 1 shows how many percent of the temperature difference that is left, after 1, 2, 3 and 4 timeconstants have elapsed. After 3 timeconstants have elapsed, 95 % of the temperature difference is vanished, which can be a practical limit for the stationary level. For example : if the temperature difference is ΔTa = - 10°C and the timeconstant is 20 hours, the temperature would have rised 9.5 °C of possible 10°C after 60 hours.

Table 1 : Shows how much of the temperature difference (ΔTa) is left after 1, 2, 3 and 4 timeconstants (τ)

Elapsed time after number of timeconstants	1	2	3	4
% of temperature difference left	36.8	13.5	5.0	1.8

3.2 Stationary term

The stationary term is given by :

$$T_\infty = \frac{(H_{ext} + H_{vent})\overline{T}_e + \overline{q}}{H_{ext} + H_{vent}} \qquad (°C) \qquad (13)$$

As can be seen from (27) the stationary temperature is independent of the thermal capacity of the room. Furthermore we see that large mean heat gain ($\overline{q}$), and high mean external temperature ($\overline{T}_e$) leads to high stationary temperature (T_∞).

3.3 Periodic term

The periodic term :

$$\hat{T}\cos\left[\omega\left(t - \tau_{lag} - t_{max}\right)\right] \qquad (°C) \qquad (14)$$

gives raise to periodic temperature oscillation. Equation (28) gives periodic oscillation with temperature amplitude $\hat{T}$, and a time-lag (τ_{lag}) . That is, the maximum temperature occur the time τ_{lag} after the time t_{max} , which is the time for maximum heat gain (and external temperature). Both $\hat{T}$ and τ_{lag} depends strongly on the thermal inertia of the room. Rooms with large thermal inertia gives small temperature amplitudes (as expected), and an increase in the time-lag. $\hat{T}$ is also depending on the heat gain amplitude and external temperature amplitude.

3.4 Maximum possible room temperature

The maximum possible temperature in the room occur when diurnal stationary condition is reached, together with daily maximum temperature amplitude :

$$T_{max} = \overline{T}_{\infty} + \hat{T} \qquad (°C) \qquad (15)$$

This maximum possible temperaterature is likely to occur in rooms with small thermal inertia (small heat capacity and large heat loss). For rooms with large thermal inertia (large heat capacity and small heat loss) stationary condition is seldom reached before the heat wave is over (or has been reduced). In offices or other rooms with a weekly 5 day occupation, the simulation period is often set to 5 days. In these rooms the maximum possible temperature (T_{max}) is unlikely to occur (also see the accompanying paper, part 2).

4.0 Conclusion

- We have presented a model which simulates temperature and cooling loads in rooms with diurnal variation of the heat gain
- The model is an alternative to use of advanced simulation tools, in an early stage of the thermal design phase of a room
- In rooms with small thermal inertia (small heat capacity and large heat loss) diurnal stationary condition ia reached fast (2 - 4 days) (see part 2)
- In rooms with large thermal inertia (large heat capasity and small heat loss) diurnal stationary conditions are seldom reached during a normal heat wave(5- 10 days), or during a normal 5 days working week (see part 2)
- Large heat capasity reduce the daily temperature variation to a large extent, and reduce the cooling demand (caused by high peaks in the heat gain e.g. solar gain)
- This implicate that use of heavy building structure can reduce temperature problems and mechanical cooling demand summertime, provided that the temperature is allowed to fluctuate

References

\1\ Børresen B.A., "Room temperature variation and cooling loads. A simplified calculation method, Tempo", Energy Conservation in the built Environment, CIB. Copenhagen 1979

\2\ Zill G.Z.; "A first Course in Differential equations", 5th ed., PWS-KENT 1992

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Appendix : Mathematical derivation

Equation (6) is a first order nonhomogeneous differential equation which can be solved with a variety of methods, such as ; Undetermined coeffisients Laplace transform, D-operator method, integrating factor method and substitution methods.

In all cases the general solution to (6) is the sum of the complementary solution, related to the left hand side (homogeneous part) , and the particular solution related to the right hand side (nonhomogeneous part).

Complementary solution

The left hand side in equation (6), which is called the homogeneous part, can easily be separated and integrated :

$$\frac{dT_{i,C}}{dt}+\frac{T_{,Ci}}{\tau}=0 \Leftrightarrow \frac{dT_{i,C}}{T_{i,C}}=-\frac{dt}{\tau} \Leftrightarrow T_{i,C}(t)=C_1 e^{-\frac{t}{\tau}} \quad , \; C_1=\text{constant} \tag{A.1}$$

Particular solution

The particular solution can be found by using the method of undetermined coefficients. We then assume a particular solution in the same algebraic form as the right hand side of equation (6) :

$$T_{i,P}=A+B\cos\omega\left(t-t_{\max}\right)+C\sin\omega\left(t-t_{\max}\right) \tag{A.2}$$

Differentiating (A.2) and substituting into (6) and collecting coefficients, we find A, B and C to be :

$$A=\beta\tau=T_\infty \quad ; \quad B=\frac{\alpha\delta}{\alpha^2+\omega^2} \quad ; \quad C=\frac{\omega\delta}{\alpha^2+\omega^2} \tag{A.3}$$

General solution

The general solution is then given by the sum of the complementary and the particular solution :

$$T_i(t)=C_1 e^{-\frac{t}{\tau}}+T_\infty+\frac{\delta}{\tau^{-2}+\omega^2}\left(\omega\sin\left[\omega\left(t-t_{\max}\right)\right]+\frac{1}{\tau}\cos\left[\omega\left(t-t_{\max}\right)\right]\right) \tag{A.4}$$

(A.4) can be written more conveniently by a trigonometric identity[2] as :

$$T_i(t)=C_1 e^{-\alpha t}+T_\infty+\hat{T}\cos\left[\omega\left(t-\tau_{lag}-t_{\max}\right)\right]$$

(A.5)

$$\tau_{lag}=\frac{\arctan(\tau\omega)}{\omega} \quad ; \quad \hat{T}=\frac{\delta}{\sqrt{\tau^{-2}+\omega^2}}$$

The constant C_1 can be determined by the initial condition $T_i(t=0) = T(0)$:

$$C_1=\Delta T_a=T_a(0)-\overline{T}_{a,\infty}-\hat{T}_a\cos\left[\omega\left(t_{\max}-\tau_{lag}\right)\right] \tag{A.6}$$

---

[2] We have that : $A\sin w+B\cos w=\sqrt{A^2+B^2}\cos(w-\phi) \quad ; \quad \tan\phi=\frac{A}{B}$

Thermal analysis of rooms with diurnal periodic heat gain, ThermSim. Part 2 : Practical use and comparison

Synopsis

Temperature and cooling demand in a room summertime are influenced by numerous factors, like internal gains, ventilation, solar gain, behaviour of occupants, thermal inertia of the room and outdoor conditions (climate).

