



**EXERGY
EFFICIENT
BUILDING
DESIGN**

POPPONG SAKULPIPATSIN

EXERGY EFFICIENT BUILDING DESIGN

Proefschrift

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door

Poppong SAKULPIPATSIN

MSc in Computer Science, Chulalongkorn University
MSc in Energy Conversion and Management, University of Applied Sciences Offenburg
geboren te Suphan Buri, Thailand

Dit proefschrift is goedgekeurd door de promotor:

Prof.ir. P.G. Luscuere

Samenstelling promotiecommissie:

Rector Magnificus	Voorzitter
Prof.ir. P.G. Luscuere	Technische Universiteit Delft, promotor
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Prof.dr.ir. A.H.C. van Paassen	Technische Universiteit Delft, reservelid

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1. Introduction

The research work is original in the sense of applying the exergy concept to building and building services design. The applicability of existing exergy-related definitions is systematically investigated in built-environment conditions (e.g. smaller temperature differences between a system and environment), incorporated to existing exergy calculation models.

This chapter begins with the context of the dissertation to describe a relation of the research to previous research. The problem definitions, the research objectives and the research questions, are subsequently presented. After that, the research approach and methodology are described, starting with a brief overview of the exergy concept and relevance of the exergy concept to building and building services design. At the end of this chapter, the dissertation outline is given.

1.1. Context of this dissertation

This dissertation is a compilation of five peer-reviewed papers presenting the results of the first doctoral research done in the Netherlands on exergy analysis applied to buildings and building services. This topic is relatively new worldwide. In addition to publication in peer-reviewed journals, the results of this work have been presented in numerous international conferences, as listed in the publication list.

Prior to the research, there has been pioneering work done by Prof. Shukuya (1994, 1996), an architectural engineer by background, who has been studying different aspects including fenestration, building services and more recently the human body.

The exergy concept has been applied to the built environment (Shukuya, 1994; Boelman, 2002; Asada and Boelman, 2004; Sakulpipatsin et. al., 2005, 2006; Schmidt and Shukuya, 2003). Some researchers (Wall, 1986, 1990; Rosen and Dincer, 2001) have also used the exergy concept in a context of sustainable development. In the last few years, a working group of the International Energy Agency has been formed within the Energy Conservation in Buildings and Community Systems programme: “Low Exergy Systems for Heating and Cooling of Buildings” (Ala-Juusela M. (ed.), 2004; Annex37, 2002). The overall objective of the IEA Annex 37¹ was to promote the rational use of energy by means of low-valued and environmentally sustainable energy sources. This annex is being followed up by the international LowExNet group, which works towards providing knowledge on and tools for exergy analyses to be applied in the built environment (LowExNet, 2004). In addition, some of the researchers have been active in current international research projects: IEA Annex 49 and COST Action 24.

¹ The International Energy Agency has supported an annex on low-exergy systems for heating and cooling of buildings.

1.2. Problem definitions

This item discusses necessities of exergy application for building and building services design, and gives some examples of exergy efficiencies of some HVAC² systems. Some impediments of using the exergy concept for building and building services design are given at the end of the item.

Buildings account for ca. 40% of final energy use in the European Union (EuroACE³, 2005), and heating and cooling amount more than 50% of the yearly energy demand of buildings in the operational phase (EC⁴, 2001). The need for energy efficiency improvement in the building sector has been addressed in the European Directive on the Energy Performance of Buildings. Buildings rely primarily on high-exergy fossil fuels for HVAC functions. Their exergy efficiency is usually less than 10% (Kilkis, 2006; Rosen and Dincer, 1997). Fossil fuels are in general employed to produce low-temperature heat. Since the fossil fuels burn at very high flame temperatures up to 2000K (Dincer and Cengel, 2001), the available work obtained by the fossil fuels is largely wasted when the fossil fuels are utilised for hot water heating, space heating, or even industrial steam production. Indoor space heating boilers have an estimated exergy efficiency of 6% and heat pumps when combined with conventional HVAC systems is not much better: 9% (Kilkis, 2006). It is unfortunate that this problem, known for a relatively long time, has not yet been addressed: the building sector with a dominant share in the annual energy use has a very low exergy efficiency of energy utilisation and thereby is polluting the atmosphere in an unnecessary way. An effective way to address this is to make use of low-exergy waste and alternative energy resources directly in temperatures compatible with new HVAC systems yet to be developed. The building sector in general has a high potential for improving the quality match between energy supply and energy demand, partly because high exergy sources are used for meeting low temperature and thereby low exergy needs.

HVAC systems can be exergy efficient if their operation temperatures are directly compatible with temperatures of low-exergy energy resources and temperatures of indoor air. At present, a radiant panel system is an alternative, which can operate at very moderate supply temperatures. But the system is limited in its ability to handle latent loads (TIAX, 2002). This limitation requires additional convective HVAC for humidity control. The hybrid HVAC System is an optimum solution as it uses different radiant and convective equipment in the same indoor space. Although the hybrid HVAC system seems to be an option for the better utilisation of low-exergy renewable and waste energy resources, they cannot eliminate equipment over-sizing and temperature conditioning. For example, the use of 45°C waste water in such a hybrid HVAC system requires 60% equipment over-sizing and a boiler to peak the resource temperature from 45°C to 55°C (Kilkis, 2006). For cooling applications, there have been efforts in Japan to develop radiant panel systems that allow surface condensation (Hirayama, 2004; Hirotsu et. al., 2005).

² HVAC stands for Heating Ventilation Air Conditioning.

³ EuroACE is the European Alliance of Companies for Energy Efficiency in Building.

⁴ EC stands for European Commission.

Nowadays, energy systems in buildings are designed based solely on the energy conservation principle. Nevertheless, this principle alone does not provide a full understanding of important aspects of energy use in buildings (Schmidt, 2004; Boelman and Asada, 2002, 2003; Sakulpipatsin et. al., 2006, 2007a; Itard, 2005). From this viewpoint, exergy analysis (Kotas, 1985; Szargut et. al., 1988; Ahern, 1980) can quantify the potential for improving this match, and the contribution of this match to better energy resource utilisation.

1.2.1. Why an exergy approach to building design?

Many researchers and practicing engineers refer to exergy methods as powerful tools for analysing, assessing, designing, improving and optimizing systems and processes. Benefits of exergy analysis are numerous, especially compared to energy analysis. For example, exergy methods can assist for evaluation of the thermodynamic values of the energy products. Exergy losses clearly pinpoint the locations, causes and sources of deviations from ideal circumstances in a system. Exergy efficiencies are measures of the approach to ideal. Nevertheless, exergy analysis is used only by a small group of those people. Rosen (2002) collected some reasons why it is not widely accepted by industry at present. Exergy methods might seem cumbersome or complex (e.g. choosing a suitable reference environment) to some people, and the results might seem difficult to interpret and understand.

Moreover, the analysis (Alefeld, 1988; Moran, 1989; Wall, 1990; Krakow, 1991; Bejan 1997) uses many concepts and definitions (e.g. efficiency, reference conditions) that originated in the electric power and chemical industries. Systematic analysis is required to establish the applicability of these concepts to the built environment. Also, exergy is often perceived as a highly complex concept. Furthermore, some practicing engineers have simply disbelieved exergy methods to lead to tangible, direct results.

Consequently, concrete examples of exergy analyses and calculation frameworks specifically developed for the built environment are needed to make the concept more familiar and usable to the building profession.

1.3. Research objectives and research questions

This research aims at developing knowledge into the applicable domains and potential added values of exergy analysis in the built environment, by studying under what conditions exergy could function as a useful concept for the built environment.

Research question Q1, as the main research question, is concerned with the potential added values of developing exergy analysis for buildings and building services, in particular HVAC systems. Research questions Q2 to Q4, as the specific research questions, are in line with the research approach and methodology (described in item 1.4), and address the development of specific knowledge and insight into potentially applicable domains. These research questions are answered in chapter 7, based on the results of the work described in chapters 2 to 6.

- Q1. Under what conditions could exergy function as a useful concept for the built environment?
- a. What is the potential relevance of the exergy concept for integrating building and HVAC system design?
 - b. What are possible advantages and disadvantages of incorporating exergy analysis into energy building system designs and indoor climate conditions?
 - c. What can building designers learn from an exergy analysis that they could not learn from an energy analysis?
- Q2. Which metrics can be used to quantify and express exergy values in buildings and HVAC systems?
- Q3. To what extent do existing exergy knowledge and definitions require adaptation in order to be meaningfully applied in buildings and HVAC systems?
- Q4. Which are the relevant parameters, precision and aggregation levels required by a calculation framework comprising energy and exergy analyses for integrated building and HVAC system design?

1.3.1. Energy, exergy, and built environment

The growing concern of environmental problems has amplified both the significance of all kinds of energy saving measures, and the inevitability for an increased efficiency in all forms of energy utilisation. Despite plenty of efforts made to improve energy efficiency in buildings, the issue of gaining an overall assessment and comparing different energy sources still exists. At present, analysis and optimisation methods do not differentiate between different qualities of energy flows in building-related applications (Schmidt, 2004).

The exergy analysis method is well known for optimisation of energy conversion in large industrial and power plants (Zhang et. al., 2006; Zvolinschi et. al., 2006). Exergy analysis can help building designers meet functionality and comfort requirements while keeping the associated energy resource depletion to a minimum (Alpuche et. al., 2005; Prek, 2006). Exergy provides a common basis for comparing the energy performance of systems associated to buildings and to building services (Schmidt, 2004; Annex 37, 2002; Action C24, 2006). For example, exergy analysis allows a designer to compare on the same basis between heat supplied by a fuel (e.g. through a boiler) and by solar heat (e.g. through a window). It also allows comparison between e.g. the electricity required by a mechanical ventilation system and the thermal energy savings resulting from the use of a heat recovery unit (Sakulpipatsin et. al., 2007a). This information can assist designers in integrating building and building services design, so as to meet user requirements with a minimum depletion of energy resources.

In the theory of thermodynamics, the concept of exergy is stated as the maximum work that can be obtained from an energy flow or produced by a system. The exergy content expresses the quality of an energy source or flow. This concept can be used to combine and compare all flows of energy according to their quantity and quality. Unlike energy, exergy is always destroyed because of the irreversible nature of

energy conversion process. The exergy concept enables us to articulate what is consumed by all working systems (e.g. man-made systems like thermo-chemical engines and heat pumps, or biological systems including the human body) when energy and/or materials are transformed for human use.

Exergy analysis can give insight into the extent to which the quality levels of energy supply (e.g. high-temperature combustion) and energy demand (e.g. low-temperature heat) are matched. High-valued energy such as electricity and mechanical work consists of pure exergy. Energy which has a very limited convertibility potential, such as heat close to room air temperature, is low-valued energy. Low exergy heating and cooling systems allow the use of low-valued energy, which can be delivered by sustainable energy sources (e.g. Kilgis, 2006; Xiaowu and Ben, 2005; Torres et. al., 1998). Most of the energy needed for heating and cooling is used to maintain room air temperatures around 20°C. In this sense, because of the low temperature level, the exergy demand for applications in room conditioning is naturally low. In most cases, however, this demand is met with high quality sources, such as fossil fuels or using electricity. Exergy analysis provides us with additional information on where and when the losses occur. It helps us to see in which part of the energy chain the biggest savings can be achieved (Schmidt, 2004).

This also explains partly the resistance which is felt by engineers and consultants to use exergy as a tool. It clearly shows the sometimes extreme low exergy efficiencies of common systems like burning gas to heat at near environmental temperatures. In these cases exergy analysis is however at its strongest. It leads to the inevitable conclusion that certain processes or systems, however widely accepted and applied, are fundamentally wrong and should be replaced by more exergy efficient ways. This however contradicts the interests of huge industries and gas companies.

1.4. Research approach and methodology

This item gives an overview of the exergy concept and presents some definitions of exergy from literature, followed by an approach adopted in this research. The approach rests on three main pillars. These pillars are integrated into two levels of HVAC and building systems. Details of the research approach are given in the second part of this item.

1.4.1. Thermodynamics, exergy and buildings

The basis of thermodynamics is stated in the first and second laws. The first law is concerned with the conservation of energy, whereas the second law is concerned with the dissipation of energy (Bruges, 1959). The first law of thermodynamics states that energy is conserved, and makes no distinction between different energy forms (e.g. heat and work). The second law, on the other hand, allows energy quality levels to be quantitatively valued (Kyle, 1999) and rank-ordered. It also asserts that accessible work potential is always lost in any real process, and provides a measure of the loss in all real energy transformation processes (Connely and Koshland, 2001).

Exergy is not subject to a conservation law, but can be lost when or where the quality of energy is degraded, due to irreversibility in any process. Exergy analysis is a method that applies the conservation of mass and conservation of energy principles together with the second law of thermodynamics for the design and analysis of energy systems. The exergy analysis is used to estimate the theoretically ideal operating conditions of a system, and the extent to which a real system deviates from the corresponding ideal performance (Bejan, 1997). The exergy method can be suitable for furthering the goal of more efficient energy resource use, for it enables the locations, type and true magnitudes of wastes and losses to be determined (Connely and Koshland, 2001).

Generally speaking, exergy is essentially related to work potential and quality changes of energy and matter in relation to a pre-defined environment. Nevertheless, many various authors choose to emphasize specific aspects in their definitions, depending on the objective and scope of their analysis.

“Exergy is the maximum theoretical work that can be extracted from a combined system consisting of the system under study and the environment as the system passes from a given state to equilibrium with the environment - that is, passes to the dead state at which the combined system possesses energy but no exergy.” (Moran, 1989)

“Exergy is the minimum theoretical useful work required to form a quantity of matter from substance present in the environment and to bring the matter to a specified state. Exergy is a measure of the departure of the state of the system from that of the environment, and is therefore an attribute of the system and environment together.” (Bejan, 1997)

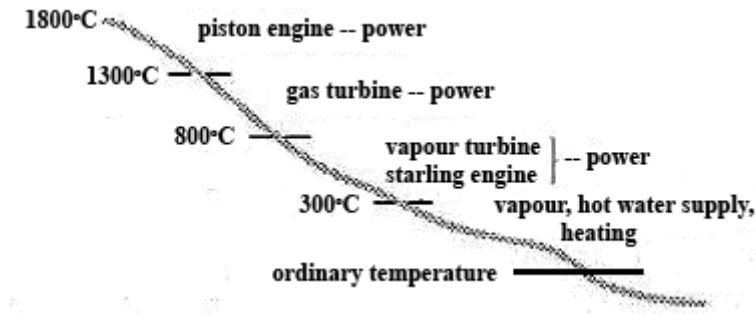
“The property exergy defines the maximum amount of work that may theoretically be performed by bringing a resource into equilibrium with its surroundings through a reversible process.” (Connely and Koshland, 2001)

“Exergy is the concept, which quantifies the potential of energy and matter to disperse in the course of their diffusion into their environment, to articulate what is consumed within a system.” (Ala-Juusela M. (ed.), 2004)

The classical exergy concept enables us to pinpoint the location, to understand the cause, and to establish the true magnitude of waste and loss upon energy conversion. Exergy analysis approach is therefore a vital tool for system designs since it provides designers with answers to two important questions of where and why system losses occur. The designers can then proceed forward and work on how to improve the system.

Exergy often appears as heat and cold; thermal exergy can be in general described by temperature differences from the environment in some outdoor climate conditions (Sakulpipatsin et. al., 2007b). Exergy reflects better than energy that heat or cold becomes more valuable at temperature levels further from the environment. Figure 1.1 shows that high-temperature heat can be converted into electric power, and also illustrates how close hot water supply and space heating temperatures are to environmental temperature.

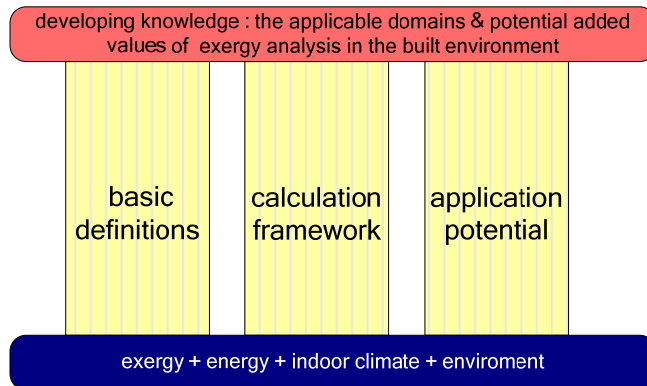
Figure 1.1 The usefulness of heat at different temperatures (Hirata, 1997)



1.4.2. Approach adopted in this research

The research rests on three main pillars, as shown in Figure 1.2.

Figure 1.2 The research scheme



1. Basic definitions: many exergy-related definitions (e.g. exergy efficiency and reference environment) have been developed for use in the electric power and chemical industries. Their applicability to built-environment conditions (e.g. smaller temperature differences between a system and environment) is investigated in pillar 1.
2. Calculation framework: existing exergy calculation models tend to allow detailed investigation of parameters related to either the building or the HVAC systems, but not to both. An energy and exergy calculation framework is developed in pillar 2 for use with a number of integrations between building design concepts and HVAC systems (e.g. a heat recovery unit in balanced ventilation systems, low-temperature heating and high-temperature cooling systems in a number of building design concepts).
3. Application potential: in a small but important pillar, the calculation models are applied to integrated systems in buildings (e.g. heat recovery of dwelling ventilation systems, district heating systems and cooling machines) and HVAC components (e.g. heat exchangers and heat pumps). This provides concrete

examples of insights that can be gained from exergy analysis, and shows how these insights differ from what can be learned from energy analysis.

