Enhancing the Design and Operation of Passive Cooling Concepts

Monitoring and Data Analysis in Four Low-Energy Office Buildings with Night Ventilation

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Abstract

As the energy demand of new office buildings has been reduced during the last years, there is a rising interest for heating and cooling systems based on renewable sources of energy. While conventional office buildings require complex technical devices which associate a high energy demand, the reduced energy demand of low-energy office buildings can be supplied mostly by natural heat sources in winter or heat sinks during the summer.

Low-energy office buildings can successfully use passive cooling concepts in order to achieve a comfortable indoor climate during the summer, without a high energy demand for air-conditioning. However, not every office building can be passively cooled. Cooling and building concept has to complement one another. Passive cooling concepts for the Mid European summer minimise the solar and internal heat gains and utilise the building’s thermal inertia. The reduced and smoothed heat gains are counterbalanced only by natural heat sinks, i.e. earth-to-air heat exchanger, concrete slab cooling or night ventilation.

This thesis focuses on the heat dissipation by night ventilation. However, the whole passive cooling system has to be analysed in order to ascertain the night ventilation efficiency. For this investigation, the realised passive cooling systems are analysed in four low-energy office buildings. Using the wide data pool from a two years’ monitoring, the night ventilation concepts are analysed comparatively. Short-term measurements in each building complement the information and are used for the thermodynamic models.

Different techniques for data analysis are applied to the monitored data. A building simulation provides an explicit thermodynamic model of the building. As the simulation deals with a vast number of input parameters and variables, the determination of separate effects is difficult. A parametric model reduces the influences to a concise number of parameters and variables. Proceeding, the energy balance summarises the results to the bare minimum. However, the energy balance can only be drawn, if all input parameters and influences are known.

The realised passive cooling systems are analysed based on the same thermodynamic model. Besides the separate conclusions from the data evaluation in each building, universal conclusions concerning the night ventilation efficiency are drawn: Night ventilation operates well, if the passive cooling concept is accurately designed and sufficiently operated. However, a passive cooling system cannot ensure that the room temperatures meet the comfort criteria everytime.

This thesis aims to enhance the design and operation of passive cooling concepts. A comprehensive data analysis shows not only the night ventilation potential in realised buildings, but also how design tools should be used in order to accurately design and how to operate the passive cooling concept with night ventilation. The use of statistically distributed input parameters enhance greatly the significance and clarity of simulation results. A precise realisation and an accurately implementation of the concept are essential for the proper operation of night ventilation and the functionality of the passive cooling system.
Acknowledgement

This thesis presents research results from my work at the Solar Building Group of the Fraunhofer Institute for Solar Energy Systems ISE in Freiburg, which lasted from April 2001 to March 2004. The Fraunhofer ISE provided me with excellent research facilities and equipment.

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## Nomenclature

### Latin letters

<table>
<thead>
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<th>Symbol</th>
<th>Meaning</th>
<th>Unit</th>
<th>Symbol</th>
<th>Meaning</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>surface / opening area</td>
<td>m²</td>
<td>a</td>
<td>thermal diffusivity</td>
<td>m²/s</td>
</tr>
<tr>
<td>ACH</td>
<td>air change rate</td>
<td>h⁻¹</td>
<td>a</td>
<td>amplitude (in Fourier series)</td>
<td>variable</td>
</tr>
<tr>
<td>Bi</td>
<td>Biot number</td>
<td>–</td>
<td>a, b</td>
<td>parameter in Logit function</td>
<td>–</td>
</tr>
<tr>
<td>C</td>
<td>heat storage capacity</td>
<td>Wh/(m³ K)</td>
<td>c p</td>
<td>heat storage capacity</td>
<td>J/(m³ K)</td>
</tr>
<tr>
<td>COP</td>
<td>coefficient of performance</td>
<td>–</td>
<td>d</td>
<td>thickness (of solid)</td>
<td>m</td>
</tr>
<tr>
<td>Dh</td>
<td>degree hours</td>
<td>Kh</td>
<td>d</td>
<td>diameter (of pipe)</td>
<td>m</td>
</tr>
<tr>
<td>G</td>
<td>specific heat gain</td>
<td>W/m²</td>
<td>f</td>
<td>any user-defined function</td>
<td>variable</td>
</tr>
<tr>
<td>Gr</td>
<td>Graßhoff Number</td>
<td>–</td>
<td>g</td>
<td>gravitation constant</td>
<td>9.81 m/s²</td>
</tr>
<tr>
<td>H</td>
<td>heat loss factor</td>
<td>W/(m² K)</td>
<td>g s</td>
<td>solar gain coefficient</td>
<td>–</td>
</tr>
<tr>
<td>I</td>
<td>solar radiation</td>
<td>W/m²</td>
<td>g²</td>
<td>effective solar aperture</td>
<td>m²</td>
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<tr>
<td>L</td>
<td>characteristic length</td>
<td>m</td>
<td>h</td>
<td>heat transfer coefficient</td>
<td>W/(m² K)</td>
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<tr>
<td>NTU</td>
<td>number of transfer units</td>
<td>–</td>
<td>h</td>
<td>height</td>
<td>m</td>
</tr>
<tr>
<td>P</td>
<td>electric performance</td>
<td>W</td>
<td>l</td>
<td>length (of pipe)</td>
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<tr>
<td>Q</td>
<td>specific internal heat gain</td>
<td>W/m²</td>
<td>p</td>
<td>probability</td>
<td>%</td>
</tr>
<tr>
<td>Q</td>
<td>thermal energy</td>
<td>J</td>
<td>q</td>
<td>specific (thermal) energy</td>
<td>J/m²</td>
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<td>R</td>
<td>hydraulic resistance</td>
<td>m³/(h Pa²)</td>
<td>s</td>
<td>error</td>
<td>%</td>
</tr>
<tr>
<td>Rₜ</td>
<td>temperature ratio</td>
<td>–</td>
<td>t</td>
<td>time</td>
<td>h</td>
</tr>
<tr>
<td>S</td>
<td>slope (in linear function)</td>
<td>–</td>
<td>t₀</td>
<td>cycle period</td>
<td>h</td>
</tr>
<tr>
<td>T</td>
<td>temperature</td>
<td>°C</td>
<td>x</td>
<td>position</td>
<td>m</td>
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<tr>
<td>U</td>
<td>heat transmission coefficient</td>
<td>W/(m² K)</td>
<td>x,y</td>
<td>partial length</td>
<td>m</td>
</tr>
<tr>
<td>V</td>
<td>volume</td>
<td>m³</td>
<td>z</td>
<td>standardised dynamic thickness</td>
<td>–</td>
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</table>

Time variables are identified by a dot, e.g.  $\dot{Q}$ for heat flow or  $\dot{V}$ for volume air flow.

### Greek letters

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Meaning</th>
<th>Unit</th>
<th>Symbol</th>
<th>Meaning</th>
<th>Unit</th>
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<tbody>
<tr>
<td>Δ</td>
<td>amplitude</td>
<td>variable</td>
<td>λ</td>
<td>heat conductivity</td>
<td>W/(m K)</td>
</tr>
<tr>
<td>Δ</td>
<td>(mean) bias</td>
<td>variable</td>
<td>ρ</td>
<td>density</td>
<td>kg/m³</td>
</tr>
<tr>
<td>Δp</td>
<td>pressure difference</td>
<td>Pa</td>
<td>σ</td>
<td>(thermal) penetration depth</td>
<td>m</td>
</tr>
<tr>
<td>ΔT</td>
<td>temperature difference</td>
<td>K</td>
<td>σ</td>
<td>standard deviation</td>
<td>variable</td>
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<tr>
<td>θ</td>
<td>heat exchanger efficiency</td>
<td>–</td>
<td>τ</td>
<td>time constant</td>
<td>h</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>ω</td>
<td>angular frequency</td>
<td>h⁻¹</td>
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### Subscripts and Superscripts

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<th>Meaning</th>
<th>Symbol</th>
<th>Meaning</th>
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</thead>
<tbody>
<tr>
<td>AT</td>
<td>ambient air temperature</td>
<td>m</td>
<td>mean</td>
</tr>
<tr>
<td>a</td>
<td>adjusted (mechanical ventilation)</td>
<td>m</td>
<td>mechanical (ventilation)</td>
</tr>
<tr>
<td>a</td>
<td>ambient / outdoor air</td>
<td>NV</td>
<td>night ventilation</td>
</tr>
<tr>
<td>c</td>
<td>corrected (g-value)</td>
<td>n,i</td>
<td>counter</td>
</tr>
<tr>
<td>c</td>
<td>comfort (temperature)</td>
<td>n-a</td>
<td>not adjusted (ventilation)</td>
</tr>
<tr>
<td>cD</td>
<td>discharge factor (ventilation)</td>
<td>op</td>
<td>operation (time)</td>
</tr>
<tr>
<td>c_p</td>
<td>pressure coefficient (wind)</td>
<td>out</td>
<td>outlet (temperature)</td>
</tr>
<tr>
<td>EAHX</td>
<td>earth-to-air heat exchanger</td>
<td>s</td>
<td>time step</td>
</tr>
<tr>
<td>eff</td>
<td>effective</td>
<td>sf</td>
<td>surface</td>
</tr>
<tr>
<td>elt</td>
<td>electric</td>
<td>T</td>
<td>temperature</td>
</tr>
<tr>
<td>HVAC</td>
<td>heating, ventilation and air-conditioning</td>
<td>t</td>
<td>at this time</td>
</tr>
<tr>
<td>h</td>
<td>hottest month</td>
<td>th</td>
<td>thermal</td>
</tr>
<tr>
<td>i</td>
<td>indoor</td>
<td>w</td>
<td>wind</td>
</tr>
<tr>
<td>i,n</td>
<td>counter</td>
<td>`</td>
<td>adiabatic</td>
</tr>
<tr>
<td>in</td>
<td>inlet (temperature)</td>
<td>~</td>
<td>mean or expected value</td>
</tr>
<tr>
<td>in</td>
<td>heat flow into solid</td>
<td>∞</td>
<td>infinite</td>
</tr>
<tr>
<td>MCS</td>
<td>Monte Carlo-simulation</td>
<td>∞</td>
<td>ideal</td>
</tr>
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</table>
1 Introduction

Ever since humans have moved into shelters, they have looked for ways to improve indoor conditions. This aspiration for comfort has not changed, but the design and the retrofit of office buildings should take not only the indoor comfort but also energy efficiency, sustainability and economics into consideration. Low-energy office buildings with passive cooling realise these demands.

In the perfect office building, occupants find high workplace quality (thermal, visual and acoustic comfort). The building is energy efficient and sustainable (e.g. building materials and layout) and the investment and operation costs are low.

How can we synthesise these design goals? In recent years, new building materials and the return to lean architecture made energy and cost efficient buildings with a high workplace comfort possible. One promising approach to reduce the energy demand and to improve the thermal comfort is passive cooling by night ventilation.

This thesis deals with monitoring of passive cooling in low-energy office buildings in order to evaluate the thermal comfort in passively cooled buildings and to analyse night ventilation efficiency.

Why do we analyse the thermal performance of buildings? The principles of passive cooling are well-known and available in comprehensive handbooks, cf. Santamouris [1-1]. Furthermore, architects and building engineers have access to various design tools. (And this thesis cannot enlarge the physical models or the existing design tools.) However, restraints and flaws may only be recognised and understood from an analysis of realised energy concepts which is applied to a wide range of buildings and time periods.

In this context, the thesis aims at the improvement of the design and operation of passive cooling concepts.
1.1 Energy and buildings

Buildings provide shelter and aim to create adequate working / living conditions for their inhabitants / occupants. Apart from these functional aspects, buildings serve as a means of cultural identification and social representation. To satisfy all these diverse expectations, financial, material and energy resources are required to construct and maintain a building. Sustainable design of new and retrofitted office buildings should consider both the demand for energy efficiency and comfort and the protection of resources.

In addition to the scientific and economic evaluation of low-energy office buildings, the European directive on the energy performance of buildings [1-2] sets new, politically coordinated benchmarks for low-energy office buildings taking the overall energy demand for heating, hot water, cooling, ventilation, lighting and technical services into account.

Previous research clearly indicates that the primary energy demand of new office buildings is dominated by the electricity demand for lighting, ventilation and air conditioning, cf. Weber [1-3]. Some recent trends can be derived from Nilson’s study on Swedish office buildings [1-4]: Office buildings have experienced a continuous decrease in heating demand following the development of stricter building codes stimulating better thermal insulation. However, this increased efficiency of the thermal envelope has been accompanied by higher electricity demands for technical services and office equipment. These data can be transferred to the Central European building stock. An evaluation of German low-energy office buildings coordinated by Voss [1-9] verifies this trend, shows that the energy consumption for heating and electricity varies in a wide range, and underlines the experience that expenses for energy saving and use of renewable energy affect the overall costs less than the general standard of equipment.

Fig. 1-1 shows the total primary energy demand and its composition for different types of office buildings. Since Knissel’s study [1-6] is mainly based on planning values from the national building standards and the two guidelines SIA 380 [1-7] and VDI 3807 [1-8] but not on monitored data, it outlines the building’s energy demand of representative building stocks rather than on realised buildings.

![Fig. 1-1](image1.png)

**Fig. 1-1:** Composition of the energy demand in office buildings on the basis of simulation studies. All buildings analysed in this thesis are low-energy office buildings.
Lean buildings pursue a concept which aims at a low energy demand with reduced technical devices [1-9]. The solar heat gains in winter and the natural heat sinks in summer are utilised efficiently by a suitable building and energy concept. Fig. 1-2 shows that passive cooling is an important feature of these “lean buildings”.

Fig. 1-2: Conventional buildings need either active heating or cooling during the whole year. Lean buildings have a clearly reduced electric and heating energy demand, a shorter heating period and do not need an air-conditioning system.

1.2 Passive cooling by night ventilation

Passive cooling covers all natural techniques of heat dissipation, overheat protection and related building design techniques, providing thermal comfort without the use of optional mechanical cooling. Passive cooling includes several well-known methods, enhanced with new technology advances and better understanding of the physical process involved. The potential efficiency of both traditional and modern applications can be optimised through a more comprehensive coupling and coordination of the available techniques and systems with architectural design of the building in its environment.

Passive cooling refers to techniques used to prevent and modulate heat gains, including the use of natural heat sinks. The following design techniques are used for protection from or prevention of heat gains:

- microclimate and site design,
- solar control,
- building form and layout,
- thermal insulation,
- user behaviour and occupancy patterns and
- internal gain control.

Heat gain modulation is achieved by proper use of the building’s thermal inertia mass. The heat is absorbed during the day and will be returned at a later time by night ventilation or dissipated by another natural heat sink:

- cool night air (night ventilation),
- ground cooling (earth-to-air heat exchanger),
- evaporative cooling,
radiative cooling (use of ground water) during the day or  
concrete slab cooling (use of chilling tower) during the night.

In moderate climates, one promising approach to reduce the energy demand for cooling without reducing comfort is passive cooling by night ventilation. Fig. 1-3 outlines the principle of night ventilation, which makes use of the cool night air and the building’s thermal inertia mass.

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**Fig. 1-3:** Principle of night ventilation: Energy balance.

However, only a certain amount of heat can be dissipated by night ventilation due to the available nocturnal temperature level, the limited time for night ventilation, the practically realisable air change rate, and the effectively utilisable heat storage capacity. As a result, the heat gains and the heat storage capacity must be balanced with the amount of heat that can be dissipated. Since this balance must be maintained, night ventilation affects many aspects of the design, construction and operation of an office building: night ventilation concepts can be realised only by an integrated design approach.

### 1.3 Integrated design

The idea of *integrated design* summarises many different project management concepts. In any case, integrated design takes ideas from different sectors (e.g. economics, ecology and architecture) into account. In the context of interdisciplinary cooperation, the design of lean buildings requires integrated solutions since the renunciation of technical devices requires suitable architectural measures. In the following, "integrated design" describes the design and project coordination concerning night ventilation.

The design of passive cooling affects the architectural design of the building, the building physical properties and the decision for / against a specific energy supply system. Starting from the given preconditions, Fig. 1-4 outlines the creative and the iterative optimisation of building concepts.
boundary conditions of the site
climate, use / equipment / working time, requirement on indoor air climate, layout, orientation, external shading, urban / rural site

- buildings envelope
  1. heating: minimising thermal losses, maximising solar gains (with sufficient heat storage)
  2. cooling: minimising thermal losses, maximising solar gains (and high thermal inertia)

- realisation of this strategy in construction elements
  1. specification of construction materials
  2. specification of windows and facade system
  3. implementation of variable sun-shading devices

- design of an appropriate energy supply system
  1. heating system
  2. air-conditioning / (passive) cooling concept

Fig. 1-4: Design strategy to minimise the energy demand for cooling and to choose the cooling concept according to Nytsch-Geusen [1-10].

As the design of passive cooling of buildings needs to consider many factors, energy and comfort targets need to be specified clearly in advance. Only after that specification should the design process go into detail. If contrary intentions (e.g. the economical realisation of a particular technical device) cannot be reconciled with a specific principle (e.g. the total energy demand for technical services), the sophisticated design process may result in dead ends.

Fig. 1-5 shows exemplarily some players involved in the design of passive cooling systems. The architect is responsible for the coordination of the investor’s request and the diverse requirements placed upon the building and the cooling concept.

Passive cooling by night ventilation is an elementary concept: Open windows – cool room. However, the design procedure is challenging and connected with several restraints.
1.4 **Restraints to night ventilation**

Some typical restraints and barriers to night ventilation can be derived from realised (and in the end not realised) projects:

- **Technical barriers**: Acoustics (i.e. open spaces for natural ventilation), fire protection (e.g. in atria), safety aspects (e.g. open windows during night), day lighting and sun-shading (i.e. electric and solar heat gains) or required space (e.g. large conduits for low pressure losses).

- **User’s doubts** concerning draft effects, dust, low room temperatures in the morning, high room temperatures in the afternoon or confidentiality (e.g. sound insulation).

- **Project management**: Lack of interaction in the design team, no stimulation of the fee structure for architects and engineers (e.g. payment according to the investment cost) or economical aspects (e.g. high specific investment costs in relation to the supplied cooling performance).

- **Validation of design tools**: The main difficulties with design tools and simulation are the small heat flow densities in summer (e.g. transmission through the external wall due to the small indoor – outdoor temperature difference or heat transfer at internal surfaces due to small air – surface temperature differences) and the changing heat flows (e.g. summer day / night). As most simulation programs and design tools have been validated for the heating period and the associated higher heat flows, the existing programs need to be reviewed especially for the summer situation.

- **Use of design tools**: Though some design tools have become available in recent years, there is a lack of knowledge concerning the input parameters. For example, the user behaviour may turn the balance for/against the efficiency of the passive cooling system for a specific building:
  - How is the user behaviour taken into account by design tools?
  - How can a designer model realistic user behaviour during the summer in the simulation program?

- **Quality assurance**: Probably, the main restraint to night ventilation systems is the lack of quality standards and, therewith, of quality control procedures.
Taking these restraints into consideration, Fig. 1-6 outlines how project management may account for the crucial points in the design of passive cooling concepts.

**Fig. 1-6:** Design of passive cooling by night ventilation: Some crucial points at each stage of development. (HOAI is the German fee structure for architects and engineers [1-12].)

## 1.5 Hypotheses

Experience shows, that well-known passive cooling techniques can be used successfully to provide a comfortable indoor climate (cf. Proceedings of the International Symposium "Passive Cooling in Buildings", 1995 [1-13]). However, passive cooling systems are not applied throughout the building stock where it may be possible:

- On the one hand, the theory of passive cooling with the physical models involved is available and widely used. Furthermore, many separate measurements have been carried out in recent years in order to analyse the night ventilation potential in passive cooling systems. Several examples of contemporary office buildings show that new technologies allow for a high comfort with passive cooling.

- On the other hand, the main restraints to passive cooling by night ventilation are (1) uncertainties concerning the design tools and the user behaviour and (2) the lack of quality control tools and experience with passively cooled buildings.

*What is the contribution of this thesis to a wider use of passive cooling concepts?* Data from a two years-monitoring in four buildings are analysed with a universal method. The analysis enhances the certainty concerning the design process and the operation of passively cooled buildings by universal conclusions.

Starting from these insights and in anticipation of an extensive review, the following hypotheses are formulated. These hypotheses accompany the thesis and are discussed in the concluding Chapter 10:

- **(h1) Feasibility:** Low-energy office buildings with passive cooling can achieve a comfortable indoor climate when architecture and building physics are harmonised with the building services.

- **(h2) Calculability:** The thermal building performance can be calculated accurately. This is the precondition for the design of passive cooling concepts.

- **(h3) Availability:** All passive cooling concepts are known and can be applied to office buildings in ordinary use.
Reproducibility: An essential engineer’s and scientist’s law states, that the same cause results in the same effect. Though it can be difficult to find this principle in the complex thermal performance of a building, the impact of heat gains, heat losses and the building’s thermal inertia on the thermal comfort can be consistently derived from field measurements.

Functionality: Passive cooling systems can be operated robustly.

1.6 Thesis outline

The main objective of this thesis is to overcome technical barriers and design restraints to passive cooling by night ventilation. Fig. 1-7 outlines the content of the thesis:

- In Part A, the methodology for data analysis used in this thesis is developed. An introduction into the underlying theory is given in Chapter 2 followed by a performance evaluation of earth-to-air heat exchangers in Chapter 3. Earth-to-air heat exchangers use the ground as a natural heat sink and exemplify, like night ventilation, air-driven passive cooling technique. The data analysis is very similar to the data analysis of night ventilation, since unknown or uncertain thermodynamic parameters are calculated by a parameter identification. As the thermal performance of earth-to-air heat exchangers are less complex than the energy balance of a building, this is a good example for the model-based data evaluation.

- In Part B, results from long-term monitoring campaigns and short-term measurements in four office buildings are presented. As each monitoring campaign aims at special aspects concerning the thermal performance of the building, the data evaluation differ from one project to the next. In Chapter 4 an extensive simulation study for the DB Netz AG building is carried out in order to estimate uncertainties in monitoring and the measurements. In Chapter 5 measured data from a comprehensive experiment at the Fraunhofer ISE building are discussed. Both the simulation and a parametric model are validated against the measured data. The modelling of heat storage capacity will be discussed in detail in Chapter 6: The data evaluation for the Pollmeier building is mainly based on the parametric model. In Chapter 7 the impact of user behaviour and air distribution on night ventilation efficiency will be further examined in the Lamparter building.

From these studies, validated simulation models (with knowledge concerning the accurate consideration of input parameters), a parametric model focusing on the fundamental parameters and detailed information on user behaviour, heat transfer, interzonal air change and air distribution in a room are available.

- In Part C, the previous findings are combined by statistical simulations and a cross-section analysis. A statistical approach concerning how to consider uncertainties and user behaviour in the design procedure is proposed in Chapter 8. In Chapter 9 the results are summarised by a universal method in terms of a cross-section analysis.
| Chap. 1 | Introduction |
| Chap. 2 | Basics and Methodology |
| Chap. 3 | Evaluation of Earth-to-Air Heat Exchangers |
|         | DB Netz AG, Fraunhofer ISE, Hans Lamparter GBR |
| Chap. 4 | Merging Short and Long-Term Measurements |
|         | DB Netz AG |
| Chap. 5 | Night Ventilation Experiments |
|         | Fraunhofer ISE |
| Chap. 6 | Data Evaluation with Simplified Models |
|         | Pollmeier Massivholz GmbH |
| Chap. 7 | Room Air Flow Distribution + User Behaviour |
|         | Hans Lamparter GBR |
| Chap. 8 | Statistical Simulation of User Behaviour |
|         | Fraunhofer ISE |
| Chap. 9 | Cross-Section Analysis |
| Chap. 10 | Conclusion |

Fig. 1-7: Thesis outline.

### 1.7 References


Part A

As this thesis is based on the data analysis in low-energy offices and focuses on an universally valid model for night ventilation, a consistent physical model and the methodology of model-based data analysis are introduced.

The design of passive cooling and night ventilation concepts is associated with many differing requirements: energy demand, thermal comfort or investment and operational costs. As passive cooling concepts have to take solar and internal heat gains into account, the architectural and utilisation concept of an office building is affected by its energy concept. Moreover, the building concept should follow the requirements defined by the passive cooling concept, i.e. facade system, energy saving office equipment, the building’s thermal inertia mass, the ventilation system or interzonal air flow through the building. In designing passive cooling systems, the modelling of night ventilation – the heat dissipation in the energy concept – is complex. These complex interrelations can be integrated by a model-based data analysis.

Fig. 2-1: Functionality and operation of night ventilation.
2.1 Literature review

A literature review concerning the modelling of night ventilation, its design and comfort criteria shows the state of the art and refers to the future work.

While this literature review ascertains the basic conditions for the evaluation of passive cooling systems, separate literature reviews are discussed where necessary: in Chapter 3 concerning the modelling of earth-to-air heat exchangers, in Chapter 5 concerning the design of night ventilation experiments, in Chapter 6 concerning the characterisation of meteorological data and in Chapter 7 concerning the calculation of heat transfer coefficients and the user behaviour.

2.1.1 Modelling night ventilation efficiency

Fig. 2-1 shows the operation of night ventilation: The building’s thermal inertia mass (in particular the ceiling) absorb heat during the day. During the night, cool air flows through the room and cools down the ceiling. Thus the air flow, the heat transfer and the heat storage have to be calculated in the modelling of night ventilation efficiency. Different physical-mathematical models are available for ventilation, heat transfer and heat storage:

- **Natural, hybrid and mechanical ventilation.** Due to the different driving forces (wind, buoyancy and fan-driven ventilation), the design of free and hybrid ventilation and the calculation of the interzonal air change is complex. Therefore, sophisticated design tools should be used in order to determine the air change rates.
  - Heiselberg (ed.) [2-37] introduces the principles of hybrid ventilation. This ventilation concept uses natural driving forces every time when possible and is supported by a mechanical ventilation system at times when necessary.
  - Feustel [2-31] presented a survey of air-flow models for multizone structures. Most air-flow network programs work with an approach which is based on hydraulic resistances: Aynsley [2-8] introduced a resistance approach for the analysis of natural ventilation air-flow networks. An extensive overview of the air flow through openings, ventilation, infiltration and interzonal air change was given by Allard [2-3].
  - Often the discharge coefficients for openings are unknown. Some measurements of discharge coefficients and conclusions on air flow through buildings were summarised by Flourentzou [2-33].
  - In urban areas, the environmental impact on passive cooling by ventilation has to be taken into account. A survey is given by Kolokotroni et al. [2-48].
  - In addition to the interzonal air-flow models, the air movement in naturally ventilated buildings should be taken into account. The air movement has been investigated by Awbi using CFD [2-7] and by Eftekhari using measurements [2-23].
  - The air-flow network model used by the building simulation program ESP-r [2-16], which is used for data analysis in this thesis, is based on Walton's model [2-72] and was implemented by Hensen [2-38].

- **Heat transfer.** As the night ventilation cools down the building, an accurate modelling of the convective heat transfer coefficient is essential for the simulation of night ventilation. A good survey on different models for building applications was given by Dascalaki et al. [2-17]: The standard deviation in different models for the
calculation of heat transfer coefficients is around 20 – 40 % (comparable to the uncertainty in modelling).

From the multitude of models, the data correlations by Alamdari and Hammond [2-2] and Khalifa and Marshall [2-47] will be used for calculating the night ventilation efficiency since these models result in the best agreement between measurement and simulation.

☐ **Heat storage.** The heat storage capacity of a room consists of the thermally utilisable heat storage capacity corresponding to all boundary surfaces and the furniture. It is dependent on (1) the thickness of each component, (2) its thermal properties, (3) the cycle period of the room air temperature (i.e. 24 hours) and (4) the heat transfer at the surface. The amplitude of the temperature oscillation diminishes with the depth into a component and with it the amplitude of the heat conduction in the solid. Starting from the heat flow calculation, the analytical solution deals with the thermal penetration depth, cf. Baehr and Stephan [2-9].

A good survey on the convective heat balance of a room and the energy stored in a concrete slab was given by Koschenz [2-50]. Hauser [2-36] verifies that a high heat storage capacity damps room temperature fluctuations in summer while the heat storage capacity influences the room temperature in winter marginally.

Carslaw and Jaeger [2-15] show how the thermally active heat storage capacity is calculated analytically. This method is similar to the numerical solution, which is specified by EN ISO 13786 [2-25], and results in the same heat storage capacity.

In the following, the heat storage capacity of a homogenous solid with one (ersatz) layer is calculated analytically according to this approach.1

In this thesis, various thermodynamic models and calculation methods are used: The modelling and measurement of *air flows in buildings* will be discussed in detail in Chapter 4. Measurements of air change rates and simplified models are also discussed in Chapter 5, 6 and 7. The impact of the *internal heat transfer coefficients* on the night ventilation efficiency is described in Chapter 4, 5 and 6 and will be closer looked at in Chapter 7. The *heat storage capacity* is taken into account by each data analysis in Chapter 4, 5 and 7 and is discussed in detail in Chapter 6.

### 2.1.2 Design criteria, guidelines and tools

As the night ventilation efficiency is limited (cf. Chapter 1.2), the daily heat gains should be reduced and the heat storage capacity of the room / building should be designed large enough. Different design tools are available to balance daily heat gains, nocturnal heat losses and the heat storage.

In designing passive cooling by night ventilation, a decision on the general ventilation concept and design goals has to be made. Different textbooks on air-conditioning (e.g. Rietschel [2-63]) and ventilation (e.g. Etheridge and Sandberg [2-29]) explain the basics of air-driven cooling. The following literature review gives a survey on available information:

☐ Each building service engineer chooses from a *vast number* of design handbooks, guidelines and tools: In addition to handbooks concerning conventional HVAC components and systems, there are guidelines (e.g. Pfafferott [2-62]), case studies (e.g. Zimmermann [2-76]) and design tools (see below) for the design of passive cooling.

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1 Building simulation programs solve Fourier’s heat conduction formula numerically.
In order to reduce the building energy consumption while maintaining high levels of thermal comfort, building research has been oriented towards the appropriate use of natural heat sinks:

- Santamouris et al. [2-1] provides comparative information regarding the performance of passive and hybrid cooling techniques involving the use of natural heat sinks.

- Kalogirou et al. [2-32] examined natural and controlled ventilation, solar shading, various types of glazing, orientation, shape of buildings and thermal mass using building simulation. Each measure provided a cooling load reduction between 10 and 30% for maintaining the test house at 25 °C. As the different measures are not correlated, the interaction between (minimised) heat gains and (optimised) heat losses is not evaluated.

Three textbooks go into detail concerning innovative ventilation concepts:

- Santamouris and Allard [2-4] give design guidelines for natural ventilation. In this thesis, the modelling of natural and hybrid ventilation is principally geared to the AIOLOS software which is presented in this book.

- According to Gids’ remarks [2-35], hybrid ventilation concepts use natural forces, i.e. wind and temperature difference as much as possible, and apply fans if natural forces cannot provide the required ventilation level. Heiselberg [2-37] provides a survey on the principles of hybrid ventilation. Hybrid ventilation should be designed as a two-mode system, which is controlled to minimise the energy consumption while maintaining acceptable indoor air quality and thermal comfort. The two modes refer to natural and mechanical driving forces.

- Liddament [2-52] summarises aspects on energy-efficient ventilation with the focus on mechanical ventilation.

Some planning tools for passive cooling / night ventilation have been established recently:

- Keller [2-45] presents an analytical method for the thermal design of buildings, which is based on characteristic building parameters. These parameters are derived from a complete mathematical solution of the energy balance of a room and is discussed in detail in Chapter 2.2.1.

- A similar model based on the non-linear coupling between thermal mass and natural ventilation in buildings has been presented by Yam et al. [2-73].

- The design tool LESOCOOL [2-64] merges an air-flow model with a heat transfer model. This simplified model for passive cooling can calculate the cooling potential, temperature evolution and air change rates for given heat gains and losses.

- A comparable program was introduced by Rousseau [2-65].

- The NatVent program [2-55] takes infiltration, ventilation and thermal into account. It serves as a pre-design tool that can be used early in the design process before explicit data about the building and the ventilation system are available.

The Swiss EMPA published a handbook on passive cooling [2-75]. This handbook summarises boundary conditions for passive cooling and criteria for the design of different passive cooling techniques. The minimum temperature difference between day and night is 5 K for night ventilation. The overall cooling load should not ex-
ceed 150 Wh/(m² d), if the temperature difference between day and night is lower than 10 K, and 250 Wh/(m² d), if the difference is higher than 10 K.

- For cooling with free ventilation, Zeidler [2-74] gives a similar limit for heat gains of 30 W/m² during the working hours, i.e. 240 Wh/(m² d), in an office with typical use. Higher cooling loads cause discomfort.

- The estimation of internal heat gains is uncertain since occupant patterns and equipment may change during the long operation time of an office building. Basic design parameters concerning the internal heat gains can be taken from Ref. [2-51] or from SIA 380 [2-67].

- Night ventilation should be designed according to national guidelines and standards: The German standard DIN 4108 [2-21] specifies the requirements for heat protection in summer. The guideline VDI 2078 [2-69] specifies the calculation procedure for the cooling load.

- The design of advanced night ventilation concepts requires often an extensive building simulation to calculate the thermal building performance and the temperature behaviour in the offices. Various building simulation programs are widely used to model the energy flows within buildings: Prominent examples are ESP-r [2-16], TRNSYS [2-11] and SMILE [2-10]. These building simulation programs can be used not only for the design of free and mechanical night ventilation, but also to evaluate other strategies to avoid over-heating in summer, e.g. optimised window-to-wall ratio, solar gain coefficient, shading devices, hybrid ventilation with an earth-to-air heat exchanger, minimised internal heat gains or different control strategies.

- Finally, an uncertainty analysis should complete the design process and should be taken into account during the modelling of night ventilation. Macdonald [2-53] showed the practical application of uncertainty analysis in building design.

Most current design tools and building simulation programs respond to “what happens if”-questions. However, in conceptual and preliminary design stages, consultants and HVAC-engineers need answers to a “what to do”-question. While the building simulation solves closed problems, pre-design tools should respond to an open problem. In this thesis, the data evaluation of monitored data aims to draw conclusions on “what happened when”-questions.

As the energy consumption for cooling and the temperature performance in summer depends on extensive correlations between heat gains, heat losses and heat storage, a universal method to calculate building performance indicators is required:

- Kolokotroni et al. [2-12] and [2-49] developed a simplified calculation tool with three output parameters: maximum dry resultant temperature during the occupied period, dry resultant temperature at the start of the occupied period and energy savings.

- Keller [2-45] used only three essential parameters to determine the thermal behaviour of a building. Using these parameters, an universally valid strategy for the building design can be deduced from functions in the frequency domain.

The data analysis in this thesis is mainly based on Keller’s model (Chapter 5, 6 and 7) and on building simulation using the ESP-r program (in particular Chapter 4 and 5). Though the design tools / calculation models are thermodynamically correct, the real use involves uncertainties which are discussed in detail in Chapter 8. Furthermore, the user behaviour should considered since the user behaviour may be the decisive factor whether a passive cooling system works or not. Statistical models for attendance, internal gains, sun protection and use of windows are discussed in detail in Chapter 7 and 8.
2.1.3 Comfort criteria

As night ventilation lowers the room temperature in order to improve the thermal comfort, the data analysis and the evaluation of night ventilation concepts should deal with comfort criteria.

Though humans can acclimatise to changing climate conditions, a comfort range can be specified clearly in which humans feel generally comfortable. However, the comfort range cannot be specified in close limits since various factors influence comfort, e.g.: room air and wall temperatures, clothing, sex, physical health and food, age, season of the year, kind of work, lighting, noise, smell, contact with the environment and cultural or social characteristics. Psychological elements have been discussed recently, too. Nevertheless, it is possible to define a comfort range in which most people feel thermally comfortable. Fanger [2-30] defined the thermal comfort dependent on:

- Clothing and level of activity.
- Air temperature and its local distribution, wall temperatures, air humidity and air movement.

Different comfort criteria and their limitations are discussed in the literature, cf. a special issue on thermal comfort [2-56]:

- Since the 1920’s hygienists have developed various calculation models for thermal comfort which take the four parameters (air and wall temperatures, air humidity and movement) into account. According to Fanger’s investigations, the thermal comfort is defined in DIN ISO 7730 [2-22] by the PMV (predicted mean vote) and the PPD (predicted percentage of dissatisfied) index. The ISO standard specifies a complex calculation procedure which is analysed for typical conditions in design handbooks to derive practical design guidelines (e.g. Recknagel-Sprenger [2-66]).

- An excellent introduction to thermal comfort standards is given by Olesen [2-58]. Olesen’s study shows how non-steady state thermal environments, local thermal discomfort and the occupants’ opportunity for control of their environment can be taken into account by a proposed new version of EN ISO 7730.