The thermal environment and cooling demand summertime are often analysed using detailed computer programs, which take into account the factors mentioned above (among others). Often the overview, transparency and some of the physical insight is lost using these advanced computer programs.

In a predesign phase of a project it is preferable to do simple calculations of the thermal behaviour of a room. These simple calculations often gives more physical insight and overview than using computer programs. Simple calculations also gives a quality assurance of later computer analysis of the room.

This is part 2 of two related papers concerning a simplified method for thermal design of rooms, called *ThermSim*. Part 1 (the accompanying paper) is concerned with derivation and interpretation of the model.

This paper is concerned with practical guidance in choosing appropriate input to the model. Comparison to the advanced simulation program BRIS is also presented.

The model shows good agreement with computer analysis when the model assumptions is fulfilled.

List of symbols

Symbol	Description	Unit
A_{fac}	Facade area	m²
A_{floor}	Floor area	m²
A_{win}	Area for whole window construction (including frame)	m²
C_{air}	Heat capacity of air (can be set to 0.34 Wh/m³K)	Wh/m³K
F_{sh}	Effective total shading factor	-
L''	Normalized mechanical or natural air flow rate	m³/hm²
n	Air infiltration in ACH	1/h
n_{per}	Occupation time for persons	h
$n_{l\&a}$	Operation time for lighting and appliances	h
q_{per}	Heat gain from persons	W
$q_{l\&a}$	Heat gain from lighting and appliances	W
q''_{sol}	Solar intensity through a vertical pane	W/m²
Q''_{sol}	Daily sum of solar gain through a vertical pane	Wh/m²
$\overline{T}_e$	Mean daily external temperature	°C

$\hat{T}_e$	Daily amplitude external temperature	°C
ΔT_{fan}	Temperature rise over the supply fan	K
U_{fac}	U-value facade construction	W/m²K
U_{win}	U-value window construction	W/m²K
V	Air volume	m³

1 Introduction

This is part 2 of two related papers concerning a simplified method for thermal design of rooms. Part 1 (the accompanying paper) presents a method/model, called *ThermSim*, that can be used in the thermal design of rooms.

This paper is concerned with practical guidance in choosing appropriate input to the model. In addition this paper gives examples in the use of the model, and comparison to advanced computer simulation.

2 Guidance in choosing appropriate input data

This sections gives guidance in selecting appropriate input to the model. Most of the input can be "normalized" by dividing the value with the floor area. E.g. internal loads in Watt can be normalized into Watt per m² floor area (W/m²), which gives a much smaller range for the value. With this approach it is possible to make tables where normalized input values can be picked from, making the calculation process quick, and reducing the chance for calculation errors.

Values presented in the tables below are related to "Scandinavian" building standards, and may not be representative for other countries and other climates. It should however be easy to modify the tables to other standards and climates.

To exemplify how values in the tables have been determined, an office room is used as a case study throughout the section.

2.1 Normalized specific external loss

The specific loss to the external is comprised of window losses, facade losses and loss due to infiltration. In modern (Scandinavian) buildings the external walls are well insulated, and heat loss through windows are dominating. In less airtight buildings (older buildings) might infiltration have some impact on the external loss.

Given an office 3 x 4 m (12 m²) large, with a facade area of 9 m² (incl. window) and a window area of 2 m². U-value for the facade construction is 0.25 W/m²K (15 cm mineral wool) and the U-value for the window is 2 W/m²K. Infiltration is estimated to 0.3 ACH (Ceiling height is 3 m). Normalized specific external loss is then given by :

$$H''_{ext} = \frac{U_{fac}A_{fac} + U_{win}A_{win} + C_{air}nV}{A_{floor}} = \frac{(9-2)\cdot 0.25 + 2\cdot 2 + 0.34\cdot 0.3\cdot 36}{12} = 0.785\ W/m^2K$$

This figure is quite typical in office rooms, it normally lies between 0.4 W/m^2K and 1.2 W/m^2K. Table 1 presents typical values of normalized specific external loss, as a function of normalized windows- and facade loss, and infiltration (in ACH).

Table 1 : Specific external loss

Wind&Facad/ Infiltration	None	0.2W/m²K Low	0.5 W/m²K Medium	1.0 W/m²K High	3.0 W/m²K Very high
n = 0.1 ACH (Low)	0.1	0.3	0.6	1.1	3.1
n = 0.2 ACH	0.2	0.4	0.7	1.2	3.2
n = 0.3 ACH (Med)	0.3	0.5	0.8	1.3	3.3
n = 0.5 ACH	0.5	0.7	1.0	1.5	3.5
n = 0.7 ACH	0.7	0.9	1.2	1.7	3.7
n = 1.0 ACH (High)	1.0	1.2	1.5	3	4.0
n = 1.3 ACH	1.3	1.5	1.8	2.3	4.3

2.2 Normalized total specific loss

The total specific loss is the sum of the specific external loss and the ventilation loss. Air flow rate is often given in m^3/h per m^2 floor area (normalized air flow), which is convenient here.

The room in subsection 2.1 is ventilated (balanced mechanical vent.) with 10 m^3/hm^2 (120 m^3/h). The total specific loss is the given by :

$$H''_{tot} = H''_{ext} + C_{air} L''_{vent} = 0.785 + 0.34 \cdot 10 = 4.185\, W/m^2K$$

Table 2 gives normalized total specific loss as a function of ventilation rate and normalized specific external loss.

Table 2 : Normalized total specific loss

External loss/ Ventilation rate	None	0.5W/m²K Low	1.0 W/m²K Medium	2.0 W/m²K High	4.0 W/m²K Very high
0 m³/hm²	0	0.5	1.0	2.0	4.0
3 m³/hm² (Low)	1.0	1.5	2.0	3.0	5.0
5 m³/hm²	1.7	2.2	2.7	3.7	5.7
8 m³/hm² (Medium)	2.7	3.2	3.7	4.7	6.7
11 m³/hm²	3.7	4.2	4.7	5.7	7.7
15 m³/hm² (High)	5.0	5.5	6.0	7.0	9.0

2.3 Normalized specific heat capacity, timeconstant and time-lag

The effective heat capacity of the room can be treated in the same manner as the specific losses. The effective heat capacity of a building construction exposed to a 24 hours cycle temperature variation, can be limited to the inside 10 cm of the construction, or inside the insulating layer. If heavy material as concrete or brick is covered with insulating materials (i.e. carpet or lowered ceiling), the accumulating layer is reduced considerably. These "rules" gives specific (per m^2) heat capacity of : ~ 50 Wh/m^2K for a massive concrete wall, ~ 35 Wh/m^2K for a massive brick wall, ~ 4 Wh/m^2K for a insulated composite wall with gypsum board or wood panelling, ~15- 25 Wh/m^2K for a concrete slab covered with carpet or lowered ceiling.