These three pillars are integrated into two levels: namely “HVAC components and systems” and “building systems”. A brief overview of the main tasks is given below; in relation to the research questions (item 1.3).

HVAC components and systems:

This part entails the set-up of a conceptual analysis framework and application of the exergy concept to HVAC component and system design. It collaborates with some outputs from the IEA Annex 37. This part focuses on research questions Q1 to Q3.

Critical analysis of basic exergy definitions and their applicability to HVAC systems in built environment conditions is systematically carried out at component level (e.g. heat exchanger and heat pump) and at system level (e.g. mechanical exhaust ventilation with natural air supply and balanced ventilation with heat recovery). Results of the analysis are discussed, by using defined metrics, to potential relevance and application possibility of the definitions to the built environments.

Building systems:

This part integrates the conceptual analysis framework, developed in the previous part, with a conceptual analysis framework of exergy in buildings. This part targets research questions Q1 to Q4.

An analysis framework to study the influence of possible definitions of a reference environment is introduced to determine the exergy of air in buildings. Then calculation models of exergy uses in buildings and building services are developed and make use of an extended built-up model in which the energy balance is considered from the demand side to the supply side, developed by Sakulpipatsin et. al. (2006) and Bezuijen (2006). The calculation models are applied for sensitivity analysis of thermal exergy demands in a building to changes of building envelope properties, and sensitivity analysis of exergy losses in building services to changes of system operations like temperature levels, in the climate of the Netherlands. At the end, some analysis results of energy and exergy of the building and building services are given in order to summarise the uses of the exergy concept in buildings and the built environment.

1.5. Dissertation outline

For the purposes of this dissertation, the exergy concept can be understood as a potential of matter to cause change, as a result of not being entirely stable relative to a reference environment. Its operational definition for this thesis is defined in chapter 2, item 2.

The quantity of exergy depends on the state of the system and on the condition of the environment. The state of the reference environment must be given for exergy analysis. This is regularly done by specifying the temperature, pressure and chemical composition of the reference environment. Past research (Wepfer et. al.,

1979; Liley, 2002) lacks a clear and accessible framework for quantifying exergy of humid air allowing for changes in environmental air temperature. In chapter 2, an analysis framework to study the influence of possible definitions of a reference environment is introduced to determine the exergy of air in buildings. Chapter 2 analyses the influence of possible definitions of the standard state of air, to determine the exergy of air in buildings, taking into account thermal, chemical and mechanical contributions. It discusses the importance of these contributions and the possibilities to determine the conditions at which it is allowable to assume that air contains no water vapour. In addition, the exergy calculations of dry air are compared with exergy values based on the assumption of using annual statistical values of the indoor and outdoor air temperatures. This chapter is related to research questions 3 to 4. This analysis framework has been accepted to be published in the international journal of exergy.

Exergy efficiencies are often defined considering the intended application of a given system under specific conditions, and therefore the definitions frequently lack uniformity. Several authors have provided definitions for exergy efficiencies (Semenyuk, 1990; Sorin and Brodyansky, 1992; Tsatsaronis, 1993; Kotas, 2001) on the large scale of energy supply systems. Woudstra (2002) distinguishes two different kinds of exergy definitions: the universal ones in which gross exergy inputs and outputs are considered, and the functional ones in which net exergy flows are considered respectively. To the best of current knowledge, there is a deficiency of systematic approach to be able to apply exergy efficiency definitions for buildings and building services, and there is a very limited knowledge on the efficiency behaviour for buildings and building services at near environmental conditions. Chapter 3 and chapter 4 critically analyse the exergy efficiency definitions for all-air HVAC system components operating at near environmental and indoor conditions. Chapter 3 deals with investigation of which relevant information the functional exergy efficiency definition provides for selection and operation of sensible heat exchangers for indoor climate control in space heating applications. It focuses on the exergy analysis of a simplified sensible heat exchange process for heating applications, by varying temperatures and heat transfer rates, considered simply in terms of exchanger heat transfer effectiveness. Chapter 4 critically analyses the universal and functional exergy efficiency definitions for a simple vapour-compression heat pump cycle for space cooling applications, by varying temperatures and internal irreversibility, considered simply in terms of the second-law efficiency. A dimensionless temperature is used to illustrate the analysis results, and to discuss the sensitivity of the exergy efficiency definitions to temperature variations for the HVAC system components. These chapters are related to research questions 1 to 4. The analysis results of the exergy efficiency definitions for the air-to-air heat exchangers have been accepted to be published in the international journal of exergy.

In cold and moderate climates, improvements in building shell insulation and air-tightness imply a shift in heating loads from transmission and infiltration towards ventilation. Heat recovery from ventilation airflow plays an increasingly important role in minimising energy needs. Such heat recovery systems rely on the input of electric power (to drive fans, heat pumps, etc.) in order to recover thermal energy. Since electricity input is relatively small compared to the amounts of thermal energy recovered, such systems are efficient from an energy viewpoint. One important yet

often overlooked aspect, however, is the difference in ‘quality’ between the high-grade electricity input and the lower grade thermal energy recovered. Chapter 5 presents steady-state energy and exergy analyses for dwelling ventilation with and without air-to-air heat recovery, and discusses the relative influence of heat and electricity on the exergy demand by ventilation airflows. Energy and exergy analysis results for De Bilt, the Netherlands, are presented in terms of heat and electricity, on an instantaneous and a daily basis. Chapter 5 is related to research question 1. The steady-state energy and exergy analyses for dwelling ventilation have been published in the international journal of ventilation.

Chapter 6 introduces an integrated and dynamic method for energy and exergy analysis of buildings and building services, since at present there is no ready-to-use dynamic model for exergy calculation over the entire energy demand and supply chain in the built environment. The method is intended to enable building designers (and building engineers) to compare between the impact of improvements in the building envelope and in building services. The method is demonstrated with a building in a cold climate and used for investigation of thermal exergy and thermal energy demands of the building and thermal energy and thermal exergy losses in the building services when some parameter values of the building and the building services are changed. This study is an initial attempt of the sensitivity analysis of the exergy values in a building and building services. This chapter is related to research question 1.

Chapter 7 finally recapitulates the findings from the previous chapters and concludes with recommendations for further research.

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The influence of possible definitions of a reference environment to determine the exergy of air in buildings

Abstract: This paper critically analyses the influence of possible definitions of the standard state of air to determine the exergy of air in buildings. Three different contributions are considered related to differences in temperature, pressure, and humidity of air inside and outside the building envelope. The possibility to calculate the exergy of air in buildings, based on only one or two of these contributions, for example expressed by a characteristic air temperature and/or air as dry air, is explored for three different locations on earth. These values are compared to those calculated using hourly statistical climate data during one year.

Keywords: exergy; reference environment; indoor air.

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Brief biographical notes: P. Sakulpipatsin is a Doctoral Researcher in Climate Design Group, at the Faculty of Architecture, the Delft University of Technology, The Netherlands. He obtained two Master Degrees: in 2003 MSc in Energy Conversion and Management from the University of Applied Sciences Offenburg, Germany; and in 2001 MSc in Computer Science from Chulalongkorn University, Thailand.

H.J. van der Kooi is an Assistant Professor in the Field of Applied Thermodynamics and Chemical Engineering at the Delft University of Technology, The Netherlands.

L.C.M. Itard is a Researcher in the field of sustainable buildings and HVAC equipment and leads the group sustainable and healthy building of the Research Institute OTB at the Delft University of Technology, The Netherlands.

E.C. Boelman has a background in Adsorption Cooling, Building Engineering and Management, and has been working on renewable energy and exergy analysis applied to buildings for several years.

1. Introduction

Buildings in the European Community require approximately 40% of final energy use. Energy for heating and cooling purposes accounts for more than 50% of the annual building energy demand in the operational phase. A building's heating demands can be met by low-grade heat sources, since the required temperatures are mostly between 18°C and 26°C. However, high-temperature processes (e.g., fossil fuel combustion) are often used to deliver the low-grade heat required by end-users in buildings. The temperature of heat delivery to indoor spaces (e.g., by radiating panels) is often also higher than what would be required in terms of human thermal comfort. Exergy analysis has the potential to optimise the building energy demand, since exergy consumption can, to a certain extent, be minimised. As an initial step of exergy application for building design, a reference environment of buildings needs to be properly defined. Since properties of (indoor) air play a vital role in indicating indoor thermal comfort, health and energy use, air seems to be the most important and appropriate medium for investigating some of the possible definitions of the reference environment to determine the exergy value of air in buildings. The most reasonable reference environment for calculating the exergy of air in buildings is the actual environmental conditions of the air outside the buildings. However, building designers may find using the actual outdoor environmental condition too complex for exergy calculations. In the practice of building design, the exergy of air in a building should be estimated in an easy, less time-consuming way, and as precisely as possible.

Several authors (e.g., Wepfer et al., 1979; Liley, 2002; Qureshi and Zubair, 2003; Alpuche et al., 2005; Mina et al., 2005; Alhazmy, 2006) have used an exergy approach to evaluate thermodynamic processes in HVAC¹ systems, but most of them used fixed or time-independent values of outdoor climate conditions (e.g., temperature at 273.15 K and pressure at 1.01325 bar) as a reference environment. The outdoor climate conditions vary in reality continuously, all over the year. The use of pre-defined standard conditions instead of this dynamic reference environment could lead to inaccurate results. Besides, the exergy calculation methods rely on several properties of the reference environment, like temperature, pressure and chemical composition. Most research into exergy and buildings only takes account of the thermal exergy of air (Schmidt, 2004; Asada and Boelman, 2004; Shukuya and Hammache, 2002; Sakulpipatsin et al., 2006; Itard, 2005). Chemical exergy caused by differences in water vapour content between indoor and outdoor air and mechanical exergy caused by pressure differences between indoor and outdoor air are ignored. From the work of Wepfer et al. (1979) and Szargut (1988) it appears however that these assumptions may lead to less inaccurate results.

The objective of this paper is to determine the exergy of air in buildings, considering also the mechanical and chemical contributions, and to illustrate differences in exergy values of air in buildings in three different places on earth located in different climate zones. The paper critically analyses the influence of possible definitions of the reference state to determine the exergy of air in buildings, considering three different parameters (air temperature T , humidity ratio of humid air W and air pressure P). This paper takes the

¹ HVAC stands for Heating Ventilation and Air Conditioning.

real environment of the buildings as the reference state to calculate the thermal, chemical and mechanical contributions to the exergy value of air. The paper first considers all three exergy contributions, and then goes on to discuss the influence of not or only partly taking into account of the humidity of air in exergy calculations in order to investigate the possibility of considering the indoor air and the outdoor air as dry air for exergy calculations. The paper also examines a large pressure range to determine the mechanical contribution to the exergy value of the indoor air. The pressure difference between the indoor air and the outdoor air is considered to vary between ± 100 Pa. The exergy calculations use specific indoor climate conditions of a general thermal comfort zone, and use outdoor climate conditions of three cities in different climate zones. The exergy calculations are made on an hourly basis, using hourly indoor and outdoor climate data. The exergy results are presented as average values and used for the discussions. Furthermore, for the exergy calculations of the dry air, this paper compares exergy values of air based on annual statistical values of the indoor air temperatures and of the outdoor air temperatures, with the exergy values of air based on the actual (changing) outdoor air temperature. This results in finding the conditions under which the much simpler possibility exists to use a static reference environment instead of a dynamic reference environment.

2. Exergy of a substance: physical and chemical exergy

The magnitude of the exergy value of a substance can be regarded as the sum of the physical and the chemical contributions. The physical exergy refers to the departure of the physical state of the system, at a certain pressure and temperature, compared with that of the reference environment. The chemical exergy refers to the departure of the chemical composition of a system, at the reference temperature and pressure, from that of the reference environment.

2.1. Physical exergy

Physical exergy (Ex_{ph}) is equal to the maximum amount of mechanical work obtainable when a substance is brought from its initial state temperature T and pressure P , to the state of the reference environment defined by T_o and P_o . For a process that brings the substance from the initial state to the state of the reference environment, the change in physical exergy (dEx_{ph}) of the substance in the process can be calculated by using equation (1). The process is considered under steady state conditions, via reversible processes which only exchange heat with the reference environment.

$$dEx_{ph} = dH - T_o dS + dKE + dPE \quad (1)$$

In equation (1) H is the enthalpy², S is the entropy³, KE is the kinetic energy and PE is the potential energy for that process. KE and PE are zero when the substance is in the state of the reference environment. Subscript o indicates that the properties are in the state of the reference environment. Exergy calculations are generally often performed

² Enthalpy (H) is the sum of the internal energy of the system plus the energy associated with work done by and on that system which is the product of the pressure and volume.

³ Entropy (dS) is defined as ' $\delta Q_{rev}/T$ ', where δQ_{rev} is the amount of heat absorbed reversibly by the system at temperature T .

under conditions where the kinetic and potential terms can be ignored (Ahern, 1980). It is assumed that potential and kinetic energy contributions to the exergy value of air in the buildings can be ignored. Equation (1) then reduces to equation (2) (Moran and Shapiro, 1998).

$$dEx_{ph} = dH - T_o dS \quad (2)$$

Since the enthalpy (H) and the entropy (S) are dependant on the air temperature (T) and the air pressure (P), equation (2) is expressed in the partial differential term of the air properties (T and P), as equation (3).

$$dEx_{ph} = \left(\left(\frac{\partial H}{\partial T} \right)_P dT + \left(\frac{\partial H}{\partial P} \right)_T dP \right) - T_o \left(\left(\frac{\partial S}{\partial T} \right)_P dT + \left(\frac{\partial S}{\partial P} \right)_T dP \right) \quad (3)$$

By assuming that air in buildings is an ideal gas: $\left(\frac{\partial H}{\partial T} \right)_P = c_p$; $\left(\frac{\partial H}{\partial P} \right)_T = 0$;

$\left(\frac{\partial S}{\partial T} \right)_P = \frac{c_p}{T}$; and $\left(\frac{\partial S}{\partial P} \right)_T = -\frac{R}{P}$. c_p and R are the molar isobaric heat capacity and

the molar gas constant respectively. The molar physical exergy value dEx'_{ph} follows from equation (4):

$$\begin{aligned} dEx'_{ph} &= c_p dT - T_o \left(\frac{c_p}{T} dT + \left(-\frac{R}{P} dP \right) \right) \\ &= c_p dT - \frac{c_p T_o}{T} dT + \frac{RT_o}{P} dP \end{aligned} \quad (4)$$

Therefore, the molar physical exergy value with the reference environment T_o and P_o is

$$Ex'_{ph} = c_p \left((T - T_o) - T_o \ln \left(\frac{T}{T_o} \right) \right) + RT_o \ln \left(\frac{P}{P_o} \right) \quad (5)$$

In equation (5), the first term, in the large brackets and multiplied by c_p , can be referred to as a thermal exergy value (Ex_{th}) and the second term can be called a change in exergy related to pressure differences, a mechanical exergy value (Ex_{me}).

2.2. Chemical exergy

Chemical exergy (Ex_{ch}) is equal to the maximum amount of mechanical work obtainable when a substance under consideration is brought from the state of the reference environment to the dead state by processes involving heat transfer and exchange of substances only with the dead state (Kotas, 1985). The final state will be what is called 'dead state', which means that all substances are in thermal, mechanical and chemical equilibrium in this state.

The exergy of the outdoor air, which is the reference environment at P_o and T_o , is strictly speaking not zero; work could be obtained if the substance were to come to thermal, mechanical and chemical equilibrium. The reference environment is not in the equilibrium (or in the dead state), because it is possible that there are still processes of diffusion taking place among the substance's chemical components in the reference environment. In addition, this reference environment is not infinitely large. Many chemical reactions in the reference environment are however blocked because the activation energy is so great that the chemical reactions to more stable substances cannot occur at outdoor conditions (Rosen and Dincer, 1997). In this work, chemical exergy is defined as the difference between the exergy content of air in buildings and the exergy content of the outside air (as the reference environment), which is not in the dead state. The air components considered in this study are dry air and water vapour. The other components in the air (CO_2 , N_2 etc.) are assumed identical in indoor and outdoor conditions. Their contribution to the exergy can therefore be neglected.

The contribution to the chemical exergy value of air due to mixing of dry air and pure water vapour in the case of an ideal mixture (air at P_o and T_o) is given by equation (6) (Interduct, 2002; Wepfer et al., 1979; Szargut, 1988).

$$\Delta Ex'_{\text{mix}} = RT_o \sum_{i=1}^n x_i \ln(x_i) \quad (6)$$

where x_i is the mole fraction of the i -th substance and R is the molar gas constant ($8.314 \text{ Jmol}^{-1}\text{K}^{-1}$).