- The ProKlimA study [2-13] deals with the “Sick Building Syndrome” (disturbance of working efficiency, comfort and health concerning the indoor climate) in 8 office buildings with and 6 office buildings without air-conditioning with a total of 6,800 occupants. Concerning the thermal comfort, the study concludes that occupants may feel more comfortable in not air-conditioned buildings even if the monitored comfort is worse than in air-conditioned buildings. This can be attributed to the fact that occupants want to be in contact with and want to control individually their environment.

- The German directive on health and safety at work [2-5] demands that the room temperature should not exceed 26 °C but can exceed 26 °C exceptionally at high ambient air temperatures.

- The German standard DIN 1946 [2-20] defines limits for the operative room temperatures in mechanically ventilated buildings: The operative room temperature should lie between 22 and 25 °C up to an ambient temperature of 26 °C. Room temperatures between 20 and 26 °C are still considered to be acceptable. The comfortable room temperature is higher at ambient temperatures above 26 °C. Though this norm is specified and valid only for mechanically ventilated and air-conditioned buildings, it is also used for passively cooled buildings.

- The purpose of ASHRAE standard 55 [2-6] is “to specify the combinations of indoor space environment and factors that will produce thermal environmental conditions...
acceptable to 80% or more of the occupants within a space. Based on Fanger’s PMV model, indoor comfort temperatures $T_{i,c}$ are calculated as a function of the outdoor air temperature $T_a$:

- for HVAC buildings: $T_{i,c} = 22.6 + 0.037 \cdot T_a \,[^\circ C]$
- and for naturally ventilated buildings: $T_{i,c} = 21.6 + 0.12 \cdot T_a \,[^\circ C]$

(2-1)

Dear [2-18] analysed raw data compiled from field studies in 160 buildings with and without air-conditioning located on four continents in varied climatic zones. While the predicted comfort temperature (ASHRAE 55) corresponds to the observations in HVAC buildings, a clear deviation between predicted and observed comfort temperatures is noticed in naturally ventilated buildings:

$T_{i,c} = 17.8 + 0.31 \cdot T_{a,f} \,[^\circ C]$

(2-2)

Thermal comfort is not an exact concept and human responses with regard to comfort do not occur as a response to an exact temperature. Humphrey’s [2-42] adaptive model predicts the indoor comfort temperature $T_{i,c}$ from the mean monthly outdoor temperature $T_{a,m}$ and – for cooled buildings – the average daily maximum temperature of the hottest month $T_{a,h}$:

- for HVAC buildings: $T_{i,c} = 12.4 + 0.32 \cdot T_{a,h} \,[^\circ C] + 0.0065 \cdot T_{a,m} \,[^\circ C]^2$
- and for naturally ventilated buildings: $T_{i,c} = 11.9 + 0.53 \cdot T_{a,m} \,[^\circ C]$

(2-3)

The prediction for free-running / passively cooled buildings has a standard error of $1 ^\circ C$ and applies a range of $10 \, ^\circ C \leq T_{a,m} \leq 34 \, ^\circ C$.

Due to the lack of active cooling devices, passively cooled buildings cannot provide for a specific indoor temperature every time. According to Rouvel’s study [2-19], the room temperature should not exceed a given temperature limit for more than 10% of the working time. The temperature limit is dependent on a pre-defined climate region: 25 °C for regions with maximum monthly mean temperature $\leq 16.5 \, ^\circ C$ (e.g. Hamm, Germany, Chapter 4), 26 °C for regions with maximum monthly mean temperature $> 16.5 \, ^\circ C$ and $< 18 \, ^\circ C$ (e.g. Weilheim and Creuzburg, Germany, Chapter 6 and 7) and 27 °C for regions with maximum monthly mean temperature $\geq 18 \, ^\circ C$ (e.g. Freiburg, Germany, Chapter 5). As this criterion depends on the ambient temperature, the evaluation of measured data is site-specific. It can be noticed from an ongoing survey at a few office buildings [2-34], that these temperature limits are not restricted enough. Kempton’s surveys [2-46] in air-conditioned residential buildings from the late eighties affirm that room temperatures should not exceed 25 °C. Thus, Rouvel’s comfort criterion is used with a temperature limit of 25 °C independent of the climate region.

The cooling power of an air-conditioning system is designed for a maximum indoor air temperature of 22 °C according to the VDI 2078 guideline [2-69] for the crucial meteorological conditions: High ambient air temperature in July and low altitude of the sun in September. However, most air-conditioning systems are designed for a

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2 Though the analysis is based only on buildings that reached statistical significance in the derivation of their own neutral temperature, the regression analysis of both HVAC buildings and naturally ventilated buildings yield in a weak statistical significance ($R^2 \leq 70\%$). This uncertainty should be kept in mind in the context of data analysis in Parts B and C of this thesis.

3 The range of comfortable temperatures is very large: While the mean temperature in August of 16.7 °C for Hamm results in a comfort temperature of only 20.75 °C, the August temperature in August 2003 of 26.0 °C for Freiburg results in a comfort temperature of 25.1 °C.
maximum ambient air temperature of 26 °C according to Recknagel-Sprenger [2-66], such that the indoor temperature increases at higher outdoor temperatures. Though these technical guidelines are no comfort criteria in an actual sense, they are often used as those by HVAC designers and building owners.

Next to the minimum requirements to thermal insulation, minimum requirements to the heat protection in summer are specified by DIN 4108 [2-21]. If these requirements are considered in the design of a building without mechanical cooling, the requirements concerning the indoor climate should be satisfied. Admittedly, the room temperature may exceed the maximum room temperature specified by the other specifications during the ordinary building operation.

Next to these steady state comfort criteria, dynamical and local comfort criteria are discussed:

- In a survey of individual buildings [2-41], the predicted comfort votes differed markedly and systematically from the actual mean vote, both for naturally ventilated and for air-conditioned spaces. The authors demand that dynamical criteria should be considered by ISO 7730 [2-22] in order to improve the prediction of thermal comfort.
- Parsons [2-59] carried out experiments to analyse the effect of acclimation (over some days) on thermal comfort and concludes that the changes in thermal comfort responses are small and unlikely to be of practical significance.
- Nicol [2-57] proposes a method to take the time (during one day) as a factor in the specification of comfort temperatures into account.
- McCartney [2-54] analysed questionnaires and investigated the validity of fixed set point temperatures. The results also show that occupants in not air-conditioned buildings feel comfortable even at temperatures outside those suggested by the current thermal comfort standards, cf. also ProKlimA-study [2-13].
- Jones [2-43] estimated the uncertainties involved in thermal models for use in thermal comfort standards and concluded, that thermal comfort can hardly be predicted by building simulation programs if they are not specially adapted to a thermal comfort model.

Various design parameters with regard to thermal comfort can be taken from guidelines: For example, more than 40 pages in the HVAC-handbook [2-66] describe the hygienic basics and the thermal comfort from a designer’s point of view. As the building standards have become stricter in recent years and the specifications with regard to thermal comfort are very complex, a discussion on practically applicable comfort criteria started in Germany in 2003 [2-70].

As is often the case when making the transition from the researcher’s world to that of the practitioner, certain sacrifices in precision are necessary in order to make models simple and useful. Thus, the room temperature is used as single comfort criterion in everyday design of buildings and energy concepts. Fig. 2-2 shows some of the above mentioned comfort criteria concerning the room temperature.

**Conclusion.** This thesis deals with passive cooling to improve thermal comfort. As low-energy office buildings are designed for and built with a high building physical standard, draught effects and discomfort due to high temperature gradients can be neglected. Therefore the data evaluation focuses only on comfort criteria, which are based on the operative room temperature. Taking the experience from various studies into account, the comfort criteria according to the DIN 1946 standard is applied to the data evaluation since this comfort range covers most of the other comfort criteria and
takes uncertainties, local and time variations into account. A first evaluation of user acceptance [2-34] confirms this approach: In low-energy office buildings with a high building physical standard, the room temperature dominates the perception of comfort.

Fig. 2-2: Comfort criteria: While DIN 1946 [2-20] takes a comfort range into account, the ASHRAE 55 standard [2-6], its revised formula [2-18] and the criterion from VDI 2078 [2-69] (with a limited design temperature) give explicit correlations. The temperature limits from the directive on health and safety at work [2-5] and from Rouvel’s approach [2-19] are 25, 26 and 27 °C.

2.2 Methodology

This thesis deals with great many monitored data, which are to be analysed by a universal method. The methodology is shortly described as model-based data analysis.

The new approach regarding the data analysis is based on building simulation and (1) a parametric model. In the calculation of the thermal building behaviour, (2) the modelling of air flow through buildings is connected with uncertainties and must be modelled accurately. In an other Subchapter, (3) the used physical models are introduced shortly and their errors are estimated. Furthermore, (4) field measurements and the monitoring in buildings are not very precise and the accuracy should be estimated.

2.2.1 Parametric model: room temperature and energy balance

The night ventilation potential can be derived from an energy balance which takes heat gains, heat losses and heat storage into account. Thus, the aim is to identify characteristic building parameters which determine heat gains, losses and storage, i.e. the heat gain G, the heat storage capacity C, and the thermal loss factor H.

In this thesis, the data analysis focuses on the mean indoor temperature and its variation in time. Particular periods may be characterised by mean heat flows and their corresponding variation in time, too. As this model is based on a harmonically oscillating heat flow and temperature variation, some simplifications are considered which will be discussed in the following Sections:
1. Mean g-value of glazing: The total solar energy transmittance is independent of the incident angle. Variable sun-shading is not taken into account.

2. Constant air change rate: Consideration of day and night ventilation by an effective air change rate.

3. No transient thermal building performance: Selection of periods with (nearly) uniform boundary conditions.

4. Fixed heat transfer coefficients.

5. User behaviour is calculated by a specific time profile depending on the working time.

6. The time shift between ambient air temperature, solar radiation and internal heat gains is ignored by this simplified model.

7. No heat flow through internal walls. Solar heat gains at and the heat conduction through external walls are estimated for steady periodic boundary conditions considering the thermal inertia of the external wall.

The parametric model is based on Keller’s model [2-45] and supplies an universally valid thermodynamic description of the thermal building performance. This model reduces the multitude of parameters to a reasonable number, such that a clear representation of the relationship between climate, energy flows and indoor temperature emerges which takes the operation of the building and the user behaviour into account.

**Temperature variation T**

The time variation $T(t)$ can be modelled by a Fourier series for the indoor and the outdoor temperature:

$$T(t) = \frac{a_0}{2} + \sum_{n=1}^{\infty} a_n \cdot \sin \left( \frac{2\pi \cdot n}{t_0} \cdot (t + t_0) \right)$$  \hspace{1cm} (2-4)

For a data set with $n$ time steps, the maximum number of frequencies is $n/2$ due to numerical uniqueness.

The time variation of the indoor temperature $T_i$ approximately satisfies a single sinusoidal oscillation for the period considered for analysis. In most applications, the cycle duration $t_0$ is 24 h. $a_0/2$ is the mean temperature $T_m$ and $a_n$ the temperate amplitude, $n/t_0$ the frequency and $t_n$ the phase shift of the $n^{th}$ frequency. A good approximation is already reached after the first element of the Fourier series.

$$T_i(t) = T_{i,m} + \Delta T \cdot \sin(\omega \cdot t) \quad \text{with} \quad \omega = 2\pi / t_0$$  \hspace{1cm} (2-5)

If all boundary conditions oscillate regularly, the variation of temperature with time can be estimated from characteristic building parameters:

- The mean indoor temperature $T_{i,m}$ can be calculated with the mean ambient air temperature $T_{a,m}$, the mean heat gain $G_m$, and the loss factor $H$.

$$T_{i,m} = T_{a,m} + \frac{G_m}{H}$$  \hspace{1cm} (2-6)

- Starting from a given loss factor $H$, the amplitude of the indoor temperature $\Delta T_i$ increases with the amplitude of the ambient air temperature $\Delta T_a$ and the heat gain $\Delta G$ and decreases with an increasing time constant $\tau$.

$$\Delta T_i = \frac{t_0}{2 \cdot \pi \cdot \tau} \left( \Delta T_a + \frac{\Delta G}{H} \right) \quad \text{with} \quad \tau \approx C/H$$  \hspace{1cm} (2-7)
Obviously, the three characteristic parameters \( G, H \) and \( C \) can be merged into two characteristics. Keller [2-45] use the time constant \( \tau = C/H \) and the gain-to-loss ratio \( \gamma = G/H \) in order to characterise the thermal building performance for design purposes. As the single effects shall be discussed separately in this thesis, three separate building parameters will be used.

**Heat gain \( G \)**

The heat gain \( G \) summarises solar and internal heat gains and is related to the surface area of the external wall \( A_{\text{ext.wall}} \). The mean heat gain \( G_m \) increases with the solar heat gain \( g \cdot I_m \) and the internal heat gains \( Q_m \). The solar heat gain is the product of the solar radiation \( I_m \) (global, on the facade), the transparent surfaces/ windows \( A_{\text{trans.surface}} \) and their solar gain factors \( g \).

\[
G_m = \frac{g \cdot I_m + Q_m}{A_{\text{ext.wall}}} \quad \text{with} \quad g = \sum_{i=1}^{k} A_{\text{trans.surface},i} \cdot g_{c,i} \tag{2-8}
\]

The amplitude of heat gains \( \Delta G \) is calculated similar to the mean heat gain \( G_m \) but with the amplitude of the solar radiation \( \Delta I \) and the internal heat gains \( \Delta Q \).

\[
\Delta G = \frac{g \cdot \Delta I + \Delta Q}{A_{\text{ext.wall}}} \tag{2-9}
\]

**Heat loss factor \( H \)**

The heat loss factor \( H \) describes the coupling between indoor and outdoor temperature and contains the heat losses due to transmission through the external wall and ventilation. The heat loss factor \( H \) is related to the surface area of the external wall \( A_{\text{ext.wall}} \). The transmission is calculated for each part of the external wall and its U-Value. The ventilation loss is a function of the air change rate, the air volume of the office and the properties of air.

\[
H = \frac{1}{A_{\text{ext.wall}}} \left[ \sum_{i=1}^{n} A_{\text{ext.wall},i} \cdot U_i + ACH_{\text{eff}} \cdot V_{\text{office}} \cdot \frac{Q_{\text{air}}}{3600} \right] \tag{2-10}
\]

The calculation is similar to the calculation methods in EN 832 [2-24] and EN ISO 13789 [2-26].

As night ventilation operates with an air change rate during the night which is higher than the air change rate during working hours, a variable air change rate has to be taken into account. Burmeister [2-14] proposed an extra factor \( f_k \) as a measure for night ventilation. Similar to this approach, the effective air change rate \( ACH_{\text{eff}} \) can be calculated from hourly data of the air change rate \( ACH_{t} \), indoor \( T_{1,t} \) and ambient air temperature \( T_{a,t} \).

---

4. The solar gain factor \( g \) is a function of the incident angle, the shading devices, the pollution, and the external shading. The "corrected" value \( g_c \), which is used to calculate the daily solar heat gains, can be estimated according to DIN 4108 [2-21]. The solar heat gains can be calculated more accurately by a building simulation.

5. As the solar radiation and the internal heat gains do not follow a single sinusoidal oscillation, the amplitudes have to be derived from a Fourier analysis. Reference values can be taken from Ref. [2-44].
As the indoor air temperature has to be known, this approach can be used only for data analysis from measurements. A good approximation for design purposes provides Eq. (2-11) with $T_{a,n}$ in place of $T_{i,t}$.

**Heat storage capacity $C$ of a solid**

A comprehensive description of the underlying theory is given by Carslaw and Jaeger [2-15] and by Keller [2-44]. This Section outlines the calculation procedure as roughly as appropriate for the data analysis. In general, the heat storage capacity is calculated from two terms:

$$C = C_w \cdot C'$$  \hfill (2-12)

The heat storage capacity of a semi-infinitesimal solid with surface temperature as a harmonic function of the time. The heat storage capacity $C_w$ of a room consists of the heat storage capacity of each internal wall / structure $A_{int.wall}$ and is related to the surface area of the external wall $A_{ext.wall}$. The specific heat storage capacity of each structure is the product of its volumetric heat storage capacity $c_p$ and its thermally effective thickness $d_{eff}$.

$$C_w = \frac{1}{k} \left( \sum_{i=1}^{n} (A_{int.wall,i} \cdot c_p \cdot d_{eff,i}) \right)$$  \hfill (2-13)

The calculation is similar to the calculation method in EN 832 [2-24].

The thermally effective thickness is a function of the thermal properties (heat conductivity $\lambda$, and volumetric heat storage capacity $c_p$) and the cycle duration of the periodic heat excitation $t_0$ (usually 24 hours).

$$d_{eff} = \frac{1}{\sqrt{2}} \sigma \quad \text{with} \quad \sigma = \sqrt{\frac{t_0}{\pi}} \cdot a \quad \text{and} \quad a = \frac{\lambda}{c_p}$$  \hfill (2-14)

At the periodic heat wave penetration depth $\sigma$, the amplitude of surface temperature is reduced to $1/e$. $\sigma$ is calculated for a semi-infinitesimal and homogenous material. The thermally effective thickness $d_{eff}$ is derived from the heat flow at the surface (index: sf) of a semi-infinitesimal layer for the temperature oscillation $\Delta T_{sf}$ of the periodic heat excitation $t_0$ at its surface:

$$q_{sf}(t) = \sqrt{2} \cdot \frac{\lambda}{\sigma} \cdot \cos \left( \omega t + \frac{\pi}{4} \right) \cdot \Delta T_{sf} \quad \text{with} \quad \omega = \frac{2\pi}{t_0}$$  \hfill (2-15)

An extensive but not sophisticated integration over time with positive heat flow results in the heat storage $q_{in}$ during a cycle period. The formula can be written with the volumetric heat storage capacity $\rho c$ and the temperature amplitude $2\Delta T_{sf}$ which affects the heat flow at the surface during a cycle period:

$$q_{in} = \int_{5/8t_0}^{9/8t_0} q_{sf}(t) dt = \sqrt{2} \cdot \frac{\lambda}{\sigma} \cdot \frac{t_0}{2\pi} \cdot 2\Delta T_{sf} = c_p \cdot \frac{\sigma}{\sqrt{2}} \cdot 2\Delta T_{sf}$$  \hfill (2-16)

Thus, the thermally effective thickness is $d_{eff} = \sigma / \sqrt{2}$. For thin structures, the thermally effective thickness $d_{eff}$ can be larger than the geometric thickness $d$. For
plausibility reasons, the following side condition may be considered: If \( d < \sigma / \sqrt{2} \), then \( d_{\text{eff}} = d \).

The heat storage capacity of a finite solid with finite heat transfer at its surface. EN ISO 13786 [2-25] shows how to determine the heat storage capacity of a solid. The following calculation correspond to this approach whereby only the difference between the ideal capacity \( C_\infty \) and the real capacity \( C \) is calculated. The factor \( C' \) determines how much heat can be supplied and dissipated during a single cycle period, if the heat transfer at the surface and a finite thickness is considered: The dimensionless geometric ratio \( z \) describes the modulation of the heat flow due to geometric limits and the Biot number \( B_i \) describes the reduction of the heat flow into the structure due to the finite heat transfer at the surface. The combined heat transfer coefficient \( h_{c+lw} \) takes the convective \( h_c \) and the long wave heat transfer \( h_{lw} \) into account. Typical \( h \)-values are between 6 and 9 W/(m² K). Fig. 2-3 shows the results from Eq. (2-17).

\[
C' = f(z, B_i) \quad \text{with} \quad z = \frac{d_{\text{eff}}}{\sigma} \quad \text{and} \quad B_i = \frac{h_{c+lw} \cdot d_{\text{eff}}}{\lambda}
\]

\[
C' = \frac{1}{\sqrt{2}} \cdot \frac{(1+i) \cdot \tanh \left[ z \cdot (1+i) \right]}{1+z \cdot \frac{1}{B_i} \cdot (1+i) \cdot \tanh \left[ z \cdot (1+i) \right]}
\tag{2-17}
\]

Differing from the underlying theory, \( z \) and \( B_i \) are calculated with the effective thickness \( d_{\text{eff}} \) and not with the thickness \( d \) of the structure: An extensive model comparison between numerical simulation and this analytical solution of Fourier’s heat transfer equation [2-61] shows that this approach has no impact on the calculation of thin structures but yields better results for thick structures.

![Adiabatic heat storage function](image)

**Fig. 2-3:** Adiabatic heat storage function. The backside of the finite layer is adiabatic. (According to the theory, the standardised dynamic thickness is calculated with the thickness \( d \) of the solid.)

**Heat storage capacity \( C \) of a room**

Due to the building’s thermal inertia, the temperature variation is attenuated. In other words, the temperature variation in a room corresponds to its heat storage capacity. Starting from the periodic indoor and outdoor temperature variation, the time constant \( \tau \)
of a room can be calculated with the heat storage capacity C and the heat loss H. The calculation of C is defined by and regularised in different standards:

- EN ISO 13786 [2-25] presents a consistent calculation procedure on the basis of complex resistances similar to Eq. (2-17), as mentioned above. The simplified calculation method uses the thickness d of the layer for thin layers and the thermally effective thickness \( d_{\text{eff}} \) for thick layers. The heat transfer is taken by an approximate formula into account. The simplified model can result in wide differences from the detailed calculation.

- EN 832 [2-24] (energy use for heating) admits the calculation of C according to the simplified calculation method from EN ISO 13786 as a good approximation where-by the maximum thickness is 0.1 m, independent of the material properties.

- DIN 4108 [2-21] (energy use for heating) demands the calculation of C according to EN ISO 13786 for components (Part 2) and according to EN 832 for the room (Part 6).

- EN ISO 13791 [2-27] (room temperatures in summer) proposes a numerical simulation of the room. The local and time discretisation are functions of the material properties of each wall.

- EN ISO 13792 [2-28] (room temperatures in summer) presents a simplified approach based on EN ISO 13791 and calculates the heat transfer between the internal wall and the room air with a simplified model from C, whereby C is calculated according to EN ISO 13786.

Obviously, there are competing approaches in different standards. For this reason, a numerical investigation on different calculation procedures has been carried out for several configurations of material properties, thickness and heat transfer in order to get an accurate model. Noteworthy, none of the analytical models result in the heat storage capacity C calculated by the numerical simulation for any configuration. Though each analytical model calculate the heat storage capacity C accurately for particular configurations, the best agreement for all configurations provides the consistently physical model with differences between the numerical and the analytical solution in the range of 10 % for typical configurations and in the range of more than 20 % for very thick layers or very high heat transfer coefficients.

The time constant \( \tau \) in Eq. (2-7) can be specified with these investigations: The heat storage capacity C is calculated according to Eq. (2-12) for the room and the heat loss factor H according to Eq. (2-10).

\[
C = \tau \cdot H
\]  

(2-18)

**Summary**

The parametric model represents a thermodynamic description of the energy balance and the resulting room temperature. This model has been developed on the basis of Keller’s approach and is mathematically derived from physical models. For this consistent derivation, the parametric model can be used as a straightforward model for data analysis and is universally valid. The model has been compared to and verified with different approaches from international standards.

### 2.2.2 Air flow through buildings

Only if the infiltration and the interzonal air change is known or calculated, the night ventilation efficiency can be calculated accurately. As the calculation is uncertain due to many imprecise input parameters, the air change rates have been measured during
the monitoring campaigns carried out in the context with this thesis. The underlying theory is outlined in the following.

The air volume flow $V_{\text{opening}}$ through an opening is generally described by:

$$V_{\text{opening}} = sgn(\Delta p) \cdot R \cdot |\Delta p|^{n}$$  \hspace{1cm} (2-19)

with the hydraulic resistance $R$ and the pressure difference $\Delta p$. The exponent $n$ is a function of the flow formation: $n=0.5$ for turbulent flow through large openings, $n=2/3$ for small openings (e.g. cracks) and $n=1$ for laminar flow. The hydraulic resistance $R$ is calculated as a function of the geometry. The most important resistance model for night ventilation purposes is

$$R_{\text{large opening}} = \frac{2}{\rho_{\text{air}}} \cdot c_{D} \cdot A_{\text{opening}}$$  \hspace{1cm} (2-20)

with the discharge factor $c_{D}$. The total resistance $R_{\text{total}}$ of $i$ parallel resistances $R_{i}$ is calculated according to

$$\frac{1}{R_{\text{total}}} = \sum \frac{1}{R}$$  \hspace{1cm} (2-21)

and the total resistance of two parallel openings according to

$$c_{D} \cdot A = \sqrt{\frac{1}{\left(c_{D,1} \cdot A_{1}\right)^2 + \left(c_{D,2} \cdot A_{2}\right)^2}}$$  \hspace{1cm} (2-22)

Two driving forces generate the pressure difference $\Delta p$ for natural ventilation:

1. The density difference in two isothermal air columns with the height $h$ and different temperatures $T_{1}$ and $T_{2}$ causes a pressure difference, if one air column is exposed to the other:

$$\Delta p = \rho_{m} \cdot c_{p} \cdot g \cdot h \cdot \left(1 - \frac{T_{1}}{T_{2}}\right)$$  \hspace{1cm} (2-23)

2. The wind $v_{w}$, which flows around a building, creates a pressure difference due to the static and dynamic pressure at its surface. The wind profile, the wind turbulence and the shape of the building affects the pressure difference which is described by the pressure coefficient $c_{p}$. The pressure coefficient $c_{p}$ can be taken from measurements (e.g. wind tunnel readings) or is calculated by numerical CFD simulations or – if possible – with analytical models.

$$\Delta p = c_{p} \cdot \rho_{m} \cdot \frac{v_{w}^2}{2}$$  \hspace{1cm} (2-24)

Additionally, a ventilation system may enhance the air change rate whereby the ventilation rate is divided into an adjusted part $V_{n}$ (supply = exhaust air flow) and a not-adjusted part $V_{n-a}$ (supply ≠ exhaust air flow). The mechanical is superposed by the natural ventilation due to the thermal buoyancy $V_{T}$ and the wind $V_{w}$. The total air flow is calculated according to the vectorial addition:

$$V_{\text{total}} = V_{m,a} + \sqrt{V_{T}^2 + V_{w}^2 + V_{m,n-a}^2}$$  \hspace{1cm} (2-25)

Though the elementary formulas are simple, the calculation procedure is complex due to the complex correlations within the network and the interactions between the natural ventilation and the temperature difference. For further details the reader is referred to
Ref. [2-4], [2-38] or [2-33]. In Part B of this thesis, the calculation procedure is applied to air-flow networks.

2.2.3 Physical models and error estimation

Physical models have to be exact and coherent. Nevertheless, (1) air flow and (2) energy balance models can only predict the reality as accurately as the boundary conditions are known. Furthermore, there may be inaccuracies due to (3) numerical estimations. These single errors result in (4) an overall error of the thermal building model. (5) Though the parametric model is universally valid, its manageability has been obtained by some model simplifications:

1. Air-flow model. Air-flow models are used in both (numerical) building simulation and (analytical) calculations. Due to the interaction between the driving forces and the thermal behaviour of the room, the air-flow and the thermal building model have to be calculated simultaneously, cf. Hensen [2-40]. The air-flow models used in this thesis are validated. However, the results may be uncertain due to the inaccurate determination of input parameters (i.e. geometry, \( c_D \) and \( c_p \)-values). Flourentzou [2-33] shows that the precision in field measurements is in the order of 20 %.

2. Energy balance model. The calculation procedures for the transmission through a wall \( \dot{Q}_{\text{trans}} \) with the thermal resistance \( 1/U \) and ventilation with the temperature difference \( \Delta T \) are well-known and have been validated repeatedly. However, the (dynamical) thermal resistance, cf. Petzold [2-60], and the thermally effective air volume flow have to be calculated accurately:

\[
\dot{Q}_{\text{trans}} = U \cdot A_{\text{wall}} \cdot \Delta T
\]  

(2-26)

\[
\dot{Q}_{\text{vent}} = \dot{V}_{\text{air}} \cdot c_p \cdot \Delta T
\]  

(2-27)

The internal heat gains are usually defined by measured / estimated time patterns:

\[
\dot{Q}_{\text{internal}}(t) = \dot{Q}_{\text{equipment}}(t) + \dot{Q}_{\text{person}}(t)
\]  

(2-28)

The calculation of solar heat gains through the transparent surface area \( A_{\text{trans}} \) due to the solar radiation \( G_{\text{solar}} \) is complex (i.e. solar gain factor \( g \) dependent on the incident angle, sun-shading, solar obstruction, reflection from inside to outside, multi-reflection in the window etc.) and affects the modelling of night ventilation indirectly.

\[
\dot{Q}_{\text{solar}} = g \cdot A_{\text{trans}} \cdot I_{\text{solar}}
\]  

(2-29)

In this context, the error is in the order of 10 % for the heat gains and 20 % for the ventilation. The application of these formulas will be discussed in detail in Part B.

3. Numerical calculation of heat conduction. The one dimensional heat conduction equation may be written in the form

\[
\frac{\partial^2 T}{\partial t^2} = a \cdot \frac{\partial^2 T}{\partial x^2}
\]  

(2-30)

and can be calculated numerically by the Crank-Nicholson scheme (with \( \gamma=0.5 \)) with the time \( \Delta t \) and local discretisation \( \Delta x \).

\[
T_n^{t+1} - T_n^t = \gamma \cdot F_0 \cdot \left( T_{n+1}^{t+1} - 2T_n^{t+1} + T_{n-1}^{t+1} \right) + (1-\gamma) \cdot F_0 \cdot \left( T_n^t - 2T_{n+1}^t + T_{n-1}^t \right)
\]  

(2-31)

where \( F_0 \) is the Fourier number \( F_0 = \frac{a \cdot \Delta t}{\Delta x^2} \).
for internal nodes; while for boundary nodes:

\[
T_{n+1}^h - T_n^h = \gamma \cdot 2Fo \cdot (Bi T_{n+1}^h - (1 + Bi) T_n^h + T_{n+1}^h) + (1 - \gamma) \cdot 2Fo \cdot (Bi T_{n-1}^h - (1 + Bi) T_n^h + T_{n+1}^h)
\]

where \(Bi\) is the Biot number \(Bi = \frac{h \cdot \Delta x}{\lambda}\) (2-32)

At each finite step in time there will be a difference between the exact solution of Eq. (2-19) and the numerical approximation according to Eqs. (2-20) and (2-21). In order to reduce the difference, the Fourier number should be in the order of 1 according to Baehr and Stephan [2-9] and the Biot number larger than 0.1 according to Hensen [2-39]. For example: Taking the properties of concrete, these requirements results in \(\Delta x = 0.05m\) (Fourier number) and for the boundary layer \(\Delta x > 0.03m\) (Biot number) for a 1 hour-time step. Hence, a concrete ceiling with \(d=0.2m\) should be divided in minimum 3 and maximum 6 layers. A model comparison shows that the differences are below 3% concerning the heat conduction through the ceiling according to Clarke [2-16] and around 10% concerning the heat storage in the ceiling, cf. Ref. [2-61].

4. Error propagation. The energy balance summarises the heat gains and can be expressed by the mean temperature and its daily amplitude. As the single heat flows have different inaccuracies, the overall error \(s_T\) can be estimated for the function \(f_T\) with \(i\) variables \(x\) and the single errors \(s_x\) from the total differential \(\frac{\partial f_T}{\partial x_i}\) according to the error propagation equation

\[
s_T = \sqrt{\sum_{i=1}^{n} \left( \frac{\partial f_T}{\partial x_i} \right)^2 \cdot s_x^2}
\]

(2-33)

Starting from Eqs. (2-9) and (2-10) for \(s_T\), the overall error is estimated for typical values (from Chapter 5) and is 9% for the mean temperature and 24% for the temperature amplitude.

5. Parametric model. There are some simplifications in the parametric model whose inaccuracies are estimated as follows: (1) The parametric model uses only one heat storage capacity \(C\) for the room whereby the temperature variation \(\Delta T_i\) can only be predicted for a single cycle period since the penetration depth \(\sigma = \sigma(t_0)\) is a function of the cycle period. (2) The air change rate is considered as a fixed value and does not depend on the temperature difference \(T_{a-Ti}\) whereby natural ventilation can be taken only in an iterative procedure into account due to the natural ventilation \(ACH = ACH(T_{a-Ti})\). (3) Solar and internal heat gains are calculated by harmonic oscillations whereby time patterns are approximated. (4) Though the model is valid for an infinitesimal Fourier series, it is reduced to one harmonic oscillation. Keller [2-44] estimated that the inaccuracy of these simplifications are smaller than 20% which is in the same order of error as the building simulation.

2.2.4 Monitoring in low-energy office buildings and its accuracy

In moderate climates, one promising approach to reduce the energy demand of office buildings for air conditioning without reducing comfort is passive cooling by night ventilation. The thermal performance of four low-energy office buildings is evaluated for the summer situation. In this context, (1) the design ideas are discussed and (2) the order of error associated with the measurements are estimated.
Survey of low-energy office buildings

The four low-energy office buildings, which are surveyed in Part B, use passive cooling and have been designed, constructed and monitored for two years in the framework of the German research programme, “Solar Optimised Buildings”, SolarBau for short [2-68].

The general benchmark of this programme for funding is a total primary energy demand below 100 kWh/(m² a) for heating, ventilation, cooling and lighting. The primary energy consumption is calculated from the end energy use (natural gas and electricity) with a primary energy conversion factor. The funding includes subsidies for energy-related optimisation during the design phase and detailed monitoring of the building in operation. The absence of investment subsidies ensures that all design solutions (with excellent visual and thermal comfort) have to be implemented under representative economic conditions.

Concerning the temperature performance in summer, the focus usually lies on avoiding unwanted solar gains in the temperate German climate. The remaining internal loads can often be counterbalanced by controlled ventilation, additional night ventilation or by earth-to-air heat exchangers, cf. Voss [2-71]. Thus each building was designed under similar targets.

Estimation of errors associated with measurements

The quality of the measurements is described in Part B in connection with the data analysis. In general, all sensors have been calibrated together with the data acquisition equipment. The measuring error of all thermometers is less than \( \pm 0.1 \) K. (This accuracy is necessary since the indoor temperature variation is in the order of 1 – 5 K.)

The main uncertainties and errors with regard to the analysis of the passive cooling system lie in temperature and air change measurements:

- Though the monitoring equipment has been designed and inspected carefully, there are errors in the ambient air temperature (e.g. insufficient protection against solar radiation), in the room temperature due to the sensor position (e.g. draught effect, height and position of the sensor or impact of the wall temperature on the measurement of the air temperature) and in the wall temperatures (i.e. position of the sensor). After all known inaccuracies have been corrected as well as possible, the order of error is estimated by \( \pm 0.2 \) K.

- The measurement of air change rates in buildings with natural or hybrid ventilation is rather difficult due to countless impacts on the free ventilation. For this reason, air change measurements are carried out repeatedly and the measuring error is documented for each result.

Taking not only the largest measuring uncertainties (temperature and air change rates) but also the inaccuracies of secondary measuring points (e.g. ventilation system, solar radiation, use of sun-shading, electricity consumption or occupancy patterns) into account, the overall error concerning the energy balance of an office is in the order of 10 %.

While these errors can be corrected, more problematic are the inaccuracies we are not aware of. For this reason, the monitoring equipment has been checked repeatedly in each building. It is assumed that there are no unrecognised errors.
Basics and Methodology

2.3 References


3 Evaluation of Earth-to-Air Heat Exchangers

Part A

Both night ventilation and earth-to-air heat exchangers (EAHX) are air-driven passive cooling concepts. While the night ventilation uses the cool night air as natural heat sink, EAHX’s use the ground as such. The data evaluation of three EAHX’s introduces a model based data analysis which takes long-term monitoring and short-term measurements into account. From this data analysis of EAHX’s, some lessons can be learned for the data analysis of passive cooling systems since the thermodynamic system is less complex than a building but also contains uncertainties.

In designing an earth-to-air heat exchanger, a decision on design goals has to be made. If the air flow is given by the ventilation system and the construction site is known, the question is: Is it more important to achieve a high specific energy performance based on the surface area of an EAHX, a high adaptation of air temperature to ground temperature or a very small pressure loss?

This Chapter deals with the performance of three EAHX’s in service, with the aim of characterising their efficiency. A general method to compare EAHX’s in operation will be introduced. First, the temperature behaviour is described by plots over time and characteristic inlet / outlet air temperature diagrams, and compared by duration curves of air temperatures. Second, the energy gain is illustrated by monthly graphs and as function of the ambient air temperature. Third, a parametric model is used to provide general efficiency criteria. Thermal efficiency is defined by both the dynamic temperature behaviour and the energy performance.