Given the room in section 2.1 with concrete floors covered with carpet, mineral wool lowered ceiling (beneath concrete construction) and brick walls in facade and partition walls. The normalized heat capacity of the room can be calculated to :

$$C''_a = \frac{C_{air}V + \sum C''_a A}{A_{floor}} = \frac{0.34 \cdot 36 + 12 \cdot 20 + 12 \cdot 20 + (9-2) \cdot 35 + (3+4+4) \cdot 3 \cdot 35}{12} = 158\ W/m^2K$$

With the normalized heat capacity and normalized total specific loss, the timeconstant and time-lag can be readily calculated :

$$\tau = \frac{C''_a}{H''_{tot}} \qquad \tau_{lag} = \frac{\arctan[\tau\omega]}{\omega}$$

Table 3 gives timeconstant and time-lag values as a function of normalized heat capacity and total specific loss.

Table 3 :Timeconstant/time-lag

Total specific loss/ Normalized heat capacity	1.0 W/m²K Low	3.0 W/m²K	5.0 W/m²K Medium	7.0 W/m²K	9.0 W/m²K High
20 Wh/m²K (Very light)	20/5.3	7/4.1	4/3.1	3/2.5	2/1.8
40 Wh/m²K (Light)	40/5.6	13/4.9	8/4.3	6/3.8	4/3.1
80 Wh/m²K (medium)	80/5.8	27/5.5	16/5.1	11/4.7	9/4.5
140 Wh/m²K (Heavy)	140/5.9	47/5.7	28/5.5	20/5.3	16/5.1
260 Wh/m²K (Very heavy)	260/5.9	87	52/5.7	37/5.6	29/5.5

Example : Total specific loss : 3.0 W/m²K and specific heat capacity : 80, gives a timeconstant of 27 hours and a time-lag equal to 5 hours and 30 minutes.

2.4 Normalized internal load and solar gain

Heat gain from persons, light and appliances is often normalized with the floor area. In addition to the maximum instantaneous heat gain (to determine the amplitude heat gain), we have to estimate the diurnal mean heat gain. If balanced mechanical ventilation is used, we also have to estimate the heat gain from the supply fans.

The room in subsection 2.1 is occupied by one person (gain : 100 W) 8 hours a day. Lighting (120 W) and a computer (50 W) gives a mean heat gain of 170 W, and both are operated 8 hours a day. The supply fan rise the supply air flow 1 Kelvin (the fans are operated 24 hours a day). The normalized heat gain amplitude related to the internal load is then given by :

$$\hat{q}_{\text{int}} = \frac{q_{per} + q_{l\&a}}{2 \cdot A_{floor}} = \frac{100 + 170}{2 \cdot 12} = 11.25\ W/m^2$$

The normalized mean heat gain related to the internal load becomes :

$$\bar{q}_{\text{int}} = \frac{q_{per}n_{per} + q_{l\&a}n_{l\&a}}{24 \cdot A_{floor}} + C_{air}L''_{vent}\Delta T_{fan} = \frac{100 \cdot 8 + 170 \cdot 8}{24 \cdot 12} + 0.34 \cdot 10 \cdot 1 = 10.9\ W/m^2$$

Table 4 gives normalized amplitude heat gain and mean heat gain related to internal loads. It is given as a function of persons per 10 m² floor area (a normal office room) and the normalized gain from lighting and appliances. Heat gain from supply fan is included in the figures (air flow 10 m³/hm² and temperature rise 1 K). Operation of lighting and appliances is assumed to be 10 hours, and effective occupation time is set to 6 hours.

Table 4 : Normalized heat gain amplitude/mean heat gain

Lighting&Applianc./ Person density	5 W/m² Low	7 W/m²	10 W/m² Normal	15 W/m²	25 W/m² High
0.5 pers/10 m² (low)	5/7	6/8	8/9	10/11	15/15
1 pers/10 m² (office)	8/8	9/9	10/10	13/12	18/16
1.5 pers/10 m²	10/9	11/10	13/11	15/13	20/18
2 pers/10 m²	13/10	14/11	15/13	18/15	23/19
3 pers/10 m² (meet.room)	18/13	19/14	20/15	23/17	28/21
5 pers/10 m² (high)	28/18	29/19	30/20	33/22	38/26

Example : Normal lighting and appliances (10 W/m²) and 2 persons per 10 m², gives a amplitude heat gain of 15 W/m² and a mean heat gain of 13 W/m²

2.5 Specific solar gain

According to ,\1\, the maximum solar intensity through a vertical pane on a clear day can be approximated to 700 W/m². This figure can be used for facades facing East through South to West. Daily sum for the same facades can be approximated to 4700 Wh/m².

Solar shading in form of venetian blinds, curtains and building extensions can reduce the solar gain considerably. The shading effect for these shading devices is often taken as a constant shading factor. Typical values are 0.1 - 0.25 for external venetian blinds, 0.3 - 0.7 for inside venetian blinds and 0.5 - 0.8 for curtains. These values are for two pane windows with no coating, and has to be adjusted if low emessivity coating, reflective coating, absorbing glass or more panes are used. Shading from building extensions and nearby vegetation or buildings has to be estimated from case to case.

In the room from subsection 2.1, the facade is facing south, and there is inside venetian blinds with a shading factor of 0.5. The normalized amplitude gain can be estimated to :

$$\hat{q}_{sol} = \frac{q''_{sol} A_{win} F_{sh}}{2 \cdot A_{floor}} = \frac{700 \cdot 2 \cdot 0.5}{2 \cdot 12} = 29\ W/m^2$$

The normalized mean solar gain can be estimated to :

$$\bar{q}_{sol} = \frac{Q''_{sol} A_{win} F_{sh}}{24 \cdot A_{floor}} = \frac{4700 \cdot 2 \cdot 0.5}{24 \cdot 12} = 16\ W/m^2$$

Table 5 gives normalized solar gain (amplitude and mean) as a function of window area per m² floor area and total shading factor. Values are valid for facades facing east to west.