3. Exergy of air in buildings

The molar exergy value of humid air in buildings can be calculated as the total of its physical exergy ($Ex'_{ph,\text{humidair}}$) and its chemical exergy ($Ex'_{ch,\text{humidair}}$), using equation (7) and equation (8) (Wepfer et al., 1979; Smith et al., 1996; Interduct, 2002). Equation (7) is derived from equation (5) and equation (8) from equation (6).

$$Ex'_{ph,\text{humidair}} = \left((1-x_s)c_{p,\text{dryair}} + x_s c_{p,s} \right) \left((T-T_o) - T_o \ln\left(\frac{T}{T_o}\right) \right) + RT_o \ln\left(\frac{P}{P_o}\right) [\text{J/mol}] \quad (7)$$

$$Ex'_{ch,\text{humidair}} = RT_o \left(x_s \ln\left(\frac{x_s}{x_{s,o}}\right) + (1-x_s) \ln\left(\frac{1-x_s}{1-x_{s,o}}\right) \right) [\text{J/mol}] \quad (8)$$

where x_s is the mole fraction of water vapour in indoor air, $c_{p,\text{dryair}}$ is a constant molar isobaric heat capacity of dry air, $c_{p,s}$ is a constant molar isobaric heat capacity of water in the vapour phase, and T and P are the temperature and pressure of the indoor air. Subscript o indicates that the properties are in the state of the reference environment.

Because building engineers and designers prefer to use kilograms instead of moles, equation (7) and equation (8) are hereunder converted to J/kg. The mole fractions of water vapour in air x_s and of dry air x_{dryair} are related to the humidity ratio W (ASHRAE, 1993). In this work, the humid air is considered as a two-component mixture of dry air and water vapour. Then the mole fraction of water vapour in air x_s can be formulated in terms of the humidity ratio W [kg water vapour/kg dry air], as shown in equation (9).

$$x_s = \frac{W}{0.62198 + W} \quad [\text{mol water vapour/mol humid air}] \quad (9)$$

Besides, the molar mass of the humid air m_{humidair} [kg/mol] can be calculated by using equation (10), where the molar mass of dry air m_{dryair} and of water vapour m_s are 0.0290 kg/mol and 0.0180 kg/mol respectively (ASHRAE, 1993).

$$m_{\text{humidair}} = \frac{1 + W}{34.5224 + 55.5081W} \quad [\text{kg/mol}] \quad (10)$$

Substituting equation (9) into equation (7) and equation (8) and dividing the R value by the molar mass of humid air (equation (10)), the exergy value of humid air in buildings (per kilogram of humid air) can be calculated, as functions of air temperature T and humidity ratio W , by using equation (11) and equation (12).

$$Ex_{ph,\text{humidair}} = \left(\frac{0.62198c_{p,\text{dryair}} + Wc_{p,s}}{0.62198 + W} \right) \left((T - T_o) - T_o \ln \left(\frac{T}{T_o} \right) \right) + R \left(\frac{1 + W}{34.5224 + 55.5081W} \right)^{-1} T_o \ln \left(\frac{P}{P_o} \right) \quad (11)$$

$$Ex_{ch,\text{humidair}} = R \left(\frac{1 + W}{34.5224 + 55.5081W} \right)^{-1} T_o \left(\left(\frac{W}{0.62198 + W} \right) \ln \left(\frac{W}{W_o} \right) + \ln \left(\frac{0.62198 + W_o}{0.62198 + W} \right) \right) \quad (12)$$

ASHRAE (1993) recommends $c_{p,\text{dryair}}$ and $c_{p,s}$ as constant values (1.006 kJkg⁻¹K⁻¹ for dry air and 1.805 kJkg⁻¹K⁻¹ for water vapour). However, the molar isobaric heat capacity c_p is a function of temperature. An empirical equation for calculating c_p (Smith et al., 1996) is shown in equation (13).

$$\frac{c_p}{R} = A + BT + \frac{D}{T^2} \quad (13)$$

where A , B and D are coefficients of the equation. The coefficients for water vapour and dry air are shown in Table 1.

Table 1 The coefficients for calculating c_p for water vapour and dry air

Substance	A	B [K ⁻¹]	D [K ²]
Water vapour	3.470	1.450 x 10 ⁻³	0.121 x 10 ⁻³
Dry air	3.355	0.5750 x 10 ⁻³	-0.016 x 10 ⁻³

For air temperatures between $\pm 50^\circ\text{C}$, the average value of $c_{p,\text{dryair}}$ is 29.0167 Jmol⁻¹K⁻¹ (1.0017 kJkg⁻¹K⁻¹) and the average value of $c_{p,s}$ is 33.5408 Jmol⁻¹K⁻¹ (1.8618 kJkg⁻¹K⁻¹).

The total change of $c_{p,dryair}$ is 2% and the total change of $c_{p,s}$ is 0.8%. The average values of $c_{p,dryair}$ and $c_{p,s}$ are used for this work.

4. Approach

The exergy value of indoor air (per kilogram of humid air) at specific indoor climate conditions is investigated as a function of different outdoor climate conditions. Reference environments, used for the exergy calculations, are actual outdoor climate, time-dependent, and from three cities in different climate zones. Differences in air properties (temperature T , humidity ratio W and pressure P) between the indoor air and the outdoor air are variables for this study. Figure 1 illustrates a state of indoor air compared with a reference environment.

Indoor climate conditions, used in the exergy calculations, are air temperature T_i between 20-26°C, relative humidity RH_i between 30-60% and air pressure P_i equal to atmospheric pressure. The indoor climate conditions are commonly applied for an indoor thermal comfort zone. Figure 2 shows the area of the conditioned indoor climate on the ASHRAE psychometric chart (ASHRAE, 1993).

A characteristic of the indoor climate conditions is that they are allowed to vary in a window of indoor air temperatures T_i and humidity ratios of the indoor air W_i . W_i can be obtained by using the ASHRAE psychometric chart or equations (ASHRAE, 1993), as a relation of T_i and RH_i . Therefore W_i [kg water vapour/kg dry air] are between 0.0044-0.0088 where T_i is 20°C and RH_i is between 30-60%. W_i are between 0.0064-0.0128 where T_i is 26°C and RH_i is between 30-60%. T_i is determined before determination of W_i , since W_i has a relation to T_i and RH_i . To determine T_i the following rules are used.

- if $T_o \geq 26^\circ\text{C}$, then $T_i = 26^\circ\text{C}$ (or, in other words, the temperature inside the building is maintained at 26°C by HVAC equipment when the outside temperature is above 26°C)
- if $T_o \leq 20^\circ\text{C}$, then $T_i = 20^\circ\text{C}$ (or, in other words, the temperature inside the building is maintained at 20°C by HVAC equipment when the outside temperature is below 20°C)
- if $20^\circ\text{C} < T_o < 26^\circ\text{C}$, then $T_i = T_o$ (the indoor and outdoor temperatures are equal).

Figure 1 A state of indoor air compared with a reference environment

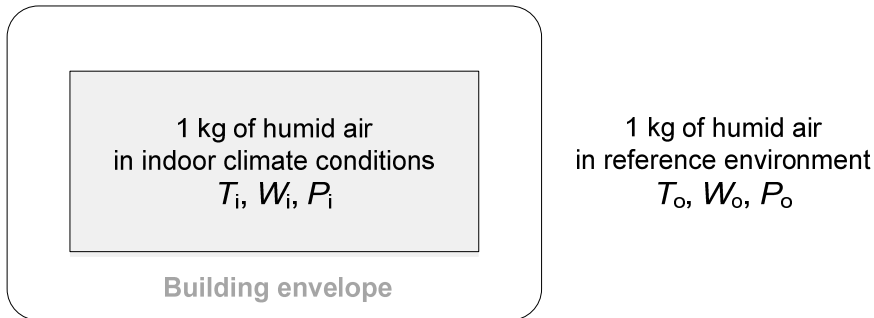
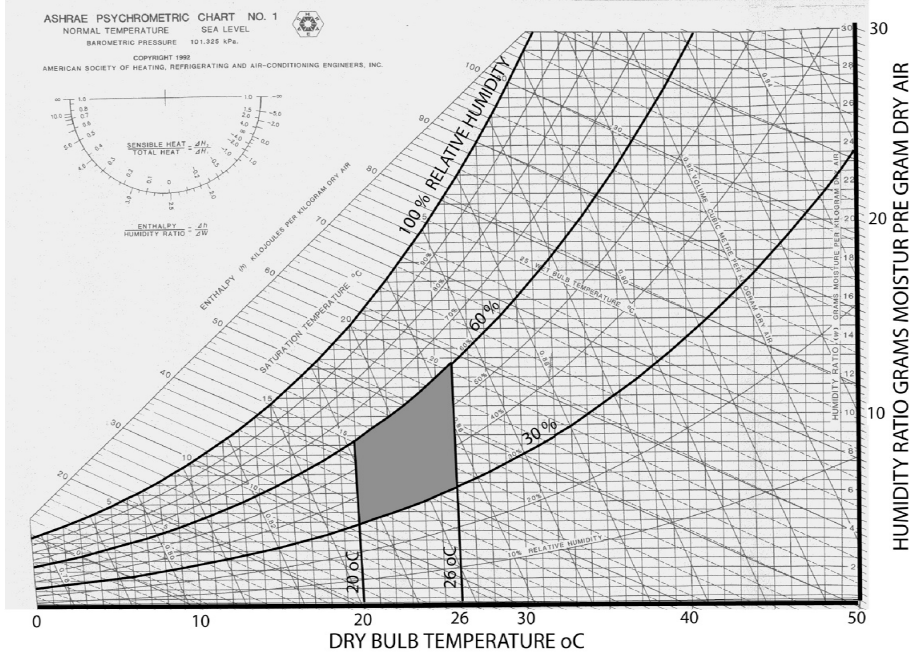


Figure 2 The conditioned indoor climate area on the ASHRAE psychrometric chart



To determine W_i identical rules are used, but the upper and lower bounds of W_i must be determined at $T_i = 20^\circ\text{C}$ and then at $T_i = 26^\circ\text{C}$. W_i can then be determined in the same way as T_i before.

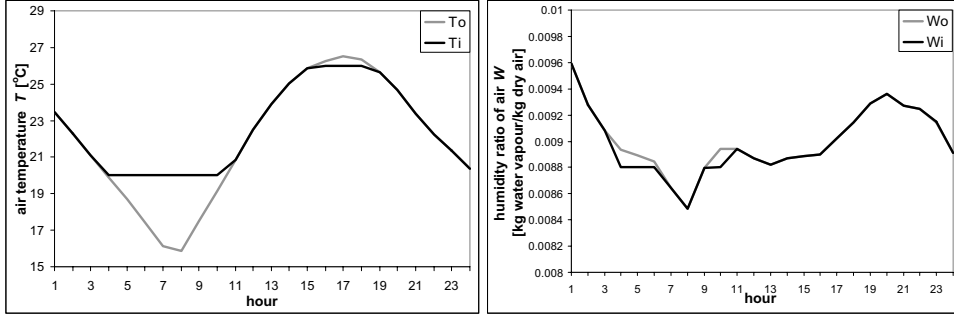
- if $W_o \geq W_{i,\max}$, then $W_i = W_{i,\max}$
- if $W_o \leq W_{i,\min}$, then $W_i = W_{i,\min}$
- if $W_{i,\min} < W_o < W_{i,\max}$, then $W_i = W_o$.

Generally speaking, when the outdoor climate conditions are within the shaded range in Figure 2, the exergy of the indoor air is zero because it is at the same condition as the outdoor air.

To visualise the above defined indoor climate conditions, hourly profiles of the air properties (T and W) inside and outside buildings in Lisbon on 21 July of the Typical Meteorological Year (TMY) are given in Figure 3. The hourly outdoor climate data comes from the TMY2 data (NREL, 1995). In Figure 3 (left), from 4:00 to 10:00, $T_o < T_i$ and then $T_i = 20^\circ\text{C}$. From 16:00 to 18:00, $T_o > T_i$ and then $T_i = 26^\circ\text{C}$. For the remaining period of the day, T_o values are between 20-26°C and then $T_i = T_o$. In Figure 3 (right), from 4:00 to 6:00 and at 10:00, W_i is 0.0088, because during these times T_i is equal to 20°C and W_i should be then between 0.0044-0.0088 (corresponding to RH_i between 30-60%). The outdoor air should be dehumidified until which the humidity ratio of the air is equal to 0.0088, before entering the buildings. For the remaining period of the day, W_i corresponds to RH_i and T_i at the hours of the day, equal to W_o .

Outdoor climate conditions, used in the exergy calculations, are from three different places in different climates (a temperate sea climate, Lisbon PT; a cold climate, De Bilt NL; and a hot and humid climate, Bangkok TH). Climate data from these sites are taken from the TMY2 data (NREL, 1995). The data represent air temperatures, humidity ratios and atmospheric pressures, on an hourly basis. Characteristics of the outdoor climate conditions are presented in the next chapter.

Figure 3 Hourly profiles of the air properties (T and W) inside and outside buildings in Lisbon on 21 July of the Typical Meteorological Year



The analysis framework in Figure 4 is used to study the influence of possible definitions of a reference environment on the exergy of air in buildings. The analysis framework considers exergy values of air in buildings, calculated by using different given reference environment alternatives, to investigate what reference environment alternative might be able to substitute the actual reference environment depending on all air properties (T , W and P). The analysis framework also considers the exergy values of air in buildings at different levels of humidity, by assuming $W_i = W_o$ and $W_i = W_o = 0$, with the other reference environment parameters remaining the same defined conditions of the indoor air and outdoor air. The aim is to investigate the possibility of considering indoor air and outdoor air as dry air for exergy calculations.

The exergy calculations are performed for 4 indoor environment alternatives:

- *indoor environment alternative A* $Ex(T_i, W_i, P_o)$: exergy calculation of humid air in buildings, assuming $P_i = P_o$, using equation (11) and equation (12), i.e. neglecting pressure differences between indoor and outdoor air.
- *indoor environment alternative B* $Ex(T_i, W_o, P_o)$: exergy calculation of humid air in buildings, assuming $P_i = P_o$ and $W_i = W_o$, using equation (14), i.e. neglecting differences in pressure and humidity ratio between indoor and outdoor air
- *indoor environment alternative C* $Ex(T_i, 0, P_o)$: exergy calculation of dry air in buildings, assuming $P_i = P_o$ and $W_i = W_o = 0$, using equation (15)
- *indoor environment alternative D* $Ex(T_i, W_i, P_i)$: exergy calculation of humid air in buildings, assuming $P_i - P_o$ is between ± 100 Pa, using equation (11) and equation (12).

Equation (14) and equation (15) are derived from equation (11) and equation (12).

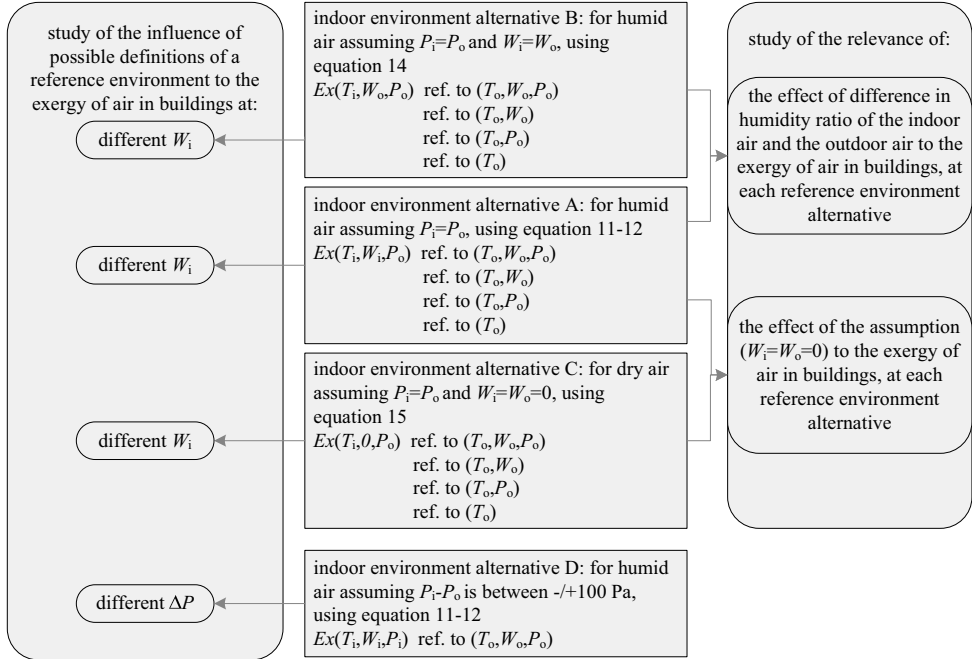
$$Ex = \left(\frac{0.62198c_{p,dryair} + W_o c_{p,s}}{0.62198 + W_o} \right) \left((T - T_o) - T_o \ln \left(\frac{T}{T_o} \right) \right) \quad (14)$$

$$Ex = c_{p,dryair} \left((T - T_o) - T_o \ln \left(\frac{T}{T_o} \right) \right) \quad (15)$$

Four possible reference environment alternatives are defined as combinations of the air parameters (T , W and P), to be used in this framework. The reference environment alternatives considered are the following:

- *reference environment alternative 1* (T_o, W_o, P_o): air temperature T_o , humidity ratio of air W_o and air pressure P_o
- *reference environment alternative 2* (T_o, W_o): air temperature T_o and humidity ratio of air W_o
- *reference environment alternative 3* (T_o, P_o): air temperature T_o and air pressure P_o
- *reference environment alternative 4* (T_o): air temperature T_o .