Fig. 3-1: Boundary conditions for the design and operation of EAHX’s: Climatic and project specific boundaries (normal), heat flow at the surface (italic) and in the ground (bold). The heat transfer between ground and air depends strongly on the position of the earth-to-air heat exchanger (here: EAHX 1 – 3).
3.1 Introduction

Due to the high thermal inertia of the soil, the temperature fluctuations at the surface are attenuated in the ground and the time lag between the surface and the ground temperature increases with the depth. Therefore, at a sufficient depth, the ground temperature is lower than the outside temperature in summer and higher in winter. When ambient air is drawn through buried pipes, the air is cooled in summer and heated in winter, before it is used for ventilation. Thus, EAHX’s can fulfil both purposes demanded above: (pre-)heating in winter and (pre-)cooling in summer.

The main advantages of the system are (1) its simplicity, (2) its high cooling and pre-heating potential, (3) low operational and maintenance costs and (4) the saving of fossil fuels and related emissions. Pre-heated fresh air supports a heat recovery system and reduces the space heating demand in winter. In summer, in combination with a good thermal design of the building, the EAHX can eliminate the need for active mechanical cooling and air-conditioning units in buildings. EAHX’s are hence a passive cooling option in moderate climates.

The energy performance of EAHX’s is described by the interaction of heat conduction in the soil and the heat transmission from the pipe to the air. Fig. 3-1 shows the main boundary conditions for the design and operation of EAHX’s. The thermodynamic model used in this Chapter reduces the parameters to some characteristic values in order to show the relationship between operation, climate and project specific boundaries clearly.

Different parametric and numerical models for EAHX’s have been published recently. Some of them are mentioned below. Simulation models can be classified as models with an analytical or a numerical solution of the ground temperature field, and mixed models:

- **Analytical models.** Albers [3-1] developed a parametric model for steady air flow based on a form factor to model the three-dimensional temperature profile. Sedlbauer [3-2] published a model based on a heat capacity model to take changes in air flow during operation into account. An evaluation of eight simulation models by Tzaferis [3-3] reached the conclusion that almost all proposed algorithms can predict the outlet air temperature with sufficient accuracy. A very new approach based on a parametric model using deterministic techniques is given by Mihalakakou [3-4].

- **Numerical simulation models** calculate the thermal performance of EAHX’s with algorithms describing the coupled and simultaneous transfer of heat in soils under a temperature gradient. A complete numerical model for a single-pipe EAHX is introduced by Mihalakakou [3-5]. This model is validated with long-term measurements and is used to describe the thermal influence of the key variables, pipe length, pipe diameter, air velocity and pipe depth [3-6]. A numerical model for a two-pipe EAHX is described by Bojic [3-7]. A numerical model for multiple-pipe EAHX’s was validated by Hollmuller [3-8].

- **Mixed simulation models** are resistance-capacity models based on a numerical solution for the earth temperature near the pipe and an analytical calculation of boundary conditions. Evers [3-9] used such a model to predict the energy performance of EAHX’s for different design parameters. In the framework of an EU project, a design tool was developed under the guidance of AEE Gleisdorf and Fraunhofer ISE by 15 engineering companies [3-10]. The simulation model is based on an extended, validated and well-tested resistance-capacity model by Huber [3-11].
The accuracy of parametric and numerical models can be tested against experimental data for the thermal performance (energy and temperature) of EAHX’s by introducing a dimensionless temperature difference. A comparison showed good agreement, Ref. [3-12]. Against this background, a design tool based on four models with different levels of detail was developed and validated at Fraunhofer ISE, Ref. [3-13]: The first model is a nomogram (set of characteristic curves) for design decisions without detailed information about the thermal soil properties and operation management, the second model is based on an analytical solution of the undisturbed three-dimensional temperature field with a form factor, the more detailed third model is based on a heat-capacity model and takes the operation time into account, and the most detailed fourth model is a complete numerical solution of the three dimensional heat and moisture transfer and the changes in the air temperature and humidity. Starting from this design tool, an analytical model was enhanced [3-14]: The model combines a form-factor model with a heat-capacity model and accurately predicts the thermal performance of EAHX’s. This model will be used to compare the thermal performance of EAHX’s in operation.

As the ground temperature is dependent on many boundary conditions, its modelling is rather difficult: Mihalakakou [3-15] used the energy balance equation to predict ground surface temperatures. In addition to this deterministic approach, a data-based approach is described by Mihalakakou [3-16]: The influence of the input climatic parameters on the ground surface temperature was investigated using a neural network approach. Starting with the calculated surface temperature, the ground temperature can be estimated.

The ground temperature profile is strongly influenced by the material parameters of the soil. Usually, determination of the heat conductivity, heat capacity and density is difficult. A detailed summary of practical models is given by Dibowski [3-17]. In addition to the ground cover and the material parameters of soil, possible thermal influence of a building or ground water should be taken into account, see Ref. [3-14]. In this Chapter, a data-based model is used to predict the undisturbed ground temperature according to Mihalakakou’s model [3-18].

There are some publications concerning the design and operation of EAHX’s, see Reise [3-10], Dibowski [3-19], Henne [3-20] and Zimmermann [3-21]. The heat and cooling energy gain by an EAHX should meet the actual energy demand of the building’s ventilation system:

- If the heating or cooling energy gain is lower than the corresponding energy consumption, the supply air has to be heated up or cooled down additionally.
- If the energy gain is higher than the corresponding energy consumption, the supply air has to be cooled down (in winter) or heated up (in summer) respectively, though this would not be necessary if the EAHX were not in operation. This means an unwanted energy dissipation.

An additional demand on EAHX’s in office buildings is that an EAHX should meet the complete cooling energy demand for sufficient thermal comfort in order to substitute a mechanical cooling device. Santamouris [3-22] presented an integrated method to calculate the energy contribution of EAHX’s to the cooling load of thermostatically controlled buildings. Bojic [3-23] introduces an energy ratio to calculate, whether an EAHX provides more or less heat or cooling energy than it needs to reach a given comfort temperature inside the building. In this Chapter, the contribution of an EAHX to the building energy consumption is characterised by a frequency distribution for the heating and cooling energy supply.

Besides the heat gain, the electricity demand should be taken into account. De Paepe [3-24] developed a method to optimise the energy efficiency of an EAHX by reducing
the pressure drop for a given thermal efficiency. According to this method, not the electricity demand but the dissipation energy caused by the pressure loss is used to calculate the coefficient of performance COP, the ratio of thermal energy supplied to mechanical dissipation energy.

In spite of many publications and available design tools, there is a lack of comparative analysis. The presented method includes an evaluation of temperature behaviour, energy gain and energy efficiency by universal performance criteria.

3.2 Description of evaluated earth-to-air heat exchangers

The new method for data evaluation is applied to monitoring data from three EAHX’s.

Projects. The thermal performance of three EAHX’s is compared. The projects are presented in detail in Ref. [3-25]: The EAHX at Fraunhofer ISE (Freiburg) is located left and right to the foundation slab of the building (cf. Fig. 3-2, a) and was designed for an air flow of 9,000 m³/h. The design idea was to achieve a high energy performance to assist the ventilation system with cooled or pre-heated air. The ventilation system for the office building of DB Netz AG (Hamm) is designed for an air flow of 12,000 m³/h. For cooling in summer, a maximum outlet temperature of 19 °C was nested during designing, at 36 °C ambient air temperature. The total surface area was designed quite large to cool the air almost to the undisturbed earth temperature, which reaches approx. 18 °C in late summer. The large register had to be positioned in the small excavation at the construction site (cf. Fig. 3-2, b). The EAHX for the Lamparter office (Weilheim) was designed for an air flow of 1,200 m³/h and for small pressure losses. It is installed around the office building (cf. Fig. 3-2, c). In summer, cool air assists the passive cooling by night ventilation. In winter, pre-heated air supports the heat recovery system, reduces the heat energy demand and prevents freezing of the heat exchanger. In all three cases, no further active cooling is needed to achieve a sufficient thermal comfort, cf. Chapter 9.

Design criteria. While the design criteria are different, the demand on operating time is similar in each building. The characteristic attribute is the surface area per air flow, it varies from 0.075 to 0.18 m²/(m³/h). The key information are given in Table 3-1.

Monitoring. The data evaluation deals with 5-minute data for air flow, ambient temperature, inlet and outlet air temperature and ground temperature. Hourly data are generated for the mean air flow, mean ambient temperature and mean ground temperature. Inlet and outlet air temperatures are averaged over the operating time. Operation is defined by a minimum air flow of 3,000 m³/h (DB Netz AG), 2,500 m³/h (Fraunhofer ISE) and 500 m³/h (Lamparter).

Fig. 3-2: From left to right: a) EAHX under construction, Fraunhofer ISE (Freiburg). b) Drawing of the EAHX, DB Netz AG (Hamm). c) EAHX under construction, Lamparter (Weilheim).
Table 3-1: Description of evaluated earth-to-air heat exchangers.

<table>
<thead>
<tr>
<th></th>
<th>DB Netz AG</th>
<th>Fraunhofer ISE</th>
<th>Lamparter</th>
</tr>
</thead>
<tbody>
<tr>
<td>number of ducts</td>
<td>26</td>
<td>7</td>
<td>2</td>
</tr>
<tr>
<td>length of ducts</td>
<td>67 m – 107 m</td>
<td>Approx. 95 m</td>
<td>each 90 m</td>
</tr>
<tr>
<td>diameter</td>
<td>200 mm and 300 mm</td>
<td>250 mm</td>
<td>350 mm</td>
</tr>
<tr>
<td>depth of ducts</td>
<td>2, 3 and 4 m, around foundation slab</td>
<td>2 m, partly below foundation slab</td>
<td>2.3 m, around foundation slab</td>
</tr>
<tr>
<td>mean air flow</td>
<td>10,300 m³/h</td>
<td>7,000 m³/h</td>
<td>1,100 m³/h</td>
</tr>
<tr>
<td>total surface area of ducts</td>
<td>1,650 m²</td>
<td>522 m²</td>
<td>198 m²</td>
</tr>
<tr>
<td>specific surface area</td>
<td>0.16 m²/(m³h⁻¹)</td>
<td>0.075 m²/(m³h⁻¹)</td>
<td>0.18 m²/(m³h⁻¹)</td>
</tr>
<tr>
<td>air speed</td>
<td>approx. 2.2 m/s</td>
<td>5.6 m/s</td>
<td>1.6 m/s</td>
</tr>
<tr>
<td>pressure loss at mean air flow</td>
<td>40 Pa (measured)</td>
<td>166 Pa (measured)</td>
<td>12 Pa (calculated)</td>
</tr>
<tr>
<td>soil type</td>
<td>dry, rocky</td>
<td>dry, gravel</td>
<td>moist, clay</td>
</tr>
<tr>
<td>ventilation system</td>
<td>hybrid ventilation with supply and exhaust air</td>
<td>hybrid ventilation with supply air</td>
<td>hybrid ventilation with supply and exhaust air</td>
</tr>
<tr>
<td>heat recovery system</td>
<td>yes, 65 % (design)</td>
<td>no</td>
<td>yes, 80 % (measured)</td>
</tr>
<tr>
<td>EAHX bypass</td>
<td>yes, but not used</td>
<td>temperature controlled (open loop)</td>
<td>temperature controlled (closed loop)</td>
</tr>
<tr>
<td>control strategy</td>
<td>time controlled</td>
<td>temperature controlled (open loop)</td>
<td>temperature controlled (closed loop)</td>
</tr>
</tbody>
</table>

All ducts are made of polyethylene.

3.3 Analysis of temperature behaviour

The temperature behaviour is described by plots over time and characteristic curves, separately for every project. The temperature behaviour of different projects can be compared with duration curves and a dimensionless ratio of temperature variation.

3.3.1 Ground temperature

In the design process, the undisturbed ground temperature is a main input parameter. However, its accurate modelling is difficult because the soil parameters are often unknown and changes over time due to the changing soil humidity. Additionally, the definition of undisturbed ground temperature is problematic due to the thermal influence of a building or different soil properties at an EAHX. In the following, the “undisturbed ground temperature” is influenced by the building but not by the EAHX and is defined for mean soil properties. The undisturbed ground temperature is hence a hypothetical value. Fig. 3-3 shows in comparison with Table 3-4 that the monitored ground temperatures are not undisturbed but influenced by both the building and the EAHX: At DB Netz AG, the maximum ground temperature is higher than the maximum outlet air temperature; at Fraunhofer ISE, the ground temperature in summer is sometimes higher than the outlet air temperature; and at Lamparter, the outlet air temperature in winter is sometimes lower than the earth temperature.

Obviously, these measurements cannot be used to evaluate the heat exchange efficiency since they are not plausible as “undisturbed ground temperature”. Therefore the undisturbed ground temperature is calculated, cf. Chapter 3.5. Anticipatory, Fig. 3-3 shows the calculated ground temperature.
3.3.2 Time variation curves and characteristic lines

Due to the damped oscillation of the ground temperature, the outlet air temperature is higher than the inlet air temperature in winter and lower in summer:

- Fig. 3-4 (left) shows hourly mean air temperatures during operation for a whole year. These plots illustrate the working principle of EAHX's but a more significant conclusion is given by the statistical analysis in Chapter 3.5.

- The thermal behaviour of an EAHX can be taken from its temperature characteristic. In Fig. 3-4 (right), vertical lines divide the temperature field into three virtual zones: the heating period for low energy office buildings with inlet temperatures below 12 °C, the cooling period with inlet temperatures above 22 °C and the passive period in between, without heating and cooling. Due to different energy concepts and ventilation strategies, there are different demands on the air-flow rate and supply air temperature. Therefore, these temperature limits are defined only as a benchmark for comparison but not as a quality criterion.
Fig. 3-4: Ambient and outlet air temperatures at DB Netz AG (only time controlled), Fraunhofer ISE (time and open loop temperature control) and Lamparter (time and closed loop temperature control). The right graphs show additionally the compensation lines (grey arrows) for high and low ambient air temperatures, cf. Eq. (3-1).

According to the building energy demand for heating or cooling, at a specific ambient temperature the inlet air should be either heated or cooled. However, in operation, there is sometimes an unwanted temperature decrease in winter or increase in summer due to the non-ideal control. Besides its design and operation, the control strategy has a strong impact on the temperature performance of an EAHX:

- The EAHX at DB Netz AG is operated during working days between 8 a.m. and 5 p.m. As the EAHX is operated without controls, there is a wide temperature range with unwanted (below 12 °C and above 22 °C) or unusable (between these temperature limits) heating and cooling.
Due to the high specific air-flow rate, the EAHX at Fraunhofer ISE has the highest range in outlet temperature. As the air flow is controlled by the inlet air temperature, there is no operation between 12 °C and 16 °C. Due to the open loop control, there is only a very small temperature overlap with unwanted heating or cooling.

The offices at Lamparter are ventilated with a constant supply air temperature of 22 °C during the working hours. Accordingly, the EAHX warms up or cools down the incoming air to 22 °C. As the closed loop control is realised without a temperature hysteresis, the EAHX is operated at every ambient temperature.

Fig. 3-5 illustrates two special aspects: The graph for Fraunhofer ISE shows why the outlet temperature varies at different inlet temperatures. Due to the high thermal inertia, the ground temperature lags behind the ambient air temperature. Hence, the ground is cool at the begin of summer with the first warm days. But at the end of summer, the ground is warmed up and the inlet air during the last summer days (e.g. with 20 °C) cannot be cooled as down as during the first summer days (e.g. with 20 °C). This is valid analogously for the winter. A similar behaviour occurs with room temperatures due to the building’s thermal inertia, cf. Fig. 6-11 in Chapter 6. The graph for DB Netz AG indicates exemplarily how an optimised control can reduce the operation time to a minimum. The limits for the inlet temperature are 12 and 20 °C and the EAHX is not operated when the air is heated during the summer or cooled during the winter. Besides the reduced energy consumption, an advantage over the only time-controlled operation is that the ground can regenerate thermally when the EAHX is not in use.

![Fig. 3-5: Inlet and outlet air temperatures at Fraunhofer ISE (with monthly mean temperature) and at DB Netz AG (with optimised control).](image)

### 3.3.3 Duration curve of outlet temperature

Time variation curves and characteristic lines illustrate the temperature behaviour of a single EAHX. Fig. 3-6 compares the outlet air temperatures for all of the three projects. The outlet air temperature is dependent on both the inlet air temperature and the ground temperature (not shown in Fig. 3-6). As the duration curve depends on the climate, a dimensionless ratio of temperature variation $R_T$ is introduced.

$$R_T = \frac{T_{\text{out,max}} - T_{\text{out,min}}}{T_{\text{in,max}} - T_{\text{in,min}}} \quad (3-1)$$

$R_T$ is independent of the climate if the inlet air temperature is related to same limits ($T_{\text{in,max}}$ and $T_{\text{in,min}}$). Therefore, the inlet temperature range is defined from –15 to 35 °C. The minimum and maximum outlet temperature for each EAHX is derived from a compensation function (gradient for the minimum and maximum outlet temperature) at
low and high inlet air temperatures in Fig. 3-4. The ratio calculated with Eq. (3-1) increases with decreasing specific surface area. Therefore, the EAHX at DB Netz AG achieves the strongest and the EAHX at Fraunhofer ISE the smallest damping of inlet air temperature, see Table 3-6.

The $R_T$-value can be used to estimate the effect of an EAHX on the thermal influence of a building (equalised temperature level for ventilation) and to design the thermal performance for the ventilation system (smaller dimension of heat exchangers, air coolers – if necessary – and heaters).

![Fig. 3-6: Duration curve of air temperature difference for all of the three projects, hourly data.](image)

### 3.4 Energy gain and supply

The energy gain is illustrated by time variation curves and is sorted by the ambient temperature, separately for each project. The energy gain from an EAHX is not identical with its actually utilised energy gain to a building due to non ideal control of the operation. The utilisable energy gain of an EAHX can be derived from a frequency distribution for the heat and cooling energy gain if temperature limits for the heat and cooling energy demand are taken into consideration.

#### 3.4.1 Yearly energy gain

The energy performance $Q_{\text{air}}$ of an EAHX is calculated from the air flow $V_{\text{air}}$, the volumetric heat capacity $\rho c_{\text{air}}$ and the air temperature difference between inlet and outlet:

$$Q_{\text{air}} = \dot{V}_{\text{air}} \cdot \rho c_{\text{air}} \cdot (T_{\text{out}} - T_{\text{in}}) \quad (3-2)$$

Taking the operation time into account, the yearly heat and cooling energy gain can be calculated from Eq. (3-2). The specific energy gain (in relation to surface area from Table 3-1) is compared for different EAHX’s in Table 3-2: As expected, the specific energy gain increases with operation time and the available temperature difference between earth and air but with decreasing specific surface area. Thus, the EAHX at Fraunhofer ISE has the highest specific energy gain. As the energy gain is dependent on climate and operation time of the EAHX, the specific energy gain is suitable to evaluate the energy saving potential of an EAHX but is not suitable as a general efficiency criterion (cf. Chapter 3.5).
Table 3-2: Heating and cooling energy gain.

<table>
<thead>
<tr>
<th></th>
<th>DB Netz AG</th>
<th>Fraunhofer ISE</th>
<th>Lamparter</th>
</tr>
</thead>
<tbody>
<tr>
<td>hours of operation</td>
<td>3,701 h</td>
<td>4,096 h</td>
<td>3,578 h</td>
</tr>
<tr>
<td>heating energy gain</td>
<td>27,700 kWh/a</td>
<td>26,800 kWh/a</td>
<td>3,200 kWh/a</td>
</tr>
<tr>
<td>specific (pipe surface)</td>
<td>16.8 kWh/(m² EAHXa)</td>
<td>51.3 kWh/(m² EAHXa)</td>
<td>16.2 kWh/(m² EAHXa)</td>
</tr>
<tr>
<td>cooling energy gain</td>
<td>22,300 kWh/a</td>
<td>12,400 kWh/a</td>
<td>2,400 kWh/a</td>
</tr>
<tr>
<td>specific (pipe surface)</td>
<td>13.5 kWh/(m² EAHXa)</td>
<td>23.8 kWh/(m² EAHXa)</td>
<td>12.1 kWh/(m² EAHXa)</td>
</tr>
</tbody>
</table>

3.4.2 Monthly energy gain and specific energy performance

Depending on the operation time, ground temperature and inlet air temperature, there is a heating energy gain in winter and a cooling energy gain in summer which is illustrated in Fig. 3-7 (left, with different scaling of y-axis).

Compared to typical weather, May 2001 was warm (cf. DB Netz AG and Lamparter) and November 2001 cold (cf. Fraunhofer ISE and Lamparter). Thus, there are comparatively high energy gains in these months because of the high temperature difference between the inlet air and ground, as the ground is – contrary to the air – cool in May and warm in November.

Each EAHX is integrated into a ventilation system. By approximation, there is either a heating or a cooling energy demand at a given ambient air temperature. Depending on the ambient temperature, the supply air should be heated or cooled. If the energy performance of an EAHX is sorted by the ambient temperature, the thermal performance at a specific temperature can be calculated. For better comparison, the mean energy performance is related to the total surface area in Fig. 3-7 (right, with same scaling of y-axis). Due to the smaller specific surface area, the specific energy performance at Fraunhofer ISE is clearly higher than at DB Netz AG or Lamparter which are similar to each other.
3.4.3 Energy supply sorted by ambient air temperature

The energy gain by an EAHX does not necessarily save heating or cooling energy. Sometimes, an EAHX cools down the incoming air at low ambient temperatures and heats the incoming air at high ambient temperatures. As this is contrary to the original aim of an EAHX, a control strategy is needed to supply energy only when it is actually needed. Starting from fictitious temperature limits for heating (12 °C) and cooling (22 °C) energy demand, the effectively usable energy supply can be estimated from Fig. 3-8:

- At DB Netz AG (only time control), only 82 % of heating energy is supplied below 12 °C and only 43 % of cooling energy above 22 °C.
If an EAHX is not only time-controlled but also temperature-controlled, more energy is supplied when it is actually needed. At Fraunhofer ISE (open loop control), 96% of heating energy is supplied below 12 °C and 72% of cooling energy above 22 °C.

At Lamparter (closed loop control without hysteresis), 88% of heat energy is supplied below 12 °C and 48% of cooling energy above 22 °C.

Monitored data from EAHX’s can be concisely evaluated with these curves (cumulative frequency distribution). During the design process, this diagram can illustrate data from a simulation in order to compare the temperature performance of different EAHX designs.

![Figure 3-8: Frequency distribution for heat and cooling energy supply.](image)

### 3.5 A model for comparison of efficiency

A parametric model is used to provide general efficiency criteria. Time variation curves for both inlet and outlet air temperatures and ground temperature $T(t)$ can be analysed by a regression function using a mean temperature $T_{\text{mean}}$, a temperature amplitude $\Delta T$ and a phase shift $t_\phi$:

$$T(t) = T_{\text{mean}} - \Delta T \cdot \sin\left(2\pi \cdot (t + t_\phi) / 8760\right)$$  \hspace{1cm} (3-3)

$T$ can be replaced by the inlet air $T_{\text{in}}$, the outlet air $T_{\text{out}}$ and the undisturbed ground temperature $T_{\text{ground}}$. As inlet and outlet air temperatures are known from measurements, Eq. (3-3) is used only for the undisturbed ground temperature.

The outlet air temperature $T_{\text{out}}$ can be calculated from the ground temperature $T_{\text{ground}}$, the inlet air temperature $T_{\text{in}}$ and the dimensionless NTU (number of transfer units). The temperatures are time dependent and NTU is a function of the operation time $t_{\text{op}}$:

$$T_{\text{out}}(t) = T_{\text{ground}}(t) + \left[ T_{\text{in}}(t) - T_{\text{ground}}(t) \right] \cdot \exp\left[-\text{NTU}(t_{\text{op}})\right]$$  \hspace{1cm} (3-4)

---

6 The "operation time” is the time during actual operation and starts at the operation begin, not the average daily operation time.
During operation, the ground near the pipe is discharged thermally. Starting from the undisturbed ground temperature, the effective temperature difference between earth and air decreases with operation time. As NTU takes both the thermal capacity of the ground and the heat transfer from the ground to the air into account, it is dependent on operation time.

NTU is the quotient of the convective heat transfer at the surface $A_{EAHX}$ (with the pipe diameter $d_{EAHX}$ and the length $l_{EAHX}$) and the heat transport. As the heat transfer between the ground and the air changes during the operation time, the heat transfer coefficient $h_{EAHX}$ is a function of the operation time. However, the thermodynamic model takes a constant NTU into account.

$$\text{NTU}(t_{\text{op}}) = \frac{h_{EAHX}(t_{\text{op}}) \cdot A_{EAHX}}{V_{\text{air}} \cdot c_{\text{air}}} = \text{const} \quad \text{with} \quad A_{EAHX} = \pi d_{EAHX} \cdot l_{EAHX} \quad (3-5)$$

These three formulas summarise the complex thermal interrelations from Fig. 3-1.

### 3.5.1 Energy performance of an earth-to-air heat exchanger

As the measured ground temperatures are obviously not undisturbed (Chapter 2.1), four parameters are unknown in Eq. (3-4) with Eq. (3-3): $T_{\text{mean,ground}}, \Delta T_{\text{ground}}, t_{\phi,\text{ground}}$ and NTU. Using the least-squares method, these parameters can be calculated using a non-linear regression analysis. The results are summarised in Table 3-3:

- The profiles of undisturbed ground temperatures are similar. However, due to the construction under the foundation slab, the undisturbed ground temperature at Fraunhofer ISE is higher and shows a shorter phase shift.
- As the specific surface area at DB Netz AG is higher (small mass flow rate), its NTU is higher than at Fraunhofer ISE. The NTU at Lamparter is smaller than that at DB Netz AG because of its comparatively large pipe diameter (small convective heat transfer coefficient).

<table>
<thead>
<tr>
<th>Parameter</th>
<th>DB Netz AG</th>
<th>Fraunhofer ISE</th>
<th>Lamparter</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_{\text{ground}}$ (°C)</td>
<td>12.8</td>
<td>13.8</td>
<td>11.9</td>
</tr>
<tr>
<td>$\Delta T_{\text{ground}}$ (K)</td>
<td>5.1</td>
<td>5.9</td>
<td>5.3</td>
</tr>
<tr>
<td>$t_{\phi,\text{ground}}$ (days)</td>
<td>55</td>
<td>21</td>
<td>50</td>
</tr>
<tr>
<td>Mean NTU</td>
<td>2.59</td>
<td>1.57</td>
<td>1.74</td>
</tr>
</tbody>
</table>

The main characteristic describing the energy performance is the overall heat transfer coefficient $h$. Using Eq. (3-5), the mean heat transfer coefficient can be calculated from the monitored air temperatures and air volume flow. The results are shown in Table 3-6: The overall heat transfer coefficient at Lamparter is smaller than at DB Netz AG and Fraunhofer ISE, though a better heat transfer has been expected at Lamparter than at DB Netz AG or Fraunhofer ISE due to the higher heat conductivity of the moist earth (cf. Table 3-1). Obviously, the convective heat transfer between the piping and air dominates the overall heat transfer from the ground to the air.

### 3.5.2 Temperature ratio for an earth-to-air heat exchanger

Using Eq. (3-3) for a regression analysis, the time variation curves can be specified by a mean air temperature, an air temperature amplitude and a phase shift, with the results listed in Table 3-4. Using the results from Table 3-4 and the earth temperature...
from Table 3-3, the temperature ratio $\Theta$ can be calculated in addition to the temperature ratio $R_T$

$$\Theta = \frac{T_{in} - T_{out}}{T_{in} - T_{ground}} \quad (3-6)$$

$\Theta$ is a “scale unit” for the temperature behaviour of an EAHX, see results in Table 3-6:

- Due to its large specific surface area, the temperature ratio $\Theta$ for the EAHX at DB Netz AG is higher than at Fraunhofer ISE.
- The temperature ratio $\Theta$ for the EAHX at Lamparter is smaller than at DB Netz AG though the specific surface area is similar: As the heat transfer is worse (small $h_{EAHX}$), the temperature ratio is smaller – in spite of a similar specific surface area.

Table 3-4: Regression analysis for inlet and outlet temperatures.

<table>
<thead>
<tr>
<th></th>
<th>DB Netz AG</th>
<th>Fraunhofer ISE</th>
<th>Lamparter</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>inlet</td>
<td>outlet</td>
<td>inlet</td>
</tr>
<tr>
<td>air temperature $T_{air}$</td>
<td>11.78 °C</td>
<td>12.80 °C</td>
<td>12.94 °C</td>
</tr>
<tr>
<td>amplitude $\Delta T_{air}$</td>
<td>8.00 K</td>
<td>5.27 K</td>
<td>11.21 K</td>
</tr>
<tr>
<td>phase shift $t_{\varphi,air}$</td>
<td>26 days</td>
<td>52 days</td>
<td>2 days</td>
</tr>
</tbody>
</table>

### 3.5.3 Thermal energy gain and mechanical dissipation energy

The energy gain of EAHX’s is associated with an energy demand for the fan. The energy efficiency of an EAHX is the ratio between its energy gain and the electricity demand for the fan to draw the air through the EAHX. As not the fan but the mechanical dissipation energy is a characteristic parameter of an EAHX, the coefficient of performance COP is calculated with the overall energy gain $[\text{kWh}\_\text{th}]$ supplied by the EAHX and the mechanical dissipation energy $[\text{kWh}\_\text{mech}]$ during operation time:

$$\text{COP} = \frac{\sum_{t_{\text{operation}}} (Q_{\text{heat}} + Q_{\text{cool}})}{\sum_{t_{\text{operation}}} (\Delta p \cdot V)} \quad (3-7)$$

The mechanical dissipation power at mean air-flow rate is 114 W at DB Netz AG, 322 W at Fraunhofer ISE and only 20 W at Lamparter. Due to the large pipe diameter and low air speed, the pressure loss at Lamparter is very small. Due to the slightly higher air speed and more bends, the pressure loss at DB Netz AG is higher. In contrast, the pressure loss at Fraunhofer ISE is much higher since the EAHX had to be integrated into the air-inlet resulting in more flow resistances. The results for each EAHX are shown in Table 3-6.

Though there are large differences in COP, it should be mentioned that each COP is high enough to save both end and primary energy, cf. Table 3-5. The energy saving can be calculated from the mechanical dissipation power and the thermal energy gains. Taking a typical fan efficiency of around 70 % into account, the end energy saving is 70 % smaller than the COP. Starting from this end energy saving and taking a primary energy conversion factor for electric energy (i.e. 3 kWh\_prim.energy/kWh\_end energy) into account, the primary energy saving is 1/3 of the end energy saving.
Table 3-5: Coefficient of performance COP with regard to the end and the primary energy demand.

<table>
<thead>
<tr>
<th></th>
<th>DB Netz AG</th>
<th>Fraunhofer ISE</th>
<th>Lamparter</th>
</tr>
</thead>
<tbody>
<tr>
<td>COPend energy</td>
<td>61.5</td>
<td>20.1</td>
<td>53.2</td>
</tr>
<tr>
<td>COPprimary energy</td>
<td>20.5</td>
<td>6.7</td>
<td>17.8</td>
</tr>
</tbody>
</table>

3.5.4 Characteristics of an earth-to-air heat exchanger

Of course, the usable heat and cooling energy supply is the most important result from the evaluation of the data. However, the energy supply is dependent on both the hours of operation and the climate. In order to characterise the thermal efficiency of an EAHX, the main characteristics of an EAHX should be independent of those changeable parameters. In addition to the energy gain in Table 3-2, Table 3-6 summarises the four main characteristics of EAHX’s from the previous Sections.

Table 3-6: Main characteristics for earth-to-air heat exchangers.

<table>
<thead>
<tr>
<th></th>
<th>DB Netz AG</th>
<th>Fraunhofer ISE</th>
<th>Lamparter</th>
</tr>
</thead>
<tbody>
<tr>
<td>temperature ratio RT</td>
<td>0.28 K_{cal}/K_{in}</td>
<td>0.47 K_{cal}/K_{in}</td>
<td>0.36 K_{cal}/K_{in}</td>
</tr>
<tr>
<td>heat transfer h_{mean}</td>
<td>5.5 W/(m²K)</td>
<td>5.0 W/(m²K)</td>
<td>3.2 W/(m²K)</td>
</tr>
<tr>
<td>efficiency ϕ</td>
<td>0.944 K_{real}/K_{ideal}</td>
<td>0.766 K_{real}/K_{ideal}</td>
<td>0.804 K_{real}/K_{ideal}</td>
</tr>
<tr>
<td>performance COP</td>
<td>88 kWhth/kWhmech</td>
<td>29 kWhth/kWhmech</td>
<td>76 kWhth/kWhmech</td>
</tr>
</tbody>
</table>

3.6 Long-term monitoring and short-term measurements

In each EAHX, the air temperatures were measured at distances of 10 m (DB Netz AG and Fraunhofer ISE) and 9 m (Lamparter) during some weeks. These short-term measurements can be used to validate the results from Table 3-6. Starting from Eqs. (3-4) and (3-5), the local air temperature at $x_{EAHX}$ can be calculated:

$$T_{\text{air}}(x_{EAHX}) = T_{\text{ground}} + (T_{\text{in}} - T_{\text{ground}}) \cdot \exp\left[-\text{NTU} \cdot \frac{x_{EAHX}}{l_{EAHX}}\right]$$

(3-8)

As the mean NTU is known, the local temperature variation should be predicted accurately: Fig. 3-9 shows exemplarily the results from the EAHX (segment west and east) at the Fraunhofer ISE building for a mean winter and summer day. Fig. 3-10 shows results from a comparison at a particular time for the EAHX at the DB Netz AG and the Lamparter building. The parameters for the calculation of the ground temperature and the mean NTU is taken from Table 3-3. Using the geometric data from Table 3-1 and the inlet air temperature and the air flow from measurements, the temperature variation is calculated by Eq. (3-8).

The measured temperature variation (short-term measurement) is predicted accurately by the calculation model (long-term monitoring: NTU from a yearly data analysis) under different conditions (summer/winter, geometric data, air flow or mean/current temperature). In other words, short-term measurements and long-term monitoring result in the same characteristic values at any time for each EAHX.
3.7 Conclusions

The thermal performance (temperature behaviour and energy efficiency) of earth-to-air heat exchangers EAHX’s has been calculated using four different approaches ($R_T$, $h_{mean}$, $\Theta$ and COP). But, which EAHX is the best concerning thermal performance?

The application of passive heating and cooling is a multilayered process and there is a wide range of design criteria and demands on ventilation systems. Thus, the evaluation of an EAHX is dependent on project-specific criteria. Each of the evaluated EAHX’s at DB Netz AG, Fraunhofer ISE or Lamparter is the best from a specific point of view:

- The EAHX at DB Netz AG narrows the outlet air temperature close to the undisturbed earth temperature.
- The EAHX at Fraunhofer ISE supplies the highest specific energy gain based on the total surface area.
- Taking a lower heat transfer into account, the EAHX at Lamparter has the highest COP.
The method for data evaluation of EAHX’s in operation gives some noteworthy considerations for design and operation of EAHX’s:

- The influence of earth parameters (soil and surface) and of the building on the earth temperature is as important as the pipe diameter on the thermal efficiency.
- In spite of a high heat flow density, the influence of the density of a pipe register on the energy gain is small: Obviously, thermal regeneration is good enough to prevent a reduced energy performance.
- Pipe lengths up to 100 m and pipe diameters around 250 mm are profitable. If the EAHX aims at a high specific energy performance, a small specific surface area should be reached using fewer pipes. If the EAHX aims at a high temperature ratio, a high specific surface area should be reached using more pipes.

However, the construction site itself often determines the dimensions of an EAHX.

In operation, the control strategy plays a decisive role for the actually usable energy supply by the EAHX. A temperature control is important to prevent unwanted heating in summer and cooling in winter. Of course, a better utilisation of energy supply is achieved by a closed loop control but its programming is difficult because of long dead times in EAHX’s. An open loop control runs robustly but usually its programming should be adjusted after the first year of operation, when the temperature behaviour is known. Though the operation of each evaluated EAHX could be improved, each EAHX supplies more heating and cooling energy than the primary energy it uses for fans.

The heating energy demand for ventilation is small if a heat recovery systems with high efficiency is operated (in this analysis: Lamparter). But in heat recovery systems with high efficiency there is the danger of freezing because the humid exhaust air is cooled down under the freezing point at very low inlet air temperatures. If an EAHX is operated in series with a heat recovery system with high efficiency it should be large enough to prevent freezing at low ambient air temperatures in any case.