Table 5 : Normalized solar gain amplitude/mean heat gain (W/m²)

Window area pr m² / Total shading factor	0.05 m²/m² Low	0.10 m²/m²	0.2 m²/m² Normal	0.3 m²/m²	0.5 m²/m² High
0.85	15/8	30/16	60/33	89/48	149/80
0.75	13/7	26/14	52/29	79/42	131/70
0.55	10/5	19/10	39/22	58/30	96/50
0.40	7/4	14/8	28/16	42/24	70/40
0.25	4/2	9/5	18/10	26/15	44/25
0.10	2/1	4/2	7/4	11/6	18/10

Example : With normal window area (0.2 m²/m²) and a total shading factor of 0.25, gives amplitude solar gain of 18 W/m² and a mean heat gain of 10 W/m²

Total heat gain (daily mean and amplitude) is the sum of the internal gain and solar gain.

2.6 Climatic data

In addition to the maximum solar intensity and daily solar gain treated in the previous subsection, we need to estimate the mean external temperature and its daily variation (amplitude). We also have to estimate the time for maximum heat gain (and external temperature).

The mean external temperature is normally found in meteorological journals. E.g. the highest five day mean temperature for the location in question could be used. This has to be evaluated against the use of the room from case to case.

The external temperature amplitude in Scandinavian climate varies between 5 - 7°C. If accurate information is not available a value of 6 °C can be used. If maximum external temperature isn't corresponding with the maximum heat gain, the temperature amplitude can be reduced a bit (0.5 - 2 °C).

If solar gain is dominating (compared to internal gains), which is one of the main assumptions in the model, the time for maximum heat gain is determined by the facade/window orientation. With daylight saving time (in Oslo) maximum solar gain is occurring : between 12.00 and 13.00 for south facades, between 9.00 and 10.00 for east facades, and between 17.00 and 18.00 for west facades. For other countries adjustment for time zone, longitude and daylight saving time has to be done.

3 Case study; comparison

A room which has been used in validation analysis of a computer program (TeknoSim), will be used as case study here, and compared to results from the widely used simulation program BRIS ,\2\. The room has width , depth and height equal to : 3.6 m x 4.2 m x 2.7 m (A_{floor} = 15.12 m², V = 40.82 m³) . The room has one facade, facing west, with one window (A_{win} = 1.92 m², U_{win} = 2.0 W/m²K). Infiltration is 0.2 ACH, and the room is ventilated continuously with 72 m³/h (4.8 m³/hm²). The room is occupied with one person (9 hours a day), and heat gain from lighting and computer is 270 W (9 hours a day). The supply fan rises the supply temperature 1 °C.

Two different building constructions has been simulated : one heavy room with concrete floor, ceiling and external wall; and one light room with insulated composite construction covered with gypsum boards or particle boards. Partition walls are insulated composite walls with gypsum board in both cases.

Calculation

Normalized total specific loss is calculated to : H'_{tot} = 2.18 W/m²K (both cases). Normalized heat capacity for the two cases are calculated to : $C''_{a,h}$ = 31.2 Wh/m²K (light room) and $C''_{a,h}$ = 140.6 Wh/m²K (heavy room). Timeconstant and time-lag for the light- and heavy room then become : τ_{light} = 14.3 hours, $\tau_{lag,light}$ = 5 hours (light) and τ_{heavy} = 64.6 hours, $\tau_{lag,heavy}$ = 5.8 hours (heavy). Normalized mean heat gain and heat gain amplitude is respectively : $\overline{q}$ = 20.4 W/m² and $\hat{q}$ = 31.0 W/m². Mean external temperature and temperature amplitude are respectively : $\overline{T}_e$ = 22 °C and $\hat{T}_e$ = 6.5 °C . This gives a stationary temperature of : T_∞ = 31.7 °C (both cases), and a temperature amplitude for the light and heavy room of respectively : $\hat{T}_e$ = 1.2 °C (light) and $\hat{T}_e$ = 5.3 °C (heavy). Transient temperature differences are calculated to : ΔT_{light} = -10.8 °C (light) and ΔT_{heavy} = -9.9 °C (heavy). Both cases are simulated for a period of five days.

Simulated operative temperature the fifth simulation day in BRIS is shown in figure 1 (light room) and figure 2 (heavy room), and is compared to calculation with ThermSim (fifth day). The operative temperature in BRIS is used for the comparison, since the calculated temperature in ThermSim is a "merged" room-, surface- and structure temperature.

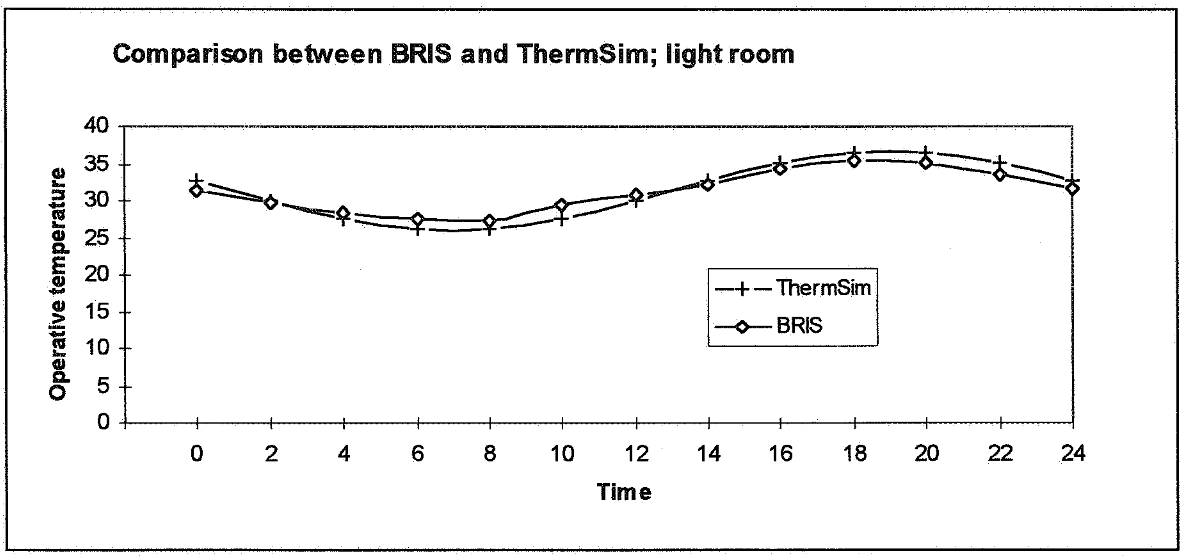

Figure 1 : Comparison between simulation in the advanced computer program BRIS and calculation with ThermSim, light room