Figure 4 Analysis framework to study the influence of possible definitions of a reference environment to determine the exergy of air in buildings



Differences between exergy results, made by using an indoor environment alternative with all reference environment alternatives, are investigated in order to identify what reference environment alternative might be able to substitute the actual reference environment (reference environment alternative 1), within the assumptions of the indoor environment alternative. Chemical contribution to the exergy value of the air can be obtained as difference between the exergy results using reference environment alternative 1 and 3, or using reference environment alternative 2 and 4. Mechanical contribution to the exergy value of the air can be obtained as difference between the exergy results using reference environment alternative 1 and 2, or using reference environment alternative 3

and 4. The rest of the contributions to the exergy value of the air is the thermal contribution.

Differences between exergy results, obtained by using indoor environment alternatives A and B with a similar reference environment alternative, are investigated to identify the possibility of neglecting the difference between W_i and W_o for the exergy calculations.

Differences between exergy results, obtained by using the indoor environment alternatives A and C with a similar reference environment alternative, are investigated to identify the possibility of considering indoor air and outdoor air as dry air for the exergy calculations.

The mechanical contribution to exergy of indoor air is then studied by using indoor environment alternative D, in a range of air pressure differences between indoor air and outdoor air ΔP , between ± 100 Pa. This is made to investigate under which conditions the mechanical contribution of the exergy value may be neglected or not. This is of particular interest when mechanical exhaust ventilation systems are used, which create an underpressure in the building, or when mechanical supply systems are used, which create an overpressure.

The exergy calculations are made for every hour in the TMY, by using hourly changing indoor and outdoor climate data. Hourly indoor climate data are determined by using the hourly outdoor climate data, as explained above. The exergy results are presented as average values per season and for the TMY (8760 hours). In this study, the TMY is divided into 2 seasons: Season I is defined as the period from 1 May to 30 September of the TMY (3672 hours); Season II is defined as the remaining period of the TMY (5088 hours). The average values of the exergy results are used for the discussions (see Figure 4).

Furthermore, the possibility of using a static reference environment, instead of a dynamic reference environment, for the exergy calculations of dry air is examined. Possible combinations of annual statistical (average, mode and median) values of T_o and of T_i are used for exergy calculations of air in buildings. The exergy results are then compared with the annual average values of exergy results of the TMY, calculated by using indoor environment alternative A and reference environment alternative 1.

5. Case study

In Sections 5.1-5.3 the climate characteristics of three different locations are shown, as well as the tabulated results of the exergy calculations. These results are analysed and compared in Section 5.4.

5.1. Temperate sea climate, Lisbon PT

Lisbon is located at 38°43' north, 9°8' west. Average, mode, median and standard deviation values of air temperatures T and of humidity ratios W at the city, per season and of the TMY, are given in Table 2. These values are derived from the climate data at Lisbon, taken from the TMY2 data (NREL, 1995).

Table 2 Air temperature and humidity ratio at Lisbon

		Air temperature T [°C]			Humidity ratio W [-]		
		Season I	Season II	Year	Season I	Season II	Year
Outdoor	Average	20.91	13.85	16.81	0.0101	0.0074	0.0085
	Mode	20.60	10.55	10.55	0.0091	0.0066	0.0091
	Median	20.85	13.70	16.43	0.0099	0.0072	0.0084
	Standard deviation	4.35	3.97	5.41	0.0018	0.0015	0.0021
Indoor	Average	21.96	20.14	20.90	0.0095	0.0073	0.0082
	Mode	20.00	20.00	20.00	0.0088	0.0088	0.0088
	Median	20.85	20.00	20.00	0.0091	0.0072	0.0084
	Standard deviation	2.28	0.66	1.80	0.0015	0.0013	0.0018

Exergy values of air in buildings in the city are calculated for every hour in the TMY, by using combinations of the indoor environment alternatives and the reference environment alternatives, mentioned in the previous section. An example of an exergy calculation is given below.

Exergy of air in buildings in Lisbon at the first hour of the TMY is calculated by taking indoor environment alternative A with reference environment alternative 1 (using equation (11) and equation (12)) and using the following data. The outdoor climate data are taken from the TMY2 data (NREL, 1995). The indoor climate data are derived from the conditions explained in the previous section.

Outdoor air: $T_o = 287.15$ K, $W_o = 0.00722$, $P_o = 101100$ Pa

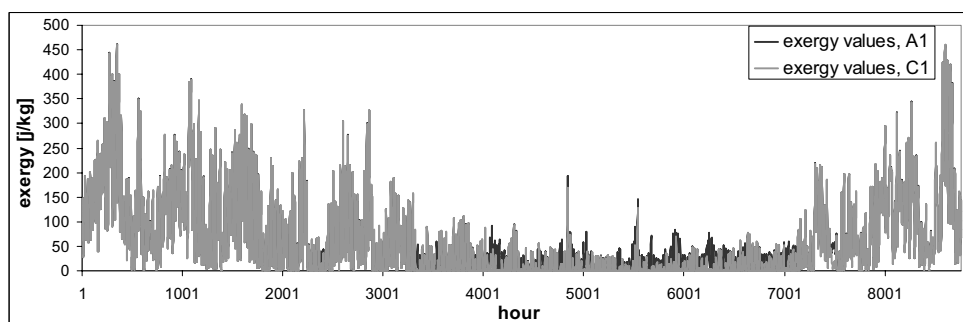
Indoor air: $T_i = 293.15$ K, $W_i = 0.00722$, $P_i = 101100$ Pa

Constant values: $c_{p,dryair} = 1.0017$ kJkg⁻¹K⁻¹, $c_{p,s} = 1.8618$ kJkg⁻¹K⁻¹, $R = 8.314$ Jmol⁻¹K⁻¹

$$\begin{aligned}
 Ex_{ph,humidair} &= \left(\frac{(0.62198)(1.0017) + (0.00722)(1.8618)}{0.62198 + 0.00722} \right) \\
 &\quad \left((293.15 - 287.15) - (287.15) \ln \left(\frac{293.15}{287.15} \right) \right) \\
 &\quad + (8.314) \left(\frac{34.5224 + (55.5081)(0.00722)}{1 + 0.00722} \right) (287.15K) \ln \left(\frac{101100 \text{ Pa}}{101100 \text{ Pa}} \right) \\
 &= 0.06254 \text{ kJkg}^{-1} \\
 Ex_{ch,humidair} &= (8.314) \left(\frac{34.5224 + (55.5081)(0.00722)}{1 + 0.00722} \right) (287.15K) \\
 &\quad \left(\left(\frac{0.00722}{0.62198 + 0.00722} \right) \ln \left(\frac{0.00722}{0.00722} \right) + \ln \left(\frac{0.62198 + 0.00722}{0.62198 + 0.00722} \right) \right) \\
 &= 0 \text{ Jkg}^{-1} \\
 Ex_{humidair} &= Ex_{ph,humidair} + Ex_{ch,humidair} \\
 &= 0.06254 + 0 \text{ kJkg}^{-1} \\
 &= 0.06254 \text{ kJkg}^{-1}
 \end{aligned}$$

This calculation is then repeated for each hour of the TMY year. Figure 5 shows the hourly exergy calculation results for the TMY year, calculated by taking indoor environment alternative A with reference environment alternative 1 (line A1) and indoor environment alternative C with reference environment alternative 1 (line C1). The differences in the exergy values between A1 and C1 are visible mostly between hour 3625 and hour 7269 that approximately corresponds to the period between June and October. In the other periods the average value of the differences is 3.99%. The differences in percentage are compared to the exergy values calculated by taking indoor environment alternative A with reference environment alternative 1.

Figure 5 Hourly profiles of exergy calculation results for the TMY year, calculated by taking indoor environment alternative A with reference environment alternative 1 (line A1) and indoor environment alternative C with reference environment alternative 1 (line C1)



Average values of the exergy values of air in the buildings per season and of the TMY are calculated from the sum of the hourly exergy calculation results, shown in Table 3.

Table 3 Exergy of air in buildings in Lisbon (in average values, J/kg) for indoor environment alternatives A, B and C

Indoor environment alternative	Reference environment alternative	Season I	Season II	Year
A (humid air)	1,2	18.45	93.39	61.98
A (humid air)	3,4	12.66	92.16	58.84
B (humid air; $W_i=W_o$)	1,2,3,4	12.67	92.16	58.84
C (dry air)	1,2,3,4	12.64	92.01	58.74

Two notices for the exergy calculations are the following. First, exergy calculations using reference environment alternatives 1 and 2, give similar exergy results, since there is no mechanical contribution to the exergy results: pressure difference between indoor air and outdoor air is not considered in indoor environment alternatives A, B and C, but in indoor environment alternative D. This also happens to exergy calculations using reference environment alternatives 3 and 4. The mechanical contributions to the exergy values of indoor air are discussed at the end of item 5.4. Second, exergy calculations using indoor environment alternative B with one of the reference environment alternatives, give similar exergy results, since there is no chemical contribution to the exergy results. This also happens to exergy calculations using indoor environment alternative C. Table 4 illustrates the two notices by presenting relations of exergy results calculated by using combinations of indoor environment alternatives A-C and reference environment

alternatives 1-4. The shadings represent possible alternatives. In addition, these notices are applicable to the other case studies as well.

Table 4 Relations of the exergy results calculated by using combinations of indoor environment alternatives A-C and reference environment alternatives 1-4

Indoor environment alternative	Reference environment alternative			
	1 (T_o, W_o, P_o)	2 (T_o, W_o)	3 (T_o, P_o)	4 (T_o)
A $Ex(T_i, W_i, P_o)$	A1 = A2		A3 = A4	
B $Ex(T_i, W_o, P_o)$	B1 = B2 = B3 = B4			
C $Ex(T_i, \theta, P_o)$	C1 = C2 = C3 = C4			

5.2. Cold climate, De Bilt NL

De Bilt is located at 52°12' north, 5°18' east. Average, mode, median and standard deviation values of air temperatures T and of humidity ratios W at the city, per season and of the TMY, are given in Table 5.

Table 5 Air temperature and humidity ratio at De Bilt

		Air temperature T [°C]			Humidity ratio W [-]		
		Season I	Season II	Year	Season I	Season II	Year
Outdoor	Average	14.99	5.31	9.37	0.0081	0.0046	0.0061
	Mode	13.55	0.60	6.50	0.0087	0.0037	0.0037
	Median	14.84	5.35	9.34	0.0079	0.0045	0.0057
	Standard deviation	4.84	5.28	6.99	0.0020	0.0015	0.0024
Indoor	Average	20.38	20.00	20.16	0.0077	0.0051	0.0062
	Mode	20.00	20.00	20.00	0.0088	0.0044	0.0044
	Median	20.00	20.00	20.00	0.0079	0.0045	0.0057
	Standard deviation	1.17	0.02	0.78	0.0016	0.0010	0.0018

These values are derived from the climate data at that site, taken from the TMY2 data (NREL, 1995). The average values of the exergy values of air in buildings in De Bilt per season and of the TMY are calculated from hourly exergy calculation results, shown in Table 6.

Table 6 Exergy of air in buildings in De Bilt (average values, in J/kg) for indoor environment alternatives A, B and C

Indoor environment alternative	Reference environment alternative	Season I	Season II	Year
A (humid air)	1,2	85.37	439.60	291.12
A (humid air)	3,4	81.12	426.42	281.68
B (humid air; $W_i=W_o$)	1,2,3,4	81.12	426.33	281.63
C (dry air)	1,2,3,4	81.00	425.93	281.34

5.3. Hot and humid climate, Bangkok TH

Bangkok is located at 13°45' north, 100°31' east. Average, mode, median and standard deviation values of air temperatures T and of humidity ratios W at the city, per season and of the TMY, are given in Table 7. These values are derived from the climate data at the site, taken from the TMY2 data (NREL, 1995).

Table 7 Air temperature and humidity ratio at Bangkok

		Air temperature T [°C]			Humidity ratio W [-]		
		Season I	Season II	Year	Season I	Season II	Year
Outdoor	Average	28.30	27.28	27.71	0.0185	0.0166	0.0174
	Mode	27.90	27.85	26.50	0.0205	0.0150	0.0150
	Median	28.25	27.33	27.75	0.0184	0.0164	0.0175
	Standard deviation	2.70	3.37	3.15	0.0019	0.0026	0.0025
Indoor	Average	25.71	25.18	25.40	0.0126	0.0123	0.0124
	Mode	26.00	26.00	26.00	0.0128	0.0128	0.0128
	Median	26.00	26.00	26.00	0.0128	0.0128	0.0128
	Standard deviation	0.70	1.49	1.25	0.0004	0.0010	0.0008

Average values of exergy values of air in buildings in Bangkok per season and of the TMY are calculated from hourly exergy calculation results, shown in Table 8.

Table 8 Exergy of air in buildings in Bangkok (average values, in J/kg) for indoor environment alternatives A, B and C

Indoor environment alternative	Reference environment alternative	Season I	Season II	Year
A (humid air)	1,2	169.00	110.32	134.91
A (humid air)	3,4	20.17	16.71	18.16
B (humid air; $W_i=W_o$)	1,2,3,4	20.20	16.73	18.19
C (dry air)	1,2,3,4	20.11	16.66	18.10

5.4. Analysis and comparison of the results

Under the assumption of identical indoor and outdoor air pressure, considering humid indoor air (indoor environment alternative A) with humid outdoor air as reference environment (alternatives 1 and 2) leads to the most accurate determination of the exergy contents of indoor air. When the humidity of the reference environment is neglected (alternatives 3 and 4), differences in average values of the hourly exergy calculation results for season I may be very high (31.4% for a temperate sea climate, 88.1% for a hot and humid climate, but only 5.0% for a cold climate). For season II and for the whole year, differences in average values of the exergy results are low for the temperate sea climate (1.3% and 5.1% respectively) and for the cold climate (3.0% and 3.2% respectively), but very high for the hot and humid climate (84.9% and 86.5% respectively). Therefore, when dealing with humid air inside, with a humidity ratio different from the reference environment, the reference environment should also consists of humidity ratio of air W , because the chemical contribution to the exergy value is relatively high compared with the thermal contribution. This is particularly important for

buildings with dehumidifying/humidifying equipment, air-cooling and also for buildings with a high occupancy level and a low level of moderate ventilation rate, because of the humidity production of occupants.

Exergy calculations using indoor environment alternatives B and C, with reference environment alternative 1, gives more or less the same exergy results, since average values of W , per season and for the year, are very small, less than 0.0101.

When the humidity is assumed to be identical indoor and outdoor (indoor environment alternative B) which will be the case of well-ventilated buildings with a low occupancy (hence a low water vapour production), it is not necessary to chose humid air as the reference environment. A reference environment consisting of dry air is accurate enough and the indoor air itself may be considered as dry air too (indoor environment alternative C). These results are valid for all the studied types of climate.

The exergy content of dry air (indoor environment alternative C), calculated with the annual statistical (average, mode and median) values of the indoor and outdoor air temperatures given in Tables 2, 5 and 7 is compared with the exergy results of the dynamic simulations given in Tables 3, 6 and 8 (indoor environment alternative A with reference environment alternative 1). The exergy calculation results of dry air that have the smallest differences from the exergy results of the dynamic simulations are given, per season and for the TMY, in Table 9.

Table 9 The exergy calculation results of dry air (indoor environment alternative C) that have the smallest differences from the exergy results of the dynamic simulations (indoor environment alternative A with reference environment alternative 1)

		Static calculation ¹		Dynamic simulation ²		Difference [%]
		T_i	T_o	Ex [J/kg]	Ex [J/kg]	
Lisbon	Season I	Average	Mode	3.17	18.45	82.84
	Season II	Average	Median	68.33	93.39	23.35
	Year	Average	Median	28.79	61.98	44.50
De Bilt	Season I	Average	Mode	80.46	85.37	5.76
	Season II	Average	Average	375.91	439.60	14.49
	Year	Mode	Mode	317.10	291.12	-8.93
Bangkok	Season I	Average	Average	11.24	169.00	93.35
	Season II	Average	Mode	11.99	110.32	89.13
	Year	Average	Median	8.93	134.91	93.13

¹ Static calculation is carried out, as calculation using annual statistical (average, mode or median) values of T_o and of T_i

² Dynamic simulation is carried out, as summation of results from hourly calculations for all hours in the TMY year. More explanation is in item 5.1.

For the whole TMY year, the use of the median outdoor temperature and the average indoor temperature will cause a severe underestimation of the exergy content for the temperate sea climate (<44.50%) and the hot and humid climate (<93.13%). For the cold climate, the use of mode outdoor and indoor temperature will cause an overestimation of the exergy content (>8.93%).