An important characteristic for passive cooling applications is the temperature ratio $R_T$ which describes the temperature damping between inlet and outlet temperature. The smaller $R_T$ is, the more cooling energy is supplied to the building. As expected, the EAHX at DB Netz AG reaches the smallest $R_T$ due to the high specific surface area (high conductive heat transfer between soil and pipe) and the comparatively small pipe diameter (high convective heat transfer between pipe and air).

As assistance to night ventilation, each EAHX replaces an active mechanical cooling system in summer.

### 3.8 References


Merging Short and Long-Term Measurements

Part B

Though the monitoring of night ventilation provides information to estimate whether a building concept achieves a comfortable indoor climate or not, only a model-based data analysis responds to the question how the building thermally behaves. Building simulation is an appropriate tool not only for design studies but also for data analysis.

Monitored data from twelve offices are evaluated by a new method for data analysis based on building simulation which deals with short-term and long-term measurements and building simulation. The methodology to use simulation for data analysis can be described briefly in three steps:

1. Set-up of the model,
2. parameter identification based on short-term measurements and
3. analysis of the accuracy of the simulation with long term measurements.

The validated simulation model enhances the understanding of passive cooling systems in ordinary use and is used to optimise the operation of night ventilation in the building.

Fig. 4-1:
DB Netz AG
Hamm, Germany
completion: 1999
4.1 Introduction and building description

The office building of DB Netz AG in Hamm (Germany) has been designed and constructed as a low-energy office building for 190 occupants without air-conditioning [4-1]. The passive cooling concept aims at a high thermal comfort with a reduced electricity consumption for technical services. The passive cooling concept merges night cooling by free ventilation, an earth-to-air heat exchanger and hybrid day ventilation. Fig. 4-1 shows a view of the building, with the air inlet (earth-to-air heat exchanger) and the atrium.

Within the design process, thermal and daylight simulations were used to optimise the energy performance of the building [4-2]. The main objective of the design concept was to primarily use architectural solutions for ventilation, cooling and lighting of the offices and to minimise HVAC systems and artificial lighting. Adequate thermal insulation and moderate window dimensions are pre-requisites for a low heating and cooling energy demand. A central atrium serves as a buffer zone for solar energy gains during the winter. In summer, solar loads are minimised by efficient shading systems and cross ventilation in the atrium roof. In the offices, artificial lights are dimmed and external venetian blinds protect the building against the sun. Shading systems in the peripheral offices with separately adjustable blinds in the upper part of the windows still enable the use of daylight when the lower part is closed fully.

Fig. 4-2 gives an overview of the building, the location of the monitored offices and the ventilation scheme. The ventilation strategy depends on the ambient temperature (free night ventilation), operation time (mechanical day ventilation) and user behaviour (free day ventilation):

- **Day ventilation** uses both natural ventilation from the outside or the atrium and supply air from a mechanical ventilation in the offices. The multifunctional corridors are mechanically ventilated with supply air. The exhaust air is drawn from the corridor and is equal to the supply air to the offices and the corridor, such that the air flows from the offices to the corridor. This air flow is superposed or enhanced by the natural ventilation.

- **Night ventilation** is used to activate the thermal capacity of the bare concrete ceiling. Fresh air flows through opened flaps due to the stack effect caused by the atrium. In summer, the night ventilation is automatically activated by the building management system from 2 a.m. to 8 a.m.

The building’s energy performance was monitored using more than 260 sensors over two years with a time resolution of 5 minutes. Concerning the energy balance in the offices and their temperature behaviour the following data are taken from the monitoring: In twelve offices, the operative room and ceiling temperatures, the electricity consumption and the supply air flow and temperature are measured. Occupancy has not been monitored but correlates strongly to the working hours. Concerning the central building service, the energy demand for heating and ventilation and the air flow due to mechanical ventilation is monitored. The meteorological data and air temperatures in the adjacent atrium specify the boundary conditions of the monitored offices. The monitored data from the office building are complemented by information from the building management system about the position of louvers and the shading devices.
Merging Short and Long-Term Measurements

Fig. 4-2: Left: Floor plan and cross-section, with position of the monitored offices in the 3rd floor (from outside to atrium: 3A009, 3A024 and 3A014). Right: Typical ventilation scheme (dotted line: natural night ventilation; solid line: hybrid day ventilation) and position of the monitored sensors.

Table 4-1 gives the key information on the building and its energy consumption, climate and internal loads which result from measurements.

Table 4-1: Key building information on geometry and energy performance for 2001.

<table>
<thead>
<tr>
<th></th>
<th>gross volume</th>
<th>net floor area (DIN 277)</th>
<th>working hours (Mon – Fri)</th>
<th>daily internal load (mean)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>25,705 m³</td>
<td>5,974 m²</td>
<td>8 a.m. – 6 p.m.</td>
<td>6.4 W/m²</td>
</tr>
<tr>
<td>heating energy demand (design)</td>
<td>65 kWh/(m² a)</td>
<td>22.3 kWh/(m² a)</td>
<td>11.2 °C</td>
<td>2.56 kWh/(m²d)</td>
</tr>
<tr>
<td>electricity demand for technical services</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>ambient air temperature (mean)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>global solar radiation (horizontal, mean)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Fig. 4-3 shows the segmentation of the energy consumption into different categories. The total end energy consumption between April 2001 and March 2002 was 65.8 kWh/(m² a) for heating and hot water and 28.9 kWh/(m² a) for electricity. Using a primary energy conversion factor of 1.1 for natural gas and 3.0 for electricity, the total primary energy consumption is 159.1 kWh/(m² a). Though the primary energy consumption exceeds the limit of the funding programme (cf. Chapter 2.2.4), the primary energy consumption of the building is still 40 % below the primary energy consumption in comparable but conventionally designed office buildings (cf. Fig. 1-1).
Fig. 4-3: Total end energy consumption of the building between April 2001 and March 2002.

The night ventilation efficiency is analysed building simulation. Fig. 4-4 outlines the methodology of the conventional and the model-based data analysis.

Fig. 4-4: Methodological approach. Data from a long-term monitoring and from short-term measurements (drawn frame) can be analysed separately (dotted frame) or via building simulation. An accuracy analysis for this simulation model provides its area and period of validity. This simulation model can be used to calculate the night ventilation efficiency from an energy balance or to optimise its operation (bold text).

4.2 Short-term measurements and long-term monitoring

An extensive monitoring, that is described in detail in Ref. [4-3], has been carried out to evaluate the thermal building performance. In the following, the data analysis focuses on the temperatures in two rooms with special respect to the air flow in a large building with an open building structure and free cross ventilation.

4.2.1 Operative room temperature

A detailed short-term measurement of the air temperature concludes that the long-term monitored temperature (Pt 100 sensors mounted on the wall) is not the air temperature but almost identical to the operative room temperature. Fig. 4-5 (a) shows that the air temperature responds more dynamically to changes in boundary conditions than the temperature measured with the wall mounted sensors.

The effect of night ventilation in the peripheral office 3A009 is higher than in the office 3A014 which is located adjacent to the atrium, cf. Fig. 4-5 (b). Due to the strong cross-ventilation, the peripheral room benefits from the cool ambient air. If the external flaps are closed, office 3A014 will be cooled by single-sided ventilation from the atrium and cooling by night ventilation will stop in office 3A009.
Merging Short and Long-Term Measurements

The results from one year of monitoring (April 2000 – March 2001) have been analysed regarding the operative room temperature. Fig. 4-6 shows the operative room temperature in the 3A009 office sorted by the ambient air temperature. Only with ambient air temperatures above 26 °C, the operative room temperature strongly correlates with the ambient air temperatures. Below 26 °C, the user behaviour concerning ventilation and shading and internal and external loads have a stronger effect on the indoor temperature.

The annual temperature distribution during office hours (Monday to Friday from 8 a.m. to 6 p.m.) can be used as a comfort criterion: The percentage with operative room temperatures above 25 °C should be lower than 10 % of working hours according Chapter 2.1.3. Fig. 4-6 shows the cumulative frequency distribution for operative room temperature in three offices: The peripheral office (3A009), the multifunctional corridor (3A024) and the internal office located adjacent to the atrium (3A014). The results show acceptable indoor climate conditions, though they do not match the comfort criteria strictly. The operative room temperature exceeds 25 °C in 3A009 for 280 h and in 3A014 for 400 h of 2,600 working hours.
The tracer gas technique with the concentration decay method is used to measure air leakage rates. The aim is to gain more detailed information on air-flow rates and flow paths in different operating states. Fig. 4-7 gives an overview of different air flow patterns during day and night from some measurements and flow visualisation by smoke: Depending on the operation of the mechanical ventilation system and the door position, either cross-ventilation or single-sided ventilation through windows dominates the air change rate. In principle, the air change is higher during the night than during the day (stack effect) and with open rather than closed doors (small flow resistance).
Merging Short and Long-Term Measurements

hybrid day ventilation

doors closed

high single-sided air exchange,
cross-ventilation approx. 1 h\(^{-1}\),
air exchange in atrium approx. 2 h\(^{-1}\)

doors opened

small single-sided air exchange,
cross-ventilation above 2 h\(^{-1}\),
air exchange in atrium approx. 2 h\(^{-1}\)

free night ventilation

high single-sided air exchange,
small free cross-ventilation,
air exchange in atrium approx. 3 h\(^{-1}\)

very small single-sided air exchange,
very high free cross-ventilation,
air exchange in atrium approx. 4 h\(^{-1}\)

Fig. 4-7: Ventilation pattern from air change measurements and visualisation of air-flow paths by smoke. Cross-ventilation is marked by simple arrows and single-sided ventilation by flow chart arrows. The size of the arrow qualitatively indicates the air change rate.

The first experiment is during the day: The hybrid ventilation in office 3A009 is evaluated by measurement with or without mechanical ventilation and with or without natural ventilation (flaps). Mechanical and natural ventilation reinforce each other, as shown in the results listed in Table 4-2.

Table 4-2: Air change rates measured by SF\(_6\) tracer gas concentration decay method in August 2000 and July 2001, wind speed remains below 5 m/s during each experiment.

<table>
<thead>
<tr>
<th></th>
<th>flaps closed</th>
<th>flaps open</th>
</tr>
</thead>
<tbody>
<tr>
<td>no mechanical ventilation</td>
<td>0.3</td>
<td>2.0</td>
</tr>
<tr>
<td>with mechanical ventilation</td>
<td>1.2</td>
<td>4.1</td>
</tr>
</tbody>
</table>

The second experiment is for night ventilation: The flaps in all rooms, at the bottom and the top of the atrium are opened. Some of the doors between the offices and multifunctional corridors are open, some are closed. The measured air flow [m\(^3\)/h] is given in Fig. 4-8. The flow network is characterised by two main properties: Due to the reduction of the hydraulic resistance, the air change in rooms with open doors is clearly higher than in rooms with closed doors. The air change rate decreases with the vertical position of the room in the building, because the stack effect is reduced when the buoyant force is smaller.

Detail information on all air change measurements is given in Ref. [4-3]. These measurements are used to set up and to validate the air-flow network for the building simulation.
4.3 Data evaluation by simulation

An adapted simulation is performed to quantify the cooling capacity of night ventilation and to improve the understanding of the thermodynamic phenomena. The aim is to derive indications

- on data evaluation from buildings in service,
- on uncertain input parameter of simulations in the design phase,
- on the optimisation potential of passive cooling and
- on the necessary complexity required for accurate simulation tools.

The aim is not primarily to achieve an accurate result for every time step but to model excellently the user behaviour and the thermal building performance for the whole summer period under changing boundary conditions.

For deeper analysis, a summer period from July 16 to August 12, 2001 is chosen, which represents different meteorological conditions concerning ambient temperatures, wind and solar radiation.

4.3.1 Set-up of the model

A detailed simulation of a cross-section of the building is performed using ESP-r [4-4]. As the free night ventilation and the hybrid day ventilation strongly influence the thermal behaviour of the offices, all air volumes connected with the simulated rooms 3A009, 3A024 and 3A014 are taken into account. The simulation scheme is shown in Fig. 4-9.
4.3.2 Sensitivity analysis and parameter identification

The measured internal loads, room / air temperatures of the adjacent rooms (from Fig. 4-2) and meteorological conditions are used as boundary conditions. Air flow is modelled by an air-flow network with 19 air flow nodes and 36 air flow connections.

Due to the lack of information on whether the doors and windows are open or closed, the user behaviour concerning free ventilation is derived from an analysis of measured temperatures to identify data sets with high temperature gradients: A high gradient usually accompanies the opening or closing of windows and doors.

Starting from this information, the influence of uncertain input parameters (such as time of occupancy, thermal properties of walls and windows, discharge coefficients of openings, g-value of glazing, reduction factor of shading devices and heat transfer coefficients) is identified by a sensitivity analysis.

The uncertain values from this sensitivity analysis are calculated by parameter identification of measured and simulated operative room temperatures. Fig. 4-10 shows the results from the parameter identification of 12 parameters for office 3A009 as an example. The value for optimisation is the arithmetic mean value of the operative room temperature (shown on the left axis) and the ceiling temperature (not shown in this figure). A robust parameter set is defined by identity of simulated and monitored temperature (target value) and a small deviation. As shown on the right axis in Fig. 4-10, a good approximation is reached after 20 simulation runs since bias and deviation are small and do not change from one simulation step to the next. Taking this set of optimised parameters as an input, there is a sufficient correlation between measurement and simulation (mean value) but shows some non-systematic inaccuracies (deviation) which are discussed below.
Fig. 4-10: Results of a parameter identification for 12 input parameters: a) Optimisation procedure with 20 simulation runs for office 3A009. b) Measured and simulated operative temperatures in offices 3A009 and 3A014. Despite some uncertainties in simulation and measurement, the operative temperatures during operation hours agree well. The period with increasing ambient temperature and solar radiation is modelled very well with both long term and short term characteristics.

4.3.3 Accuracy analysis of the simulation model

The measured operative room and ceiling temperatures are used to quantify the differences between measurement and simulation. The validated simulation model is checked in two ways: A Monte Carlo-simulation analyses how uncertain input parameters affect the simulation results. A Fourier analysis shows whether the thermal behaviour is simulated accurately or not.

Monte Carlo-simulation

A Monte Carlo-simulation [4-5] on 30 input parameters – among which are the 12 uncertain parameters from the sensitivity analysis – is performed for 300 simulations.\(^7\)

\(^7\) The input parameters are statistically distributed around the true mean value according to the Gauß error standard distribution. A good approximation to the Gauß bell-shaped curve is reached after 300 simulations runs.
A Monte Carlo-simulation provides an uncertainty analysis with consideration of the interaction between all varied parameters.

In this Chapter, only the building physical properties and parameters are varied in order to prepare an accurate building model, but not the building use or the user behaviour.\(^8\) The Gauß error distribution curve for the Gaußian variable \(x\) is defined by

\[
p(x, \bar{x}, \sigma) = \frac{1}{\sqrt{2\pi \cdot \sigma^2}} \cdot e^{-\frac{(x-\bar{x})^2}{2\sigma^2}} \text{ with the expected value } \bar{x} = x_m\]  

(4-1)

with the average value \(x_m\) and its standard deviation \(\sigma\). The thermal properties of walls (conductivity \(\lambda\) and heat storage capacity \(c_p\), the discharge coefficients \(c_D\) of the skylights and windows to the outside and the atrium, the discharge coefficients \(c_D\) of the flaps at the bottom of the atrium and the smoke-and-heat outlets at the top of the atrium, the supply air-flow in the offices and the corridor and the exhaust air-flow from the corridor and the heat transfer coefficients at the ceiling in each room. Besides the building physical properties, the time of occupancy is uncertain and is also considered as statistically distributed input parameter. All other values (e.g. g-value, status of blinds or time of window / flap opening) are known from the monitoring equipment or have been determined during the model set-up and by the sensitivity analysis.

The deviation of the input parameters was chosen in order to achieve a specific deviation on the three main building parameters \(C\), \(H\) and \(G\) according to Chapter 2.2.1. The building capacity \(C\) is varied in the range of 25 \%, the heat loss coefficient \(H\) of 20 \% and the specific heat gain \(G\) of 20 \%. The variation is given within the 2\(\sigma\)-boundaries.\(^9\) Due to the limits given by the three building parameters, the deviation parameter of the 30 single parameters is within typical limits. Thereby, the room temperature does not take extreme values caused by unrealistic parameter sets.

The results for each simulation (672 hours) are analysed by the difference \(\Delta T_{\text{MCS}}\) of the mean temperature for the whole period of comparison and the deviation \(\sigma_{T_{\text{MCS}}}\) for each time step:

\[
\Delta T_{\text{MCS}} = \frac{1}{672 \text{hrs}} \left( \sum_{t=1}^{672} T_{\text{monitoring}}(t) - \sum_{t=1}^{672} T_{\text{simulation}}(t) \right) \]  

(4-2)

\[
\sigma_{T_{\text{MCS}}} = \sqrt{\frac{1}{672} \sum_{t=1}^{672} (T_{\text{monitoring}}(t) - T_{\text{simulation}}(t))^2} / 672 \text{hrs} \]  

(4-3)

As each simulation results in a mean operative room temperature and a deviation between simulation and measurement, the results can be statistically analysed whereby the mean values for \(\Delta T_{\text{MCS}}\) and \(\sigma_{T_{\text{MCS}}}\) from 300 simulation runs are (nearly) identical to the basic simulation from the parameter identification. (The results would be exactly identical, if the symmetric variation of input parameters resulted in a symmetric response function.)

The results from the 300 simulations are shown in Fig. 4-11 and summarised in Table 4-3. Thereby, \(\Delta T_{\text{MCS}}\) characterises the overall energy balance and \(\sigma_{T_{\text{MCS}}}\) the accuracy of the simulation.

\[\Box\] The mean bias \(\Delta T_{\text{MCS}}\) is in the peripheral office 3A009 (0.05 K) smaller than in the internal office 3A014 (0.47 K). However, the variation within the 2\(\sigma\)-boundaries of

\[\Box\]

8 The user behaviour is considered by a statistical simulation in Chapter 8.

9 This variation corresponds to the uncertainties which are discussed in Chapter 2.2.4.
the mean temperature is in 3A009 (1.16 K) much higher than in 3A014 (0.29 K). The peripheral office is affected directly by the meteorological conditions while the internal office is mainly in thermal contact with the atrium which is partly affected by the outdoor climate and partly by the offices. Accordingly, the simulated mean room temperature is more precise for the peripheral office.

The standard deviation $\sigma_{T_{MCS}}$ of the room temperature is in the range of 0.4 to 0.45 K for both the peripheral and the internal office 3A014. The temperature deviation is higher in peripheral office 3A009 than in the internal office since the impact of the environment and its variation is stronger.

**Conclusion.** The stronger the impact of the environment on the indoor climate, the less sensitive are the input parameters (thermal long-term behaviour) but the more inaccurate is the simulation (thermal short-term behaviour).

![Fig. 4-11: Results from a Monte-Carlo simulation for the operative room temperature (bold lines) and ceiling temperature (thin lines) in two offices: a) Simulated mean temperature according to Eq. (4-2). Though the input parameters varies in the order of 20 %, the mean room temperature varies (only) around 0.3 K. In spite of the normal distribution of all input parameters, the results are not normally distributed. b) Standard deviation between measurements and simulation according to Eq. (4-3). The average variation around the “true” measurement is approximately 0.4 K. Obviously, the monitored ceiling temperature does not correspond to the simulated. This can be attributed to the sensor position. If the sensor is deeper in the ceiling, it cannot quantify the short-term fluctuation at the surface which results in a relatively high deviation, even if the mean value is identical (long-term fluctuations).](image-url)
Merging Short and Long-Term Measurements

Table 4-3: Results from a Monte-Carlo simulation for the operative room temperature in two offices. Simulation results are given for the mean value and the variation within the $2\sigma$-boundaries.

<table>
<thead>
<tr>
<th></th>
<th>mean value [°C] (measurement)</th>
<th>mean value [°C] ($\Delta T^{\text{MCS}}$ simulation)</th>
<th>deviation $\sigma^{\text{MCS}}$ [K] (meas – sim)</th>
</tr>
</thead>
<tbody>
<tr>
<td>room temperature – 3A009</td>
<td>24.82</td>
<td>24.81 – 24.87 – 24.97</td>
<td>0.39 – 0.42 – 0.48</td>
</tr>
<tr>
<td>room temperature – 3A014</td>
<td>25.27</td>
<td>25.63 – 25.74 – 25.92</td>
<td>0.40 – 0.40 – 0.42</td>
</tr>
</tbody>
</table>

Fourier analysis

A Fourier analysis is performed on both measured and simulated temperatures in order to quantify the building’s dynamic response and provides the frequency spectrum of a time data set for the operative room and the ceiling temperatures. The application of Fourier analysis for heat transfer phenomena is described by Keller [4-6].

If the frequency spectrum of simulated data agrees with the frequency spectrum of measured data, the overall energy balance and its variation in time is calculated accurately, cf. Chapter 2.1.3 for Fourier series. The results for the office 3A009 in Fig. 4-12 show a good correlation of measurements and simulation of both the operative room and the ceiling temperature for the long period. Table 4-4 shows that the short-term dynamics agree better for the operative room than for the ceiling temperature.

Table 4-4: Results of a Fourier analysis of the measured and simulated operative and ceiling temperatures in office 3A009.

<table>
<thead>
<tr>
<th></th>
<th>operative room temperature</th>
<th>ceiling temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>measurement</td>
<td>simulation</td>
</tr>
<tr>
<td>mean value [°C]</td>
<td>25.0</td>
<td>25.1</td>
</tr>
<tr>
<td>amplitude (672 h)</td>
<td>1.7 K</td>
<td>1.9 K</td>
</tr>
<tr>
<td>phase shift</td>
<td>132 h</td>
<td>139 h</td>
</tr>
<tr>
<td>amplitude (24 h)</td>
<td>0.6 K</td>
<td>0.6 K</td>
</tr>
<tr>
<td>phase shift</td>
<td>2.3 h</td>
<td>0.3 h</td>
</tr>
</tbody>
</table>
Summary

The results from setting up of the simulation model, a parameter identification and the accuracy analysis using Monte-Carlo simulation and Fourier analysis can be summarised as follows: The simulation does not match the measurement at every time step but describes the building parameters – including user behaviour – well.

The uncertainty analysis shows, that even high uncertainties in separate building physical properties has a low influence on the room temperature. Obviously, the thermal building behaviour is mostly affected by the building layout and its HVAC concept and the climate. The user’s influence on the thermal building behaviour is discussed in Chapter 8.

4.3.4 Energy balance from the validated simulation model

Starting from this validated model, the heat gains and losses are calculated. In this Subchapter the energy balance is drawn for the office 3A009. Fig. 4-13 gives an overview of heat gains and losses for the period from July 16 to August 12, 2001:

- Internal heat gains are higher than solar heat gains.
- The most energy is discharged by hybrid day ventilation including cooled supply air from the earth-to-air heat exchanger. A third of the heat removal is caused by the free night ventilation.
- Depending on the temperature difference between inside and outside, and the solar radiation on the external wall, there is either a heat gain or loss from the external wall. On average there is heat removal through the external wall, which is about half of the heat removal caused by the free night ventilation.
Fig. 4-13: Energy balance for heat gains, losses and storage in office 3A009 from July 16 to August 12, 2001.

Furthermore, the validated simulation model can be used to derive general conclusions concerning the thermal building performance. Fig. 4-14 shows exemplarily the influence of wind speed on the infiltration and of the use of shading devices on the solar heat gains.10

On the one hand, there is a weak correlation between infiltration and wind speed at a certain time, because cross-ventilation is strongly dependent on user behaviour (i.e. opening of doors, windows and flaps) which varies from time to time. On the other hand, a correlation on wind speed can be derived for the DB Netz AG building: If the wind speed rises from 2.5 to 4.5 m/s, infiltration will be doubled for both hybrid day ventilation and free night ventilation.

The daily solar heat gains are strongly dependent on the automatic shading control by the building management system and on the user behaviour (manual control). Two characteristic ranges can be defined: On cloudy days with less than 4 kWh/(m² d) solar radiation, the daily solar heat gains show a large variation since blinds are closed / opened often. On days with very low solar radiation, the actual solar heat gain is close to the solar heat gain with permanently open blinds. On sunny days with more than 8 kWh/(m² d) solar radiation, the blinds are closed more often by the building management system or by the occupants. On days with very high solar radiation, the actual solar heat gain is close to the solar heat gain with permanently closed blinds.

10 As the ventilation and sun-shading in the DB Netz AG building is automatically controlled (with manual override), these correlations are different from the Lamparter, the Pollmeier and the Fraunhofer ISE building.
4.4 Optimisation of night ventilation

The validated and well tested simulation model can be used to optimise the night ventilation efficiency. Only existing technical devices and facilities of the building management system may be used to achieve an optimised control strategy. Two strategies will lower the level of operative temperature. Results are given in Table 4-5.

The strategy optimised night ventilation is to control the flaps at the top of the atrium by the operative room temperature in the multifunctional zone 3A024 (set point: 20 °C, 10 p.m. – 7 a.m.) and the flaps at the bottom of the atrium by the air temperature in the atrium (set point: 18 °C, 4 a.m. – 8 a.m.). Additionally, the windows are closed automatically when the ambient temperature exceeds 26 °C. No change is made in the shading control, and no mechanical ventilation is used. This will halve the number of hours with T > 25 °C even without mechanical ventilation, because the heat gains are reduced and the building is cooled down stepwise and effectively.
The strategy additional mechanical ventilation is to control the night ventilation only according to the time without manual involvement. Additionally, the mechanical ventilation with cool supply air from the earth-to-air-heat exchanger is used at operative room temperatures in the multifunctional corridor 3A024 above 24 °C. The maximum temperature of the internal office 3A014 can be significantly reduced due to the direct cooling effect, whereas the mean temperature in the peripheral room remains higher. Obviously, the effect of free ventilation is diminished: During the day time, the free natural ventilation effect is partly disturbed by the mechanical ventilation.

In addition to the results in Table 4-5, Fig. 4-15 shows a comparison of the two strategies for office 3A009. (The effect of the control strategies on the operative room temperature in the internal office 3A014 is not shown in this figure.)

Table 4-5: Classification of temperatures for different night ventilation strategies (16/07/2001 – 12/08/2001; 672 h).

<table>
<thead>
<tr>
<th>Room</th>
<th>hrs. &gt; 24 °C</th>
<th>hrs. &gt; 25 °C</th>
<th>hrs. &gt; 26 °C</th>
</tr>
</thead>
<tbody>
<tr>
<td>night ventilation (base case)</td>
<td>3A009 471</td>
<td>3A024 630</td>
<td>3A014 654</td>
</tr>
<tr>
<td>optimised night ventilation</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>additional mechanical ventilation</td>
<td>3A009 324</td>
<td>3A024 447</td>
<td>3A014 495</td>
</tr>
<tr>
<td>ambient air temperature</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Fig. 4-15: Comparison of three different control strategies for night ventilation in office 3A009.

4.5 Conclusion

The building is designed to achieve moderate summer indoor conditions. The percentage of working hours with operative temperatures above 25 °C is around 10 %. Passive cooling by free night ventilation improves the thermal comfort without increasing the electricity demand for mechanical cooling devices.

As shown by a comparison between monitored data and simulation, simulation tools can successfully contribute to the design of large buildings with a free ventilation concept and passive cooling by night ventilation. Hybrid ventilation strategies have to
be implemented carefully in order to avoid disturbance of the natural ventilation by additional, mechanically driven air flows.

The presented methodology for data evaluation using simulation allows a deeper insight into the efficiency of night ventilation (e.g. air flow patterns, user behaviour, building parameters, heat transfer, energy balance) and gives the possibility to improve ventilation strategies. The validated building simulation can be used to optimise the operation of night ventilation. The optimised strategy has been implemented at the DB Netz AG building successfully.

4.6 References


5 Night Ventilation Experiments

Part B

The previous Chapter showed, that building simulation is a sophisticated tool which can be advantageously used for a model-based data analysis from a scientific point of view. However, the large number of input parameter and the huge time exposure make this approach inapplicable for practical use, i.e. quality-assurance measurements for take-over, straightforward building analysis or re-definition of building parameters during the implementation of the technical devices and the building management system. A simplified parametric model is introduced in this Chapter and is validated against experiments and building simulation.

The aim is to identify characteristic building parameters G, H and $\tau$ from Chapter 2.2.1 and to determine the night ventilation effect with these parameters. Therefore, experiments were carried out in two offices in order to determine the efficiency of night ventilation, depending on the air change rate, solar and internal heat gains. The experiments (one room with and one without night ventilation) are evaluated by using both a parametric model and the building simulation. Both models are merged in order to develop a method for data evaluation in office buildings with night ventilation.

At the new institute building of Fraunhofer ISE, both mechanical and free night ventilation is used for passive cooling of the offices. The results from long-term monitoring show that room temperatures are comfortable even at high ambient air temperatures.

Fig. 5-1: Fraunhofer ISE
Freiburg, Germany
completion: 2001
5.1 Introduction and building description

The new building of Fraunhofer ISE contains laboratories and offices and was designed with the goal of combining a high-quality workplace and good functionality with low energy consumption, integration of solar energy systems and an aesthetically pleasing design. Thermal insulation, protection against the sun, illumination and ventilation engineering are all tailored to a minimal energy demand. The energy supply is based on a gas-powered combined heat and power plant CHP. The waste heat serves both to heat the building and to supply the air conditioning for the laboratories via an absorption chiller (cogeneration/cooling system).

Fig. 5-1 shows a view of the building and its façade concept. For more information about the building, the reader is referred to [5-1]. The current use of the building for applied research inevitably results in a high energy demand. Therefore, the energy demand is reduced by both the building design and operation management. Table 5-1 gives the key building information.

Table 5-1: Key building information.

<table>
<thead>
<tr>
<th>gross volume</th>
<th>net floor area</th>
<th>working hours</th>
<th>occupants</th>
</tr>
</thead>
<tbody>
<tr>
<td>64,320 m³</td>
<td>13,150 m²</td>
<td>Mon – Fri 8 a.m. – 6 p.m.</td>
<td>approx. 300</td>
</tr>
<tr>
<td>U-value (mean)</td>
<td>solar gain factor g, (with and w/o shading)</td>
<td>heating energy demand (design)</td>
<td>electricity demand for technical services</td>
</tr>
<tr>
<td>0.43 W/(m² K)</td>
<td>0.51 / 0.11</td>
<td>41.2 kWh/(m² a)</td>
<td>20.0 kWh/(m² a)</td>
</tr>
</tbody>
</table>

The energy use of the technical facilities, meteorological data, the information on energy flows in the offices (electricity demand, sun-shading, occupancy and open/close contacts) and the room temperatures are monitored in a long-term measurement programme and are complemented by short-term measurements. This data analysis deals only with room temperatures in the offices in one of the three wings, cf. arrows in Fig. 5-2. The floor area is 18.4 m², the external wall area 12.4 m² and the gross air volume of the room 60.6 m³. Differing from Table 5-1, the mean U-value of the office facade is 0.7 W/(m² K).

Fig. 5-2: Floor plan. The building, which largely comprises three storeys with an additional basement, links three parallel members with one development axis and the adjacent complex. All offices in building wing C face south and the laboratories face north. The two offices investigated are on the ground floor.
Fig. 5-3 outlines the ventilation system. The laboratories (facing north) are mechanically ventilated and must have an air supply separate from the corridor for safety-at-work reasons. Due to the simultaneous use of laboratories and offices (facing south), the offices cannot be ventilated by free cross-ventilation. Thus, there is a need for mechanical ventilation which is realised only with exhaust air. The ventilation system ensures a minimum air change rate of 1 h⁻¹ during working hours. A heat recovery system recycles the waste energy to the supply air for the laboratories in winter. During summer nights, the air change rate in the offices is increased to 5 h⁻¹, if the corridor is 2 K warmer than the ambient air. As the air is well mixed during night ventilation, the corridor temperature represents the mean temperature on the corresponding floor.

![Ventilation system diagram](image)

Fig. 5-3: Ventilation of the offices with heat recovery in winter and night ventilation in summer (openable skylight). The air change can be intensified by single-sided ventilation (windows).

### 5.2 Investigations on night ventilation efficiency

The potential of night ventilation potential to improve comfort has mainly been investigated by numerical means. Santamouris [5-2] and [5-3] introduced an integrated method to calculate the energy contribution of night ventilation techniques to reduce the cooling load of a building. Kolokotroni [5-4] investigated how free and mechanical night ventilation reduce the required plant capacity and the energy consumption in air-conditioned office buildings. In consideration of these studies, this Chapter focuses on a full-scale experiment and its evaluation.

#### 5.2.1 Literature review

The experiments, which are evaluated in this Chapter, have been designed against the background of already realised experiments:

- Kolokotroni et al. [5-5] uses temperature/humidity charts for data analysis of results from a monitoring programme and from a simulation in order to generate a pre-design tool for summer cooling with night ventilation for office buildings in moderate climates.

- Geros et al. [5-8] carried out an experimental evaluation of night ventilation in four different buildings. Additionally, simulation investigations are used to determine how the air change rates, the building construction and the climatic parameters affect the night ventilation. The data evaluation deals with the nocturnal air change rates and the indoor air temperature.
Givoni [5-7] carried out experiments in light-weight and high-mass buildings in order to determine the effectiveness of night ventilation and the building’s thermal inertia in lowering the indoor daytime temperatures. A simple model for the daytime maximum temperature is derived from an extensive data analysis but does not take the user behaviour or internal heat gains into account.

Blondeau et al. [5-8] carried out full-scale measurements in a three-storey office building. The data analysis deals with both comfort criteria and energy balance to characterise the building potential for night ventilation whereby the modelling of heat transfer coefficients is difficult.

5.2.2 Design of experiments and data analysis

In two offices, experiments were carried out in order to determine the efficiency of night ventilation, depending on the air change rate, the ambient air temperature, solar and internal heat gains.

**Result analysis.** The night ventilation efficiency can be quantified by the thermodynamic cause (energy balance) and its effect (room temperature). This Chapter evaluates the night ventilation efficiency on the basis of the reduction of the room temperature and the heat dissipation by night ventilation. The experiments are evaluated by using both the parametric model and building simulation.

**Data analysis with the parametric model.** The parametric model (from Chapter 2.2.1) deals only with three building parameters G, H and \( \tau \) and limited additional boundary data. This model takes only steady periodic conditions into account. While the parametric model agree adequately with the measurements for a typical cycle period (24 hours), it does not agree well at each time step.

**Data analysis with building simulation.** As the heat flows are calculated numerically, the building simulation can take transient conditions into account. Thus, the building simulation match the experiments accurately at each time step and the night ventilation effect is calculated accurately in combination with various other effects. As the simulation program deals with many input parameters, the results must be interpreted. Only universal conclusions provide a concise description of the night ventilation efficiency.

**Generalisation of results.** As a basic principle, results from experiments in buildings cannot be reproduced: As the heat storage of the building is a transient phenomenon, the building is never in the same thermal condition it was before, even if heat gains and losses do not change. Using the building simulation, the (transient) measurements can be transferred into a (periodic) steady-state model.

**Development of simplified data models.** While the parametric model provides representative factors, the building simulation deals with all complex interrelations. With the parametric model, thermal building characteristics can be deduced from the simulation results. In other words, the vast number of input parameters and interactions in the building simulation is reduced to a manageable number of characteristic parameters.

**Comparison with long-term monitoring.** The data analysis from the experiments is compared with the results from the long-term monitoring. If the same results can be drawn either from a very detailed experiment with many information about the boundary conditions or from operational measurements without detailed information about the boundary conditions, the data analysis can be used straightforward even for very simple monitoring campaigns.
Methodical approach. A multilayered model comparison with many separate aspects provides a consistent method for data analysis. Fig. 5-4 shows how measurements are analysed. As this Chapter focuses on the validation of this approach, the flow sheet focuses on data analysis. However, this approach may be applied to the design of model-based control systems in future. In the context of this thesis, the aim is to analyse data from a long-term monitoring based on the parametric model.

![Diagram of data analysis process](image)

Fig. 5-4: Methodological approach. Data from a long-term monitoring or from short-term measurements can be analysed separately (dotted frame) or combined (dashed line). The aim is to analyse rough data from a long-term monitoring directly with the parametric model or to identify the main building parameters from short-term measurements (bold text). This Chapter (italic text) links measured data to the building simulation and to the parametric model and harmonises the models (dotted line).

5.3 Data analysis of long-term monitoring

Ordinary monitoring systems record usually only the room temperature in the offices. The aim is to interpret these temperatures with a minimum information on the boundary conditions. This data analysis motivates the design of the experiments and serves as comparison between the detailed experiments and the rough monitoring.