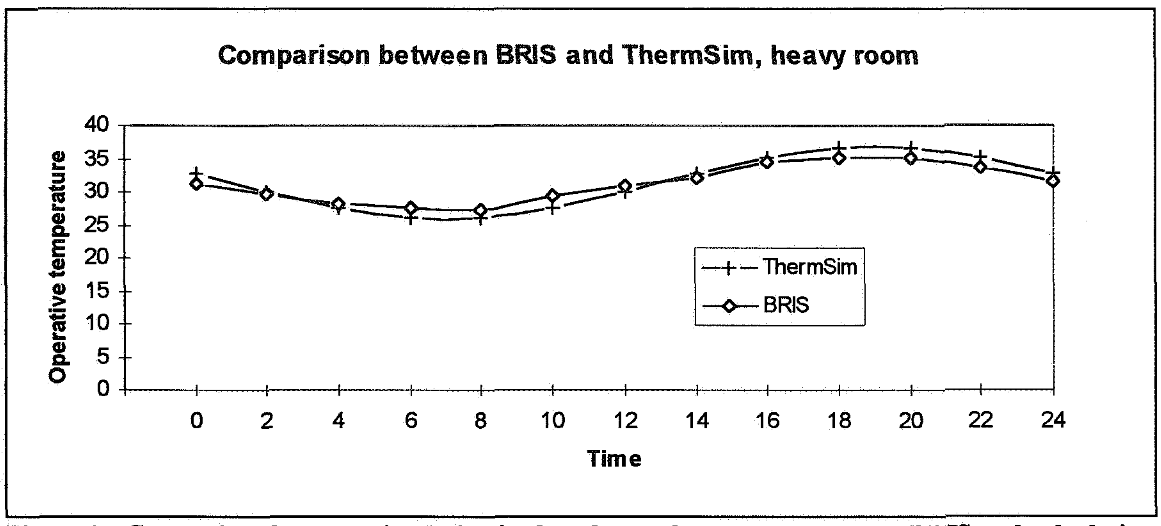

Figure 2 : Comparison between simulation in the advanced computer program BRIS and calculation with ThermSim, heavy room

5 Discussion and conclusions

- Temperature variation simulated with BRIS and calculated with ThermSim is similar, for both the light and heavy room
- Maximum temperature is somewhat higher calculated with ThermSim compared to BRIS (1.2 °C for light room and 1.7°C for heavy room).
- Maximum temperature occur a bit later in ThermSim than in BRIS ((1-2 hours in both rooms). This implicate that the calculated time-lag in ThermSim overestimate the "real" time-lag.
- Diurnal stationary conditions is reached after 5 days in the light room, but far from reached the heavy room (both with BRIS and ThermSim).
- Comparison between ThermSim and BRIS shows good agreement, and ThermSim should therefore be well suited for thermal analysis in a predesign phase of a project
- The simulations and calculations shows that large heat capacity reduce the daily temperature variation to a large extent, and prevent stationary condition being reached during a normal heat wave or a normal working week
- ThermSim is very well fitted for sensitivity analysis, because it only deals with the most important parameters affecting the thermal conditions in the room

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VENTILATION AND COOLING

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Natural Ventilation of the Contact Theatre

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1. Synopsis

This paper will describe the design and development of the natural ventilation system of the new Contact Theatre Complex Manchester, UK designed by A.Goldrick of Short Ford Associates.

The ventilation design is based on a stack dominant system using an 'H-Pot' chimney configuration. The paper describes the development of the ventilation design of both the studio theatre and main auditorium ventilation systems. These have been developed with the aid of wind tunnel and CFD testing in order to produce a strategy and design relatively insensitive to wind direction, yet providing sufficient ventilation to overcome the high heat gains expected from an audience and stage lighting. The potential for conflicts between wind and buoyancy forces have been reduced through the location and positioning of inlets and through the sizing and design of the stack and H-pot devices.

2. Introduction

Design for the natural ventilation of a building is seldom straightforward, this is particularly true for a building with a high predicted occupancy and large internal gains such as a theatre. This paper describes the evolution of the ventilation strategy of a theatre where the architect was in a position to commission both wind tunnel testing and CFD prediction at an early stage in the design proceedure and was thus easily able to encorporate necessary design alterations to optimise performance.

3. Initial design

The original contact theatre building situated in the grounds of Manchester University consisted of a 450 seat auditorium, dressing rooms and a small workshop. The rehearsal rooms and offices were situated on a different site at some distance away. The brief was to move the existing accommodation from the old site and incorporate it, together with a new studio theatre, with the existing auditorium and workshop building.

Some 32 architectural practices were shortlisted for the project and 7 were interviewed. In order to be elegible for an Arts Council Lottery award the proposed building would have to be shown to be environmentally sustainable. Short Ford and Associates are a practise with a history of environmentally responsive architecture with natural ventilation and passive cooling being incorporated in most of their past schemes.
The Contact Theatre committee were impressed with these aspects of the practice's portfolio and awarded them the contract.

4. Building form

The initial design provided natural ventilation for both the elevated studio theatre and the original auditorium, the foyer and rehearsal rooms. The studio theatre and auditorium were served by a number of large chimney stacks, four terminating in H-pot devices above the studio theatre and six for the auditorium. (Fig. 1).

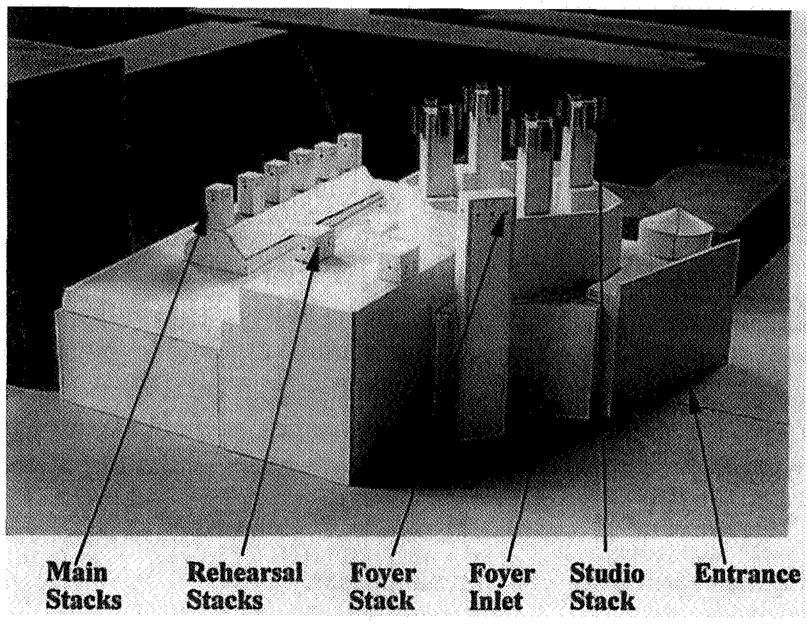

Figure 1 wind tunnel model showing main elements of building

5. Wind Tunnel Testing

The aim of the wind tunnel testing was to determine the mean wind pressure coefficients at the proposed locations of the ventilation inlet and outlets in order to assess their performance for a range of wind directions.