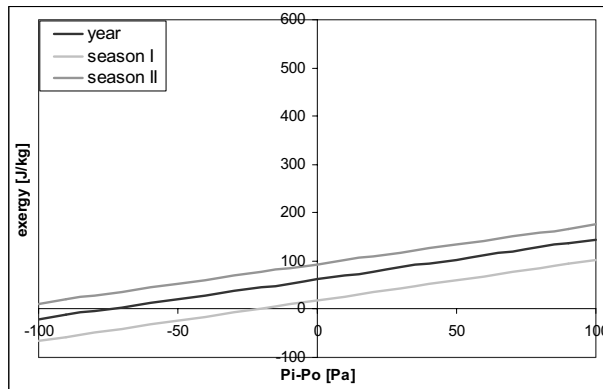
For season I and season II, the use of the average indoor temperature and a various statistical value of the outdoor temperature will also cause a severe underestimation of the exergy content. In the hot and humid climate, the exergy calculation results of dry air are 93.35% (for season I) and 89.13% (for season II) less than the exergy results of the

dynamic simulations. In the cold climate, the exergy calculation results of dry air are 5.76% (for season I) and 14.49% (for season II) less than the exergy results of the dynamic simulations.

As a result, it indicates that, in this case, using the annual statistical values of the indoor air temperatures and outdoor air temperatures is not accurate enough for the exergy calculation of indoor air.

The mechanical contributions to the exergy values of indoor air are calculated using indoor environment alternative D with reference environment alternative 1, for every hour of the TMY. The difference in air pressure between indoor and outdoor air ΔP is assumed constant during the year. Figure 6 shows the average values of the exergy values of air in the buildings, per season and of the year, at which ΔP is between ± 100 Pa, for the temperate sea climate of Lisbon.

Figure 6 Exergy of air in buildings in Lisbon, when ΔP is between ± 100 Pa (average values, in J/kg) for indoor environment alternative D



The lines in Figure 6 are almost linear because the range of ΔP is much smaller than P_o . These lines are nearly parallel since differences in average outdoor air temperatures (in Kelvin) in season I, season II and the year are all small, less than 1.50%. If a mechanical ventilation system were used, ΔP could be about 40 Pa. In this case, the mechanical contributions to the exergy values could be between 32.65-33.55 J/kg for the temperate sea climate, between 31.75 and 32.95 for the cold climate and between 34.33 and 34.45 for the hot humid climate. For the temperate sea climate, the mechanical contribution to the exergy could therefore amount for almost 35% of the total annual average exergy. When considering season I only, this percentage increases to 65%. For the cold climate the mechanical contribution to the annual average share of exergy decreases to less than 10% (27% for season I) and for the hot and humid climate it amounts 20% of the annual average exergy. This means that when the building is placed in under or overpressure through a mechanical ventilation system, the mechanical exergy losses are not negligible.

6. Conclusions

The paper presents the analysis results of the exergy determination of air in buildings. The exergy determination is examined for combinations of the outdoor climate conditions of some climate zones (a temperate sea climate zone, Lisbon PT; a cold climate zone, De Bilt NL; and a hot and humid climate zone, Bangkok TH), with specific indoor climate

conditions. The reference environment for humid air, used for the exergy analysis, is determined by three parameters of air properties (temperature T , humidity ratio W and pressure P). The influence of possible definitions of a reference environment to determine the exergy of air in buildings is studied, as is the relevance of the effect of each of the parameters of indoor air on the exergy of air in buildings.

The chemical contribution to the exergy value of air in buildings might, in some cases, be negligible for the exergy calculation of air in the buildings, depending on the season and the type of climate. In a cold climate the chemical contribution is relatively small compared with the thermal contribution. However, the chemical contribution is rather large in a hot and humid climate (and in a temperate sea climate in season I in particular). The exergy calculation in a cold country might use only environmental temperature as a characteristic of the reference environment, and assume that the outdoor air and the indoor air are completely dry. In a hot and humid (or temperate sea) country these assumptions could lead to very large discrepancies and are therefore not recommended.

It is acceptable in some climates to consider a static reference environment only, instead of a dynamic reference environment, for calculating the exergy value of air in buildings for a year. In a cold climate, exergy value of the air strongly depends on its thermal contribution. Accordingly, the outdoor air temperature might be sufficient as a reference environment for the exergy calculation. Besides, the annual mode value of the hourly outdoor air temperatures and of the hourly indoor air temperatures for the TMY might be appropriate as a static reference environment for the exergy calculation. This is not acceptable for the exergy calculation in a hot and humid (or temperate sea) climate.

The mechanical contribution to the exergy of indoor air depends on the air pressure difference between indoor and outdoor air ΔP and the outdoor air temperature T_o . The mechanical contribution has a linear relation to ΔP , where ΔP is between ± 100 Pa, since the range of ΔP is much smaller than P_o . When using a mechanical ventilation system, ΔP could be about 40 Pa. For these outdoor climates investigated here, the mechanical contributions to the exergy values could be between 10 and 35% of the total exergy for the seasons and the TMY.

The results are summed up in Table 10. Error values in % of the exergy calculations for all options are ratios between the average exergy values (seasonal and annual) of indoor air at (T_i, W_i) resulted by the different indoor environment alternatives or the static exergy calculation results (column 1; Table 10) and the average exergy values of the indoor air resulted by indoor environment alternative A with reference environment alternative 1. The average exergy values of the indoor air are results from hourly exergy calculations (see item 5.1 for more explanation), and given in Tables 3, 6 and 8. The static exergy calculation results are given in Table 9. For example, the error of the exergy calculation that considers humid indoor air and uses T_o as the reference environment, which is 31.36% (column 2, row 3; in Table 10), is a ratio between 12.66 J/kg (column 3, row 3; in Table 3) and 18.45 J/kg (column 3, row 2; in Table 3). The grey colour indicates some error values that are considered to be not acceptable.

It is recommended to pay attention to this error problem when exergy chains are calculated in buildings (e.g. Schmidt, 2004; Itard, 2005; Sakulpipatsin et al., 2006) because the share of exergy losses throughout the exergy chain could change strongly according to the chosen reference and indoor air properties. This will be the case in particular when humidifiers and dehumidifiers are used or when mechanical ventilation systems are used.

Table 10 Error values in % of the exergy calculations for all options

Exergy Calculation	Temperate sea climate (Lisbon)			Cold climate (De Bilt)			Hot and humid climate (Bangkok)		
	Season I [%]	Season II [%]	Year [%]	Season I [%]	Season II [%]	Year [%]	Season I [%]	Season II [%]	Year [%]
A3 ^{1,2} , A4 ^{1,2} (ref. to T_o only)	31.36	1.32	5.07	4.98	3.00	3.24	88.06	84.85	86.54
B1 ^{1,3} , B2 ^{1,3} , B3 ^{1,3} , B4 ^{1,3} , ($W_i=W_o$)	31.36	1.32	5.06	4.98	3.02	3.26	88.05	84.83	86.52
C1 ^{1,4} , C2 ^{1,4} , C3 ^{1,4} , C4 ^{1,4} , ($W_i=W_o=0$)	31.50	1.47	5.22	5.13	3.11	3.36	88.10	84.90	86.58
Static calculation	82.84	23.35	44.50	5.76	14.49	8.93	93.35	89.13	93.13
D1 ¹ with $P_i-P_o = 40$ Pa	64.51	25.93	34.78	26.02	6.63	9.73	16.93	23.73	20.31

¹ The exergy calculation names contain two letters. The first alphabet and the second number indicate what indoor environment alternative and what reference environment alternative the exergy calculations use respectively. For example, the exergy calculation A3 use indoor environment alternative A with reference environment alternative 3.

² Results from the exergy calculations A3 and A4 are similar. More explanation is in item 5.1

³ Results from the exergy calculations B1, B2, B3 and B4 are similar. More explanation is in item 5.1 In addition, these results do not apply in a situation where the humidities indoor and outdoor are identical all over the year.

⁴ Results from the exergy calculations C1, C2, C3 and C4 are similar. More explanation is in item 5.1

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Nomenclature

A	Coefficient [-]
B	Coefficient [-]
c_p	Constant molar isobaric heat capacity [$\text{m}^2\text{kg}^{-2}\text{mol}^{-1}\text{K}^{-1}$; $\text{Jmol}^{-1}\text{K}^{-1}$]
D	Coefficient [-]
Ex	Exergy [m^2kg^{-2} ; J]
Ex'	Molar exergy [$\text{m}^2\text{kg}^{-2}\text{mol}^{-1}$; Jmol^{-1}]
H	Enthalpy [m^2kg^{-2} ; J]
KE	Kinetic energy [m^2kg^{-2} ; J]
M	Molar mass [mol]
P	Air pressure [$\text{m}^{-1}\text{kg}^{-2}$; Pa]

The influence of possible definitions of a reference environment

ΔP	Air pressure difference between the inside and the outside of buildings [$\text{m}^{-1}\text{kgs}^{-2}$; Pa]
PE	Potential energy [$\text{m}^2\text{kgs}^{-2}$; J]
R	Molar gas constant [$\text{m}^2\text{kgs}^{-2}\text{mol}^{-1}\text{K}^{-1}$; $\text{Jmol}^{-1}\text{K}^{-1}$]
RH	Relative humidity [-]
S	Entropy [$\text{m}^2\text{kgs}^{-2}\text{K}^{-1}$; JK^{-1}]
T	Air temperature [K; °C with notation]
W	Humidity ratio [-]
x	Mole fraction of water vapour in humid air [-]

Subscripts

Ch	Chemical
Dryair	Dry air
Humidair	Humid air
i	Indoor
Me	Mechanical
Mix	Mixing
o	Reference environment state; dead state
Ph	Physical
Pure	Pure
S	Saturated water; water vapour
Th	Thermal

Abbreviations

NL	Netherlands
PT	Portugal
Ref	Reference
TH	Thailand
TMY	Typical Meteorological Year

Functional exergy efficiency and exergy consumption behaviour for air-to-air heat exchangers operating at near-environmental temperatures

Abstract: The main purpose of this paper is to investigate how changes in temperature levels affect the efficiency of thermal exergy transfer occurring at near-environmental temperatures. In this regard, this paper focuses on building ventilation as an application example involving heat transfer relatively close to ambient conditions. The analysis uses exergy efficiency and heat transfer effectiveness, combined with exergy consumption and warm/cool exergy, to gain insight into the effect of varying temperatures in heat exchange at near-environmental temperature. A dimensionless temperature T^* is proposed to identify and analyse patterns in the sensitivity of the functional exergy efficiency to temperature variations. The analysis is performed with a simple air-to-air sensible heat exchanger model for heating purposes, assuming constant values of exchanger heat transfer effectiveness. Depending on how the heat exchanger temperatures relate to the environmental temperature, different values of functional exergy efficiency can be obtained for the same exchanger heat transfer effectiveness. Furthermore, some temperature combinations can lead to unexpected values of functional exergy efficiency. The dimensionless temperature T^* is used together with the notion of warm and cool exergy to explain the results, and to identify exergy-efficient temperature combinations for heat exchange at near-environmental temperatures.

Keywords: exergy; exergy efficiency; functional exergy efficiency; heat exchanger; space heating; HVAC design.

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Brief biographical notes about authors

E.C. Boelman, Dr.Eng. MBA, has a background in Adsorption Cooling, Building Engineering and Management, and has been working on renewable energy and exergy analysis applied to buildings for several years.

P. Sakulpipatsin, MSc, is a Doctoral Researcher in Climate Design Group, at the Faculty of Architecture, Delft University of Technology, The Netherlands. He holds two master degrees: in Energy Conversion and Management from the University of Applied Sciences Offenburg, Germany (2003); and in Computer Science from Chulalongkorn University, Thailand (2001).

Dr. Ir. H.J. van der Kooi is Assistant Professor in the Field of Applied Thermodynamics and Chemical Engineering at the Delft University of Technology, The Netherlands.

Dr. L.C.M. Itard is a Researcher in the field of sustainable buildings and HVAC equipment and leads the group sustainable and healthy building of the Research Institute OTB at the Delft University of Technology, The Netherlands.

Prof. Ir. P.G. Luscuere is a professor of building services in the Climate Design Group, at the Faculty of Architecture, Delft University of Technology. He is also a guest professor at the University of Tianjing in China, and a director with the engineering firm Royal Haskoning in the Netherlands.

1. Introduction

1.1. Objective and approach

The main objective of this work is to investigate how changes in temperature levels affect the efficiency of thermal exergy transfer occurring at near-environmental temperatures.

We propose combining the notions of heat exchanger heat transfer effectiveness, functional exergy efficiency and exergy consumption with the concept of warm and cool exergy, in order to obtain new insights that may be useful when specifying the operating temperatures of air-to-air sensible heat exchangers used at near-environmental temperatures. We also propose a dimensionless temperature, which we define as a ratio: of the temperature difference between the hot stream inlet and the reference environment; to the temperature difference between the hot and the cold streams at the heat exchanger inlets. As a possible application example, we address a simplified balanced ventilation system for use in buildings. Other possible application domains include industrial processes at near-environmental temperatures, such as food manufacturing.

1.2. Literature overview

The work presented in this paper builds on the notions of exergy efficiency, exergy consumption, and warm and cool exergy. Several authors have provided definitions for exergy efficiencies (Semenyuk, 1990; Sorin and Brodyansky, 1992; Tsatsaronis, 1993; Kotas, 2001). From those viewpoints, it is possible to define exergy efficiencies in various ways, depending on the significance of various conditions such as sensitivity for

changes in a system, applicability in practice, accuracy and accessibility. Semenyuk (1990) discusses heat exchanger exergetic efficiency as a function of a dimensionless temperature, and has indicated domains of technically inexpedient heat exchanger operation. Similarly to the present study, his analysis shows that using hot thermal carriers (above environmental temperature) to heat cold thermal carriers (below environmental temperature) is irrational since this heating could be accomplished by using environmental air. However, his approach is computationally more complex, as it requires knowledge of heat exchanger inlet and outlet temperatures. Moreover, his study focuses on operating temperatures relatively far from the environmental temperature.

Wu et al. (2006) use the concept of heat transfer effectiveness (ASHRAE, 2000; Holman, 2002) to indicate the relative magnitude of the heat transfer, and perform detailed comparisons of exergy transfer effectiveness with heat transfer effectiveness, for parallel flow, counter-flow and cross-flow heat exchangers operating above and below the surrounding temperature. They analyse variations of exergy transfer effectiveness with number of transfer units (NTU), with the ratio of the heat capacity of cold fluid to that of hot fluid (C_c/C_h) and with finite pressure drops. They note that there is not an optimal combination of NTU and C_c/C_h for maximising exergy transfer effectiveness. They do not elaborate on the effects of temperature variations.

Johannessen et al. (2002) examine temperature profiles and local entropy production profiles in heat exchangers. They show that the standard counter-current heat exchanger is the best first approximation to optimal heat exchange conditions in practice, as it has qualitatively the same properties as the optimal solutions presented in their study; when the temperature difference $T_h - T_c$ between the hot and the cold fluids is approximately constant. In the present study the authors assume $T_h - T_c$ to be constant throughout the heat exchanger, for the sake of simplicity.

Hesselgreaves (2000) studies entropy generation in heat exchangers, and develops a new relationship for optimising balanced counter-flow heat exchangers for a 'long' duty, i.e. with significant temperature changes of the working fluids.

Yilmaz et al. (2001) present a review of second-law based performance evaluation criteria to evaluate the performance of heat exchangers, using entropy and exergy as two different groups of evaluation parameters. Tsatsaronis and Park (2002) discuss avoidable and unavoidable exergy destruction in components. In the exergetic analysis, the avoidable exergy destruction gives a realistic picture of the potential for improving the thermodynamic effectiveness of each component. They state that the calculation of avoidable exergy destruction and avoidable investment costs is associated with arbitrary decisions that reflect the maximal and minimal efficiency that can be achieved for the component being considered in today's technological and economic environment. In their opinion, this arbitrariness must be accepted, in order for engineers to improve their understanding of the potential for improvements. They also state that decisions required to calculate the unavoidable exergy destruction and costs, if made prudently, should not significantly affect the conclusions to be drawn from their analysis.

The work presented in this paper is based on exergy efficiency definitions presented by Woudstra (2002), who distinguishes two different types of exergy efficiency definitions: universal exergy efficiency and functional exergy efficiency. The universal exergy efficiency has been criticised in the literature (Woudstra, 2002) as not being sufficiently sensitive to changes in a system. This can occur when the exergy loss within the system is small compared to the exergy of the incoming flows. In this case, the outgoing flows still have relatively high exergy content, because most of the exergy has been fed to the system without being used by it. Also the universal exergy efficiency makes no

distinction regarding usability (exergy transferred inside the heat exchanger or discarded with the outgoing flows), or intended use in heating or cooling (Boelman and Sakulpipatsin, 2005). Functional exergy efficiency is more sensitive to changes in exergy loss within the system, and yields ‘net’ efficiency values since it excludes the exergy discarded with outgoing flows. In this study, the functional exergy efficiency is used as a measure for the results of the exergy analysis of a heat exchanger.