Mean room temperature during the year

The results from monitoring the room temperatures in 21 offices during summer 2002 show that room temperatures exceeded 25 °C for less than 8 % of the working hours, even at high ambient air temperatures. Fig. 5-5 shows the mean room temperatures in the offices in the ground floor, the 1st floor and the 2nd floor (subset of 21 offices) for the whole year 2002. Due to the thermal stratification, the room temperature increase by 0.5 K from one floor to the next.
The operative room temperatures were evaluated not only for the mean room temperature but in all 21 offices in wing C: Table 5-2 shows the percentage rate of working hours with temperatures over 25 °C for each office. In comparison with Fig. 5-5, the room temperature exceeds 25 °C less frequently in some rooms and more frequently in others. This can be attributed to the user behaviour, too.

Table 5-2: Percentage rate of working hours [hrs of 2,600 h] in 2002, when operative room temperature exceeds 25 °C. All rooms in wing C from West “-A” to East “-I”, see Fig. 5-2. In the underlined rooms, the room temperature exceeds more or less often 25 °C than shown in Fig. 5-5.

<table>
<thead>
<tr>
<th>X</th>
<th>-A</th>
<th>-B</th>
<th>-C</th>
<th>-D</th>
<th>-E</th>
<th>-F</th>
<th>-G</th>
<th>-H</th>
<th>-I</th>
</tr>
</thead>
<tbody>
<tr>
<td>RT_C3X</td>
<td>10.1%</td>
<td>10.6%</td>
<td>14.2%</td>
<td>9.5%</td>
<td>11.2%</td>
<td>8.6%</td>
<td>11.3%</td>
<td>10.0%</td>
<td></td>
</tr>
<tr>
<td>RT_C2X</td>
<td>5.4%</td>
<td>7.4%</td>
<td>5.3%</td>
<td>6.3%</td>
<td>6.1%</td>
<td>8.6%</td>
<td>8.3%</td>
<td>8.5%</td>
<td>8.7%</td>
</tr>
<tr>
<td>RT_C1X</td>
<td>8.8%</td>
<td>10.8%</td>
<td>7.2%</td>
<td>4.9%</td>
<td>not used</td>
<td>not used</td>
<td>not used</td>
<td>not used</td>
<td></td>
</tr>
</tbody>
</table>
Night Ventilation Experiments

Mean room temperature during the summer

Fig. 5-6: Operative room temperature sorted by ambient air temperature for the working hours during the summer period June 1 – August 31, 2002.

Mean room temperature and night ventilation

Fig. 5-7: Variation in time of the mean operative room temperature in the offices for the working days between June 1 and August 31, 2002.

Mean room temperature and night ventilation

Due to an error in the building management system, the night ventilation was used only between July 16 and August 31, 2002 during the summer period. Thus, the operative room temperatures can be separated into groups with and without night ventilation (June 1 to July 15, 2002). Fig. 5-8 shows the mean value of 4 room temperatures: Night ventilation reduces the mean room temperature by 0.8 K during the working hours.
Energy gain and auxiliary energy

Passive cooling concepts compete against mechanical cooling machines for an energy-efficient cooling strategy. The coefficient of performance COP is the ratio of the utilised thermal energy to the input of electric energy during night ventilation. In night ventilation concepts, the “utilised” energy is the dissipated heat during the night.

$$\text{COP}_{\text{NV}} = \frac{\int_{\text{NV start}}^{\text{NV end}} V_{\text{air}}(t) \cdot c_{\text{p,air}} \cdot (T_{\text{room}}(t) - T_{\text{ambient}}(t)) \, dt}{\int_{\text{NV start}}^{\text{NV end}} P_{\text{electric, fan}}(t) \, dt}$$  \hspace{1cm} (5-1)

The mean COP was 4.5 kWh_{th}/kWh_{el} in 2002. As the electric energy demand for ventilation is quite high 0.46 W/(m³ h⁻¹), the COP for night ventilation is quite low. Unfortunately, the reason for the high energy consumption has not been identified yet. With a typical energy demand for exhaust ventilation lower than 0.2 W/(m³ h⁻¹), the COP would increase to 10 kWh_{th}/kWh_{el} without the need for additional ventilation.

A direct system comparison between air-conditioning and night ventilation not easy: Air-conditioning operates during the day together with the regular ventilation and night ventilation off-time in addition to the regular ventilation. Air-conditioning supplies energy when needed and supplies room temperature as desired. However, often the air must be de-humidified with a high auxiliary energy demand. The COP of an energy-efficient chiller is comparable to the actual realised night ventilation with 4.5 kWh_{th}/kWh_{el}.

Fig. 5-9 shows the COP as a function of the daily mean temperature. As expected, the COP decreases when the ambient air temperature increases due to the smaller temperature difference between inside and outside during the night.
Summary

Though the long-term monitoring provides an estimation of the thermal behaviour, the night ventilation effect cannot be quantified with respect to the energy flows. Experiments with regard to the night ventilation efficiency have to be carried out in order to determine the influencing parameters.

5.4 Experiments on night ventilation

In April 2002 (March 28 – May 7, 2002), some experiments were carried out in two offices to determine the effect of night ventilation. While one room was passively cooled by night ventilation, the other room was not cooled (reference room).

Aim of the experiments. The experiments provide a data basis for the characterisation of night ventilation. Only experiments offer the possibility for the variation of the nocturnal air change rate which is very important to determine the night ventilation efficiency. The results can be used for the modelling of calculation models. While operational measurements can only deal with few data, many boundary conditions can be measured during short-term experiments, in particular the air change rate. Besides the main parameters, tributary effects can be evaluated, e.g. the local and time temperature distribution.

Measuring system. During the experiments, meteorological data, air change rates, air temperatures (incl. three-dimensional temperature field), surface temperatures (floor, ceiling, window and internal wall) and the operative room temperature (globe thermometer) were measured with a time resolution of 5 minutes. Additionally, the local temperature profile at the ceiling was measured using an infrared camera.

Measuring realisation. The experiments have been divided into five periods. In each period the nocturnal air change rate (only in the room with night ventilation) and the position of the sun-shading has been changed in order to vary the solar heat gains. The measuring periods are separated by two days with the building in free running mode.

A small ventilator was used to draw a defined air-flow rate through the office during the night from 2 a.m. to 7 a.m. The internal heat gains were “simulated” by a fan heater with a thermal energy performance of 980 W in each room from 8 a.m. to 6 p.m. corresponding to the working hours.
The day ventilation highly effects the room’s energy balance. During the hybrid day ventilation, the air change rate is mainly dependent on the status of flaps and windows in the two offices and the adjacent rooms. This interrelation is taken into account by an air-flow network to calculate the air change rate in each office. As the air flow caused by natural ventilation is the most uncertain parameter, detailed measurements were repeated several times in order to set up and validate this air-flow model. Table 5-3 shows some results from these air change measurements in ordinary operation. As these measurements only determine the boundary conditions, they are not discussed separately.

Table 5-3: Air change rates from measurements with the tracer gas technique.

<table>
<thead>
<tr>
<th>Hybrid day ventilation, all windows and flaps open</th>
<th>Hybrid day ventilation, windows and flaps open only in 1 room</th>
<th>Free day ventilation without exhaust fan</th>
<th>Night ventilation, all windows and flaps open</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.5</td>
<td>2.7</td>
<td>0.3 – 1.7</td>
<td>3.7 – 5.25</td>
</tr>
</tbody>
</table>

Data analysis. As the building is always in a transient never in a periodic steady state condition, data from the measurements must be summarised for separate time periods. For this reason, four periods with comparable boundary conditions were defined for detailed data analysis. Table 5-4 gives an overview of the main experimental configurations and boundary conditions. The night ventilation results in a temperature decay of 0.6 to 2.1 K during the day (24 hours).

As an example, the comparison between the periods 1 and 4 may reveal the correlation between heat gains and losses and the room temperature. The ambient air temperature in period 4 is 0.7 K higher than in period 1. Furthermore, in period 4 the blinds are open and not closed as in period 1. Though the solar radiation is 52 % higher in period 1, the solar gains are almost 3 times higher in period 4 than in period 1 due to the higher g-value. In spite of the higher temperature level and the higher heat gains, the room temperature in period 4 is only 0.1 K higher than in period 1 due to the higher nocturnal air change rate. Corresponding, the temperature difference between the two rooms is 0.2 K higher in period 4 (0.8 K) than in period 1 (0.6 K). In general, the temperature difference between the night ventilated room and the reference room is comparatively small since the energy balance is dominated by the internal heat gains and the heat transfer between the adjacent rooms partly compensates the temperature difference.
Table 5-4: Boundary conditions and results from measurements. (The room temperature is shown only for comparison reasons in this place.)

<table>
<thead>
<tr>
<th></th>
<th>period 1 31/03 – 08/04</th>
<th>period 2 13 – 15/04</th>
<th>period 3 27 – 29/04</th>
<th>period 4 02 – 05/05</th>
<th>period 5 19 – 26/04</th>
</tr>
</thead>
<tbody>
<tr>
<td>solar shading</td>
<td>(venetian blinds)</td>
<td>close</td>
<td>close</td>
<td>close</td>
<td>open</td>
</tr>
<tr>
<td>mean ambient temp.</td>
<td>$T_{a,m}$ [°C]</td>
<td>11.3</td>
<td>9.4</td>
<td>8.5</td>
<td>12.0</td>
</tr>
<tr>
<td>and its amplitude</td>
<td>$\Delta T_a$ [K]</td>
<td>6.6</td>
<td>3.2</td>
<td>2.6</td>
<td>4.8</td>
</tr>
<tr>
<td>global solar rad.</td>
<td>$I_{fa, m}$ [W/m²]</td>
<td>204</td>
<td>84</td>
<td>52</td>
<td>134</td>
</tr>
<tr>
<td>and its amplitude</td>
<td>$\Delta I_{fa, m}$ [W/m²]</td>
<td>327</td>
<td>143</td>
<td>100</td>
<td>176</td>
</tr>
<tr>
<td>mean heat gains</td>
<td>$Q_m$ [W/m²]</td>
<td>48</td>
<td>48</td>
<td>48</td>
<td>48</td>
</tr>
<tr>
<td>and its amplitude</td>
<td>$\Delta Q_m$ [W/m²]</td>
<td>33</td>
<td>33</td>
<td>33</td>
<td>33</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>office with night ventilation</th>
<th>yes</th>
<th>no</th>
<th>yes</th>
<th>no</th>
<th>yes</th>
<th>no</th>
<th>yes</th>
<th>no</th>
</tr>
</thead>
<tbody>
<tr>
<td>air change rate</td>
<td>3.16</td>
<td>≈1</td>
<td>5.03</td>
<td>≈1</td>
<td>2.16</td>
<td>≈1</td>
<td>5.03</td>
<td>≈1</td>
</tr>
<tr>
<td>mean indoor air temp. $T_{i,m}$ [°C]</td>
<td>21.8</td>
<td>22.4</td>
<td>21.1</td>
<td>22.6</td>
<td>20.7</td>
<td>22.8</td>
<td>21.9</td>
<td>22.7</td>
</tr>
</tbody>
</table>

**Room air and surface temperatures**

Fig. 5-10 shows measured temperatures in the office with night ventilation from 2 a.m. to 7 a.m. for April 24, 2002. As expected, the fluctuation of the air temperature is greater than that of the operative room temperature. Due to the temperature stratification, the surface temperature at the ceiling is higher than at the floor. Due to the small heat storage capacity of the internal wall, the temperature fluctuation near the internal wall (adjacent to a similar office) is greater than at the floor or the ceiling. The wall-mounted air thermometer “RT_C115” from the long-term monitoring is partly influenced by the wall temperature (adjacent to the corridor) and is not equal to the air temperature at any time but agrees sufficiently well with the operative room temperature.
Fig. 5-10: Temperature fluctuation in the office with night ventilation (air change rate 4 h⁻¹) on April 24, 2002. The mean temperature RT_C115 is 21.0 °C and the operative room temperature 20.9 °C. Thus, RT_C115 can be used as operative room temperature for long-term monitoring.

Local temperature distribution

The surface temperature at the ceiling was measured using an infrared camera. Fig. 5-11 (a) shows the local temperature field (y-axis, window – door, cf. Fig. 5-3) at different times (x-axis) during three hours starting at 8 p.m. At every time step, the ceiling temperature is higher in the centre than at the window or the door. Thus, the heat transfer at the ceiling (and also of the other surfaces) is not identical at every place, as it depends on local temperature differences.

Furthermore, Fig. 5-11 (b) shows the temperature profile before and during night ventilation: During night ventilation, the incoming, cold ambient air falls down to the floor. Thus, there is a large temperature gradient near the ceiling and the floor.

Both the local surface temperature distribution and the air temperature profile should be considered for the analysis of the heat transfer, cf. Chapter 5.6. In practical use, an appropriate heat transfer coefficient takes these local effects into account. Uncertainties can be estimated by a sensitivity analysis.
Night Ventilation

Starting from the monitored data, the energy flows can be calculated. Fig. 5-12 shows the night ventilation effect for three different air change rates during the night (cf. Table 5-4) as a function of the daily mean ambient temperature. As the experiments were carried out during spring, the ambient temperature is lower than in summer. As expected, the night ventilation effect increases with the air change rate and decreases with the ambient air temperature.

Summary

There are many data from the experiments and the long-term monitoring, but neither Table 5-4 nor Fig. 5-10 provides data for comparison of night ventilation, because
there are many thermal influences (e.g. heat flow from or to adjacent rooms, fluctuating solar heat gains or air change between the offices and the corridor) which affect the energy balance in addition to the night ventilation.

The next Section introduces two models in order to evaluate the contribution of night ventilation to the energy balance and the temperature behaviour in an office: A parametric model and building simulation. Both methods aim to separate the night ventilation from other influences and effects, such as the heat loss due to day ventilation and transmission, the building’s thermal inertia or the heat gains. The influence of user behaviour, i.e. occupancy, use of equipment or operation of sun-shading and window opening, is discussed in detail in Chapter 8.

5.5 Analysis of the night ventilation experiments

The experiments are analysed by the parametric model from Chapter 2.2.1 and by building simulation with the aim to draw an energy balance for the room and to separate the night ventilation effect from the energy balance.

5.5.1 Parametric model

The experiments can be analysed using the parametric model. Table 5-5 compares the mean temperature and the temperature amplitude from measurements using Eq. (2-4) for parameter identification with the results of the parametric model using Eqs. (2-6) and (2-7). As period 2 contains two short time periods which do not satisfy steady periodic conditions, only three of the four periods, which are defined in Table 5-4, are used for data analysis.

Table 5-5: Energy balance and air temperature for the measurement according to the parametric model.

<table>
<thead>
<tr>
<th></th>
<th>period 1</th>
<th>period 3</th>
<th>period 4</th>
<th>period 1 + 3 + 4</th>
</tr>
</thead>
<tbody>
<tr>
<td>nocturnal air change rate</td>
<td>3.16 h⁻¹</td>
<td>2.16 h⁻¹</td>
<td>5.03 h⁻¹</td>
<td>without NV</td>
</tr>
<tr>
<td>heat loss factor H [W/(m² K)]</td>
<td>4.0</td>
<td>2.9</td>
<td>5.3</td>
<td>3.74</td>
</tr>
<tr>
<td>heat gain G / ΔG [W/m²]</td>
<td>42.4 / 62.5</td>
<td>35.4 / 52.1</td>
<td>53.2 / 79.9</td>
<td>43.6 / 64.8</td>
</tr>
<tr>
<td>time constant τ [h]</td>
<td>21.9</td>
<td>33.6</td>
<td>20.6</td>
<td>28.6</td>
</tr>
<tr>
<td>mean temperature T$_{im}$ [°C]</td>
<td>21.8</td>
<td>21.9</td>
<td>20.7</td>
<td>20.8</td>
</tr>
<tr>
<td>and its amplitude ΔT$_i$ [K]</td>
<td>4.1</td>
<td>3.2</td>
<td>4.0</td>
<td>3.5</td>
</tr>
</tbody>
</table>

The calculation of H, G and τ is described in detail by Jäschke [5-9]. A brief discussion of the results shows how the parametric model can be used:

- The parametric model fits the mean temperature better than the temperature amplitude: While the heat gains and losses (mean temperature) can be calculated accurately, the modelling of the heat storage capacity (temperature amplitude) is uncertain due to the simplifications in the calculation procedure (cf. Chapter 2.2.1), local temperature gradients and uncertain heat transfer coefficients (cf. Fig. 5-11).

- The time constant of the room strongly depends on the air change rate: The higher the nocturnal air flow, the shorter is the time constant. The mean time constant of the room without night ventilation is 12 % longer than of the room with night ventilation (average of the three periods).
The higher the daily heat gains, the lower is the impact of night ventilation on the room temperature. At first view, this appears to be illogical since night ventilation dissipates the heat which entered the room during the day. However, high heat gains cause a high excess temperature which results in an increased heat loss during the day. Thus, the absolute energy dissipation due to night ventilation is higher at daily higher heat gains but the relative change in room temperature is smaller.

Though there are differences between measurement and calculation, the parametric model can be used for data analysis. A higher accuracy – especially for the temperature amplitude – can be reached, if the energy flows are calculated by building simulation.

5.5.2 Building simulation

The ESP-r simulation program [5-10] was used for data analysis as an alternative to the parametric model. All input parameters and boundary conditions (i.e. climate, sun-shading control, air flow, internal heat gains, user behaviour) are well-known or can be modelled accurately.

Model set-up and parameter identification. First, an accurate simulation model is to set up with data from the experiments in April. Many influencing variables are well-known since they were measured precisely. As some building physical parameters (i.e. heat transfer coefficients at internal walls, solar absorption of the external wall, g-value of the glazing and the effective thermal conductivity of the ceiling) are uncertain, they can be fitted to the measurements by parameter identification. Fig. 5-13 compares the operative room temperature from measurement (globe thermometer) and simulation for two days. The small difference between measurement and simulation can be primarily explained with the calculation of the heat transfer between the internal walls and the room air. The heat transfer coefficients cannot be calculated with regard to the real temperature profile in the room, cf. Fig. 5-11, but only for an ideally mixed room. Furthermore, the calculation even of a mean is uncertain, cf. Chapter 7.3.2.

Model validation. Second, the accuracy of the simulation model is checked with data from July 2002. This model validation uses the model that was set up initially with data from April 2002, but with data from the offices in ordinary operation. The internal heat gains, the operation of the ventilation system, the status of flaps, windows and blinds are known from the long-term monitoring. In contrast to the simulation with the measured air change rates from the data in April, the air-flow rate (infiltration from outside, air change with adjacent zones and exhaust air from the ventilation system) is calculated by the simulation. Fig. 5-13 shows a good agreement between measurement and simulation results. Thus, all boundary conditions and thermal interactions are taken accurately into account.

---

12 The building simulation has been performed not only for the 4 periods, but for the whole experiment.
Fig. 5-13: Building simulation: a) Model set-up with measurements from the experiments in April 2002: Variation of temperature with time for two days. Additionally, the results from the parametric model with $H$, $G$ and $\tau$ from Table 5-5 are shown. b) Model validation with measurements from long-term monitoring: Comparison between measurements and simulation in July 2002 for ordinary operation.

5.6 Analysis

Whereas simplified assumptions had to be made for the parametric model, the building simulation program takes explicitly the underlying affects into account and calculates the heat flows more accurately.

As both models can analyse the thermal behaviour, a promising approach is to merge both models. Using an accurate model for building simulation, the measurements from the real, transient experiment can be evaluated for periodic steady-state conditions. In conclusion, the building parameters $G$, $H$ and $\tau$ can be derived accurately from these simulation results.

5.6.1 Model comparison

Starting from realistic assumptions, the parametric model does not agree with transient measurements at any time, while the building simulation shows adequate agreement with measured temperatures. Therefore, the accuracy of the parametric model is
checked by comparison with the building simulation. Only if both models yield the same results, they can be merged.

The comparison is based on the office depicted in Fig. 5-3. If solar heat gains are ignored, internal heat gains and air change rate are considered to be constant during the day, and the ambient temperature oscillates regularly, then both models will lead to similar results, whereby the parametric model overestimates the thermal loss factor $H$ by about $0-10\%$ as can be seen in Table 5-6. The error increases slightly with the air change rate. Though both models use constant heat transfer coefficients for this comparison study and assume ideally mixed room air, the heat loss is lower in the numerical model due to the long-wave radiation. Starting from low outside and high indoor temperatures, the light-weight components (i.e. floor) dissipate heat to the room air and are thermally discharged. Without long-wave radiation, this component would no longer affect the energy balance of the room but the floor actually absorbs heat from heavy-weight components (i.e. ceiling). This heat is dissipated to the room air, increases the room temperature and, hence, decreases the heat loss coefficient. As this effect increases with the air change rate, the difference between the analytical and the numerical model also increases. Nevertheless, the error is small and can be neglected, if combined heat transfer coefficients are considered by the parametric model.

Table 5-6: Model comparison for the thermal loss factor $H$ between the parametric model and building simulation.

<table>
<thead>
<tr>
<th>air change rate</th>
<th>$0\ h^{-1}$</th>
<th>$1\ h^{-1}$</th>
<th>$4\ h^{-1}$</th>
<th>$8\ h^{-1}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$H$ from parametric model with Eq. (2-10) [W/(m² K)]</td>
<td>1.3</td>
<td>2.95</td>
<td>8.0</td>
<td>14.5</td>
</tr>
<tr>
<td>$H$ from simulation with Eq. (2-6) [W/(m² K)]</td>
<td>1.3</td>
<td>2.9</td>
<td>7.9</td>
<td>13.2</td>
</tr>
</tbody>
</table>

The modelling of the heat storage capacity $C$ is completely different: The parametric model solves the heat conduction equation analytically and the building simulation numerically, but both models result in the same heat storage capacity for different thicknesses and heat transfer coefficients, Fig. 5-14.

![Fig. 5-14: Heat storage capacity $C$ against component thickness and heat transfer coefficient. (The "analytical" value is calculated by Eq. (2-12) and the "numerical" by ESP-r.)](image)
5.6.2 Merging the parametric model and building simulation

One promising approach for data analysis is to transfer the results from a parameter identification using the building simulation to the parametric model:

1. First, the simulation model is fitted to extensive measurements (here: April simulation).
2. This model is validated with measurements from the building under ordinary operation conditions (here: July simulation).
3. The simulation model can be used for data analysis with periodic steady state boundary conditions and operation (here: Table 5-7).
4. Finally, the simulation results are analysed using the parametric model (here: Table 5-7 and Fig. 5-15).

Thus, the parametric model can take realistic assumptions from experiments or operational measurements into account via the transient building simulation as a loop way.

In the validated simulation model from Chapter 5.5.2, adiabatic boundary condition are introduced at the internal walls, as a single office is to be characterised. The simulation has been carried out with separately fixed air change rates for day and night, sun-shading on or off and no internal heat gains. As climatic data set the day July 28, 2002 is used with $T_{a,m}=23.8\, ^\circ\text{C}$ and $\Delta T_a=5.5\, \text{K}$ and the vertical solar radiation on the south facade $I_m=160\, \text{W/m}^2$ and $\Delta I=280\, \text{W/m}^2$.

Table 5-7 shows the input parameters (air change rate and $g_{\perp}$-value) and Fig. 5-15 the simulation results with the effect of night ventilation and solar shading on the indoor air temperature: In this case, night ventilation reduces the mean indoor temperature by 2 to 3 K depending on the solar shading, by 3 to 4 K depending on night ventilation and both night ventilation and solar shading by 5.7 K.

<table>
<thead>
<tr>
<th></th>
<th>no night ventilation, but no solar shading</th>
<th>no night ventilation, but solar shading</th>
<th>night ventilation and solar shading</th>
</tr>
</thead>
<tbody>
<tr>
<td>ACH + NV</td>
<td>1 (1 am – 24 am)</td>
<td>3 (2 am – 7 am)</td>
<td>3 (2 am – 7 am)</td>
</tr>
<tr>
<td>$g_{\perp}$</td>
<td>0.51</td>
<td>0.51</td>
<td>0.11</td>
</tr>
<tr>
<td>$H$ [W/(m² K)]</td>
<td>1.73</td>
<td>3.36</td>
<td>1.67</td>
</tr>
<tr>
<td>$G$ [W/m²]</td>
<td>9.8</td>
<td>10.6</td>
<td>2.8</td>
</tr>
<tr>
<td>$\Delta G$ [W/m²]</td>
<td>17.1</td>
<td>18.5</td>
<td>4.8</td>
</tr>
<tr>
<td>$\tau$ [h]</td>
<td>46.5</td>
<td>25.1</td>
<td>39.2</td>
</tr>
</tbody>
</table>

Notice concerning the negative thermal loss factor $H$: If the office is cooled by night ventilation and the blinds are closed, the mean indoor air temperature is lower than the ambient air temperature. Eq. (2-6) demands a negative $H$-value, which is delivered by a negative, thermally effective air change rate from Eq. (2-11). A negative air change rate is thermodynamically impossible but mathematically correct in this context.

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13 This simulation is dependent on the thermal behaviour of the room but is independent of the operation. Thus, the building characteristics $G$, $H$ and $\tau$ can be derived from these simulation results.
Fig. 5-15: Simulation with generalised inputs. Different passive cooling measures provides a lower room temperature and / or a temperature modulation during the day.

5.7 Conclusions

The building simulation provides accurate results, if the input parameters and boundary conditions are well-known. However, user behaviour results in energy and temperature variations which are of the same order of magnitude as the effect of different design decisions and operation strategies.

Statistically distributed events (e.g. user behaviour, cf. Chapter 8) and transient phenomena affect the energy balance and the indoor temperatures in buildings. Thus, the data analysis of measurements in buildings under ordinary operation conditions is sophisticated and may even lead to wrong conclusions, if the data analysis is carried out too simple.

However, an extensive cross-section analysis of experiments and long-term monitoring with building simulation and the parametric model concludes that measurements can be analysed by the parametric model. This approach provides the three main building parameter $\tau$, $G$, and $H$ and yields a universal description of the thermal building behaviour and the night ventilation effect.

The method \textit{parametric model + building simulation} focuses on main building parameters and provides a simplified thermal model which can be used advantageous for data analysis. An accurate parametric model can be deduced from a procedure in three steps:

1. Short-term measurements (weather, indoor air, surface and operative room temperatures, air change rate, solar shading, internal gains, occupancy).
2. The thermal behaviour is simulated by a sophisticated building simulation with the short-term measurements as input data and known material properties (g-value, U-value, thermal properties).
3. The main building parameters are derived from the validated simulation model with typical weather data, operation and user behaviour.

This model describes the thermal behaviour of the Fraunhofer ISE building or, more generally spoken, of a specific building. With the approach discussed in this Chapter, the building behaviour can be characterised as a function of its building physical properties and for periodic changing influences like typical user patterns and control...
strategies for day and night ventilation or sun-shading. The developed method can be applied to the building design and operation:

- During the design phase, different design options are evaluated by building simulation. Due to the vast number of input parameters, the interpretation of simulation results is often difficult. The transformation into the main building parameters $G$, $H$, and $\tau$ provides a concise presentation in order to compare different techniques for avoiding over-heating in passively cooled buildings or to check the plausibility of the simulation and its underlying assumptions.

- In the context of quality assurance, the presented method (measurement – simulation – building parameters) can be used in order to evaluate whether the original design ideas were realised or whether the control strategies (i.e. night ventilation and / or solar shading) are correctly implemented.

- Moreover, the parametric model can be integrated into the building management system (e.g. adaptive or predictive controllers) for ventilation strategies or shading control. The parameters from the design process can be re-defined once during the implementation of the building or regularly during the building operation with the presented method.

### 5.8 References


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14 The statistically distributed occupancy and user behaviour with regard to ventilation or sun-shading can be considered, if the building parameters are described not only with a mean value but also with its standard deviation according to the approach in Chapter 4.3. The influence of user behaviour on the building’s energy balance and the room temperature is discussed in Chapter 8.
Part B

The previous Chapter concluded that a simplified parametric model is sufficiently accurate to calculate the thermal building performance. Based on the parametric model, the characteristic building parameters can be derived from an energy balance and an analysis of the temperature performance of the building.

This Chapter focuses on the impact of the meteorological environment and the building's thermal inertia on the indoor temperature performance. Data from a long-term monitoring campaign and short-term measurements are used to identify the correlation between ambient air temperature and room temperature and to identify the night ventilation efficiency.

In contrast to extensive simulation studies, the presented method is based on the straightforward mathematical model from Chapter 2.1.3 using Fourier analysis of temperature time series and an energy balance model which is validated with detailed measurements.

Besides the weather, the user also has a strong impact on the energy balance and thus the room temperature. In this context, the Pollmeier building is a good example for a passively cooled and freely ventilated building which has been designed robustly enough to compensate for inappropriate user behaviour.

Fig. 6-1:
Pollmeier Massivholz GmbH
Creuzburg, Germany
completion: 2001
6.1 Introduction and building description

The new administration building of Pollmeier Massivholz GmbH (Creuzburg, Germany) has been designed and constructed as a low-energy building, Fig. 6-1. Pollmeier Massivholz GmbH is an export-oriented, medium-sized company in the wood-working industry with a total staff of 400. The company constructed a new administration building [6-1] for up to 100 employees in Creuzburg, Germany. On the ground floor, several conference and service rooms and also a cafeteria are grouped around a three-storey atrium. Spacious office areas are located on the upper storeys and are enclosed by a large expanse of glass towards the atrium. The construction project fulfils the demands of stringent low-energy building standards and conforms to the ideal of a "lean building": Minimal energy requirements and a pleasant indoor climate are achieved by optimised building service facilities, which are reduced to the bare minimum.

The building has an exhaust air system with slit valves for incoming air in the façade (Fig. 6-2) and is connected to an existing wood-fired district heating system. An exhaust air heat pump supports the space heating and water heaters. Great attention was paid to high energy efficiency in building service equipment and illumination. Part of the electric power is supplied by a photovoltaic array integrated into the roof construction of the atrium. The electricity and heating energy demand, meteorological data, operating parameters of the technical building services and room temperatures were monitored for two years [6-2].

![Fig. 6-2: Floor plan and ventilation. The air flows through slit valves into the room. An exhaust fan discharges 1,280 m³/h during the night, approx. 1 air change per hour. The open-plan office is located on the 2nd floor and is highlighted.](image)

Internal and solar gains are minimised in summer, such that the remaining loads can be counterbalanced only by controlled day and hybrid night ventilation. The ventilation concept for the Pollmeier building includes opening windows, slit valves (for supply air) and an exhaust fan. Due to a low pressure drop and energy-efficient fans, the measured electricity consumption for ventilation is only 0.16 W/(m³ h⁻¹). During working hours, the fan is controlled by the CO₂-level and during summer nights by the room temperature. In winter, waste heat is recovered by a heat pump. This article focuses on the night ventilation efficiency in an open-plan office on the 2nd floor. The key information is given in Table 6-1.
Table 6-1: Key building and open-plan office specifications.

<table>
<thead>
<tr>
<th>gross volume</th>
<th>net floor area</th>
<th>compactness</th>
<th>working hours</th>
<th>occupants</th>
<th>completion</th>
</tr>
</thead>
<tbody>
<tr>
<td>16,850 m³</td>
<td>3,510 m²</td>
<td>0.32 m²/m³</td>
<td>Mon – Fri 7 a.m. – 6 p.m.</td>
<td>100</td>
<td>2001</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>height</th>
<th>net floor area</th>
<th>outside walls</th>
<th>windows</th>
<th>solar control</th>
<th>internal walls</th>
</tr>
</thead>
<tbody>
<tr>
<td>3 m</td>
<td>535 m²</td>
<td>220 m²</td>
<td>71 m²</td>
<td>outside 50 %</td>
<td>1,005 m²</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(mean U-value)</td>
<td>(g-value)</td>
<td></td>
<td>(incl. ceiling)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>1.2 W/(m² K)</td>
<td>0.58</td>
<td></td>
<td>0.2 m concrete</td>
</tr>
</tbody>
</table>

In the Pollmeier building, the data are monitored with a time resolution of 2 minutes. Data are available from the building management system and provide all information to draw the energy balance for the open plan office. This long-term monitoring was complemented with short-term measurements. The data analysis deals with hourly data. Fig. 6-3 shows how the temperature behaviour of the Pollmeier building and its energy balance are merged.

Fig. 6-3: Methodological approach. Data from the long-term monitoring are analysed with the energy balance. Detail information from short-term measurements (dashed line) are used to set-up the energy balance model. A weather data analysis shows that the building can be described accurately with the indoor / outdoor air temperatures. The temperature behaviour from the long-term monitoring and short-term measurements are used to calculate the heat storage capacity. Night ventilation efficiency and heat storage capacity (bold text) can be derived from the energy balance.

6.2 Analysis of meteorological data

Weather conditions affect the thermal behaviour of a building. Therefore, the meteorological data are characterised in order to determine the night ventilation potential.

In conventional HVAC design, the most common performance figure is the “number of cooling degree-days”, cf. Recknagel-Sprenger [6-3] or other national guidelines. Ghiaus et al. [6-4] used degree-days to evaluate natural ventilation potential. Tselepidaki et al. [6-5] showed that there is a strong correlation between daily minimum, mean and maximum ambient air temperatures and found a strong impact of different climatic data sets on the final energy need of buildings. Heinemann et al. [6-6] showed that ambient air temperature and solar radiation correlate strongly in summer. Thus, the ambient air temperature can be used to estimate the cooling requirement.
With reference to the design of passive cooling and air-conditioning systems, Santamouris et al. carried out a statistical analysis on summer ambient temperatures [6-7], on the persistence of high temperatures [6-8], and on summer conditions concerning cooling purposes in Athens (Greece) [6-9]. In addition to the ambient air temperature, comfort [6-10] and discomfort [6-11] indices were used and the results were applied to other regions [6-12]. The data analysis is based on these findings, i.e. long- and short-term dynamics, comfort criteria and cumulative duration curves.

Cooling degree-days and a statistical analysis of ambient air temperature can be used to calculate the energy demand for cooling purposes accurately. But the temperature in passively cooled rooms depends not only on the ambient temperature, but also on solar radiation, wind and the use of the building.

Keller [6-13] developed a more precise method of analysing meteorological data analysis using building-specific climate surfaces. This model is based on a Fourier analysis of typical weather periods. The duration of a period should cover three time constants of the building, i.e. approx. 5 days for a thermally heavy-weight building.

The potential for night ventilation can be estimated by the cumulative duration curve of the ambient air according to Fig. 6-4. While the daily maximum temperature is approx. 36 °C (cooling required), the daily minimum temperature never exceeds 20 °C. Thus, the building can be cooled by night ventilation even during the warmest night. Furthermore, the daily temperature difference increases, in general, with the daily maximum temperature, cf. Fig. 6-5. When cooling is required (high day temperature) to provide comfortable indoor air temperatures, the cooling potential by night ventilation is high due to the high day – night temperature difference. However, the actual night ventilation performance is not necessarily also high, cf. Chapter 6.4.3.

![Fig. 6-4: Analysis of ambient air temperature: Duration curves of minimum, mean and maximum ambient temperatures.](image-url)
6.3 Ambient and indoor air temperature

Night ventilation efficiency is often analysed using correlations between outdoor and indoor air temperatures. The following data analysis is based on and extends some previous results:

- Givon [6-14] monitored buildings with different thermal inertias under different ventilation and shading conditions. The experiments were analysed by regression analysis: The best correlation exists between the outdoor average and the indoor maximum.

- Kolokotroni et al. [6-15] used results from monitoring (air and slab temperatures during the summer) in order to investigate passive ventilation cooling for a school building. Based on the monitored conditions, a thermal model was used to quantify the effect of weather in relation to provided ventilation and shading. The investigations aim at a model-based control method using empirical models.

- Geros et al. [6-16] carried out experiments and simulations in order to evaluate night ventilation efficiency. The night ventilation efficiency is strongly related to the relative difference between indoor and outdoor temperature – mainly during the night – , the useable air-flow rate applied during the night (taking ventilation efficiency into account), and the thermal capacity of the building.

- Blondeau et al. [6-17] carried out experiments in a building in ordinary operation with various night ventilation schemes. Next to the graphical analysis of temperatures, the results have been extended by numerical simulations. The heat dissipated by night ventilation was 18 Wh/(m² day) which is much lower than in the Pollmeier building, cf. Fig. 6-18.

Though the night ventilation efficiency is calculated for each of these experiments, none of these experiments was analysed using a straightforward data model. In the following, the temperature performance (short-term and long-term dynamics) is analysed via time dependent profiles and frequency analysis. In Chapter 6.4, the temperature behaviour is merged with the energy balance of the room.
6.3.1 Frequency analysis

The fundamental assumption underlying the frequency domain method is that meteorological time series can be represented by a series of periodic cycles. In this way the weather’s influence can be represented by a steady-state term accompanied by a number of sinusoidal harmonics. The division of meteorological time series into component sinusoidal variations about a mean condition is achieved by Fourier series representation. A continuous function \( f(t) \) can be approximated by a series of sine and cosine functions or a series of sine functions taking a phase shift into account, cf. Eq. (2-1).