Wind pressure measurements were conducted on a 1:150 scale physical model of the theatre building in a boundary layer wind tunnel. The model of the theatre building itself was approximately 30x30x15 cm in size and was placed at the centre of 2 metre diameter turntable. This allowed surrounding buildings to be included in the site model up to a radius of 150 metres (Fig. 2).

Mean wind pressure coefficient measurements were made at 57 points over the building surface for 16 wind directions equally spaced around the compass. The pressure taps were located in positions identified as inlet or exhaust points, particular care was taken to monitor the conditions created inside the H-pot terminations of the chimney stacks.

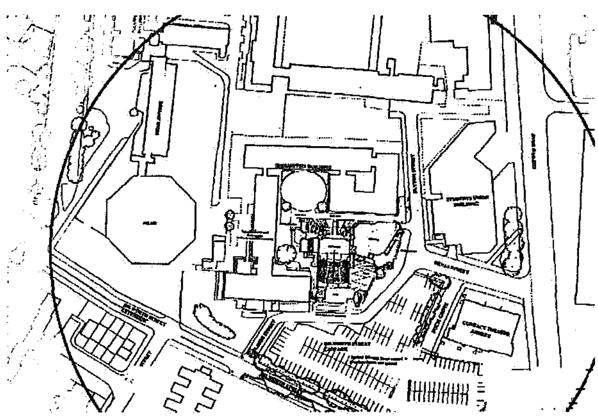

Figure 2. Indicating site model boudary

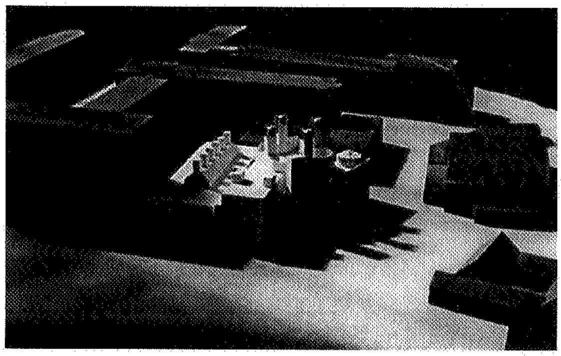

Figure 3 Wind tunnel model showing proximity of neighbouring buildings

The design intent is to naturally ventilate the main spaces. The aim of the ventilation design is to maximise the wind induced pressure differences between the inlet and outlet points for each zone while not opposing the natural buoyancy stack effects due to the heat gains from the occupants and lighting, that is, the pressure at the outlet point should be lower than that at the inlet (a positive gradient). We call the difference in measured pressure coefficient between inlets and outlets the Cp gradient. A positive Cp gradient will lead to wind flow assisting natural buoyancy driven ventilation, whereas a negative gradient would oppose it.

The wind tunnel testing was carried out in two main phases.

5.1 Phase 1

5.1.1 Main Auditorium.

For the first phase the main auditorium stacks were to exhaust through surface mounted grilles situated on their North and South (opposing) faces (Fig. 1). The inlet position was at low level on the west side of the auditorium.

Wind Direction	Cp Gradient Inlets->Outlets
North	+0.1
NE	0.0
East	*-0.1*
SE	*-0.1*
South	*-0.1*
SW	*-0.1*
West	*-0.1*
NW	+0.6

Table 1 Average Cp Gradient for Main Theatre Zone

The initial results (table 1) indicated that for all wind directions apart from NW there is no robust positive pressure gradient and there is likely to be conflict between buoyancy forces and wind effects. Examining the data more closely it became clear that the stacks above the auditorium were not providing an overall robust negative pressure, the vents on opposite sides of the stacks sometimes cancelling one another. Further, the close proximity of the taller buildings to the North and West of the theatre generated low pressures over the entire building envelope.

5.1.2 Studio Theatre

The terminating devices for the Studio Theatre consisted of four large stacks terminating in H-pot devices. The inlets were situated low down on the North facade of the building.

Results here indicated a robust positive pressure gradient for all wind directions, the data showing that the H-pot terminated stacks provide a strong low or negative pressure irrespective of the wind direction.

Wind Direction	Cp Gradient Inlets->Outlets
North	+1.1
NE	+0.6
East	+0.3
SE	+0.3
South	+0.3
SW	+0.2
West	+0.6
NW	+0.5

Table 2 Average Cp Gradient for Studio Theatre Zone

5.1.3 Foyer

The ventilation device for the Foyer space comprised of one large extract stack with surface mounted grilles, the inlet points being positioned at the foyer entrance.

Generally there are positive pressure gradients for most wind directions. The North-westerly sector indicated marginal conditions in terms of the Cp gradient, but this was considered acceptable, given the relative frequency of that wind direction.

Wind Direction	Cp Gradient Inlets->Outlets
North	+0.1
NE	+0.4
East	+0.2
SE	+0.2
South	+0.2
SW	*0.0*
West	*0.0*
NW	*-0.2*

Table 3 Average Cp Gradient for Foyer Zone

5.1.4 Rehearsal Rooms

Two short stacks with surface mounted grilles served as extract devices for the rehearsal rooms with inlet points sited low on the south and east facades.

Pressure gradients were generally positive with marginal conditions in the south and westerly quadrant.

Wind Direction	Cp Gradient Inlets->Outlets
North	+0.2
NE	+0.3
East	+0.4
SE	+0.3
South	0.0
SW	*-0.1*
West	0.0
NW	*-0.1*

Table 4 Average Cp Gradient for Rehearsal Rooms

5.1.5 Summary

The wind tunnel tests of the first phase of the Contact Theatre scheme indicate that the stack driven ventilation for the Studio Theatre should be successful with wind pressure assisting natural buoyancy effects.

The Foyer and Rehearsal rooms should experience no wind versus stack effect conflict and that for most directions, bar the westerly quadrant, the wind would provide positive reinforcement.

The results indicate that the Main Auditorium could exhibit a conflict between wind and buoyancy pressures, which occur for most wind directions.

5.2 Phase II - Design Evolution

As a result of the phase I testing, substantial alterations were made to the natural ventilation design of the Theatre, in particular in the size, location and termination of the stack outlets.