Because the focus of this paper is on operating temperatures, the heat exchanger itself is treated as a ‘black box’ model in this study. The effect of intrinsic parameters such as flow patterns and heat transfer rates (Bart, 2002) are assumed to be lumped into the heat exchanger heat transfer effectiveness ε , which is taken as an operating parameter. The exchanger heat transfer effectiveness is defined as the ratio of the actual heat transfer from either stream to the maximum possible heat transfer in the heat exchanger (ASHRAE, 2000; Holman, 2002; Wu et al., 2006). One advantage of the lumped parameter ε is that it allows heat exchanger outlet temperatures to be estimated as functions of inlet temperatures. In HVAC design practice it is customary to use empirical values of ε from heat exchanger manufacturers.

1.3. Contribution of this paper to the state of art

In the literature, links have been made between sustainability, exergy consumption and heat transfer at near-environmental temperatures. Examples include warm and cool exergy (Shukuya, 1996) and tepidology (Wall, 1990). Links have also been made between exergy resource efficiency and sustainability (de Swaan Arons et. al., 2004; Connely and Koshland, 2001). To the best of the authors’ knowledge, there have been no attempts so far to link exergy efficiency, exergy consumption and heat transfer at near-environmental temperatures.

The industrial and societal relevance is related to the possibility of improving the efficiency of thermal exergy transfer at near-environmental temperatures. By combining the notions of heat exchanger heat transfer effectiveness, functional exergy efficiency and exergy consumption with the concept of warm and cool exergy, we obtain new insights into exergy gains and losses involved in air-to-air sensible heat exchange at near-environmental temperatures. Possible applications include balanced ventilation in buildings and industrial processes at near-environmental temperatures, such as food manufacturing. (de Swaan Arons et. al., 2004)

2. Analysis

2.1. Key concepts

This item presents some key concepts used in this paper. First, exergy is defined for the purposes of this work, essentially in relation to a pre-defined reference environment, and the concept of warm and cool exergy is presented. Second, the simplified model of sensible heat exchangers, the main relevant parameters and the working definition of heat transfer effectiveness are introduced. After that, an example of thermal exergy profiles and entropy generation in a counter-flow heat exchanger is shown. At last, exergy efficiencies relevant to this work are defined.

2.1.1. Exergy and environment

Exergy is always evaluated with respect to a reference environment. The intensive properties of the reference environment determine the exergy of a system. The exergy of a system becomes zero when the system is in equilibrium with the reference environment. Many researchers have examined characteristics of the reference environment (Gaggioli and Petit, 1977; Wepfer et. al., 1979; Sussman, 1981; Ahrendts, 1980; Rodriguez, 1980; Rosen, 1986; Sakulpipatsin et. al., 2007b). The reference environment acts as an infinite system, and is a sink and source for thermal energy and substances. Only internally reversible processes in which the intensive state of the system remains unaltered (i.e. its temperature, pressure and chemical composition remain constant) are assumed to take place in, and in interaction with this reference environment.

In the built environment, physical and chemical properties of exergy carriers are needed for selecting a right reference environment. Properties of environmental air (temperature, pressure and chemical compositions) are suitable for determination of a reference environment of the built environment, because properties of (indoor) air play a vital role in indicating indoor thermal comfort, health and energy use. In addition, the resulting exergy value is influenced by the season and the type of climate. In a cold climate the chemical contribution to the exergy value is relatively small when compared to the thermal contribution. However it is rather large in a hot and humid climate (Sakulpipatsin et. al., 2007b).

The exergy value of air in a building can be expressed as a function of temperature, pressure, and chemical composition. In this paper it is assumed that there is no difference (or ignorable) between air pressure inside and outside the building and that the pressure is constant and equal to 101.325 kPa. This means that the mechanical contribution to the exergy value is zero. Furthermore only dry air of standard chemical composition is considered and this leads to a chemical exergy contribution of zero as well.

To calculate the thermo-mechanical part of the exergy of a substance in a steady state process, the kinetic and potential exergy can be neglected or do not play a role at all; equation 1 can be used (Moran and Shapiro, 1998).

$$Ex_{th} = H - H_0 - T_0 (S - S_0) \quad (1)$$

where H is the enthalpy and S is the entropy of the substance both at a temperature T . H_0 and S_0 are the enthalpy and the entropy of the substance at the standard state at temperature T_0 .

For the purposes of this work, the thermal exergy per second $\dot{Ex}_{th}$ of an ideal gas stream is calculated as a function of mass flow rate $\dot{m}$, isobaric heat capacity c_p , temperature T , and standard temperature as the temperature of the environment T_e , by using equation 2 (Moran and Shapiro, 1998; Sakulpipatsin et al, 2007b).

$$\dot{Ex}_{th} = \dot{m} < c_p > \left((T - T_e) - T_e \ln \left(\frac{T}{T_e} \right) \right) \quad (2)$$

In equation 2, c_p is an appropriate mean heat capacity to be used over the temperature interval $T_e - T$.

2.1.2. Warm and cool exergy

The name ‘tepidology’, meaning the study of moderately warm systems, was suggested in 1978 for a theory to describe systems close to ambient conditions (Alfvén, quoted by Wall, 1990). Besides space heating (Asada and Boelman, 2004), other processes also operate close to ambient conditions, e.g. cooling cycles driven by low-grade heat (Boelman et. al., 1994, 1995). Food industries also use large amounts of energy for heating and cooling. Wall (1990) noted that special relations are valid for systems close to equilibrium (e.g. linear relations may often be assumed), that this implies a special treatment, and that tepidology may be an important field of further research. In the same paper he discussed the exergy of heat and cold, and showed examples of the extent whereby the exergy of systems at constant temperatures (+20°C and –20°C) changes as a result of ambient air temperature variations throughout the year.

Shukuya (1996) named “warm exergy” the exergy contained by air at a temperature higher than its environment, and “cool exergy” the exergy contained by air at a temperature lower than its environment. Warm exergy and cool exergy are quantities dependent on the thermal state of a substance with respect to the reference environment, and can be calculated according to equation 2. Supposing that air in a room in a building is defined as the system to be studied, when the temperature of air in the room is higher than the temperature of air in the environment, air in the room has warm exergy; when the temperature of air in the room is lower than that of air in the environment, room air has cool exergy. This implies that the direction of the exergy flow is always from the inside to the outside of a building, although the direction of the energy flow changes depending on whether the indoor air temperature is higher or lower than the temperature of air in the environment (Shukuya 1996).

The exergy value of air in the building is zero if the temperature of air in the room T_i is equal to the temperature of environmental air T_e . At room temperatures that are smaller (cool exergy) and larger (warm exergy) than the temperature of environmental air, the air in the room has a positive exergy value. The indoor air has a certain amount of exergy both when its temperature is higher than the environment (warm exergy) and when its temperature is lower than the environment (cool exergy) (Shukuya 1996).

2.1.3. Simplified sensible heat exchangers

For the purpose of this study, the system to be studied consists of two air streams where thermal energy is transferred from the hot air stream to the cold air stream in a counter-flow sensible heat exchanger. The system will be defined more extensively. The heat exchanger model is built as a function of heat transfer fluid (air) temperature levels, $T_{c,in}$, $T_{c,out}$, $T_{h,in}$ and $T_{h,out}$, environmental air temperature T_e , total heat capacities of both air streams C and exchanger heat transfer effectiveness ε . In order to concentrate only on thermal exergy losses and to allow the analysis to be based on as few parameters as possible, the effect of exergy loss by dissipation of mechanical energy in the heat exchanger (Bart, 2002) is neglected. In practical applications (Sakulpipatsin et. al., 2007a) this effect needs to be considered, since it affects fan power requirements.

Figure 1 schematically illustrates airflow and air temperature profiles in the simplified counter-flow sensible heat exchanger model used in this paper, if the total heat capacities of both airflows are considered to be the same ($C_h=C_c$). Tables 1 and 2 list the main relevant parameters and boundary conditions.

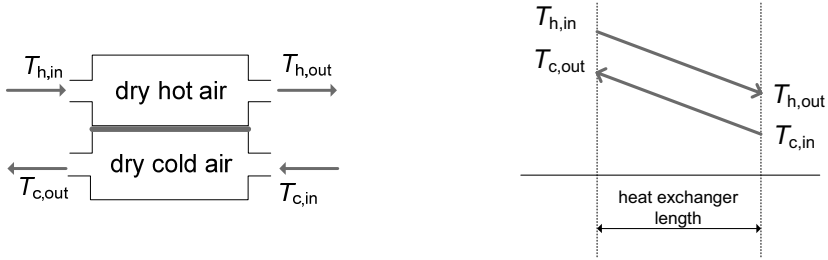


Figure 1 Schematic representation of a counter-flow heat exchanger (left) and air temperature profiles (right), assuming the total heat capacities of the airflows are the same

In order to focus on the objective of the paper, a simplified, black box model of a counter-flow heat exchanger is used. The concept of exchanger heat transfer effectiveness ε (ASHRAE, 2000; Holman, 2002; Wu et al., 2006) is used as a lumped parameter to characterise heat exchanger thermal performance. Airflow and other effects are neglected, as outlined in Table 1. More detailed studies of heat transfer and fluid flow phenomena in heat exchangers are beyond the scope of this work and can be found in the literature, e.g. Bart (2002); Heun and Crawford (1994); Vaidya et. al. (1992).

Table 1. Simplifying assumptions for the heat exchange model

<i>Simplifying assumptions</i>	
• counter-flow heat exchange	• well insulated
• only sensible heat exchange considered	• air flow effects neglected
• heat exchange between 2 dry airflows	• only thermal exergy considered
• temperatures and heat transfer coefficients constant and uniform	
• constant temperature difference $T_h - T_c$ throughout the heat exchanger	
• same mass flows for hot and cold air: 1 kgs^{-1}	
• isobaric heat capacity of dry air constant $c_{p,air}: 1.005 \text{ kJkg}^{-1}\text{K}^{-1}$	
• exchanger heat transfer effectiveness ε is a parameter, at 70%, 90% and 100%	

Environmental temperature domains are defined based on typical ranges applicable to HVAC systems in heating applications, and are given in Table 2. Heat exchanger operating temperatures are taken over a somewhat broader range than in usual HVAC applications, in order to enable the temperature combinations required to obtain dimensionless temperatures T' covering the relatively broad range of $0 \leq T' \leq 5$. The dimensionless temperature T' is defined as in equation 21 and explained in item 2.2.1.

Table 2 Temperature domains

<i>Temperature domains</i>	
• $253.15 \text{ K} \leq T_c \leq 293.15 \text{ K}$	• $293.15 \text{ K} \leq T_{h,in} \leq 373.15 \text{ K}$
• $10 \text{ K} \leq T_{h,in} - T_{c,in} \leq 130 \text{ K}$	• $283.15 \text{ K} \leq T_{c,in} \leq 363.15 \text{ K}$
• $0 \leq T' \leq 5$	

The equivalent temperatures (Çengel and Robert, 2001) of the cold air $T_{eq,c}$ and the hot air $T_{eq,h}$ are calculated from the air temperatures at the heat exchanger inlets T_{in} and outlets T_{out} , by using equation 3.

$$T_{eq} = \frac{T_{out} - T_{in}}{\ln\left(\frac{T_{out}}{T_{in}}\right)} \quad (3)$$

2.1.4. Heat transfer effectiveness

The heat transfer effectiveness ε of the heat exchanger is often used by HVAC designers as a lumped parameter to characterise the thermal performance of a heat exchanger. It can be simply defined in terms of the inlet and outlet air temperatures and the total heat capacities of the streams (ASHRAE, 2000), as shown in equation 4.

$$\varepsilon = \frac{Q}{Q_{max}} = \frac{C_c (T_{c,out} - T_{c,in})}{C_{min} (T_{h,in} - T_{c,in})} = \frac{C_h (T_{h,out} - T_{h,in})}{C_{min} (T_{h,in} - T_{c,in})} \quad (4)$$

The exchanger heat transfer effectiveness ε is different from the energy efficiency of a heat exchanger (defined as the ratio between the total energy output and the total energy input). The exchanger heat transfer effectiveness ε accounts only for thermal terms of energy, while the energy efficiency accounts for all terms of energy (e.g. thermal, chemical and mechanical terms). Therefore both measures cannot be compared in general.

In the simplified model and operating conditions for this paper, mass flow rates, temperatures and heat transfer coefficients are assumed uniform and constant through the heat exchange process, and the total heat capacities of both air streams are the same $C_h = C_c$. The effect of airflow on heat transfer performance is thus neglected. The heat exchanger to be studied is used for heating the cold air from $T_{c,in}$ to $T_{c,out}$.

Given these assumptions, the exchanger heat transfer effectiveness ε is taken as a ratio of the actual temperature rise ($T_{c,out} - T_{c,in}$) to the maximum possible temperature rise ($T_{h,in} - T_{c,in}$), as shown in equation 5.

$$\varepsilon = \frac{C_{min}}{C_c} = \frac{T_{c,out} - T_{c,in}}{T_{h,in} - T_{c,in}} \quad (5)$$

By assuming that the mass flows of the hot air and the cold air are the same at 1 kg s^{-1} (Table 1), the total heat capacities of the air flows are also the same: $C_{min} = C_h = C_c$. From the above, air temperatures at the heat exchanger outlets $T_{c,out}$ and $T_{h,out}$ are expressed in equation 6 and equation 7.

$$T_{c,out} = T_{c,in} + \varepsilon (T_{h,in} - T_{c,in}) \quad (6)$$

$$T_{h,out} = T_{h,in} - \varepsilon (T_{h,in} - T_{c,in}) \quad (7)$$

2.1.5. Thermal exergy loss and entropy generation

Figure 2 schematically shows an example of thermal exergy profiles in a counter-flow heat exchanger for a given environmental air temperature T_e , with operating temperatures of hot air T_h and cold air T_c all above the environmental air temperature T_e .

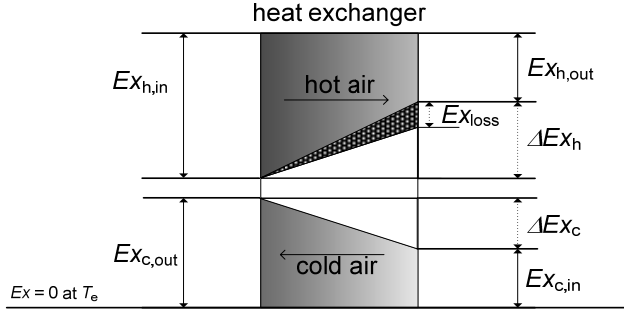


Figure 2 Simplified diagram of the thermal exergy profiles in a counter-flow sensible heat exchanger (environmental air temperature T_e , and operating temperatures (T_h and T_c) above T_e)

The exergy values are calculated according to equation 2. Since the hot air transfers thermal energy to the cold air in the heat exchange process, the temperature of the hot air decreases and of the cold air increases. This leads to an increase of the thermal exergy of the cold air by ΔEx_c while the thermal exergy of the hot air decreases by ΔEx_h . In this case exergy is transferred from the hot air stream to the cold air stream.

Such a real process always takes place with generation of entropy as an implication of the second law of thermodynamics. According to the Gouy-Stodola relation, as shown in equation 8, the ultimate corresponding exergy loss is given by the product of the temperature of the environment and the entropy generation of the system considered. The Gouy-Stodola relation (de Swaan Arons et. al., 2004) expresses that the amount of exergy loss per second is proportional to the amount of entropy generated per second, and that the proportionality constant is T_0 , the temperature of the reference environment.

$$\dot{Ex}_{\text{loss}} = T_0 \dot{S}_{\text{gen}} \quad (8)$$

Entropy generation occurs as a consequence of the finite temperature difference between the hot air stream and the cold air stream needed to realize the transfer of thermal energy from the hot air stream to the cold air stream. This loss is due to the temperature difference between the hot air stream and cold air stream at each point in the heat exchanger, operated in a steady state, $T_h - T_c$. (de Swaan Arons et. al., 2004)

Consider that, at a certain point in the hot air stream, the temperature is T_h and that a small amount of thermal energy per second $\delta \dot{Q}$ is transferred from this point to the nearby cold air stream at temperature T_c . As a consequence of this process a small amount of entropy $d\dot{S}_{\text{gen}}$ is generated, as shown in equation 9. (de Swaan Arons et. al., 2004)

$$d\dot{S}_{\text{gen}} = \delta\dot{Q} \left(\frac{1}{T_c} - \frac{1}{T_h} \right) > 0 \quad (9)$$

The exergy loss per second in the process of the transfer of a small amount of thermal energy per second can be found by multiplying equation 9 with T_0 . The result is given in equation 10.

$$\delta\dot{E}x_{\text{loss}} = T_0 \delta\dot{Q} \left(\frac{1}{T_c} - \frac{1}{T_h} \right) \quad (10)$$

The amount of thermal energy transferred per second is shown in equation 11.

$$\delta\dot{Q} = U dA (T_h - T_c) \quad (11)$$

In this expression U is the overall heat transfer coefficient, dA is the small area through which the thermal energy is transferred from the hot air stream to the cold air stream, and $T_h - T_c$ is the temperature difference at the point considered in the heat exchanger, often called the driving force for heat exchange. Including this expression in the expression for the amount of exergy loss in equation 10 gives the expression in equation 12 (Wall, 1990).

$$\delta\dot{E}x_{\text{loss}} = T_0 U dA \frac{(T_h - T_c)^2}{T_h T_c} \quad (12)$$

From this expression it is clear that the exergy loss depends quadratically on the driving force for heat exchange and that the smaller both T_h and T_c are the larger the exergy loss is. For a given exergy loss, if $T_h - T_c$ is decreased then dA must be increased in such a way that the same amount of thermal energy $\delta\dot{Q}$ can still be transferred. In the limiting case $T_h = T_c$ there is no exergy loss but no thermal energy transfer either. In order to minimise exergy losses, there will be a thermo-economic optimum for which the total heat transfer area of the heat exchanger and a mean temperature difference are determined. In general, counter-flow heat exchange entails smaller and more constant temperature differences $T_h - T_c$ than parallel-flow heat exchange. Hence, counter-flow heat exchange will give the possibility to realize smaller exergy losses, especially when the total heat capacities of both air streams are the same. This is the reason why only counter-flow heat exchange will be considered in this paper, in particular the case of the total heat capacities of both streams are the same.