In a control theoretical sense, the building responds to periodic excitation functions (i.e. outdoor temperature, solar radiation, internal heat gains or ventilation) with a periodic response function (i.e. the indoor temperature). Thus, not only the meteorological data, the internal heat gains and the ventilation but also the indoor temperature can be represented by Fourier series.

Furthermore, the weather can be characterised only by the outdoor temperature as a good approximation (cf. Chapter 6.2) and the internal heat gains and the ventilation are (almost) independent of the weather during summer. Hence, the impact of the climate on the temperature performance can be accurately analysed by a Fourier analysis of the ambient and the indoor temperature.

**Fourier analysis for the whole year 2002**

Fig. 6-6 and Fig. 6-7 show the results of the Fourier transformation using data of the whole year (8,760 hours). The data are re-transferred from the frequency to the time domain according to Eq. (2-4) using the 12 elements of the Fourier series with the largest amplitudes since the frequencies with large amplitudes have the strongest impact on the temperature variation in time. Both input and output signal are equalised whereby characteristic periods can be identified. The oscillation of the ambient air temperature is largely attenuated due to the building’s thermal inertia. The only frequency with a noticeable amplitude of more than 1 K is the 1/d-frequency which describes the daily temperature oscillation.

![Fourier spectrum and re-transformation](image)

Fig. 6-6: Ambient air temperature: Fourier spectrum (left) and re-transformation (right). The ambient air temperature shows – besides the yearly mean temperature – two main frequencies: A yearly and a daily amplitude (3.4 K). Due to its statistical character, the ambient air temperature has a high noise factor.
Fig. 6-7: Indoor air temperature: Fourier spectrum (left) and re-transformation (right). The ambient air temperature is shows besides the yearly mean temperature – two main frequencies: A yearly and a daily amplitude (1.2 K). Due to its thermal inertia and the building, the input noise of the ambient air temperature is smoothed by the building and the room temperature has a low noise level.

**Fourier analysis for the summer of 2002**

As the building is neither actively heated nor actively cooled during the summer period, the indoor air temperature depends only on the weather, internal gains, ventilation and user behaviour with regard to the window opening. (In this context, only the correlation between outdoor and indoor air temperature is analysed.) Table 6-2 shows the results of the Fourier analysis for the summer period June 1 – August 31, 2002 with 92 days. Both ambient and indoor air temperature can be accurately represented by the mean temperature and two frequencies. Short fluctuations of the ambient air temperature are more strongly attenuated (26 %) than long fluctuations (only 41 %). The two corresponding time-dependent profiles are shown in Fig. 6-8 (short-term characteristic) and Fig. 6-11 (long-term characteristic) for the summer period.

<table>
<thead>
<tr>
<th>component frequency</th>
<th>corresponding period of amplitude</th>
<th>0 / 92 days (mean value)</th>
<th>92 / 92 days (daily amplitude)</th>
<th>4 / 92 days (23-day amplitude)</th>
</tr>
</thead>
<tbody>
<tr>
<td>ambient air temperature</td>
<td></td>
<td>18.78 °C</td>
<td>5.08 K</td>
<td>2.12 K</td>
</tr>
<tr>
<td>indoor air temperature</td>
<td></td>
<td>22.69 °C</td>
<td>1.33 K</td>
<td>0.87 K</td>
</tr>
</tbody>
</table>

**6.3.2 Short-term dynamics**

The short-term dynamics of the indoor air temperature can be derived from Fig. 6-8. During working hours, the indoor air temperature increases due to internal and external heat gains. As the building receives heat gains, the mean indoor temperature (23.06 °C) is higher than the mean ambient air temperature (21.06 °C) during this week. Due to the building's thermal inertia, the outdoor temperature fluctuation is attenuated.
Fig. 6-8: Ambient and indoor air temperature during a summer week. (Grey columns mark working hours: 8 a.m. – 5 p.m.)

Though the Pollmeier building is passively cooled, the limits for actively cooled buildings according to DIN 1946 (cf. Chapter 2.1.3) are used for analysis of the thermal building behaviour. (As the operative room temperature is not measured the data analysis deals with the indoor air temperature.) Fig. 6-9 shows that the indoor air temperature during working hours exceeds the comfort level only for a few hours. In addition, the time-dependent behaviour is shown for the warmest days: July 9, 2002 (high outdoor temperature) and August 26, 2002 (high indoor air temperature). Self-evident, both ambient and indoor temperatures increase during the day. However, due to the high mass of the building, high ambient temperatures do not necessarily cause high indoor air temperatures. Correspondingly, the highest room temperatures are measured at the end of the summer when the building’s thermal mass has been heated up during the summer.

Fig. 6-9: Indoor air temperature versus ambient air temperature. Working hours from 8 a.m. to 5 p.m. During the summer, the room temperatures exceed the comfort range (grey lines, cf. DIN 1946 in Chapter 2.1.3) rarely. Two days show exemplarily that the room temperature are generally higher at the end of summer since the building structure is heated.
The typical short-term characteristic of a building (cycle period: 24 hours) can be derived from a regression analysis for the mean day according to Eq. (2-5) during the period June 1 – August 31, 2002. Fig. 6-10 shows the indoor temperature increase due to heat gains as well as the attenuation of amplitude and the phase shift as a result of the thermal mass. As the results agree well with those from the Fourier analysis Eq. (2-4), a simple harmonic oscillation can be used for data evaluation.

![Figure 6-10: Mean summer day with regression analysis.](image)

**Table 6-3**: Mean temperature, amplitude and phase shift from a regression analysis according to Eq. (2-5) for June 1 – August 31, 2002. The results are similar to the results from the Fourier analysis (cf. Table 6-2 and Fig. 6-10).

<table>
<thead>
<tr>
<th></th>
<th>$T_{\text{mean}}$</th>
<th>$\Delta T$</th>
<th>phase shift $\phi$ / time lag</th>
</tr>
</thead>
<tbody>
<tr>
<td>ambient air temperature</td>
<td>18.50 °C</td>
<td>-4.65 K</td>
<td>0.49 $T_{\text{max}}$ at 4:10 p. m.</td>
</tr>
<tr>
<td>indoor air temperature</td>
<td>22.67 °C</td>
<td>-1.23 K</td>
<td>0.87 $T_{\text{max}}$ at 5:55 p. m.</td>
</tr>
</tbody>
</table>

### 6.3.3 Long-term dynamics

In addition to the short-term dynamics, the building has a long-term dynamic behaviour as outlined in Fig. 6-9: The building components store heat during the day and longer periods. Fig. 6-11 shows a time shift between outdoor and indoor temperatures of approx. 2 days.
Fig. 6-11: Daily mean temperature (24 hrs.) of ambient and indoor air temperature. (Grey columns mark working days: Monday – Friday.)

Fig. 6-12 shows the daily mean indoor air temperature as a function of the daily mean outdoor air temperature. Two summer weeks show that the indoor temperature can increase from one day to the next even if the ambient temperature decreases. Heat gains and losses are decoupled due to the high thermal storage capacity.

Fig. 6-12: Daily mean temperature for working days in summer 2002 during the working hours from 8 a.m. to 5 p.m. As the building structure is warmer in July, the room temperatures are higher than in June.

6.4 Data evaluation using the energy balance

As the temperature performance responds to heat gains, losses and storage, the temperature performance can be merged with a thermal building model. In the following, the energy balance is solved according to the parametric model from Chapter 2.2.1.

As the cycle periods of the excitation functions acting upon the building determine the cycle period of heat flows and indoor air temperature, the time period used for data
analysis in the frequency domain has to be defined. The length of the time period should be longer than the time constant of the system but short enough to separate characteristic periods concerning weather, user behaviour or building management. Hence, the time period is defined as follows: June 1 to August 31, 2002.

6.4.1 Energy balance

In order to draw the energy balance, each heat flow has to be calculated. While solar and internal heat gains and heat transmission can be calculated directly from measurements and construction details, the ventilation effect can only be calculated on the basis of a model.

- Time-dependent solar radiation on three facades and solar shading / solar control has to be taken into account in order to calculate the solar gains accurately.
- The electricity consumption is measured. Taking the efficiency of each electric appliance into account, the internal heat gains can be calculated. The daily occupancy can be estimated from some samples during summer 2002.
- The heat transmission through the external wall can be determined according to the European standard EN 832 and the German guideline VDI 2078, under consideration of the construction with its properties and time constant.
- More uncertainties are associated with the air infiltration. Thus, the air change rate was measured. An adapted model for the use of windows is applied to the measured data in order to determine the air infiltration at each time step.

Measurement and air-flow model. The air change rates were measured under different conditions using the tracer gas technique, details in Ref. [6-22]. The ventilation efficiency can be estimated from a flow visualisation (cf. Fig. 6-16) and the measured temperature profile from the floor to the ceiling (cf. Fig. 6-17). A similar flow model on cross-ventilation was presented by Schmidt et al. [6-23].

User model. Nicol et al. [6-24] carried out an extensive survey on the use of controls in naturally ventilated buildings. Of all available controls in naturally ventilated buildings, windows have the largest effect on indoor climate. Though the use of windows varies from person to person, the proportion of windows open is closely related to the indoor and outdoor temperatures. Thus, a model dependent on the ambient air temperature is suitable for this purpose.

The energy balance for a whole summer period is derived from an hourly energy balance, if each heat flow is integrated over the time June 1 – August 31, 2002. The heat flow into the building structure (heat storage) is integrated only for positive heat flows. The ventilation heat flow is divided into the ventilation during the working hours and off-time. For comparison reasons, the energy is related to the summer period (92 days) and the net floor area (535 m²).

\[
\frac{dQ_{\text{storage}}}{dt} = \dot{Q}_{\text{solar}} + \dot{Q}_{\text{internal}} + \dot{Q}_{\text{day ventilation}} + \dot{Q}_{\text{day ventilation}} + \dot{Q}_{\text{transmission}}
\]

(6-1)

Fig. 6-13 shows the energy balance for the mean day in summer 2002: Solar and internal gains are similar, 2/3 of daily heat gains are stored in the building components and are dissipated during the night using hybrid ventilation. Due to the temperature

15 In order to accurately predict the performance of a building in the time domain (e.g. building simulation), there is a need for finely resolved meteorological data, with a length of the time-step being considerably shorter than the time constant of the system being monitored, cf. Hensen [6-19].
equilibrium (mean value for all summer days), heat transmission and day ventilation hardly affect the energy balance.

Fig. 6-13: Energy balance for the summer 2002. Day ventilation from 7 a.m. to 7 p.m. and night ventilation from 7 p.m. to 7 a.m. (All daily energy flows per net floor area.)

If the energy balance from Eq. (6-1) is analysed not for the whole period but for the hours of the day, the thermal building performance can be described. Fig. 6-14 shows the daily heat flows for the working days: In the early morning, the building is cleaned and operation in the production area starts which results in a peak load at 6 a.m. In addition to these internal gains, there are solar gains (depending on solar radiation, shading and user behaviour). Due to the small temperature differences, ventilation cannot be used for passive cooling during the working hours. Heat gains are dissipated, in general, during the night. According to heat gains and losses, the heat flow from the components into the room (heat storage) is negative during the working hours due to heat gains and positive during the night due to the heat dissipation.

Fig. 6-14: Heat flow for the mean summer day 2002.

According to the parametric model, both the heat loss coefficient $H$ and the heat gain $G$ and $\Delta G$ can be derived from a regression analysis of the energy balance. Results are shown in Table 6-4. The mean indoor air temperature can be accurately calculated with these parameters from Eq. (2-6):

□ The measured mean indoor air temperature is 22.69 °C, cf. Table 6-2 (Fourier analysis).
Using the coefficients $H$ and $G$ in Eq. (2-6), the calculated mean indoor air temperature is 22.67 °C, cf. Table 6-3 (regression analysis).

While the mean temperature can be calculated directly from the energy balance, the calculation of the temperature fluctuation and the heat storage capacity $C$ according to Eq. (2-7) is more sophisticated and is discussed in the following Subchapter.

Table 6-4: Heat loss and gain coefficients from a sinusoidal regression analysis of the energy balance shown in Fig. 6-14.

<table>
<thead>
<tr>
<th></th>
<th>heat loss $H$</th>
<th>heat gain $G$</th>
<th>heat gain $\Delta G$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0 / 24 h (mean value)</td>
<td>4.79 W/m²ext.wall/K</td>
<td>18.65 W/m²ext.wall</td>
<td>–</td>
</tr>
<tr>
<td>24 / 24 h (daily amplitude)</td>
<td>–</td>
<td>–</td>
<td>29.43 W/m²ext.wall</td>
</tr>
</tbody>
</table>

6.4.2 Heat storage capacity

Fig. 6-15 gives an overview of methods to determine the heat storage capacity: Either the heat storage capacity is calculated from an energy balance or from construction details.

From energy balance. The time constant $\tau$ of the room is calculated from the energy balance and the temperature performance for the two periods, Fig. 6-15 (left side). The two main cycle periods for the ambient air temperature are 1 day and 23 days (results from the Fourier analysis). The heat gains oscillate daily.

From construction details. The calculation of the heat storage capacity from construction details, Fig. 6-15 (right side), depends strongly on the heat transfer at the construction components (see below). As all input parameters are known (construction details in Ref. [6-22]), the heat storage capacity can be calculated for the two main cycle periods, 24 hrs and 552 hrs, from the construction details.

Results. The heat storage capacity for each cycle period is accurately calculated by two different models, cf. Table 6-5: The difference is 10 % for the short-term dynamics.
and is negligible for the long-term dynamics whereby the calculation procedure is validated.

Table 6-5: Heat storage coefficient C.

<table>
<thead>
<tr>
<th>Cycle Period</th>
<th>C from Energy Balance</th>
<th>C from Construction Details</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 day</td>
<td>138 Wh/(m² ext.wall K)</td>
<td>154 Wh/(m² ext.wall K)</td>
</tr>
<tr>
<td>23 days</td>
<td>1,026 Wh/(m² ext.wall K)</td>
<td>1,034 Wh/(m² ext.wall K)</td>
</tr>
</tbody>
</table>

On the modelling of the heat transfer. Due to the air flow patterns in the room and the stack effect, a temperature profile becomes established from the bottom to the top which affects the heat transfer at the surface: While free convection occurs at the bottom, the air is stably stratified at the ceiling.

Fig. 6-16 visualises the air flow patterns with smoke at the window. Due to the slow air-flow rate (air speed below 0.3 m/s) and high temperature difference (approx. 8 K) during the night ventilation, the air flow falls off 0.8 m behind the window. Thereby, the air is distributed on the floor. The lake of cold air reaches a height of approx. 1.0 m. (Similar experiments, with higher air velocities and temperature differences, were carried out by Heiselberg et al. [6-26].)

Fig. 6-16: Air flow pattern near the window and at the floor, visualisation with smoke (July 4, 2002, 10:00 p.m.).

Fig. 6-17 shows the temperature profile from short-term measurements in July 2002. Surface temperatures were measured using an IR camera, and air temperatures were measured using three thermometers at heights of 0.1 m (floor), 1.1 m (desk) and 2.9 m (ceiling). Taking the temperature gradients into account, the heat transfer coefficients can be estimated using the Khalifa and Marshall model [6-27].
6.4.3 Night ventilation

The night ventilation efficiency between 8:00 p.m. and 6:00 a.m. can be separated from the energy balance. As windows are manually opened by the user, there are nights with many, less or no open windows. If the windows are closed, fresh air enters through slit valves (cf. Fig. 6-16), and the mechanical air change rate is approx. 1 h⁻¹. If the windows are open, the air change rate varies according to the open area and the additional driving forces (wind and temperature difference).

Fig. 6-18 shows the night ventilation efficiency as a function of the percentage of open windows and the daily mean ambient air temperature. As expected, the night ventilation efficiency increases with the ratio of open windows. Though the daily temperature differences are higher at warm days (high night ventilation potential, cf. Fig. 6-5), the night ventilation performance decreases at high ambient air temperatures due to the small difference between indoor and outdoor temperatures, cf. Fig. 6-9.

6.5 Conclusions

Extensive data analysis of the temperature performance and the energy flows in an office lead to the conclusion that indoor air temperatures and energy flows can be
merged in a consistent data model. It is self-evident, that a thermodynamically correct energy balance can be drawn for each time step. The developed model is specifically applied to analysis of measured data from a building in ordinary use. It takes user behaviour, building management strategies and meteorological data into consideration. So it is possible to determine the building’s main characteristics not only from simulations (cf. Chapter 4 and 5) but also directly from measurements:

- The mean indoor air temperature is calculated from a steady-state energy balance.
- Short-term and long-term dynamics can be evaluated on the basis of a validated energy balance.

According to and extending from the literature, universal graphs can be used in order to characterise the thermal building behaviour. The results from this graphical data analysis can be merged with the energy balance, establishing a comprehensive building, which can be used

- in the design phase in order to quantify uncertainties and to estimate the effect of long and short-term variations in the energy balance to the temperature behaviour.
- in a building management system. For this purpose, a building model can be adapted on the basis of short-term measurements from the initial operation period.

### 6.6 References


7 Room Air-Flow Distribution and User Behaviour

Part B

In the previous Chapters, the use of the building simulation (Chapter 4), a parametric model (Chapter 5) and the energy balance equation (Chapter 6) have been discussed. These validated models are applied to the data analysis in this Chapter in order to evaluate the impact of user behaviour and heat transfer on the night ventilation efficiency.

Passive cooling by night ventilation is often designed by using building simulation. In the design procedure, the building simulation can be used to specify the maximum solar and internal heat gains, to define the required heat storage capacity or to design window openings or the air flow through the building. However, two input variables for the building simulation are very uncertain in everyday design: Heat transfer coefficients and user behaviour concerning night ventilation.

This Chapter presents a building with a successfully operating passive cooling system. The data evaluation is based on long-term monitoring and short-term measurements and focuses on the air-flow distribution in the room (in order to calculate heat transfer coefficients) and the user behaviour (in order to estimate the frequency of typical airflow conditions). Following this data analysis, two preliminary models are presented, which may reduce uncertainties during the design process: A model for the heat transfer coefficient during night ventilation and a user model for a building with mechanical day and free night ventilation.

Fig. 7-1:
Hans Lamparter GBR
Weilheim a.d. Teck, Germany
completion: 2000
7.1 Introduction and building description

Introduction. Passive cooling by night ventilation has been investigated repeatedly by measurements and numerical approaches. In recent years, different numerical models and design tools have become available in simulation programs, handbooks and the literature. Though building physicists and HVAC designers access this knowledge, the design of night ventilation is still challenging.

The design tools – based on an air-flow network – calculate the air flow volume through a room as a function of the climatic and thermal boundary conditions. The air distribution in the room emerges from the thermodynamic parameters: Air change rate, surface temperature and air temperature. However, two input parameters for sophisticated design tools concerning the free ventilation are very uncertain:

- **What is the value of the heat transfer coefficient?** During night ventilation, the heat is dissipated from the building structure by the cool night air. Therefore, the convective heat transfer coefficients need to be calculated accurately.

- **When and how often do users open their windows?** The windows in most passively cooled buildings are manually opened by the occupants. A statistical user model is necessary in order to calculate ventilation rates that are close to reality.

This Chapter focuses on the influence of the air distribution in the room and the user behaviour on the heat dissipation by night ventilation. Therefore, results from long-term monitoring and short-term measurements are evaluated by a CFD simulation (air distribution and heat transfer coefficients) and by a statistical analysis.

Building description. The administration building for Hans Lamparter GbR, a company of consulting engineers and surveyors, was rigorously planned from the very beginning as a “passive building” with a heating energy demand lower than 15 kWh/(m² net floor area a). The compact building is divided into a two-storey structure with wood panelling and a recessed attic storey separated by reinforced concrete slabs. Due to its extremely good thermal insulation standard, its compact structure, passive use of solar energy and a ventilation system with highly efficient heat recovery and use of heat from the ground, the building has a specific heating energy demand of just 12 kWh/(m² net floor area a), which means that it can be heated solely by heating the incoming air. In summer, reduced internal heat gains, effective sun shading, an earth-to-air heat exchanger (day ventilation), passive cooling by night ventilation, and a high heat storage capacity (shift of heat dissipation from day to night) prevent high room temperatures from occurring during the working hours. For further details the reader is referred to Ref. [7-26].

Fig. 7-1 shows a photograph of the Lamparter building (SW façade). At this moment, sun-shading is closed on the top floor and in some offices on the ground and the 1st floor. The Lamparter building is situated in Weilheim a.d. Teck near Stuttgart, Germany, and consists of offices along the south-west (SW) and north-east (NE) sides.

Table 7-1 gives the key building information. Additionally, the main meteorological data and the internal heat gains due to the electricity consumption of lighting and equipment in the offices summarise the basic conditions for both the heating and ventilation system and the passive cooling system.
Table 7-1: Key building information.

<table>
<thead>
<tr>
<th>gross volume</th>
<th>heated net floor area</th>
<th>working hours</th>
<th>occupants</th>
</tr>
</thead>
<tbody>
<tr>
<td>25,705 m³</td>
<td>1,000 m²</td>
<td>7 a.m. – 6 p.m.</td>
<td>30 – 35</td>
</tr>
<tr>
<td>ambient temperature</td>
<td>heating degree days</td>
<td>global solar radiation</td>
<td>internal heat gains</td>
</tr>
<tr>
<td>[average, 2003]</td>
<td>[total, 2003]</td>
<td>[total, 2003]</td>
<td>[total, 2003]</td>
</tr>
<tr>
<td>11.8 °C</td>
<td>2,925 Kd</td>
<td>1,237 kWh/(m² hor a)</td>
<td>138 Wh elt/(m² office d)</td>
</tr>
</tbody>
</table>

Fig. 7-2 illustrates the ventilation concept: During working hours, each office is mechanically ventilated with 30 m³/hr per person. Fresh air is drawn through an earth-to-air heat exchanger. The supply air is pre-heated by the earth-to-air heat exchanger and a heat recovery system in winter. An air heating register supplies the remaining heat energy demand. During summer nights, the thermal building mass is cooled by free night ventilation using the buoyancy effect: Louvers at the top of the building are automatically opened. Cool night air flows through the manually opened skylights into the offices and through the open doors and the staircase to the air-outlet at the top.

Fig. 7-2: Hybrid ventilation system. (Day ventilation is indicated with black arrows and night ventilation with grey arrows.)

Fig. 7-3 shows the interzonal air flow during night ventilation and some results from an air change measurement using the tracer-gas technique [7-17].

Fig. 7-3: Free night ventilation in the office building and some results from air change measurements with the tracer-gas technique on July 13, 2002. The air change is given in [h⁻¹]. The shaded areas indicate the two offices that were monitored in detail.
The lean building concept results in a low energy consumption. Table 7-2 shows the results from the monitoring programme for 2003.

Table 7-2: Energy performance (January 1 – December 31, 2003).

<table>
<thead>
<tr>
<th>Description</th>
<th>Consumption</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heating energy demand</td>
<td>29.3 kWhth/(m² a)</td>
</tr>
<tr>
<td>supplied by heat recovery</td>
<td>16.0 kWhth/(m² a)</td>
</tr>
<tr>
<td>supplied by air heating</td>
<td>13.3 kWhth/(m² a)</td>
</tr>
<tr>
<td>Heating energy consumption</td>
<td>14.4 kWhth/(m² a)</td>
</tr>
<tr>
<td>Electricity consumption</td>
<td>11.4 kWhth/(m² a)</td>
</tr>
<tr>
<td>auxiliary power</td>
<td>0.8 kWhth/(m² a)</td>
</tr>
<tr>
<td>ventilation</td>
<td>5.1 kWhth/(m² a)</td>
</tr>
<tr>
<td>lighting</td>
<td>5.5 kWhth/(m² a)</td>
</tr>
<tr>
<td>Primary energy consumption</td>
<td>50.1 kWhth/(m² a)</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Description</th>
<th>Consumption</th>
</tr>
</thead>
<tbody>
<tr>
<td>PV electricity production</td>
<td>830 kWhth/kWp</td>
</tr>
<tr>
<td>installed power</td>
<td>8 kWp</td>
</tr>
<tr>
<td>Solar thermal heating</td>
<td>400 kWhth/m²</td>
</tr>
<tr>
<td>installed area</td>
<td>3.9 m²</td>
</tr>
</tbody>
</table>

Based on the experience from the previous Chapters 4 – 6, the monitored data can be analysed by the energy balance. Detail information concerning the heat transfer in the room and the user behaviour can be derived from this energy balance and additional short-term measurements. Fig. 7-4 shows how these separate aspects are harmonised.

Fig. 7-4: Methodological approach: Input data are printed in rectangles and models in ellipses. The data from the long-term monitoring are analysed with regard to the temperature behaviour and the energy balance. The short-term measurements are analysed by the building simulation, which provides the model for the energy balance, and provide the input parameter for the CFD model. Starting from this preparatory work, the night ventilation efficiency can be evaluated. The aim is to identify models (bold) for the user behaviour and the heat transfer coefficients which can be used in design of passive cooling.
7.2 Long-term monitoring

The energy performance of the Lamparter building was monitored between August 1 to July 31, 2003. In this Chapter, data from 39 sensors are used in order to evaluate the thermal building behaviour in summer:

- Meteorological station.
- Heating and ventilation system and earth-to-air heat exchanger.
- In two 2-person offices on the 1st floor, which are shown in Fig. 7-3: Air temperature, electric energy consumption, occupancy, window status (open / closed), supply air volume flow and temperature.

7.2.1 Temperature behaviour

The building and energy concept aims at comfortable room temperatures without air-conditioning. Did the building meet the specifications in the summer periods studied?

Fig. 7-5 shows the cumulative distribution curves for the room and the ambient air temperatures during the working hours in 2002 and 2003. As the summer of 2003 was warmer than the summer of 2002, the room temperature is higher in 2003. A simple criterion to compare the summer situation is the number of hours with room temperatures higher than 25 °C, cf. Chapter 2.1.3. The summer of 2002 can be characterised by short hot periods and the summer of 2003 by long periods with outdoor day temperatures above 25 °C:

- Consequently, the room temperature exceeds 25 °C in 2003 (warm summer) more often than in 2002.
- As the building could compensate for the high ambient temperature in 2002 but not in 2003 (long periods of warm weather), the room temperature exceeds the ambient temperature more often in 2003 than in 2002.

![Cumulative duration curves for the ambient and room air temperatures](image)

**Fig. 7-5:** Cumulative duration curves for the ambient and room air temperatures: In 2002 (2,520 working hours), the ambient air temperature exceeded 25 °C for 6.1 % and the room temperature for 3.8 % of the total office hours. In 2003 (2,480 working hours), the ambient air temperature exceeded 25 °C for 11.2 % and the room temperature for 11.8 % of the working hours. (The ambient air temperature falls below 10 °C during the rest of the time.)

Though Fig. 7-5 shows that the passive cooling system operated well in both years, the night ventilation efficiency can only be derived from an energy balance.
7.2.2 Energy balance for the summer period

All energy flows were calculated using the ESP-r building simulation program [7-4]: The input parameters for the integrated thermal and air-flow simulation are the meteorological data, the occupancy sensors and the monitored electricity consumption. Both the mechanical and the free ventilation are calculated by an air-flow network using the meteorological data, the air temperatures in each zone, the information about the mechanical ventilation system and the status of the window flaps. The status of the sun-shading devices (venetian blinds) is adjusted according to Reinhart’s user model [7-21], which was determined originally from measurements at the same site in 2000.

Fig. 7-6 shows the energy balance for the summer period June – August 2001. Heat gains are shown on the right and heat losses on the left side. The heat storage is the sum of all heat flows into and out of the partitions during a day:

- Though the solar radiation on the SW facade is higher than on the NE facade, the solar heat gains are similar in both rooms, as the venetian blinds are closed more often in the SW office (glare protection) than in the NE office. Due to technical equipment, the internal gains are higher in the NE office.

- The heat loss due to day ventilation is similar in both offices, as the mechanical air change and the supply air temperature (cool air from the earth-to-air heat exchanger) is identical. The higher heat gain during the day results in a higher heat loss during the night in the NE office.

- Due to the higher heat flows in the NE office, the heat storage is higher than in the SW office.

![Energy balance for each office](image)

Fig. 7-6: Energy balance for each office. The comparatively small heat losses / gains due to transmission and interzonal air change are included in day / night ventilation.16

7.2.3 Energy balance and room temperature

Due to the higher heat gains, the NE office is slightly warmer than the SW office: The mean room temperature is 23.5 °C in the NE and 23.3 °C in the SW office during the office hours (June 1 – August 31, 2002).

16 As the heat gains are higher in the NE office than in the SW office, the heat storage and the heat loss due to night ventilation are also higher.
Room versus ambient air temperatures during the office hours in 2002. Due to the air heating, the room air temperature is higher during the heating period than during spring and autumn (moderate ambient air temperatures). During the summer, the room temperatures exceed the comfort range (black lines, cf. DIN 1946 in Chapter 2.1.3) rarely.

Using a simulation model, the contribution of night ventilation to the energy balance, cf. Fig. 7-6, and to comfortable room temperatures, cf. Fig. 7-7, can be evaluated from monitored data.

But how can the night ventilation efficiency be calculated during the design process? For this purpose, the designer needs to consider the convective heat transfer (Chapter 7.3) and the user behaviour (Chapter 7.4) accurately in the simulation model.

7.3 Short-term measurements and computational fluid dynamics

The convective heat transfer can be estimated based on a validated computational fluid dynamics (CFD) model, in this study for the SW office on the 1st floor. The CFD model was designed with GAMBIT and simulated with FLUENT [7-10]. The air flow and temperature distribution is simulated on July 13, 2002 at 3 a.m. for an air change rate of 7 h⁻¹. Fig. 7-8 shows the temperatures used in the CFD model:

- The mean room air temperature is used in order to validate the CFD model with the overall energy balance of the room.
- Boundary conditions 1: The ambient air temperature and the temperature in the adjacent corridor (combined zone).
- Boundary conditions 2: The surface temperatures at each wall were measured using an IR-camera and are used in the CFD model. (The graph shows only the average surface temperature.)
Chapter 7

12.07.02 13.07.02 14.07.02

Fig. 7-8: Ambient, room air and surface temperatures for a typical summer period in 2002. The CFD simulation was carried out for the night between July 12 and 13, 2002 (grey frame).

As the CFD model is used to calculate the convective heat transfer between each surface and the air separately, it does not take long-wave radiation between the surfaces into account. This simplification is thermodynamically correct because the surface temperatures are used as fixed boundary conditions.\textsuperscript{17}

In order to get reliable heat transfer coefficients, a high resolution of the CFD model is necessary close to the surfaces. The boundary layers are modelled according to the logarithmic boundary layer velocity profile. Concerning the local discretisation within the CFD model, Gritzki [7-12] found that the ventilation efficiency in a room with single-sided ventilation is almost independent of temperature differences and air change rates but is strongly influenced by the geometry of the window. Therefore, Fig. 7-9 shows a high density of control volumes near to the surfaces and close to the window.

\textsuperscript{17} The convective heat transfer and the long wave radiation are taken separately into account by the building simulation since the surface temperatures are not known during the design process and have to be calculated.)
7.3.1 Validation: Air flow and temperature distribution

**Qualitative validation.** The CFD model can be validated by a qualitative comparison between an air-flow visualisation using smoke and the calculated air-flow paths. Fig. 7-10 shows a smoke visualisation of the interzonal air change between the office and the corridor (back flush) due to the temperature difference and the CFD simulation at that time.

![Fig. 7-10: a) Air-flow visualisation with smoke across the door on July 13, 2002, 3 a.m. (date and time of simulation). The office is on the left, the corridor is to the right. b) Air streamlines simulated with FLUENT. Corresponding to the air flow visualisation, air leaves the office at the bottom and enters the office from the corridor at the top of the door.](image)

**Quantitative validation.** A comparison of local air velocities and local air temperatures is used for a quantitative validation: Fig. 7-11 shows the air distribution (air velocity and temperature) in the room. The cool night air enters the room at the ceiling, flows under the ceiling for 1 to 1.5 m and falls down to the floor. At warm walls, the air flows up again and creates a convection roll perpendicular to each wall. The back flush in the door results in another convection roll in the main flow direction. A complex air-flow distribution emerges from these different convection rolls.

The accuracy of the air-flow model and the underlying thermal model is verified with the measured local air temperature and velocity at the desk: At this selected point, the simulated air temperature and velocity are similar to the measured values.
7.3.2  Convective heat transfer coefficient at the ceiling

As shown in Chapter 2.2.1 and in Fig. 5-14 the convective heat transfer coefficient \( h_c \) affects the effective heat storage capacity of each enclosure (i.e. walls, floor and ceiling but also the furniture) and hence of the room. As the ceiling dominates the room's thermal inertia due to its high mass, the heat transfer at the ceiling must be modelled as accurately as possible.

In the previous Sections, the data analysis in each building dealt with \( h_c \)-models according the literature. As the available \( h_c \)-models are not strictly valid for night ventilation, the \( h_c \)-value must be determined from measurements with parameter identification. A sensitivity analysis can successfully be used in order to estimate the error associated with \( h_c \). However, \( h_c \) is unknown during the design process and must be calculated directly. For this design reasons, a \( h_c \)-model for night ventilation is developed by CFD simulation which has been validated with measurements.

The heat flow between a fluid (here: air) and a solid surface (here: wall, ceiling or floor) increases with the temperature gradient \( \partial T / \partial y \) at the surface. The heat transfer
coefficient \( h_c \) is an artificial operand which is used to calculate this heat flow from the surface temperature and the mean air temperature:

\[
\dot{q}_{\text{surface}} = -\lambda_{\text{air,surface}} \cdot (\partial T_{\text{air}} / \partial y)_{\text{surface}} = h_c \cdot (T_{\text{surface}} - T_{\text{air,\infty}})
\] (7-1)

Results from the CFD simulation

While the CFD model calculates the temperature gradient according to a detailed physical model (very small air volume cells in the CFD model), the heat transfer coefficient \( h_c \) is used in heat transfer calculations (only one air volume zone in the building simulation). The heat transfer coefficient can be derived from the CFD model using Eq. (7-1) because the heat flow at each surface and the mean air temperature \( T_{\text{air,\infty}} \) are calculated by FLUENT and the surface temperature \( T_{\text{surface}} \) is known:

**According to Fig. 7-6**, the heat flow due to ventilation (here: heat loss during the night) can be calculated from the overall energy balance for the room:

\[
\dot{Q}_{\text{heat,loss}} = \dot{V}_{\text{air}} \cdot \rho_{\text{air}} \cdot (T_{\text{air,\infty}} - T_{\text{air,\in}})
\] (7-2)

Using the input parameters (air volume flow=400 m³/h and \( \Delta T_{\text{inside-outside}}=2 \) K) in Eq. (2), the heat loss is 267 W.

**This heat flow must be equal to the convective heat flow from all surfaces to the air:**

\[
\dot{Q}_{\text{heat,loss}} = \sum_{i=1}^{6} \dot{h}_c,i \cdot A_i \cdot (T_{\text{surf},i} - T_{\text{air,room}})
\] (7-3)

The heat flow between all surfaces and the air is calculated by the CFD model and is 245 W. The convective heat transfer at the ceiling is \( h_{c,\text{ceiling}}=2.4 \) W/(m² K).

The difference between the overall energy balance and the detailed CFD model is less than 9 %. (The difference may be explained with the assumption in Eq. (7-2) that the room air is ideally mixed. This results in the approximation \( T_{\text{air,\infty}}=T_{\text{air,room}} \).) For comparison, typical uncertainties in CFD models are in the range of 20 %, cf. Posner [7-18]. As the difference is sufficiently small, the CFD model can be used to calculate heat transfer coefficients.

A brief literature review

Dascalaki [7-5] gives a comprehensive review of models for the calculation of heat transfer coefficients for building applications. Obviously, each constellation of temperature and flow distribution requires a special model: Models for walls, ceilings and floors take a stably stratified temperature distribution or the buoyancy effect into account. As the heat transfer at the ceiling is very important for night ventilation (high thermal inertia), this study focuses on the heat transfer coefficient at the ceiling. A comparison of different \( h_c \)-models was carried out [7-19] and showed that the model from Awbi and Hatton [7-2] (with the Graßhoff number \( Gr \) and a characteristic length \( L \)) and from Alamdari and Hammond [7-1] (with the temperature difference \( \Delta T \) between surface and fluid and a characteristic length \( L \)) yield the best agreement.

\[
h_{c,\text{Awbi&Hatton}} = 3.52 \cdot Gr^{0.16} \cdot \lambda_{\text{air}} / L \approx 1.824 \cdot \Delta T^{0.16} / \sqrt{L}
\] (7-4)

\[
h_{c,\text{Alamdari&Hammond}} = 0.6 \cdot (\Delta T / L^2)^{0.2}
\] (7-5)

Using the input parameter from Fig. 7-8 and \( L=4.2 \) m (length of the ceiling from Fig. 7-3), Eq. (7-4) gives \( h_c=0.35 \) W/(m² K) and Eq. (7-5) \( h_c=0.9 \) W/(m² K). The accuracy of both models is insufficient: The heat transfer coefficient from the validated CFD
simulation is $h_c=2.4 \text{ W/(m}^2\text{ K)}$. Both the air change rate and the temperature difference between inside and outside affect the air-flow distribution in the room (e.g. Coanda effect, formation of convection rolls and temperature stratification) and, hence, the heat transfer at the surfaces and in particular at the ceiling.