The modifications made to the existing phase I model were as follows (figure 4) :-

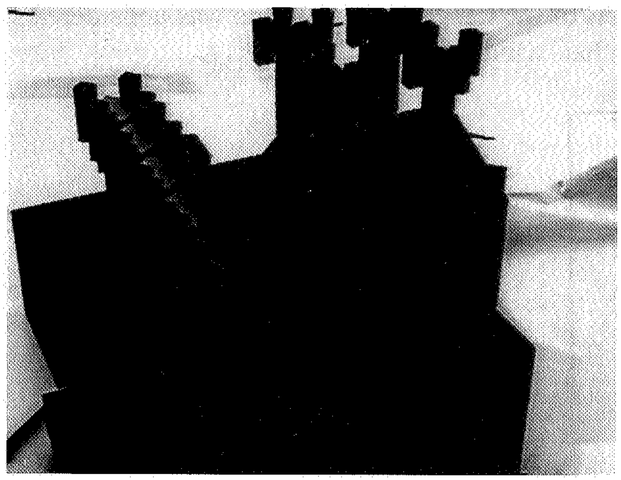

Figure 4 Final design configuration

- the extract stacks for the auditorium were significantly modified with four H-pots per stack, with the pots mounted on the chimney corners. This increase in pot number was necessary to handle the predicted volume of extracted air.

-the extract stacks on the studio theatre were also increased in size and height. These were modelled to determine their effect when upwind of the auditorium.

-additional inlets for the main auditorium were positioned on the north side of the building.

5.2.1 Main Auditorium

The results for the Main Auditorium are presented in Table 4 The pressure gradients measured for eight main wind directions. In the table below "old inlet" refers to the proposed inlet position of the phase I tests.

Wind Direction	Final Model Cp Gradient New Inlets->Outlets	Final Model Cp Gradient Old Inlets->Outlets	Old Model Cp Gradient Old Inlets->Outlets
NE	+0.2	+0.4	+0.1
East	+0.1	+0.1	0.0
SE	+0.0	+0.2	-0.1
South	+0.1	+0.2	-0.1
SW	*-0.1*	*-0.1*	-0.1
West	+0.2	+0.1	-0.1
NW	+0.5	+0.6	-0.1
North	+0.6	+0.9	+0.6

Table 4 Average Cp Gradient for Main Theatre Zone

The alterations for the Phase II design indicated that for most wind directions the auditorium stacks were effective with positive pressure gradients. The south westerly wind direction providing the only problem producing a slightly negative pressure gradient. This was considered acceptable for the final design of the building.

6. CFD Analysis

Wind tunnel testing indicated that for the auditorium, in its final configuration most wind directions would provide positive pressure gradients between the inlet and outlet points. This indicates that natural buoyancy driven stack ventilation could be considered the worst case condition and any wind would assist ventilation. A Computaional Fluid Dynamics (CFD) analysis was therefore carried out on the main theatre auditorium and studio theatre to assess the ventilation performance resulting from the buoyancy affects due to heat gains from occupancy and lighting.

The CFD model used was DFS-AIR (code produced by Design Flow Solutions). The code was used to simulate internal air movement and heat distribution for the Main Auditorium and Studio Theatre for summer conditions. The simulation predicted the air flow distribution from its entry point into the undercroft areas of both spaces, the cooling effect of the air in contact with the undercroft thermal mass, the air entering the performance spaces, the heating effects of the occupants and lights and the air being exhausted through he stacks.

Figures 5 and 6 show typical results ofr air movement in the undercroft of the Studio Theatre (Figure 5) and the in the Main Auditorium (Figure 6).

The CFD analysis predicted satisfactory conditions for internal air temperature and air speed and ventilation rates or the theatre under normal patterns of use.

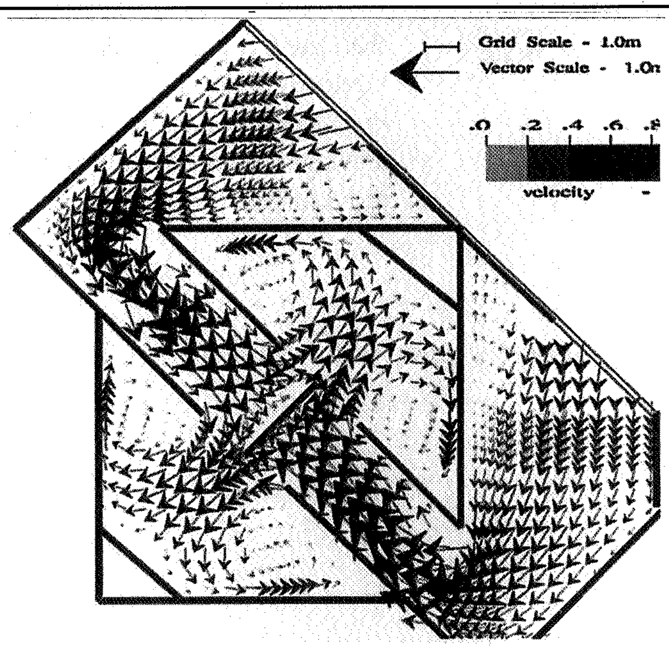

Figure 5 Air flow predicted for the undercroft of the Studio Theatre.

7. Discussion and Conclusions

In determining the natural ventilation design for an inner-city building with high internal gains, the balancing of wind and buoyancy forces becomes of great importance.

Inlet positions for natural ventilation systems invariably have to be sited low on the building envelope where they are more prone to the turbulence created by surrounding buildings and are therefore more difficult to fine tune. Extract devices tend to be situated high on the building in less turbulent air and are therefore easier to modify effectively.