2.1.6. Universal exergy efficiency

The universal exergy efficiency is defined by Woudstra (2002) as a ratio between the gross exergy output and the gross exergy input, as in equation 13.

$$\eta_{u,l} = \frac{Ex_{c,\text{out}} + Ex_{h,\text{out}}}{Ex_{c,\text{in}} + Ex_{h,\text{in}}} \quad (13)$$

This definition can be useful when sources and products cannot be clearly defined, e.g. in the case of complex systems consisting of several processes.

Equation 13 can be further reformulated as equation 14, by substituting equation 2 and the total heat capacities of the cold air stream and the hot air stream.

$$\eta_{u,I} = \frac{C_c \left(T_{c,out} - T_e - T_e \ln \left(\frac{T_{c,out}}{T_e} \right) \right) + C_h \left(T_{h,out} - T_e - T_e \ln \left(\frac{T_{h,out}}{T_e} \right) \right)}{C_c \left(T_{c,in} - T_e - T_e \ln \left(\frac{T_{c,in}}{T_e} \right) \right) + C_h \left(T_{h,in} - T_e - T_e \ln \left(\frac{T_{h,in}}{T_e} \right) \right)} \quad (14)$$

Temperature sensitivity of the universal exergy efficiency $\eta_{u,I}$ is discussed in (Boelman and Sakulpipatsin, 2005). The results show that, in the reversible limiting case (which can be approximated by $\varepsilon=100\%$, see item 2.1.4), the temperature difference between both air streams at every point in the heat exchanger can decrease to zero and then the universal exergy efficiency $\eta_{u,I}$ reaches one. This is because the temperature of the cold air stream increases to $T_{h,in}$ and the temperature of the hot air stream decreases to $T_{c,in}$.

Alternatively, the universal exergy efficiency $\eta_{u,I}$ can be adapted. If the cooled hot stream at $T_{h,out}$ is not used further, then the heated cold stream at $T_{c,out}$ is the only used exergy output. The adapted exergy efficiency $\eta_{u,II}$ is expressed in equation 15.

$$\eta_{u,II} = \frac{Ex_{c,out}}{Ex_{c,in} + Ex_{h,in}} \quad (15)$$

In the reversible limiting case (which can be approximated by $\varepsilon=100\%$, see item 2.1.4), when the total heat capacities of both streams are equal, $\eta_{u,II}$ is expressed in equation 16.

The second universal exergy efficiency $\eta_{u,II}$ is more sensitive to changes of $T_{c,in}$, $T_{h,in}$ and T_e than $\eta_{u,I}$, in the reversible limit case (which can be approximated by $\varepsilon=100\%$, see item 2.1.4), because $\eta_{u,II}$ accounts only for $Ex_{c,out}$ as its product output. This definition could be useful when the value of $Ex_{h,out}$ is unknown, e.g. because this unused stream is being discarded into the environment without being used.

$$\eta_{u,II} = \frac{1}{1 + \frac{T_{c,in} - T_e - T_e \ln \left(\frac{T_{c,in}}{T_e} \right)}{T_{h,in} - T_e - T_e \ln \left(\frac{T_{h,in}}{T_e} \right)}} \quad (16)$$

2.1.7. Functional exergy efficiency

The functional exergy efficiency η_f of the heat exchanger can be defined as a ratio of all product outputs $\sum Ex_{product}$ to all source inputs $\sum Ex_{source}$, as shown in equation 17 (Woudstra, 2002).

$$\eta_f = \frac{\Sigma Ex_{\text{product}}}{\Sigma Ex_{\text{source}}} \quad (17)$$

When the goal is to increase thermal exergy of the cold air stream by means of exergy transfer from the hot air stream, thermal exergy increase of the cold air ΔEx_c is considered as the net product output, and the absolute value of the thermal exergy decrease of the hot air $|\Delta Ex_h|$ is considered as the net source input. In this case equation 17 can be rearranged to equation 18.

$$\eta_f = \frac{\Delta Ex_c}{|\Delta Ex_h|} = \frac{Ex_{c,\text{out}} - Ex_{c,\text{in}}}{Ex_{h,\text{in}} - Ex_{h,\text{out}}} \quad (18)$$

By assuming that the heat exchanger is well insulated and the airflow effects are neglected (Table 1), thermal energy is assumed to be completely transferred from the hot air to the cold air ($Q_h = Q_c$). Equation 18 can be rewritten as a function of the temperatures ($T_{c,\text{in}}$, $T_{h,\text{in}}$, $T_{c,\text{out}}$, and T_e) and the total heat capacities of the air streams (C_h and C_c), as shown in equation 19.

$$\eta_f = \frac{C_c (T_{c,\text{out}} - T_{c,\text{in}}) - C_c T_e \ln \left(\frac{T_{c,\text{out}}}{T_{c,\text{in}}} \right)}{C_c (T_{c,\text{out}} - T_{c,\text{in}}) + C_h T_e \ln \left(1 - \frac{C_c (T_{c,\text{out}} - T_{c,\text{in}})}{C_h T_{h,\text{in}}} \right)} \quad (19)$$

In the reversible limiting case (which can be approximated by $\varepsilon=100\%$, see item 2.1.4), and when the total heat capacities are the same, the functional exergy efficiency η_f reaches one, since $T_{h,\text{out}}$ approaches $T_{c,\text{in}}$ and $T_{c,\text{out}}$ approaches $T_{h,\text{in}}$.

When the heat exchange is for cooling purposes, in the normal case where all temperatures of the air streams in the heat exchanger are below the temperature of the environment, thermal exergy decrease of the cold air stream is considered as the net source input and thermal exergy increase of the hot air stream is considered as net product output. The thermal exergies of the hot and cold air streams is considered as cool exergy, for defining functional exergy efficiency for the cooling purposes $\eta_{f,\text{cooling}}$ (equation 20).

$$\eta_{f,\text{cooling}} = \frac{\Delta Ex_h}{|\Delta Ex_c|} = \frac{Ex_{h,\text{out}} - Ex_{h,\text{in}}}{Ex_{c,\text{in}} - Ex_{c,\text{out}}} \quad (20)$$

2.2. Exergy analysis

The analysis aims at providing a quantitative indication of how changes in temperature levels affect the efficiency of thermal exergy transfer occurring at near-environmental temperatures. To this end, a simplified counter-flow heat exchanger model is used, considering heating purposes. First a broad analysis is made, focusing mainly on the

effect of temperature combinations on exergy efficiency. Then an application example is presented, for air-to-air sensible heat exchange in balanced ventilation in buildings.

In the first part of this chapter, exergy analysis of the heat exchanger model is performed in relation to exchanger heat transfer effectiveness and temperature variations of heat exchanger and environment. Functional exergy efficiency is calculated and a dimensionless temperature is defined.

As outlined in Table 1 and Table 2, the following parameters are allowed to vary: environmental air temperature T_e ; hot and cold air temperature at the heat exchanger inlets $T_{c,in}$ and $T_{h,in}$; and exchanger heat transfer effectiveness ε . A number of heat exchanger inlet air temperature combinations are defined, within the ranges given in Table 2. Air temperatures at the heat exchanger outlets are then determined, as simple functions of inlet air temperatures and exchanger heat transfer effectiveness. Cold air temperatures at the heat exchanger outlet $T_{c,out}$ are calculated by using equation 6. The total heat capacities of both air streams are the same $C_h=C_c$. $T_{c,in}$, $T_{h,in}$ and ε are varied within the ranges indicated in Table 1 and Table 2. Hot air temperatures at the heat exchanger outlet $T_{h,out}$ are calculated by assuming that $Q_h = Q_c$, because of the assumption that the heat exchanger is well insulated.

According to the assumptions in Table 1, there is no difference in the chemical contribution to the exergy values of the cold air at the heat exchanger inlet $Ex_{ch,c,in}$ and outlet $Ex_{ch,c,out}$, because the air is assumed to be dry and to have a constant chemical composition. Also there is no difference in the mechanical contribution to the exergy values of the cold air at the inlet and the outlet ($Ex_{me,c,in}$ and $Ex_{me,c,out}$), because pressure drops in the heat exchange process is neglected. Therefore the exergy values of the cold air at the inlet and outlet points are equal to their thermal contributions. This happens with the hot air as well. In this study, the exergy analysis thus considers only the thermal contribution to the exergy values.

The dimensionless temperature T' is introduced in item 2.2.1 for analysis of the exergy results. Item 2.2.2 briefly discusses exchanger heat transfer effectiveness ε in the context of this work, and shows how functional exergy efficiency η_f can be expressed in relation to ε .

2.2.1. Dimensionless temperature

A dimensionless temperature T' is defined (Boelman and Sakulpipatsin, 2004, 2005, Sakulpipatsin et. al., 2007c), in order to analyse functional exergy efficiency calculation results as a function of environmental air temperature T_e in relation to inlet air temperatures ($T_{c,in}$ and $T_{h,in}$). The dimensionless temperature expresses a temperature difference between the hot inlet air $T_{h,in}$ and the environmental air T_e , relative to the inlet air temperature difference ($T_{h,in}-T_{c,in}$). T' is defined for the heat exchanger model in heating mode in equation 21.

$$T' = \frac{T_{h,in} - T_e}{T_{h,in} - T_{c,in}} \quad (21)$$

Similarly, for the heat exchanger model in cooling mode, a dimensionless temperature T'' can be defined to express a temperature difference between the cold inlet air $T_{c,in}$ and

the environmental air T_e , relative to the inlet air temperature difference ($T_{h,in} - T_{c,in}$), as in equation 22.

$$T'' = \frac{T_e - T_{c,in}}{T_{h,in} - T_{c,in}} \quad (22)$$

Figure 3 illustrates how the dimensionless temperature T' can express different temperature combinations of $T_{c,in}$ and $T_{h,in}$, for a given environmental air temperature $T_e = 10^\circ\text{C}$.

For $T'=1$, $T_{c,in}$ is equal to T_e . In practice, this could correspond e.g. to a heat exchanger taking up environmental air at the cold heat exchanger inlet in order to (pre) heat it for use in balanced ventilation systems. For $T'>1$, $T_{c,in}$ is above T_e . This could be the case for ventilation air being pre-heated (e.g. by a sun room or in buried air ducts) above T_e before reaching the heat exchanger inlet.

For $0 < T' < 1$, T_e is between $T_{c,in}$ and $T_{h,in}$. This could be the case of heat exchange between air from a freezer and from an office, but is unlikely to occur in space heating applications. For $T' < 0$, T_e is above $T_{c,in}$ and $T_{h,in}$. This could be the case when heat exchange takes place between two cold air streams. This is also unlikely to be the case in space heating applications.

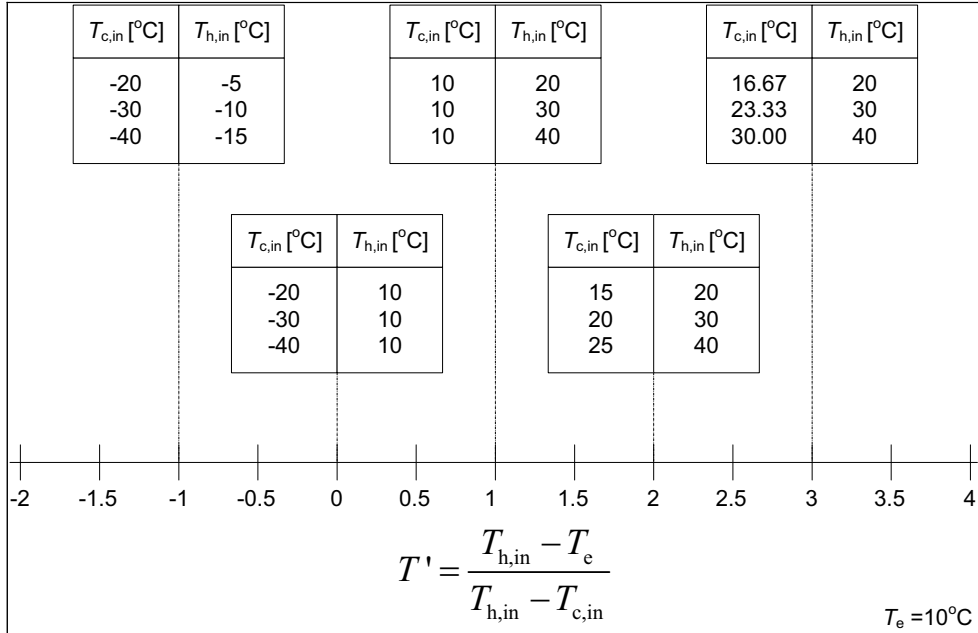


Figure 3 Examples of dimensionless temperatures in relation to $T_{c,in}$ and $T_{h,in}$, for $T_e = 10^\circ\text{C}$

Figure 4 schematically illustrates the relationship between inlet air temperature profiles of the heat exchanger model $T_{h,in}$ and $T_{c,in}$, environmental air temperatures T_e , and dimensionless temperatures T' , where $0 < T' < 1$. The functional exergy efficiency values η_f are taken from item 3.

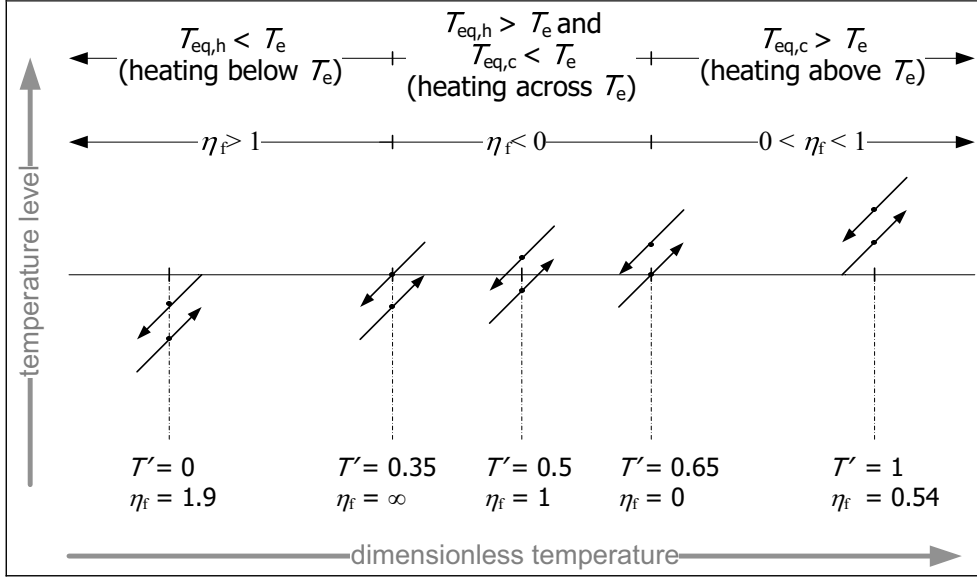


Figure 4 Comparison of temperature profiles in the heat exchanger at the same T_e

2.2.2. Functional exergy efficiency as a function of temperatures and heat transfer effectiveness

This item presents a simplified expression for calculating functional exergy efficiency, used to obtain the results discussed in item 3.2.

For well-insulated heat exchangers in balanced ventilation systems involving sensible heat exchange only, it can be reasonably assumed that $C_{\min} = C_h = C_c$ and $\dot{Q}_h = \dot{Q}_c$, and that changes in heat transfer coefficient (e.g. due to variations in temperature and mass flow) are negligible. The concept of exchanger heat transfer effectiveness ε (ASHRAE, 2000; Holman, 2002; Wu et al., 2006) can then be used to directly calculate the functional exergy efficiency η_f , without the need to first determine outlet temperatures. This can be done by substituting equation 6 and equation 7 into equation 19.

$$\eta_f = \frac{\varepsilon(T_{h,in} - T_{c,in}) - T_e \ln\left(\frac{\varepsilon(T_{h,in} - T_{c,in})}{T_{c,in}} + 1\right)}{\varepsilon(T_{h,in} - T_{c,in}) + T_e \ln\left(1 - \frac{\varepsilon(T_{h,in} - T_{c,in})}{T_{h,in}}\right)} \quad (23)$$

3. Results and discussion

This chapter presents and discusses the sensitivity of functional exergy efficiency η_f of the heat exchanger model to dimensionless temperature T' and exchanger heat transfer effectiveness ε .