**Proposal for a new $h_c$-model**

The heat transfer coefficient should take the air change rate $A(CH)$, and the indoor air temperature $T_{\text{air, indoor}}$ and outdoor temperature $T_{\text{air, outdoor}}$ into account, cf. Seitz [7-25]. In contrast to Eqs. (7-4) and (7-5), the $h_c$-model does not use the temperature difference between the surface and the air but between indoor and outdoor. This $h_c$-model is a result from preliminary studies and is validated for air change rates from $5 – 7 \text{ h}^{-1}$ and $\Delta T$ from $3 – 5 \text{ K}$ in the room according to Fig. 7-2 and Fig. 7-3:

$$h_{c, \text{night ventilation}} = 0.24 + 0.378 \cdot A(CH) \cdot h^{-1} - 0.037 \cdot (T_{\text{air, indoor}} - T_{\text{air, outdoor}}) \quad (7-6)$$

Eq. (7-6) is valid for the heat transfer coefficient at the ceiling during free ventilation in an office when the outdoor temperature is lower than the indoor temperature.

The heat transfer coefficient is affected by the air flow distribution. As the air-flow distribution is a function of the buoyancy force (specified by $\Delta T$) and the inertial force (specified by $A(CH)$) the model should take into account the Archimedes number, which is the ratio of buoyancy force divided by the inertial force. Therefore, further work has to be done to improve Eq. (7-6) which takes both effects separately into account by linear correlations. However, this preliminary $h_c$-model calculates the heat transfer coefficients at the ceiling with the main driving forces $A(CH)$ and $\Delta T$ and in an appropriate order of magnitude.

**7.4 User behaviour**

There is a broad consensus that people influence their immediate environment if they are exposed to conditions which lie outside their personal comfort range, e.g. Baker [7-3]. Fanger [7-9] specified external stimuli that may contribute to thermal comfort: temperature distribution (outdoor air, indoor air and radiant temperature), humidity and air speed. Obviously, personal variables influence the (manual) setting of windows: occupancy profile, metabolic rate, clothing insulation or noise. Thus, the opening / closing of windows depends on external parameters (i.e. thermal comfort) but is also affected by statistically distributed factors (i.e. personal variables).

Therefore, some investigations on user behaviour have been carried out for many years in order to develop user models. Most investigations refer to residential buildings and aim to determine the influence of window opening on the heating energy demand. A comprehensive AIVC-report [7-6] summarises results from different studies which were carried out between the early 1960s and the late 1980s. In addition to Fanger's criteria, the study describes temporal factors (biorhythmic patterns during the day, season of the year, day of the week and previous thermal experience of the occupants) and spatial factors (room volume, contact with the outside, lighting level, position and type of windows and doors) which influence the user behaviour in residential buildings during winter. Though the study cannot be used directly for an user model in office building in summer, it identifies important boundary conditions.
**Literature review**

Table 7-3 summarises the main findings from other studies which can be used to deduce a user model for window opening in office buildings during the summer.

Table 7-3: Some findings from previous investigations on user behaviour regarding window opening, which can be used for a user model concerning office buildings.

<table>
<thead>
<tr>
<th>Office buildings</th>
<th>Residential build.</th>
<th>Questionnaire</th>
<th>Measurements</th>
<th>Findings</th>
</tr>
</thead>
<tbody>
<tr>
<td>Raja [7-20]</td>
<td>x</td>
<td>x</td>
<td></td>
<td>Users open windows at high room temperatures or poor indoor air quality. Highest correlation with ambient air temperature: Most windows are closed below 15 °C and almost all windows are opened above 27 °C.</td>
</tr>
<tr>
<td>Nicol [7-15]</td>
<td>x</td>
<td>x</td>
<td></td>
<td>A probability algorithm relates occupant behaviour to the outdoor temperature. The „Logit model“ assumes that the probability of an open window increases with the outdoor temperature. Windows are opened more often above 10 °C.</td>
</tr>
<tr>
<td>Warren [7-27]</td>
<td>x</td>
<td>x</td>
<td></td>
<td>In summer, users prevent overheating in buildings that are not air-conditioned by opening windows. Users close windows due to noise. Wind and solar radiation have almost no influence on the user behaviour.</td>
</tr>
<tr>
<td>Fritsch [7-11]</td>
<td>x</td>
<td></td>
<td>x</td>
<td>Window opening does not correlate with indoor but with outdoor air temperature. Windows are seldom closed / opened above 18 °C and remain in a specific status (small switching frequency).</td>
</tr>
<tr>
<td>Knapp [7-13]</td>
<td></td>
<td>x</td>
<td></td>
<td>User behaviour depends on the season (spring / summer / autumn / winter). A time-dependent user profile describes window opening and switching frequency accurately.</td>
</tr>
<tr>
<td>Reiss [7-22]</td>
<td>x</td>
<td>x</td>
<td></td>
<td>Window opening depends on outdoor temperature, season and time of day. Characterisation of three user groups depending on the duration of window opening.</td>
</tr>
<tr>
<td>Russ [7-23]</td>
<td>x</td>
<td>x</td>
<td></td>
<td>Window opening times differ from 0.03 h/h in winter to 0.38 h/h in summer. However, neither the seasonal nor the daily (time-dependent) user behaviour is consistent due to very different switching frequencies.</td>
</tr>
<tr>
<td>IWU [7-7]</td>
<td>x</td>
<td>x</td>
<td>x</td>
<td>In houses with a high insulation standard and a ventilation system, there is (almost) no correlation between window opening and heat energy demand but between window opening and room temperature: Occupants rationally open windows in order to lower the room temperature in summer.</td>
</tr>
</tbody>
</table>

**Data analysis with the ambient air temperature**

The data analysis below is based on only two offices. Therefore, all conclusions are not to be taken as absolute values but as preliminary results, whereby a method for the development of a user model is discussed at the same time.18

Reasonably, a user model can assume that the probability \( p \) of an event happening (in this case window opening) increases as the intensity of the stimulus \( x \) (in this case the temperature) increases. This probability is defined according to the Logit model and is used as regression function for data analysis:

18 Though the data analysis is based on two years of monitored data, it is not possible to carry out a sensitivity analysis due to the very small sample size
\[
\log\frac{p}{1-p} = a + bx \tag{7-7}
\]

where \(a\) and \(b\) are constants. A weighted regression analysis of the Logit equation against the values of \(x\) is performed to give estimates of the values of the constants. Fig. 7-12 shows the relative frequency of open windows in both offices for two years. In addition, the calculated probability from Eq. (7-7) (\(a=2.68\) and \(b=0.25\)) and results from Nicol’s survey [7-15] on naturally ventilated office buildings (\(a=-2.3\) and \(b=0.1\)) are shown:

- In general, more windows are opened in the Lamparter building at higher ambient temperatures than expected from Nicol’s survey.
- Windows are opened more frequently when the ambient air temperature increases but windows are closed (again) at very high ambient air temperatures. However, the sample size (fewer than 50 events in each category) is small at ambient temperatures higher than 30 °C.

The difference between Nicol’s model (naturally ventilated buildings) and the Lamparter building may be attributed to the ventilation system: As the indoor air quality is good, occupants do not need to open the windows at low or high ambient temperatures for a better indoor air quality.

Fig. 7-12: Window-opening behaviour of users, based on two years of monitored data. (Fine lines: fewer than 50 events/category at ambient temperatures above 30°C.)

Fig. 7-13 shows the monthly mean value of the ambient air temperature and the relative frequency of open windows. This graph indicates clearly that the user behaviour varies depending on the season. During the three summer months, the mean probability of open windows is approximately 66%.
Data analysis for the time of the day

Based on the hourly and monthly data analysis, typical user behaviour for the summer period can be derived from the three summer months. Fig. 7-14 shows the relative frequency of open windows during weekdays. Whereas the windows in freely ventilated offices are usually open during the working hours, the windows in the (mechanically ventilated) Lamparter building are closed during working hours. This indicates rational user behaviour:

☐ The occupants keep the windows close during the day. Fresh and cool (earth-to-air heat exchanger) air is supplied by the ventilation system. The indoor air quality is good (sufficient supply air) and the noise level (from outside) is low.

☐ Approximately 70 % of all windows are open during the night to enhance the night ventilation and the cooling of the building.

Fig. 7-14: Time-dependent user behaviour for the summer of 2002 during weekdays. At weekends, the relative frequency is 59 % in the NE office and 65 % in the SW office.
Window opening and air flow through the room

The heat flow due to ventilation can be calculated by combining a user model and an air-flow model: Maas [7-14] derived a semi-empirical air-flow model for single-side ventilation, which can be combined easily with a user model. The air flow is calculated from the opening area $A$, the height $h$ of the window and the temperature difference between the indoor and outdoor air (Eq. 7-8). The heat flow (i.e. heat loss in winter) can be calculated from the window opening frequency (depending on the outdoor air temperature) and the resulting air flow. However, both the user model and the air-flow model show large uncertainties, cf. Schmidt [7-24].

\[
V_{\text{single-side ventilation}} = A \cdot \sqrt{0.0037 \cdot h \cdot \Delta T_{\text{ambient}} - \text{room}}
\]  

(7-8)

Following this approach, the user behaviour and an air-flow model are combined for the summer period in the Lamparter office building. The simplified air-flow model (Eq. 7-9) takes only the buoyancy effect into account, cf. Etheridge [7-8]. The effective opening area ($A_{\text{cD}}$) of the two serial openings is 0.21 m$^2$, cf. Ref. [7-16] for more details, and the driving height $h$ is 4.0 m. Hence, the air flow can be calculated using the room and the ambient air temperature.

\[
V_{\text{cross ventilation}} = (A \cdot c_D) \cdot \sqrt{2 \cdot g \cdot h \cdot \Delta T_{\text{ambient}} - \text{room}} / T_{\text{ambient}}
\]  

(7-9)

Assessing window opening in design. In the design of a passive cooling system, a decision on the ventilation concept has to be made: Mechanical or free night ventilation. If free – wind or buoyancy driven – night ventilation is used for passive cooling, the designer has to decide whether the windows should be opened manually or automatically. Fig. 7-15 shows the results from a statistical simulation using the air-flow model from Eq. (7-9) and the user model according to Fig. 7-14:

- During the summer period (June 1 – August 31, 2002; 2,208 hours), the night time (10 p.m. – 6 a.m.) amounts to 736 hours. During 732 of these 736 hours, the ambient air is cooler than the room air and night ventilation can be used effectively to cool down the building.

- Taking a (mechanical) air change rate of 6 h$^{-1}$ (corresponding to 360 m$^3$/h) as a reference, the average natural air change is as high as this reference almost every night.

- Users forget to open the windows during the night every fifth night on average. If windows are closed, there is only a small air change:
  - If the door is open, the air volume flow is approximately 120 m$^3$/h (mainly single-side ventilation between the office and the corridor).
  - If the door is closed, the air volume flow is reduced to 30 m$^3$/h (small amount of air infiltration due to leakage).
Fig. 7-15: Cumulative duration curves of air flow during night ventilation for summer of 2002. 92 summer days and 8 hours/night gives a total of 736 hours. During 732 of 736 hours, the ambient air is cooler than the room air (night ventilation potential).

**Conclusion.** During the summer, the user behaviour with respect to ventilation can be modelled accurately as a function of the time of the day, whereby the probability for the opening of windows during the night can be estimated from measurements. A statistical model can be used to calculate the air flow through the office.

### 7.5 Conclusions

The passive cooling system maintains a comfortable indoor climate and is integrated into a low energy building which achieves a very low energy demand. Due to the excellent combination of minimised heat gains in summer, high thermal inertia, an efficient ventilation system with an earth-to-air heat exchanger and free night ventilation, acceptable room temperatures were obtained even during long hot periods in the summer of 2003.

The data analysis focuses on parameters influencing the efficiency of night ventilation. The heat loss due to night ventilation is necessary to dissipate the heat which is stored in the building components and especially in the ceiling during the day. A detailed analysis on room air-flow distribution and user behaviour regarding the night ventilation efficiency was carried out and provides two main results:

- During night ventilation, the heat transfer coefficient at the ceiling highly depends on the air distribution in the room. As cool air enters the office under ceiling contrary to the buoyancy effect, the air-flow distribution is mainly influenced by the air change rate and the indoor – outdoor temperature difference (e.g. Coanda effect, thermal stratification and air convection rolls). Thus an $h_c$-model should take (1) the air change rate and (2) the indoor – outdoor temperature difference into account.

- The user behaviour regarding window opening can be divided in typical schedules which vary seasonally. (In this Chapter, a schedule for the summer season is presented for a building with mechanical day and free night ventilation.)

This Chapter shows how convective heat transfer and user behaviour can be used during the design of passive cooling systems.
7.6 References


[7-14] A. Maas, Experimentelle Quantifizierung des Luftwechsels bei Fensterlüftung, University of Kassel, Department of Architecture, 1995. (In German.)


8 Statistical Simulation of User Behaviour

Part C

The statistical simulations are based on conclusions drawn from the data evaluations described in Part B of this thesis. Due to constantly changing meteorological boundary conditions and user behaviour, the building’s thermal behaviour is not defined deterministically but should be determined statistically. Therefore, the deterministic simulation models can be used with statistically distributed input parameters.

A large number of design guidelines and tools are available for the design of passive cooling systems. However, the building engineer should take several uncertainties into account since the actual use of the building, the building physical properties or the user behaviour are uncertain. One promising approach to include these uncertainties in the design procedure is the use of statistical models: The design parameter is defined by a mean value and its deviation. From a control theoretical point of view, the deterministic controlled system responds to random disturbance variables by a statistically distributed response function. This Chapter shows how statistical simulations can be applied to the design process.

8.1 Introduction

In Chapter 4, the influence of material properties and parameters in the air-flow network is statistically analysed in order to prepare and to evaluate the simulation model. For this Chapter, the validated building model from Chapter 5 is used to investigate the impact of user behaviour on the thermal building performance. The following procedure describes the data analysis for the summer period: Statistical analysis of room temperatures, statistical analysis of parameters with regard to the use of the building, statistical simulation of the thermal building performance and comparison of the design simulation with the statistical simulation.

The programming of the Monte Carlo-Simulation (cf. Chapter 4.3) is briefly described in Chapter 8.3.

In preparation for the statistical simulation, several investigations on detail phenomena concerning the thermal behaviour of the Fraunhofer ISE building were carried out to provide a reliable simulation model:
1. The value of internal heat transfer coefficients has been calculated by the use of CFD-simulations, cf. Seitz [8-1].

2. The influence of wind on the natural / hybrid ventilation has been estimated by the simulation of the wind around the Fraunhofer ISE building. The wind pressure coefficients $c_p$ have been calculated by the use of CFD-simulations, cf. Matsche [8-2].

3. Additionally, the infiltration and the interzonal ventilation have been measured by the use of tracer-gas technique, cf. Qiu [8-3].

4. The heat gains can be taken directly or calculated from monitored data: Internal heat gains contain equipment (electricity consumption) and persons (occupancy sensors). Solar heat gains are calculated from window surface area, g-value, solar radiation (meteorological data) and position of the sun-shading (monitoring).

5. Heat gains, losses and storage have been summarised by means of an energy balance calculation (cf. Chapter 5).

Based on this preparatory work and the validated simulation model, the heat flows and the room temperature can be modelled statistically, taking into account user behaviour with respect to equipment, persons, sun-shading (manual control of venetian blinds) and use of windows / sky lights (outdoor) and doors / ventilation flaps (indoor). The statistical models are deduced from a long-term monitoring.

The statistical simulation is supposed to deal with uncertain input parameters and should show whether typical occupancy patterns are effective for modelling the thermal building behaviour and how far statistical models can be used successfully to take uncertainties into account. Finally, the statistical simulation should be used as a design tool. Retrospective, the statistical simulation responds to the question whether the thermal building behaviour (monitoring) is different from the assumptions on which the design process (building simulation) is based.

These questions will be answered according to the following procedure:

1. **Response function.** The room temperatures are statistically analysed.

2. **Excitation function.** The input parameters are statistically analysed with regard to the use of the building (i.e. equipment and occupancy) and user behaviour (i.e. ventilation and sun-shading). Through this comprehensive data evaluation, statistical models for the input and output of the thermodynamic system “building” were derived.

3. **Controlled system – monitoring.** Using these data series as input parameters, the thermal building performance is simulated: Many parameter combinations are considered by the simulation, and each is distributed around a true mean value. It is reasonable to assume that the parameters are normally distributed, cf. Eq. (4-1) in Chapter 4.3.3. As all input parameters are varied simultaneously, the room temperature is also distributed around a true mean value. The results from the simulation are compared with the monitored room temperatures. Starting from this validated model, the impact of the user behaviour on the thermal building behaviour can be investigated independently, without weather and other influences affecting the results.

4. **Controlled system – design.** The simulation model is used with the meteorological data from the design phase but also with the statistical model of real building use (i.e. equipment, occupancy patterns and user behaviour with regard to ventilation and sun-shading). Since both the design and the statistical model are simulated with the same meteorological data, differences in the predicted room temperatures are due to differences in the building’s physical parameters (i.e. g-value, material
properties and solar obstructions) and in the use of the building. (Anticipatory, it can be said that the difference in the building physical model is negligible. The different room temperatures are caused only by the different use of the building.) From the comparison between these simulation results and the results from the design process, the influence of the user behaviour on the room air temperatures can be defined: What is the difference between the design user and the real user?

8.2 Data evaluation

For the statistical data analysis and simulation, the time period and the sample of offices must be defined. For this data sample, the variation in time of room temperatures and heat gains is analysed. The user behaviour is analysed with special regard to the ventilation.

Time period. Starting from the hourly ambient air temperatures in 2003, a typical period can be defined. In this context, a "typical period" is characterised by the altitude / position of the sun and similar ambient air temperatures in order to apply consistent user profiles for the closing of sun protection and the opening of windows. As temperatures never fall below 12 °C between June 1 and August 31, 2003 (cf. Fig. 8-1, long period of warm weather), the simulation period should take place within this summer period. The period June 12 to July 23, 2003 is short enough to carry out many simulation runs in a row and long enough to cover different summer weather conditions since similar ambient air temperatures and sun positions in this time period will produce likely consistent user behaviour.

![Graph of ambient air temperature 2003](image)

Fig. 8-1: Ambient air temperature 2003: Summer period with ambient air temperatures above 12 °C.

Office sample. 16 offices in the Fraunhofer ISE building are considered for the statistical data evaluation and simulation. Since all rooms are located in the first and second floor in the same part of the building, the climate impact from the outside is the same. However, the mean room temperatures vary from 25.9 to 26.9 °C, which is 3 to 4 K warmer than the mean ambient air temperature (cf. Fig. 8-2), since the offices vary in the daily attendance and the office equipment (internal heat gains) and the user behaviour regarding sun protection (solar heat gains) and ventilation (heat losses).
Variation in time of room temperatures

The following procedure provides the variation in time of room temperatures and its statistical distribution.

1. Preparation of hourly room temperatures in each of the 16 offices.
2. Calculation of daily mean room temperature and daily temperature fluctuation in each office.
3. From this time series, mean room temperature, 16 and 84% quantile are identified for each day. The minimum / maximum temperatures specify the limit of variation.
4. Additionally, the hourly mean room temperature is calculated from the hourly room temperature in each of the offices.

There are no data gaps within the period considered for data analysis.

Exemplarily for this procedure, Fig. 8-3 shows the time behaviour of the daily mean room temperature (in 16 offices) and its daily deviation / variation in all rooms:

- The mean deviation is 0.4 K and varies from 0.15 to 0.9 K.
- The mean variation is 1.45 K and varies from 0.7 to 2.5 K.

As the climatic and the building physical boundary conditions are (almost) identical in all rooms, the temperature variations result partly from the thermal stratification, but mostly from variations in use of the offices. In the following, the variations in user behaviour and heat gains are evaluated in order to interpret the observed temperature variations.
User behaviour regarding ventilation

The following procedure provides the user behaviour regarding ventilation and its statistical distribution.

(1) Preparation of the hourly status (open or closed) of door, ventilation flap (indoor), sky light (outdoor) and window in each of the 16 offices. If a room has more than one window / sky light these openings will be combined by an OR-relation.

(2) The time series differentiates working days and weekends.

(3) All data lines are sorted by the time of day.

(4) From this information, relative frequencies of opening are calculated for each ventilation component and for each hour of the day.

(5) Using the hourly data regarding the status of all openings in each of the 16 offices, the local distribution of hydraulic resistances (mean value and statistical distribution) is calculated using an air-flow network with resistances in series (16 offices) and parallel (office – corridor).

There are no data gaps within the period considered for data analysis.

Exemplarily for this procedure, Fig. 8-4 shows the average user behaviour in 16 offices during 30 working days:

☐ As expected, doors are closed outside business hours. More or less than 50 % of all doors are opened during the working hours. At least 90 % of all ventilation flaps above the door are opened during the whole day. These flaps are manipulated very rarely.

☐ In general, windows are opened at arrival and are closed bit by bit when the ambient air temperature is increasing. At least 50 % of all windows are opened during night. Users rarely adjust the sky lights above the window. At least 80 % of all sky lights are opened.

☐ The exhaust fan delivers an air change rate of 1 h⁻¹ (480 m³/h for 8 offices) during the working hours and 5 h⁻¹ (2,400 m³/h for 8 offices) during night ventilation in each floor (cf. Chapter 5). If the hydraulic resistance was the same in each of the offices, each office would get the same fraction of the air-volume flow which is 1/8 or 12 % of the total air-volume flow. Due to the changing distribution of open and
closed flaps, doors and windows, the air-flow network (hydraulic resistances in series) constantly changes. Some offices get less than 4 % and other rooms up to 18 % of the air flow.

Fig. 8-4: Time dependent user behaviour regarding ventilation during the working days. Left: Manual opening of ventilation components. Right: Mechanical air change due to different user behaviour in adjacent rooms.\(^{19}\)

\(^{19}\) The corridor has one door to the magistrale and one door to the outside, cf. Fig. 5-2 in Chapter 5.1. If these doors are open, the exhaust fan does not draw the air through the offices. As both doors are usually closed for fire protection and safety-at-work reasons, their effect on the air-flow network is negligible.
Variation in time of heat gains

The following procedure provides occupancy patterns, the time variation of electricity consumption, the use / control of sun-shading and its statistical distribution.

(1) The hourly heat gains due to persons are calculated by the ultrasonic sensor and correspond to 80 W/person. Missing or incorrect data are substituted by an average time profile which is calculated from all available sensors.

(2) The total electric power consumption \([W_{\text{total}}]\) is hourly measured. Furthermore, the electricity consumption \([kWh_{\text{office}}/(m^2 \text{ d})]\) in an office is dependent on the electric connection power and the estimated operation time of the equipment, cf. Selg [8-4]. Thus, the total electricity consumption can be divided into the electricity consumption in each office, if the attendance in each office is also known. With this information, the electric power \([W_{\text{office}}]\) is calculated for each office.

(3) The hourly solar heat gains are calculated from the solar radiation on the window, which is calculated from meteorological data, taking into consideration shading from adjacent buildings and the facade, cf. Herkel [8-5], as well as the position of the venetian blinds, cf. Gorzelanny [8-6]. The data acquisition is realised by a web camera. The error-prone data evaluation by image processing results in some data gaps which are substituted depending on their length of time: If the status of the venetian blind is not recognised, the last valid value will be extrapolated. If the data gap covers several hours, the missed value will be substituted by the mean value of all known positions.

Through this process, hourly data for internal and solar heat gains are available for each office. Fig. 8-5 outlines the daily heat gains in all of the 16 offices during 30 days and its variation. The heat gains vary from one room to the next and can vary from one day to the next:

- Due to the different application of computers, the daily heat gains from equipment (grand average: 137 Wh/(m² d)) varies from 58 to 295 Wh/(m² d).
- Since the rooms can be occupied by as few as one and as many as four persons and the mean attendance in the office is variably long (e.g. laboratory workers), the mean daily heat gains from persons (grand average: 54 Wh/(m² d)) varies greatly from 26 to 103 Wh/(m² d).
- In comparison, the variation in solar heat gains (grand average: 158 Wh/(m² d)) is relatively small, “only” 30 %: The room, in which the occupants close the venetian blinds most frequently, gains on the average 106 Wh/(m² d), the room with rarely closed blinds gains meanly 210 Wh/(m² d).
- Heat gains can vary heavily from one day to the next: In some offices, the standard deviation reaches 70 % (equipment), 93 % (persons) and 43 % (solar) of the mean value.
Fig. 8-5: Daily heat gains and 1-σ deviation (working days). The mean heat gains are slightly higher in the 8 offices in the 2nd floor (362 Wh/(m² d)) than in the 1st floor (333 Wh/m² d)).

The 8 offices in the 1st floor are 0.4 K cooler than the 8 offices in the 2nd floor, cf. Fig. 5-7 in Chapter 5. As the impacts (i.e. thermal stratification, sun-shading by adjacent building parts and the higher heat gains) on the temperature difference cannot be distinguished from each other, the 16 offices are evaluated together.20

8.3 Statistical simulation for summer 2003

There are approximately $10^{18}$ possible combinations from the 16 heat gain series and the status of ventilation components within the period of 1,008 hours, but the number of simulation runs can be reduced to 1,000 by the use of statistically distributed input parameters. The Monte-Carlo simulation deals with statistical input parameters which are defined by a true average and a realistic deviation, cf. Chapter 4. All input parameters are varied at the same time. The validated simulation model from Chapter 5 is applied to the statistical simulation, which requires that all internal surfaces have adiabatic boundary conditions.

The following procedure describes the program flow of the statistical simulation according to the Monte-Carlo simulation concept. The aim is to model the heat gains as close to reality as possible and to investigate the user behaviour with regard to ventilation.

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20 Confirmatory, a sensitivity analysis shows that the internal heat gains and the user behaviour are the dominant parameters in the energy balance compared to the thermal stratification or the sun-shading by adjacent building parts.
1. Random determination of mechanical air change according to the statistical distribution in Fig. 8-4 (right).

2. Calculation of user behaviour concerning the window opening. For each hour of the day, the status of window, sky light, ventilation flap and window is calculated with the Gauß function. Mean value, cf. Fig. 8-4 (left), and standard deviation are known from the data analysis. The time series is mathematically conservative: The opening status increases or decreases over the time but does not oscillate from one hour to the next.

3. The hourly time series of the internal and solar heat gains for one room is taken from the data analysis. At the next time step, the next room is chosen. After 16 simulation runs, the procedure starts with the first room again. Mean values and standard deviation are shown in Fig. 8-5.

4. For each simulation run, the hourly room air temperatures are saved.

5. The statistical analysis corresponds to the data analysis of the monitored room temperatures according to Fig. 8-3.

Since the input variables are statistically distributed, the calculated room temperatures are statistically distributed as well. The simulated temperatures are compared with the monitored room temperatures in order to evaluate the statistical input models. Fig. 8-6 shows that the mean temperature variation in the simulation closely approximates the variation in measurements.

☐ In this 42-days period, the mean measured and simulated room temperatures differ from each other by 0.2 K.

☐ Every day, the simulated room temperatures meet the monitored temperatures within the standard deviation.

☐ As expected, the deviation (of 16 offices) is higher in the simulation than in the monitored data at each day since the balancing heat transfer between adjacent rooms is disconnected by the adiabatic boundary conditions.

Fig. 8-6: Daily mean room temperatures: The measured room temperature is 26.4 °C and the simulated 26.2 °C. The mean ambient air temperature is 22.9 °C.
While Fig. 8-6 shows a good agreement in the mean temperatures, Fig. 8-7 evaluates the dynamic temperature behaviour:

- In this 42-days period, the mean measured and simulated daily fluctuations differ from each other by 0.5 K. Since temporary fluctuations of heat gains or losses cannot be balanced by heat conduction from / to adjacent rooms due to the adiabatic boundaries, the simulation shows a higher daily fluctuation.

- The variation in time is very similar. The simulated daily temperature fluctuations match those measured during 39 of the 42 days.

- Noteworthy, the deviation (of 16 offices) in the simulation is smaller than in the monitored room temperatures on some days. High variations in measurements may be explained by the measuring system and the local position of temperature sensors (cf. Fig. 5-10 in Chapter 5).

![Graph of daily fluctuation of room temperatures](image)

**Fig. 8-7: Daily fluctuation of room temperatures:** The average measured fluctuation is 3.5 K, the average simulated fluctuation is 4.0 K and the average fluctuation of the ambient air temperature is 11.8 K.

The duration curve, according to Fig. 8-8, estimates the comfort criteria concerning the room temperatures and its occurrence frequency. The simulation comes to a similar conclusion concerning overheating but shows a slightly wider tolerance band.\(^{21}\)

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\(^{21}\) Due to the heat gains, the maximum *daily room temperature* is higher than the maximum daily ambient air temperature. However, the maximum *hourly room temperature* is below the maximum hourly ambient air temperature due to the heat storage capacity. This observation is discussed in Chapter 9, Fig. 9-1.
The passive cooling system is designed in order to hold the room temperature in the comfort range. Fig. 8-9 shows clearly that the simulation (hourly room temperature from 1,000 simulation runs) and the monitoring (hourly room temperature from 16 offices) result in the same benchmark: During this extreme weather situation, the room temperatures do not meet the comfort criteria. The room temperature exceeds the tolerable temperature during 623 hours (measurement) and 562 hours (simulation) of the 1,008 analysed hours. Additionally, Fig. 8-9 shows exemplarily the temperature variation in time for 24 hours. The dynamic thermal building behaviour is also simulated realistically.

The results of this comprehensive data evaluation reveal that the thermal building simulation model combined with the statistical user model characterise the thermal building performance accurately.

8.4 Statistical simulation with design parameters

In the design process, the user behaviour with regard to the ventilation and the manual control of the sun-shading and the internal heat i.e. gains (use of office equipment,
electric lighting and occupancy patterns) had to be estimated and were taken from the available models at the time, cf. Herkel’s report on the design of the Fraunhofer ISE building [8-5]. With the statistical simulation, these design assumptions can be checked against the real building operation – assuming that users behave in the average summer (i.e. test reference year TRY 7 [8-7]) similar to the summer 2003.

Fig. 8-10 shows that the test reference year TRY7 takes neither high ambient air temperatures nor long summer periods with warm weather into account, since the design data set differs greatly from the measurements. In the following, the predictions from the design simulation are reviewed using the statistical simulation with the same weather data set TRY 7 but also the user models derived from the data evaluation.

![Fig. 8-10: Comparison of ambient air temperature for different years: In the design simulation the test reference year TRY 7 was used. The summer 2002 was a typical summer. Obviously, the test reference year does not consider high summer ambient air temperatures. The summer 2003 was 5.2 K warmer than the test reference year and 3.7 K warmer than the summer 2002.](image)

For comparison studies, the design and the statistical simulation has to be adjusted in order to operate with the same time schedules of heat gains. As the heat gains are imported from a time series, the design model must be simulated with the calendar of the year 2003 and the weather data from the TRY 7 to accurately take the working days and weekends accurately into account. The solar heat gains in 2003 differ from TRY 7 due to different solar radiation data. Therefore, the solar heat gains in the statistic model are adapted to the test reference year by multiplication with the ratio of the solar radiation in 2003 to the solar radiation in the TRY 7.

Fig. 8-11 shows (1) that the room temperatures in the statistical simulation follow a qualitatively similar variation in time to the room temperatures predicted by the design simulation (2) but that the room temperature is clearly higher in the statistical model (realistic user behaviour). Even the coolest mean room temperature for the whole period (considering the standard deviation) in the statistical simulation is above the mean room temperature from the design simulation, cf. Table 8-1.
Fig. 8-11: Daily mean room temperature from design and statistical simulation.

What are the reasons that the room temperature is higher in the operation than predicted in the design process?
1. The design simulation operates with constant air change rates which depend on the room air temperature and the time of the day. Though the statistical simulation is based on a much more complex air-flow network which takes user behaviour regarding the ventilation into account, the air change rates are similar in the design and the statistical simulation. This is an insignificant difference.

2. While the night ventilation is interrupted in the design simulation at low ambient air temperatures, this cooling potential is used in the ordinary operation (statistical model) during the summer nights (cf. Chapter 5). Due to this different operation management, the night ventilation potential is better incorporated into the statistical than in the design simulation at low ambient air temperatures. In spite of the higher nocturnal heat dissipation, the mean room temperature calculated by the statistical simulation with the real user behaviour is 1 K higher than the room temperature predicted by the design simulation due to the heat gains.

3. Fig. 8-12 outlines that the monitored solar and the internal heat gains are higher than their estimates in the design simulation, because: (1) The work stations are located in the offices and not – according to the assumptions from the design phase – in separated IT rooms. (2) As the venetian blinds are mainly used to protect glare, they are more rarely closed in summer time than estimated. These are two significant differences concerning the internal and solar heat gains between the estimated (design simulation) and the monitored data (statistical simulation).

![Figure 8-12: Daily heat gains from design and statistical simulation.](image)

Mean temperature in the period June 12 to July 23. Table 8-1 shows the mean room temperature and the mean heat gains: In the statistical simulation, the significantly higher heat gain results in a higher (mean) room temperature since the additional heat gain cannot be counterbalanced by the heat dissipation due to the (night) ventilation.

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22 Noteworthy, the actually realised g-value of the facade is lower than the designed value: The monitored solar heat gains during the weekends are lower than the designed since the blinds are closed and no or only a few users open the blinds. Concerning the calculation of the g-value and the solar heat gains, the reader is referred to Chapter 5.5.2 and Table 5-1.
Table 8-1: Mean room temperature and heat gains from design and statistical simulation.

<table>
<thead>
<tr>
<th></th>
<th>room temperature (with standard deviation)</th>
<th>heat gains (working days)</th>
<th>heat gains (weekend)</th>
</tr>
</thead>
<tbody>
<tr>
<td>ambient air temperature</td>
<td>18.47 °C</td>
<td>–</td>
<td>–</td>
</tr>
<tr>
<td>design simulation</td>
<td>22.61 °C</td>
<td>214 Wh/(m² d)</td>
<td>75 Wh/(m² d)</td>
</tr>
<tr>
<td>statistical simulation</td>
<td>23.87 °C (+/- 1.01 K)</td>
<td>380 Wh/(m² d)</td>
<td>121 Wh/(m² d)</td>
</tr>
</tbody>
</table>

**Daily mean temperature.** The duration curve is an important criterion in the design process. Fig. 8-13 underlines the conclusion that the assumptions from the design procedure were too confident: Taking 24 °C (mean daily temperature) as a standard of comparison, the design simulation forecasts only 3 days with a higher daily room temperature, but the daily room temperature exceeds 24 °C for 20 days in the simulation based on monitored data.

**Conclusion.** Both the design and the statistical model are simulated with the same meteorological boundary conditions (here: test reference year TRY 7), but the simulated room temperatures differ greatly. As all other boundary conditions are (almost) identical, the higher room temperatures must be directly associated with the use of the building.

![Fig. 8-13](image)

**Fig. 8-13:** Duration curve of daily mean room temperatures. During a long period the actual room temperature is at least 1 K above the room temperature forecasted by the design simulation.

**Hourly room temperature.** Fig. 8-14 shows how the room temperature (design simulation and mean value of 1,000 simulation runs) increase with the ambient air temperature:

- On the left side, the hourly room temperatures for 24 hours / day are shown. Though the mean room temperature is higher in the statistical simulation with the real user behaviour than in the design simulation,
  - the highest room temperatures are predicted by the design simulation (due to the determined time series of heat gains) and
  - the lowest room temperatures are simulated by the statistical simulation (due to the uninterrupted night ventilation).

- On the right side, the hourly room temperatures during the working time and the comfort range are shown. This graph shows clearly that design and monitoring
result in different comfort benchmarks: The room temperature exceeds the tolerable temperature in 51 hours or 12 % (design) in opposite to 94 hours or 22 % (monitoring) of the 420 working hours.

**Conclusion.** As expected from the previous Sections, the non-achievement of comfortable room temperatures is caused by the use of the building and cannot be attributed to the design. In other words, the design assumptions with regard to the use of the building have been too optimistic.