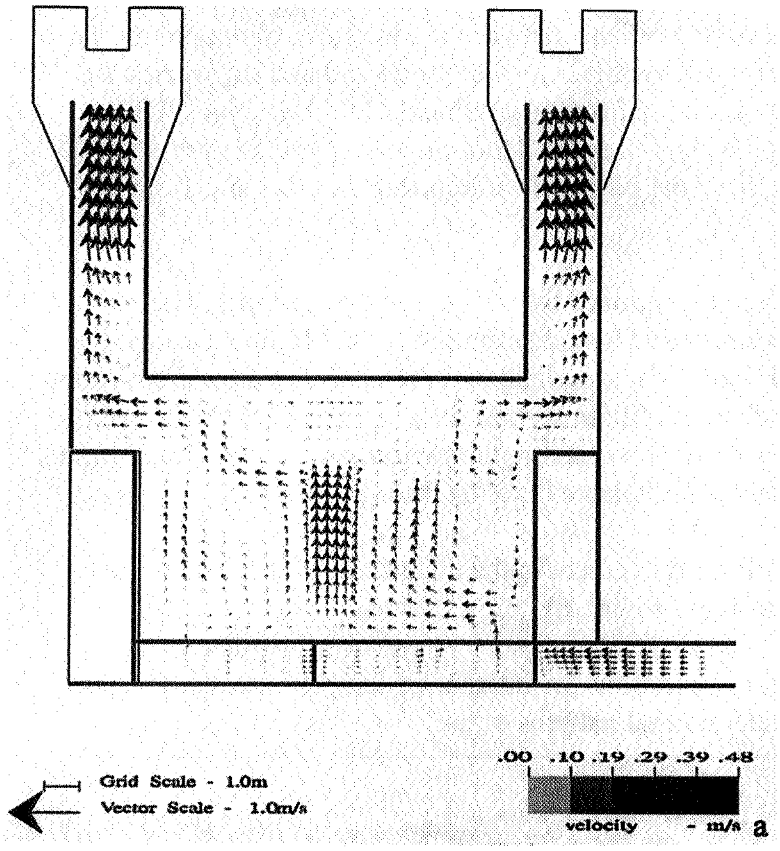

Figure 6 Air flow in the Main Auditorium

Wind tunnel and CFD analysis of early stage design concepts is possible and can make a great contribution to the design development process. It is possible to moderate the detrimental effects on ventilation outlets caused by the turbulent "wakes" encountered downwind of tall buildings by careful design and testing of outlet devices.

CFD modelling can be used with more confidence in conjunction with wind tunnel testing. The prediction of stack effect as being the "worst case" condition for example can be validated by wind tunnel testing. When wind ventilation forces are marginal then the Cp values can be used as input to CFD models to investigate the combined wind and stack effect on internal air movement.

8. Acknowledgements

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Title: SWEDISH DUCT LEAKAGE STATUS

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Swedish Duct Leakage Status

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The first fifty years

With very few exceptions all buildings in Sweden and their installations are performed in accordance with the quality requirements specified in the 'AMA-system'. AMA standing for 'Allmän Material- och Arbetsbeskrivning', i.e. General Requirements for Material and Workmanship, and was launched already 1950 and has since been extended to cover all areas within the building and infrastructure fields.

The first ventilation duct tightness requirements were stipulated in the HVAC part of the 1960 edition of AMA. In the 1966 edition the requirements where strengthened to comprise two 'tightness norms', A and B. The fulfilment of these requirements were to be spot checked by the contractor but supervised by the owner. Each spot test should cover systems with a minimum of 10 m^2 duct surface.

In the 1972 edition the requirements where transformed into two 'tightness classes' A and B (in accordance with the A and B classes used today), with B being three times tighter than A. A was then the standard requirement for the 'complete duct system in the air handling installation', i.e. including 'dampers, filters, humidifiers and heat exchangers'.

The following advises were given to the designer:

> 'The choice of tightness class should be based on economic decisions. An analysis of duct systems according to economic factors thus shows that:
> - duct systems with an air flow less than 3 m^3/s should be built according to class A,
> - systems with no treatment of the air or only air heating should also be built according to class A but, if the system was operated for more than 8 hours/day, class B should be considered,
> - all extract air systems should be built according to class A,
> - the tightness requirements should be increased parallel to the degree of used air treatment. Thus cooling, high class filters, humidification and dehumidification of the air could motivate the use of class B.'

The 1983 AMA edition, still in use today, was influenced by the rising energy costs after the first oil crisis 1972/73 and the higher tightness being possible by the use of spiral wound round ducts. Tightness class C was now added, being three times tighter than B and thus nine times tighter than A. The standard AMA requirement thus became class C for round ductwork larger than 50 m^2 duct surface, class B for smaller round duct systems as well as for all rectangular ductwork and A for visible supply and extract ducts within the ventilated room. The introduction of class C was first met with resistance from the contractors who considered class C to be too high a demand. After the first year in use it was however found that the AMA requirements were easier to fulfil than first thought, the opposition died and the demands were accepted.

The status today

AMA requires all ventilation and air conditioning systems to be carefully commissioned. The procedures include:

- measurement and adjustment of all supplied or extracted air flows in the building. The result should be within ±15% including the measurement error. The result is to be presented on standard AMA protocols,
- the duct system leakage has to be controlled, normally by the contractor as part of the contract. This is done as a spot check where the parts to be checked are chosen by the owner's consultant. For round duct systems 10% and for rectangular ducts 20% of the total duct surface has to be controlled. In case the system is found to be more leaky than required the tested system shall be tightened and another equally sized part of the system shall be controlled in the same manner. Should also this part be found to leak more than accepted the complete installation has to be leak tested and tightened until the requirements are fulfilled.

Time for tightness class D?
Today we are working with the next AMA generation, AMA 98, to be published during 1988. We now think that it is time to raise the tightness requirements once more by introducing tightness class D as the standard requirement for larger spiroduct systems. There are several reasons for this step:

- the today available technology permits it.
  The quality of modern round duct system available on the market today, with double rubber seals connections, are that tight when installed properly. (One of the main AMA principles is to raise the quality requirements if and when the technology makes it possible),

- the duct systems installed today will probably be used for at least the next twenty years. An eventual higher investment cost for a higher quality duct system should be considered on an Life Cycle Cost (LCC) basis and not just on the first cost,

- the energy prices will be higher and the demand for low energy use will increase during this period - the green house effect and the ozone layer are among the factors to be considered,

- air leaking out of or into duct systems between the fans and the ventilated rooms in the building has to be compensated by higher fan air flows (which is an old AMA requirement - the air flow considered necessary to ventilate a room should also be delivered there. Another AMA requirement is that the air flows are to be adjusted, controlled and listed for each room),

- the leakage air is of little or no use to the building but leads not only to higher operating costs but often also to disadvantages such as noise and draught complaints.

Compulsory authority requirements
The AMA requirements are made valid when they are referred to in the contract between the owner and the contractor - which is practically always the case in Sweden.

The concern about an increasing part of the Swedish population becoming allergic and asthmatic, often due to 'sick buildings' and inadequate dilution of indoor emissions by inferior ventilation systems, lead the Swedish Parliament and Government to decide on compulsory inspection of ventilation systems (Government Bill 1990/91:145 and Ordinance, SFS 1991:1273, about the performance checks on ventilation systems).

The rules for the inspection were issued by the Swedish National Board of Housing, Building and Planning (General Guidelines 1992:3 'Checking the performance of ventilation systems' based on BFS 1992:15 'Regulations about performance checks on ventilation systems').

The intervals between the checks depend on how sensitive the building occupants are and how complicated the ventilation system is. They vary from 2 years inspection intervals for day-care centres, schools, health care centres, etc. up to 9 years for one- and two-dwelling houses with balanced ventilation.

The performance checks are to be carried out by an inspector who is authorised either nationally by the Swedish National Board of Housing, Building and Planning or locally by the municipal committee(s) responsible for planning and building matters.

The inspector qualifications differ between these different buildings and systems and whether the authorisation is local or national.