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**Fig. 8-14:** Room versus ambient air temperature. Left: All hourly data and indoor = outdoor air temperature line for comparison. Right: Only working hours and comfort criteria according to DIN 1946, cf. Chapter 2.1.3.

**Fig. 8-15** shows the results from a Fourier analysis (cf. Chapter 6.3) of the ambient air temperature, the design and the statistical simulation. The statistical simulation is analysed for the hourly mean room temperature from 1,000 simulation runs:

- From a control theoretical point of view, the building acts like a low-pass filter for fast changing input parameters (i.e. excitation function) due to its thermal inertia, cf. Chapter 6.3.1. Slow variations of input parameters (here: ambient air temperature) are slightly modulated due to user behaviour since windows and blinds are closed / opened according to the room temperature and the weather. For these two reasons, the response function (here: room temperature) is described by only a few frequencies.

- Due to heat gains, the mean room temperature is higher than the ambient air temperature. Taking the real use of the building into account, the mean room temperature is 1.2 K higher than calculated by the design simulation.

- As expected, the 1/d-frequency dominates the frequency spectrum. The daily amplitude is in the same order for the design and the statistical simulation. This indicates that the dynamic building performance is accurately simulated.

- Since the design simulation is based on typical occupancy patterns, the frequency spectrum shows characteristic frequencies in the design simulation which are equalised by the statistical approach due to overlapping events (e.g. arrival and departure time at the office or manual control of windows and blinds). For example, the 1/3d-frequency in the design simulation corresponds to 8 working hours and the 1/8d-frequency to the main occupancy 3 hours before and 3 hours after lunch break. These well-defined patterns are "dispersed" by the actual building use which is taken into account by the statistically distributed user behaviour.
Fig. 8-15: Fourier spectrum from simulation results. (Simulation with weather data from the test reference year.)
Table 8-2 summarises the results from the Fourier analysis for 4 main frequencies: The average, the 1st frequency (long term dynamic), the 1/1d frequency (short term dynamic) and the 1/8hrs frequency (working time). While the amplitude attenuations are similar for the daily frequency (50 %), the long and short-term dynamics are different:

- **Average.** *The design simulation underestimates the heat gains.* The monitored room temperature is 1.3 K higher than predicted.

- **42 days.** *The design simulation overestimates the time constant.* The building responds faster to weather changes since users do not manually control windows when the ambient air temperature changes (assumption in the design simulation) rather at certain times of the day.\(^{23}\) Thus, low and high ambient air temperatures have a larger impact on the room temperature than estimated by the design simulation. This is a significant difference concerning the ventilation.

- **1 day.** *The design simulation takes the short-term behaviour accurately into account.* The daily amplitude is predicted accurately in spite of two very different approaches concerning heat gains and losses.

- **8 hours.** *The design simulation depends on an oversimplified user model.* There is no well-defined time of occupancy.

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**Table 8-2: Results from the Fourier analysis.**

<table>
<thead>
<tr>
<th></th>
<th>ambient air [amplitude]</th>
<th>design [amplitude]</th>
<th>statistical [amplitude]</th>
</tr>
</thead>
<tbody>
<tr>
<td>average</td>
<td>18.5 °C</td>
<td>22.6 °C</td>
<td>23.9 °C</td>
</tr>
<tr>
<td>cycle period : 42 days</td>
<td>2.1 K</td>
<td>1.0 K</td>
<td>1.4 K</td>
</tr>
<tr>
<td>cycle period : 1 day</td>
<td>3.6 K</td>
<td>1.85 K</td>
<td>1.8 K</td>
</tr>
<tr>
<td>cycle period : 8 hours</td>
<td>–</td>
<td>0.4 K</td>
<td>0.1 K</td>
</tr>
</tbody>
</table>

---

\(^{23}\) As a result from the data analysis, the manual control of openings is modelled as a daily time profile which is a function of the seasonal but not the actual ambient air temperature.
8.5 Conclusions

Four lessons can be learned from the statistical simulation:

1. A statistical data analysis provides adequate user models for application in building simulation.
2. The Monte-Carlo simulation is an appropriate tool to calculate the thermal building performance with a true mean value and a statistically relevant deviation.
3. At the time of the design simulation, the available user models were used. Obviously, the design assumptions with regard to the heat gains have been too optimistic and user behaviour should have been modelled in greater detail.
4. Statistical simulation can be advantageously applied to the design process and enhances the significance and clarity of simulation results.

8.6 References


[8-7] A. Jahn, Entwicklung von Testreferenzjahren TRY für Klimaregionen der Bundesrepublik Deutschland, Bundesministerium für Forschung und Technologie (Forschungsbericht T 86-051), Düsseldorf, Germany, 1986. (In German.)
Chapter 9

9 Cross-Section Analysis

Part C

In Part B of this thesis, the night ventilation efficiency is calculated for each office building separately and in connection with the individual building and energy concept. The previous Chapter shows how uncertainties and variations in the thermal performance can be estimated by a statistical approach. Hence, the monitored data can be comparatively evaluated according to the methodology from Part A.

9.1 Introduction

The thermal building behaviour is analysed by universal temperature graphs and the overall energy balance of the offices in summer. This approach reduces the information concerning the passive cooling system to the essential minimum. If the energy balance can be transferred into the temperature characteristic of a building with a few parameters, a comprehensive thermal building model can be derived from ordinary temperature measurements.

Accordingly, it has been shown in the Chapters 4 – 7 that the indoor air temperature and the energy balance of the room result in a consistent model. Starting from this model, the four low-energy office buildings DB Netz AG, Fraunhofer ISE, Pollmeier and Lamparter are analysed by the same data model for two summer periods. While the DB Netz AG building is examined for the two typical summers 2001 and 2002, the Fraunhofer ISE, Pollmeier and Lamparter buildings are analysed for the typical summer 2002 and the very warm summer 2003.

If the thermal performance in each building can be discerned accurately by this method, the data model can be practically used for quality assurance of the building simulation in the future since the complex interactions calculated by the building simulation can be verified with a few concise parameters. Furthermore, measurements in building, e.g. during the implementation phase, can be analysed using this simplified data analysis with an acceptable time effort.

9.2 Summer 2003 – its impact on the design of cooling concepts

In Germany, the three summer months June, July and August 2003 were significantly too warm. The daily mean temperature was 19.6 °C, 3.4 K above the reference temperature. This extreme summer weather can be predicted by the numerical climate models; summer conditions like those experienced in 2003 should occur statistically every 1,000 years [9-1] – even if the anthropogenic global warming is taken into account. In particular regions (e.g. Freiburg), the mean ambient air temperature was 5 K higher than the reference year.
If the summer 2003 was used as design weather for passive cooling concepts, (almost) every building would need air-conditioning to meet the comfort criteria. Though the summer 2003 is not suitable for design studies, its impact on the design of mechanical and passive cooling systems is discussed intensely, cf. Ref. [9-2]. Several adjudications which demand a maximum indoor temperature below 26 °C in offices, cf. Ref. [9-3], [9-4] and [9-5], enhance additionally the uncertainty concerning the technical realisation of passive cooling systems, though the underlying assessments do not correspond to the scientific point of view, cf. Chapter 2.1.3.

Thus, the analysis in this Chapter is not only supposed to describe the methodological approach but also to show whether passive cooling systems can be effective even under extreme weather conditions.

### 9.3 Cumulative duration curves

Fig. 9-1 compares the cumulative duration curves of the room temperature and the ambient air temperature for the four low-energy office buildings. The graphs take only working hours into account.

- The DB Netz AG graph shows clearly that the yearly frequency distribution of room temperatures is similar, if the weather (here: 2001 and 2002) is similar.

  **Conclusion 1**: As expected, if the weather (and the other boundary conditions) do not change, the thermal performance of a building remains also unchanged.

- A “summer day” is defined by a daily maximum temperature of 25 °C. Furthermore, the simplified comfort criteria from Chapter 2.1.3 refer to 25 °C. For these reasons, this temperature limit is used as a consistent standard for comparison. The graphs for Fraunhofer ISE, Pollmeier and Lamparter show generally the same behaviour: Each building exceeded 25 °C more often during the warm summer 2003 than during the typical summer 2002, which is characterised by the degree hours Dh in Table 9-1.

  **Conclusion 2**: Obviously, if the weather changes, the thermal behaviour of a building (here: degree hours over 25 °C) will change.

- However, the change of the degree hours [Kh] above 25 °C (ambient air and room temperature) from 2002 to 2003 is different for the three projects: In summer 2003, the degree hours above 25 °C for the ambient air temperature were 1.903 Kh (Fraunhofer ISE), 644 Kh (Pollmeier) and 738 Kh (Lamparter) higher than in 2002. In summer 2003, the degree hours above 25 °C for the room temperature were 1.628 Kh (Fraunhofer ISE), 365 Kh (Pollmeier) and 453 Kh (Lamparter) higher than in 2002. In the following, the ratio of the degree hours $R_T$ will also be used in order to evaluate the thermal building performance for the two summer seasons:

$$R_T = \frac{(Dh_{2003} - Dh_{2002})_{RT}}{(Dh_{2003} - Dh_{2002})_{AT}}$$

(9-1)

In actively cooled buildings the ratio $R_T$ is 0. As the cooling system provides a constant room temperature, there is no difference of degree hours above 25 °C.
Table 9-1: Degree hours over 25 °C.

<table>
<thead>
<tr>
<th></th>
<th>DB Netz AG</th>
<th>Fraunhofer ISE</th>
<th>Pollmeier</th>
<th>Lamparter</th>
</tr>
</thead>
<tbody>
<tr>
<td>$D_h_{\text{outdoor air}}$ [K]</td>
<td>328 436</td>
<td>684 2062</td>
<td>708 1240</td>
<td>501 1069</td>
</tr>
<tr>
<td>$D_h_{\text{indoor air}}$ [K]</td>
<td>203 272</td>
<td>309 1661</td>
<td>63 340</td>
<td>109 472</td>
</tr>
<tr>
<td>$R_T$ [K/K]</td>
<td>-</td>
<td>0.86</td>
<td>0.57</td>
<td>0.61</td>
</tr>
</tbody>
</table>

The difference of the degree hours approximately corresponds to the area between the 2002-line and the 2003-line for the ambient and the room air temperature, respectively. The ratio of the two areas indicates how the building can compensate for high ambient air temperatures: The smaller the ratio, the less the ambient air temperature affects the room temperature. The Fraunhofer ISE building (86 %) has coped with the summer 2003 worse than the Pollmeier (57 %) or the Lamparter building (61 %), since the yearly difference of the ambient air temperature is larger at the Fraunhofer ISE building than at the other buildings. At the Fraunhofer ISE building, the degree hours over 25 °C in 2003 were 3 times higher than in 2002, but at the Pollmeier and the Lamparter building only 1.75 and 2.13, respectively.

**Conclusion 3:** If the chronology of climate situations (here: periods of warm weather) changes, the thermal performance of a building will change.
Fig. 9-1: Cumulative duration curves of the room temperatures. The ratio of the area between the ambient and the room temperature shows how the building compensates for high ambient air temperatures. If the ambient air temperature is similar, the room temperature is also similar since these short-term variations can be attenuated. The more the duration curve for the ambient air temperature changes, the larger the ratio of the ambient and the room temperature areas. These long-term variations cannot be attenuated since the heat storage capacity is completely used causing the room temperature to increase at high ambient air temperatures.

9.4 Room versus ambient air temperature

The cumulative duration curves show how often the temperature exceeds a specific limit. However, these curves cannot be used to evaluate the comfort since the criterion “temperature limit” depends on the ambient air temperature. Fig. 9-2 shows the indoor-outdoor-temperature graph which is independent of the actual weather:

- The DB Netz AG building provides comfortable room temperatures though the actual room temperature does not always meet the exact comfort range.
- The room temperature in the Fraunhofer ISE building is too high. While the room temperature was slightly too high in 2002, the room temperature did not meet the comfort range during the summer 2003 (temporarily 3 – 4 K too warm).
- The Pollmeier (low heat gains) and the Lamparter building (earth-to-air heat exchanger) provided comfortable room temperatures in summer 2002. In 2003, the room temperature was 1 – 2 K too high.

Based on the conclusions concerning the duration curves in Fig. 9-1, the following thesis will be investigated in the following Sections: The increased failure to meet the comfort standard in 2003 can be explained by using smaller time constants $\tau$ than in 2002.
The building performance can be derived from a regression analysis for the graphs from Fig. 9-2 and further explained by the results shown in Fig. 9-3. The variation of the room temperature can be divided into two ranges:

- During the heating period, occupants use the radiators to maintain their desired room temperature. In a wide range of weather conditions (i.e. especially the ambient air temperature $T_a$) the room temperature $T_i$ remains at this constant temperature $T_{i,\text{base}}$.

- At higher ambient air temperatures the room temperature increases due to the higher temperature level. The ambient air temperature $T_{a,\text{limit}}$, at which the room temperature starts to increase, is primarily a function of the gain/loss-ratio $\gamma$. The slope $S$, with which the room temperature increases, is a function of the gain/loss-ratio $\gamma$ and the time constant $\tau$. (For further details, the reader is referred to Chapter 2.1.3 and the parametric model.)

Hence, the regression analysis is based on a linear function:

$$T_i(T_a) = T_{i,\text{base}} \text{ if } T_a \leq T_{a,\text{limit}} \quad \text{and}$$

$$T_i(T_a) = T_{i,\text{base}} + S \cdot (T_a - T_{a,\text{limit}}) \text{ if } T_a > T_{a,\text{limit}}$$

(9-2)

The graphs verify the assumption that the building parameters change with the weather conditions, while the gain/loss-ratio $\gamma$ remains essentially constant.

In contrast to the DB Netz AG building (summer 2001 and 2002 with similar weather), the temperature gradient in 2003 is higher than in 2002 in the Fraunhofer ISE, Pollmeier and Lamparter buildings. Thus, the time constant $\tau$ of a building is not a constant parameter but rather depends on the weather and its variation in time (here: short and long periods of warm weather).
Taking the parameters from Eq. (9-1), more details can be discussed supplementary to the passive cooling: The winter room temperature $T_{i,\text{base}}$ in the Lamparter building is higher than in the other buildings due to the air heating. As the Pollmeier building consists of open-plan offices, the room temperature is also slightly higher. The limit temperature $T_{a,\text{limit}}$ for the Lamparter building is higher than for the other buildings since the variation of the supply air temperature is attenuated by the earth-to-air heat exchanger. At the DB Netz AG building this cooling effect is counterbalanced by the heat gains from the atrium, which causes the room temperature to increase with the ambient air temperature. The limit temperature is also below the heating balance temperature at the Fraunhofer ISE due to high heat gains and the Pollmeier building as well due to fresh air through slit valves.

During a typical summer (here: 2002), all buildings have a similar temperature gradient $S$ from 0.22 to 0.25 $K_{RT}/K_{AT}$ at high ambient air temperatures. Due to a low air change rate, the time constant $\tau$ is higher for the Pollmeier building which results in a small temperature gradient.

The higher the gain/loss-ratio $\gamma$ and the smaller the time constant $\tau$, the higher is temperature gradient $S$. Starting from a nearly constant gain/loss-ratio $\gamma$, the temperature gradient $S$ of a building is primarily a function of the building’s time constant $\tau$. In each building, the temperature gradient $S$ was approximately 50 % higher in 2003 than in 2002. Obviously, the time constant $\tau$ is smaller in 2003 than in 2002.
Table 9-2: Parameter from a regression analysis according to Fig. 9-3.

<table>
<thead>
<tr>
<th></th>
<th>DB Netz AG</th>
<th>Fraunhofer ISE</th>
<th>Pollmeier</th>
<th>Lamparter</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_{\text{base}}$ [°C]</td>
<td>21.9</td>
<td>21.4</td>
<td>22.0</td>
<td>23.0</td>
</tr>
<tr>
<td>$T_{a,\text{limit}}$ [°C]</td>
<td>10.1</td>
<td>10.8</td>
<td>11.9</td>
<td>19.2</td>
</tr>
<tr>
<td>slope S [K RT/KAT]</td>
<td>0.23</td>
<td>0.24</td>
<td>0.25</td>
<td>0.35</td>
</tr>
</tbody>
</table>

These conclusions can be verified by a simulation study. A typical office room according to the specifications from Chapter 5 is simulated without internal and solar heat gains but with a typical air change during the day. Consequently, the mean room temperature is identical to the mean ambient air temperature since the room is in thermal balance with its environment. The weather is simulated for periodic steady state conditions whereby a mean summer day ($T_{a,m}=19.4$ °C) and a warm summer day ($T_{a,m}=26.3$ °C) alternate in certain time intervals:

- 2 average days followed by 1 warm day,
- 10 average days followed by 5 warm days and
- 40 average days followed by 20 warm days.

As in each simulation the average period is twice longer than the warm period, the mean temperature in each simulation is 21.7 °C, but Fig. 9-4 shows that the temperature distribution differs: While short-term variations are highly attenuated, the thermal inertia is not large enough to mitigate variations with a longer cycle period. Thus, the room cannot compensate for the temperature variations at very long cycle periods and the room temperature is coupled more directly to the ambient air temperature. Due to this close correlation, the slope of the regression line increases with the duration of the cycle period.

Fig. 9-4: Indoor versus outdoor temperature for a room under different weather conditions. The sketches on the right side show the cycle periods for each simulation run. Since the heat storage capacity is completely used during long periods of warm weather, the building’s thermal inertia cannot compensate for high ambient air temperatures. Therefore, the building responds more rapidly during long periods of warm weather and reaches higher room temperatures more frequently. (The grey dot shows the mean temperature for each simulation.)
9.5 Energy balance

The analysis of the temperature performance can be combined with the energy balance of the building. Table 9-3 shows the essential values required to evaluate the thermal building performance in summer:

- The heat gains can be taken directly from measurements (i.e. internal heat gains) or have been calculated (i.e. solar heat gains) from the building geometry, material properties and meteorological data.

- The heat losses are mainly caused by ventilation. Consequently, the air change rate determines the heat loss in each building.

- The heat storage capacity is (almost) identical in each building for the daily period. Since the building constructions are very similar, the heat storage capacity for longer cycle periods are (almost) identical, too. However, it was found in the previous Subchapters that all buildings responded to changes of the ambient air temperature faster in 2003 than in 2002. Obviously, the buildings’ thermal inertia could not compensate for the high ambient air temperatures because the available heat storage capacity had been already utilised completely.

Table 9-3: Energy balance (working days) room temperature (only working hours) for the summer period 2002 and 2003.

<table>
<thead>
<tr>
<th></th>
<th>Lamparter</th>
<th>Pollmeier</th>
<th>Fraunhofer ISE</th>
<th>DB Netz AG</th>
</tr>
</thead>
<tbody>
<tr>
<td>heat gains $^1$ [Wh/(m² d)]</td>
<td>252</td>
<td>184</td>
<td>282</td>
<td>145</td>
</tr>
<tr>
<td>mean air change rate (day) $^2$ [h⁻¹]</td>
<td>4 – 7 $^5$</td>
<td>2 – 3</td>
<td>3 – 5</td>
<td>2 – 4 $^6$</td>
</tr>
<tr>
<td>mean air change rate (night) $^3$ [h⁻¹]</td>
<td>6 – 8 (free)</td>
<td>1 – 6 (min. 1)</td>
<td>5 – 8 (min. 4)</td>
<td>2 – 5 (free)</td>
</tr>
<tr>
<td>heat storage capacity $^4$ [Wh/(m² K)]</td>
<td>ca. 25</td>
<td>ca. 25</td>
<td>ca. 25</td>
<td>ca. 25</td>
</tr>
<tr>
<td>ambient air temperature $^5$ [°C]</td>
<td>21,4 21,8 21,8 22,5 22,5 22,5 22,4 21,4 21,4 21,4 20,4 21,8 21,8 21,8 22,5 22,5 22,5</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>room air temperature $^5$ [°C]</td>
<td>23,4 23,2 23,2 24,6 24,6 24,6 24,6 24,6 24,6 24,6 24,6 24,6 24,6 24,6 24,6 24,6</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>temperature difference $^6$ [K]</td>
<td>2,0 1,4 1,4 0,4 2,1 0,4 2,1 0,4 2,1 0,4 2,1 0,4 2,1 0,4 2,1 0,4</td>
<td></td>
<td></td>
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</tr>
</tbody>
</table>

1 mean heat gains (sum of internal and solar heat gains)
2 typical air change rate during working hours
3 typical air change rate during the night (minimum mechanical air change rate)
4 thermally effective heat storage capacity (cycle duration: 24 hours). As the bare concrete ceiling dominates the overall heat storage capacity, the thermally effective thickness is calculated only for the concrete ceiling for better clarity.
5 mean temperature during the summer period (only working hours)
6 inclusive approximately 1.2 h⁻¹ mechanical air change rate from earth-to-air heat exchanger
7 mechanical night ventilation is enhanced by manual opening of windows

Energy balance and indoor / outdoor air temperature

Fig. 9-5 shows the regression lines for the three office buildings for the summer 2002 and 2003. These graphs are based on the same data as the analysis in Table 9-3 and differ from the regression lines in Fig. 9-3, since they take only temperatures from June 1 to August 31 into account. Though the room temperatures are higher in summer 2003, the excess temperature is smaller than in summer 2002. In other words: On the one hand, the hourly room temperatures are not only higher but also exceed the comfort criteria more often in summer 2003 than in summer 2002. On the other hand, the mean room temperatures increase less than expected.
Since the room temperature responds to the heat / loss-coefficient and the time constant of the building, this discrepancy at first view is discussed on the basis of the energy balance model from Chapter 2.2.1. For this correlation, the temperature behaviour is analysed not only for the working hours but for all hours of the day. Table 9-4 shows, that the (relative) temperature difference – at higher (absolute) temperatures – is smaller in the summer 2003 than in 2002. (This investigation has been carried out for every project but is discussed exemplarily for the Fraunhofer ISE building only.)

<table>
<thead>
<tr>
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</tr>
</thead>
<tbody>
<tr>
<td>ambient air temperature °C</td>
<td>18.6</td>
<td>20.9</td>
<td>18.8</td>
<td>20.2</td>
<td>20.0</td>
<td>23.7</td>
</tr>
<tr>
<td>room air temperature °C</td>
<td>22.7</td>
<td>24.0</td>
<td>22.7</td>
<td>23.8</td>
<td>23.7</td>
<td>26.4</td>
</tr>
<tr>
<td>temperature difference K</td>
<td>4.1</td>
<td>3.1</td>
<td>3.9</td>
<td>3.6</td>
<td>3.7</td>
<td>2.7</td>
</tr>
</tbody>
</table>
Fig. 9-5: Room air temperature versus ambient air temperature for the working hours from June 1 to August 31: Regression lines for the room air temperature during summer 2002 and 2003. The dots show the mean room and ambient air temperature in summer 2002 and 2003 from Table 9-3. For better clarity, the line ambient = room air temperature is plotted.

Fraunhofer ISE: energy balance and room air temperature

In 2002, the night ventilation was in operation only from July 15 to August 31 due to an error in the building management system, which yields a temperature difference of 1.2 K, cf. Fig. 5-8. Furthermore, Fig. 9-6 shows clearly that the weather conditions were very different: While in 2002 the daily mean outdoor air temperature sometimes falls below 20 °C, this occurs very seldom in 2003. Taking a typical user behaviour into account, the windows were closed more often in 2002 than in 2003 due to these low ambient air temperatures. This hypothesis can be verified with the monitored window opening behaviour shown in Fig. 9-7. The comparison of open windows in July shows that actually more windows were open in 2003 than in 2002.
Fig. 9-6: Indoor and outdoor air temperature during the summer period (June 1 – August 31) in the Fraunhofer ISE building in 2002 and 2003. The indoor air temperature is the mean value of 16 offices according to the statistical analysis in Chapter 8.

Fig. 9-7: Window opening behaviour in July 2002 and 2003. The mean ambient air temperature for July was 19.9 °C in 2002 and 22.2 °C in 2003. Accordingly to the user model in Chapter 7, more windows were open in July 2003 than in July 2002.

Why is the excess temperature in 2003 smaller than in 2002? With the previous investigations, this question can be answered. The mean indoor temperature $T_{i,m}$ is calculated according to Eq. (2-6) from the mean ambient air temperature $T_{a,m}$ and the gain-to-loss ratio $\gamma = \frac{G_m}{H}$. In 2003, the heat gains $G_m$ were (almost) identical to the heat gains in 2002 but more windows were open and the night ventilation was used during the whole summer period, which suggests that the heat loss factor $H$ was larger in 2003. Thus, the excess temperature $\gamma$ is smaller in 2003 than in 2002 by 1 K based on a daily comparison at the Fraunhofer ISE building, cf. Table 9-4, and by 1.7 K when only working hours are considered, cf. Table 9-3.

Why do the room temperatures exceed the comfort criteria more often 2003? The temperature amplitude $\Delta T_i$ is calculated according to Eq. (2-7) from the temperature amplitude $\Delta T_a$, the quotient of the heat gain amplitude $\Delta G$ and the heat loss factor $H$ and the time constant $\tau = C/H$. The heat gain amplitude $\Delta G$ was (almost) identical in 2002 and 2003. Fig. 9-8 shows that the daily temperature amplitudes are similar in both years, too: $\Delta T_{a,2002} = 3.8K$, $\Delta T_{a,2003} = 3.9K$, $\Delta T_{i,2002} = 1.3K$ and $\Delta T_{i,2003} = 1.2K$. The conversion of Eq. (2-7) to $C$ shows that $C$ is proportional to $H$ and hence increases with $H$ if all other input parameters are held constant. With the monitored data from 2002
and 2003, the conclusion can be drawn that the daily heat storage capacity is (almost) identical in both years.

Thus, the more frequent occurrence of high room air temperatures corresponds with the long-term behaviour of the building according to Fig. 9-4: Due to the continuous thermal exposure in 2003, the long-term heat storage of the building is heated and cannot compensate for temperature changes which continue for several days, cf. Chapter 6.3.

![Graph showing air temperature over the day for 2002 and 2003](image)

**Fig. 9-8:** Mean day for the Fraunhofer ISE. The outdoor (black lines) and indoor temperature (grey lines) are sorted by the time of the day for the summer 2002 (with triangles) and 2003.

### Energy balance in comparison

Fig. 9-9 compares the energy balance for the four office buildings. The daily heat flows \([\text{Wh/(m}^2\text{ d)}]\) differ in a wide range; the mean heat gains are only 145 at the DB Netz AG and 184 Wh/(m² d) at the Pollmeier building and reach 300 in the south-west office of the Lamparter or even 455 Wh/(m² d) office with high heat gains of the Fraunhofer ISE building. Correspondingly, the heat loss differs. As expected, the day ventilation influences the energy balance strongly in the Lamparter and the DB Netz AG building due to the earth-to-air heat exchanger and only slightly in the Fraunhofer ISE and the Pollmeier building.

In each building, the heat stored in the building structure is around the half of the heat gains. In general, the heat storage increases with the heat gains and decreases, if energy is dissipated during the day, i.e. cool supply air from an earth-to-air heat exchanger.

Finally, the night ventilation’s contribution to the passive cooling system can be qualitatively derived from these graphs. For comparison reasons, the night ventilation is defined as the heat loss not during the working hours. Due to different ventilation and control concepts, the operation time is different, e.g. the free night ventilation at the Lamparter and the DB Netz AG building or the mechanical ventilation at the Fraunhofer ISE and the Pollmeier building. However, the heat dissipation due to night ventilation is in the same range or higher than the heat energy which is stored in the building structure.
Energy balance [Wh/(m² d)] for the working days during the summer period in the four low-energy office buildings. The heat loss is drawn on the left and the heat gain on the right side. The heat loss or gain due to heat transmission and the interzonal air change is considered in the “ventilation”. At the Lamparter building, the two offices are analysed separately. At the Pollmeier building, the open plan office and at the DB Netz AG building, the peripheral office 3A009 is analysed. The energy balance at the Fraunhofer ISE is calculated for the typical room and for an office room which is exposed to the heat gains within the standard deviation σ.
9.6 Conclusions

An analysis of monitored data from summer 2001 and 2002 (typical summer weather) and 2003 (summer weather with long and extremely warm periods) reveals that office buildings in central European climate do not need to be air-conditioned, if they are accurately designed and rationally operated. However, none of the buildings utilised the passive cooling potential completely.

Concerning thermal building performance in summer, the conclusions drawn from a very simplified cross-section analysis correspond to the conclusions drawn from a more detailed analysis. The detail analyses are discussed in Chapter 4 – 7.

In a methodical context the cross-section analysis yields two main results:

☐ Since the thermal performance in several buildings has been analysed accurately under different conditions with the parametric model, this model is verified and, hence, can be used for data evaluation.

☐ Since the parametric model is based on a consistently physical description of the thermal building behaviour, it is not only valid for low-energy buildings but also for other building designs or under different boundary conditions.

Thus, the parametric model provides an important contribution to the enhancement of reliability concerning the design of passive cooling systems: The very complex interactions, which are taken into account by the building simulation, can be verified with a few concise parameters.

9.7 References


10 Conclusion

The results from the previous Chapters enhance the understanding of passive cooling with night ventilation in four low-energy buildings. Some of the findings are relevant beyond these specific projects. In this Chapter the hypotheses from Chapter 1.5 are discussed in order to generalise these findings.

Low-energy office buildings with passive cooling aim at a high workplace comfort and a low energy demand, a sustainable building and energy concept and low investment and operation costs. The passive cooling systems of four office buildings are analysed in order to evaluate how far the buildings satisfy the expectations concerning the thermal building performance and the comfort in summer with a low auxiliary energy consumption for the use of the cool night air as a natural heat sink for cooling purposes.

This thesis contributes to the understanding of passive cooling concepts and enhances their design since a comprehensive data analysis from four office buildings in operation and a universal method for data evaluation is available. Starting from this work, the certainty with regard to the design process and the operation of passively cooled building is enhanced.

This achievement can be discussed with the hypotheses from Chapter 1.5.

1. Feasibility

Experimental and numerical studies show that low-energy office buildings achieve a comfortable indoor climate only with passive cooling. Provided that architectural design and building physics are harmonised with the building services, offices can be passively cooled under Mid European climate in most cases. Specially used rooms in the office building, e.g. IT rooms or seminar rooms, may be air-conditioned separately when necessary.

Shortcomings and successes in the design and operation of passive cooling systems has been derived from an extensive data analysis. As this data analysis is based on a universal method, generalised conclusions can be drawn from several office low-energy office buildings: If the design targets are consequently realised, each of the monitored building achieves comfortable indoor climate in summer.

Improved building materials and a high quality standard at the construction site will propagate the realisation of low-energy office buildings and make a variety of architectural designs possible.
2. Calculability

The thermal building performance can be calculated accurately by numerical simulation, if all input parameters are well-known:

- The accuracy of building simulation can be strongly improved by the use of statistical models for input parameters (e.g. user behaviour).
- Uncertainties concerning the input parameters (e.g. building physical properties) can be evaluated by building simulation. A Monte Carlo-simulation and results analysis based on Fourier series are efficient tools for the assessment of uncertainty.
- Users affect the thermal performance of a passively cooled building since the energy balance cannot be influenced actively. Occupancy patterns and user behaviour with regard to ventilation (e.g. window opening) and sun-shading can be derived from long-term monitoring. This information reduce the uncertainties in the design of passive cooling systems.

CFD simulation is suitable for the visualisation and analysis of air-flow distribution in rooms. As the heat transfer influences greatly the thermal building behaviour, the heat transfer coefficients must be calculated accurately. While calculation models are available for typical daily conditions, the heat transfer coefficients during night ventilation are uncertain. The results from a CFD simulation can be analysed in order to develop a sufficient model for the heat transfer coefficient during night ventilation.

The heat storage capacity has a significant impact on the room temperatures during the working time and the night ventilation efficiency and, hence, has to be modelled precisely. The available calculation methods often oversimplify the calculation procedure. The heat storage capacity should be calculated numerically. If the heat storage capacity has to be calculated in analytical building models, the heat penetration must be calculated for the characteristic cycle periods, i.e. 24 hours, duration of a heat period in summer or the time constant of the building.

3. Availability

Several projects have been realised successfully and show, that passive cooling concepts are well-known and can be applied to office buildings in ordinary use. The separate modules in a passive cooling concept (e.g. night ventilation, facade systems, natural ventilation or earth-to-air heat exchangers) are well-established in construction experience.

4. Reproducibility

An essential engineer's and scientist's law says, that the same cause results in the same effect. Though it can be difficult to find this principle in the complex thermal performance of a building, the impact of heat gains, heat losses and the building’s thermal inertia on the thermal comfort have been derived from field measurements.

A simplified parametric model can be used to evaluate data both from simulation studies and from long-term monitoring or short-term measurements. The results may support architects and engineers assuring the quality of passive cooling systems during the design, the commissioning and the operation of a passively cooled building.

The conclusions concerning the passive cooling concepts with night ventilation are generalisable since the universally valid models have been applied to a wide range of buildings and boundary conditions.
5. Functionality

Operation experiences from several years show that passive cooling systems can be operated robustly. However, the building management system had to be adjusted and optimised in each building after the first summer in order to use the night ventilation potential efficiently. The operation of the passive cooling system and especially the night ventilation can be improved in every building investigated in this thesis.

Building simulation and the parametric model have been successfully applied to an optimisation procedures and to the determination of the main influencing variables. The night ventilation efficiency can often be enhanced with very simple changes only in the control strategy.

Besides the building control, the user behaviour has a strong influence on the efficiency of passive cooling concepts. The user behaviour has been investigated in detail and user models for specific summer conditions have been developed for low-energy buildings. These models are not universally valid since the available data set is not large enough, but the models are generalisable for low-energy office buildings with and without auxiliary mechanical ventilation.

Provided that the building and its energy concept has been designed robustly, the building forgives its building management system or its occupants for an error in the control system and the user behaviour, respectively. A robust night ventilation design represents low internal heat gains, a self-shading facade or a semi-automatic solar control system, a sufficiently high thermal inertia, a largely designed heat dissipation (i.e. sufficiently large openings and conduits) and a for the users comprehensible building operation.
Software used in context with this thesis

Data Analysis:
- GNU awk version 3.1.0 and GNU sed version 3.02 (data handling)
- Microsoft EXCEL 2000 (data analysis)
- GNU PLOT version 3.8i (regression analysis and data visualisation)
- cornerstone 3.7.1 (statistical analysis)
- python 2.1.1 numeric (fourier analysis)

Building Simulation:
- ESP-r System Version 9 Series 4.34a (January 2002)
  (optical properties: WINDOW 4.1)

CFD Simulation:
- GAMBIT v2.1 (pre-processing)
- FLUENT v6.0 (simulation and post-processing)

Tracer gas equipment:
- AUTOTRAC 101, tracer gas: SF₆

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Data Acquisition at DB Netz AG, Hamm.
- data recording: Johnson Controls JCI Regelungstechnik GmbH (building management system), Agilent 34970A and 34901A (data logger) and Fraunhofer ISE (data logger).
- data processing: mux (Fraunhofer ISE).
- data storage: data base mySQL v3.2.2.

Data Acquisition at Fraunhofer ISE, Freiburg.
- data recording: ICP-CON devices with (1) Agilent serie 349 (HP multiplex) and (2) Fraunhofer ISE (data logger). Building management system FRIMAT (daily data transfer to database with conversion of time-variant data series into 10 minutes intervals)
- data processing: remus 1.2.5 (Fraunhofer ISE).
- data transmission: computer network via TCP/IP serial coupler
- data storage: ASCII database and netCNF (python).

Data Acquisition at Pollmeier Massivholz GmbH, Creuzburg.
- data recording: building management system Sauter EY 3600.
- data processing: Sauter NovaPro16 dBase.
- data transmission: Symantec pcAnywhere 10.
- data storage: data base Oracle 9iR2.

Data Acquisition at Hans Lamparter GBR, Weilheim a.d. Teck.
- data recording: Ahlborn Mess- und Regelungstechnik.
- data processing and data storage: AMR Data-Control 4.1.3/v5.
Publications written in context with this thesis


J. Pfafferott, BINE-Themen-Info I/03 “Passive Kühlung mit Nachtlüftung”.


Curriculum Vitae

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09/95 - 10/95 Working Student at Energiteknisk Analys, Gothenburg, Sweden.
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01/95 - 12/95 Research Assistant in the Reserach Project „Development of
12/96 - 02/97 Diploma Thesis “ Development of a Design Tool for Earth-to-Air
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March 20, 1997 Diplom-Ingenieur Technical University of Berlin
04/01 – 03/04 Enrollment in the Ph.D. Programme of the Department of
Architecture at the Technical Univeristy of Karlsruhe.

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