Item 3.1 presents and discusses examples of η_f results for a relatively broad range of T' , and identifies ranges of T' that are relevant to heating applications. Item 3.2 analyses the sensitivity of η_f to temperatures and ε , for T' ranges identified as relevant to heating applications. Based on the insights gained from this analysis, item 3.2 also discusses how operating temperatures can be selected for improving the functional exergy efficiency of the heat exchanger model.

The exergy values of dry air at the heat exchanger inlets and outlets ($Ex_{h,in}$, $Ex_{h,out}$, $Ex_{c,in}$ and $Ex_{c,out}$) are calculated by using equation 2. Exergy differences (ΔEx_h and ΔEx_c) correspond to the differences between air exergy at the inlets and outlets. Functional exergy efficiency values η_f of the heat exchanger are calculated using equation 23. The dimensionless temperature T' (equation 21) is used to identify and analyse patterns in the sensitivity of functional exergy efficiency to temperature variations.

3.1. Example of dimensionless temperature T' ranges applicable to heating

Table 3 shows an example of exergy and functional exergy efficiency calculation results for the heat exchanger model. The calculations aim at showing how functional exergy efficiency values are influenced by variations in dimensionless temperature T' over a relatively broad range of T' .

Table 3 Exergy and functional exergy efficiency calculation results for $T_e = 10^\circ\text{C}$

T'	6.00	2.00	1.00	0.80	0.70	0.65	0.60	0.50	0.35	0.35	0.30	0.00	-4.00
η_f	0.960	0.830	0.550	0.340	0.140	0.000	-0.210	-1.000	-604	1869	6.860	1.880	1.090
range of η_f	$\rightarrow 1$	< 1				0	< 0		$\rightarrow -\infty$	$\rightarrow +\infty$	> 1		$\rightarrow 1$
remarks	$\Delta Ex_c > 0, T_{eq,c} > T_e$					$\Delta Ex_c = 0$ $T_{eq,c} = T_e$	$\Delta Ex_c < 0, T_{eq,c} < T_e$						
	$\Delta Ex_h > 0, T_{eq,h} > T_e$								$\Delta Ex_h \rightarrow 0^+$ $T_{eq,h} \rightarrow T_e^+$	$\Delta Ex_h \rightarrow 0^-$ $T_{eq,h} \rightarrow T_e^-$	$\Delta Ex_h < 0$ $T_{eq,h} < T_e$		
	$ \Delta Ex_c < \Delta Ex_h $						$ \Delta Ex_c = \Delta Ex_h $		$ \Delta Ex_c > \Delta Ex_h $				
	heating above T_e (relevant range for heating)					heating across T_e (inefficient range, not recommended)					heating below T_e (non-applicable range)		
$T_{h,in}$	100.00	40.00	25.00	22.00	20.50	19.78	19.00	17.55	15.29	15.28	14.50	10.00	-50.00
$T_{h,out}$	89.50	29.50	14.50	11.50	10.00	9.28	8.50	7.05	4.79	4.78	4.00	-0.50	-60.50
$T_{eq,h}$	94.73	34.72	19.72	16.72	15.22	14.50	13.72	12.27	10.01	10.00	9.22	4.72	-55.29
$T_{c,in}$	85.00	25.00	10.00	7.00	5.50	4.78	4.00	2.55	0.29	0.28	-0.50	-5.00	-65.00
$T_{c,out}$	95.50	35.50	20.50	17.50	16.00	15.28	14.50	13.05	10.79	10.78	10.00	5.50	-54.50
$T_{eq,c}$	90.22	30.22	15.22	12.22	10.72	10.00	9.22	7.77	5.51	5.50	4.72	0.22	-59.79
$Ex_{h,in}$	11.909	1.493	0.386	0.249	0.191	0.166	0.141	0.099	0.049	0.049	0.036	0.000	7.464
$Ex_{h,out}$	9.479	0.645	0.036	0.004	0.000	0.001	0.004	0.016	0.049	0.049	0.065	0.201	10.627
ΔEx_h	2.430	0.847	0.350	0.245	0.191	0.165	0.137	0.084	0.000	0.000	-0.029	-0.201	-3.163
$Ex_{c,in}$	8.509	0.386	0.000	0.016	0.036	0.049	0.065	0.100	0.171	0.172	0.201	0.414	12.191
$Ex_{c,out}$	10.839	1.089	0.191	0.098	0.063	0.049	0.036	0.016	0.001	0.001	0.000	0.036	8.739
ΔEx_c	2.330	0.703	0.191	0.082	0.027	0.000	-0.029	-0.084	-0.170	-0.171	-0.201	-0.378	-3.452

Notes: 1. The unit of the temperatures (T_e , $T_{h,in}$, $T_{h,out}$, $T_{eq,h}$, $T_{c,in}$, $T_{c,out}$ and $T_{eq,c}$) is degree Celsius ($^\circ\text{C}$).
2. The unit of exergy values ($Ex_{h,in}$, $Ex_{h,out}$, ΔEx_h , $Ex_{c,in}$, $Ex_{c,out}$ and ΔEx_c) is kilojoules (kJ).
3. Dimensionless temperature (T') and functional exergy efficiency (η_f) are dimensionless.

The exergy calculations are made with the following assumptions. Environmental air temperature T_e is constant at 10°C. Air temperature difference at the heat exchanger inlets ($T_{h,in} - T_{c,in}$) is constant at 15°C. $T_{h,in}$ is between -50°C and 100°C. $T_{c,in}$ is between -65°C and 85°C. The numbers highlighted in bold indicate the cases where one of the inlet air temperatures and the equivalent air temperature (equation 3) are equal to T_e .

Exchanger heat transfer effectiveness ε is assumed to be constant at 70%. The simplified model neglects any variations in overall heat transfer coefficient that could result from varying airflow and temperature regimes. Although ε is assumed to be constant, the functional exergy efficiency η_f varies depending on operating temperatures and environmental temperature.

The example in Table 3 shows negative functional exergy efficiencies η_f for T' between 0.35 and 0.60. This is because the cold air is heated below environmental temperature and loses cool exergy without gaining warm exergy. For $T' = 0.35$, $\eta_f \rightarrow \infty$ because $T_{eq,h} \approx T_e$, which results in ΔEx_h becoming infinitesimally small compared to ΔEx_c . For $T' < 0.30$, η_f is positive because both exergy changes of the hot air stream and the cold air stream (ΔEx_h and ΔEx_c) are negative. Values of η_f are larger than one because the exergy change on the cold side is bigger than the exergy change on the hot side of the heat exchanger: $|\Delta Ex_c| > |\Delta Ex_h|$.

In the example shown in Table 3, the definition of η_f from equation 23 is only applicable to $T' \geq 0.65$, where $0 \leq \eta_f \leq 1$. In the range of $T' > 6$ the sensitivity of η_f to T' becomes rather low because the heat exchanger operates relatively far from environmental temperature ($T_{h,in} - T_e \gg T_{h,in} - T_{c,in}$). In the range of $0.65 \leq T' \leq 1$, η_f is relatively low ($0 \leq \eta_f \leq 0.55$), because $T_{eq,c}$ is relatively close to T_e , and hence ΔEx_c (or $Ex_{product}$) is rather low. In the range of T' between 1 and 6, η_f is between 0.55 and 0.96. This range is recommended for heat exchanger operation in heating mode. Item 3.2 discusses in more detail how operating temperatures can be selected for heating applications, e.g. in buildings, within the range of T' between 1 and 5.

3.2. Sensitivity of functional exergy efficiency η_f to temperatures and exchanger heat transfer effectiveness ε

This item analyses the sensitivity of η_f to temperatures and ε , with a view to practical application examples. Based on the insights gained from the analysis, it discusses how operating temperatures can be selected for improving the functional exergy efficiency of the heat exchanger model, considering the effect of exchanger heat transfer effectiveness ε .

The discussion is based on examples of functional exergy efficiency η_f dependence on T_e and $T_{h,in}$. The temperature ranges are narrowed down to ranges applicable to heating in buildings. Exchanger heat transfer effectiveness ε is kept constant at 70% and 90%, which is a reasonable assumption for systems used in buildings (ASHRAE, 2000).

Each plot in Figures 6 and 7 below is made by keeping constant the temperatures of environmental air T_e (-25°C, +15°C, +25°C) and hot inlet air $T_{h,in}$. Plots are made for $T_{h,in} = +20^\circ\text{C}$ (assuming indoor air from an office is used to preheat cold air) and $T_{h,in} = +40^\circ\text{C}$ (assuming the same for a sauna). The dimensionless temperature T' is varied from ca. 0.65 to 13, by varying the cold inlet temperature $T_{c,in}$ from -50°C to +28°C (for the office) and from -60°C to +39°C (for the sauna).

At a given T' , η_f can vary somewhat for a same ε value, depending on $T_{h,in}-T_{c,in}$ and T_e , as shown in Figure 5. First the hot air temperature $T_{h,in}$ is taken as 40°C (e.g. a sauna in a building). Within this range the effect on exergy efficiency η_f of varying T_e from -25°C to +25°C is rather small, but nevertheless noticeable in the range of T' between 1.5 and 5. For example, at $T'=2.5$, $\eta_f=0.882$ for $T_e = -25^\circ\text{C}$ and $\eta_f = 0.865$ for $T_e = +25^\circ\text{C}$. For a narrower temperature range, the effect of varying T_e is less noticeable. For example, taking $T_{h,in} = +20^\circ\text{C}$ (e.g. an office in a building) and $T'=2.5$, we obtain $\eta_f=0.876$ at $T_e = -25^\circ\text{C}$ and $\eta_f=0.862$ for $T_e=15^\circ\text{C}$. For the ranges considered in this example, the effect on η_f of changing T_e is insignificant, compared to the effect of changing $T_{h,in}-T_{c,in}$.

In practice, assuming the environmental temperature to be constant should not lead to significant errors within the temperature ranges normally applicable to the built environment.

Figure 5 shows that η_f is little sensitive to changes in T_e within the ranges considered. Hence, a constant value of T_e is chosen for comparing the effects of changing ε and $T_{h,in}-T_{c,in}$ on a cold winter day. Figure 6 presents plots of η_f at $T_e = -25^\circ\text{C}$ and $T_{h,in} = +20^\circ\text{C}$, for $\varepsilon=90\%$ (light line) and for $\varepsilon=70\%$ (dark line). As in Figure 5, $T_{h,in}-T_{c,in}$ is changed by varying $T_{c,in}$.

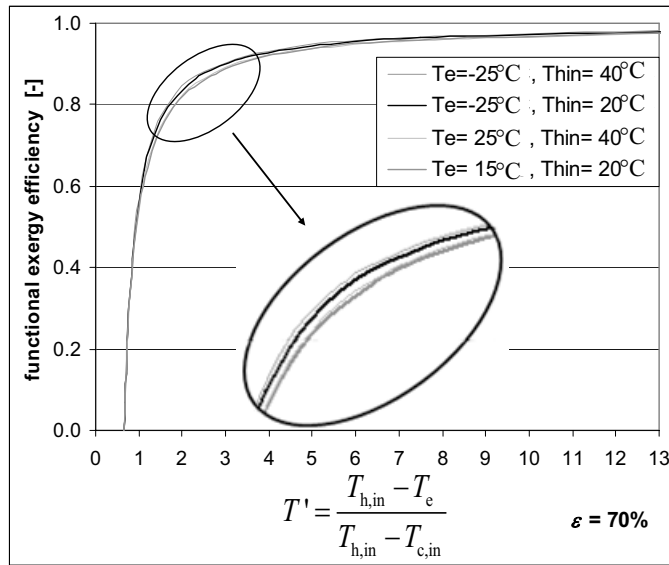


Figure 5 Sensitivity of η_f to variations in T_e and $T_{h,in}$ at $\varepsilon = 70\%$

When $\varepsilon=70\%$, η_f varies between 0.7 and 0.95 for T' between 1.27 and 5.53. In this example, $T'=5.53$ corresponds to $T_{c,in} = +11.86^\circ\text{C}$. The outside air would have to be pre-heated by 36.86 K (from $T_e = -25^\circ\text{C}$ to $T_{c,in} = +11.86^\circ\text{C}$) before entering the heat exchanger, in order to obtain exergy efficiency $\eta_f=0.95$. For $\eta_f=0.9$, pre-heating by 29.91K (from $T_e = -25^\circ\text{C}$ to $T_{c,in} = +4.91^\circ\text{C}$) would be needed. On the other hand, to obtain $\eta_f=0.7$ it is enough to pre-heat the outdoor air by 9.7K, from $T_e = -25^\circ\text{C}$ up to $T_{c,in} = -15.3^\circ\text{C}$. This could possibly be achieved by so-called passive means, such as a sunspace or buried pipes. In a real system, electricity input to drive fans and/or control devices would have to be considered as well.

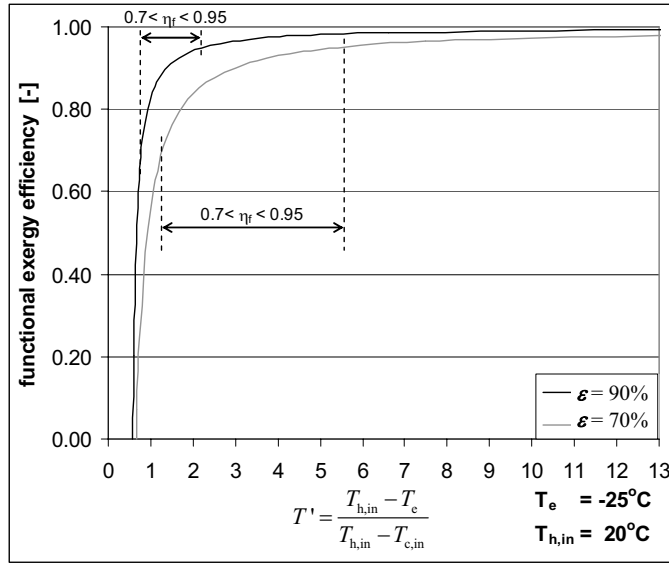


Figure 6 Sensitivity of η_f for two exchanger heat transfer effectiveness values: $\varepsilon = 90\%$ and $\varepsilon = 70\%$ at $T_e = -25^\circ\text{C}$ and $T_{h,in} = 20^\circ\text{C}$

Increasing ε to 90% for the same example means that even if cold environmental air at $T_{c,in} = -25^\circ\text{C}$ would be used directly at the cold inlet, an exergy efficiency $\eta_f = 0.83$ would be obtained for the heat exchanger model. To obtain $\eta_f = 0.9$, it would be enough to pre-heat the outdoor air ($T_e = -25^\circ\text{C}$) by 11.97K to $T_{c,in} = -13.03^\circ\text{C}$. A further increase to $\eta_f = 0.95$ would require pre-heating by 24.7K from $T_e = -25^\circ\text{C}$ to $T_{c,in} = -0.3^\circ\text{C}$. In a real system, the benefit of increased thermal performance would have to be weighed against the increased fan power normally required by heat exchangers of higher exchanger heat transfer effectiveness.

4. Conclusions

The main purpose of this paper is to investigate how changes in temperature levels affect the efficiency of thermal exergy transfer occurring at near-environmental temperatures. The analysis uses exergy efficiency and exchanger heat transfer effectiveness, combined with exergy consumption and warm/cool exergy, to gain insight into the effect of varying temperatures in heat exchange at near-environmental temperature. A dimensionless temperature is proposed to identify and analyse patterns in the sensitivity of the functional exergy efficiency to temperature variations.

The analysis in this work is performed with a simple air-to-air sensible heat exchanger model for heating purposes, assuming constant values of exchanger heat transfer effectiveness. The paper focuses on building ventilation as an example of a system involving heat transfer relatively close to ambient conditions. The approach is also applicable to other systems involving heating and cooling at near-environmental temperatures, e.g. industrial processes such as food production.

One important finding of this study is that different values of functional exergy efficiency can be obtained for the same exchanger heat transfer effectiveness, depending on how the heat exchanger temperatures relate to the environmental temperature. This is because the exchanger heat transfer effectiveness of a heat exchanger provides information on the relationship between the operating temperatures, but does not indicate whether these temperatures are near or far from the environment level. The functional exergy efficiency, on the other hand, does include information on the environmental temperature.

In combination with the dimensionless temperature T' defined in this paper and in (Boelman and Sakulpipatsin, 2004, 2005, Sakulpipatsin et. al., 2007c), η_f can be used as a guide for selecting temperatures to operate heat exchangers near environmental temperature in an exergy efficient way.

Another important finding is that the notions of heat transfer effectiveness, functional exergy efficiency and exergy consumption can be combined with the concept of warm and cool exergy, in order to select exergy efficient temperature combinations in the preliminary design of systems involving heat exchange very near to environmental temperatures. For the example of balanced ventilation and heating in buildings considered in this paper, the analysis shows that for exergy efficient operation, it is recommended to select temperature combinations corresponding to $T' \geq 1$. In this range, the equivalent temperatures of hot and cold air are above environmental temperature T_e . Although the hot air loses warm exergy, the heat exchange is exergy efficient because there is sufficient gain of warm exergy by the cold air. On the other hand, heat exchanger operation is not recommended for $T' < 1$. In this range, the equivalent temperature of cold air is below T_e and heating this cold air implies losing its cool exergy. The cool exergy of the cold air could be better used for cooling another flow of matter, at or below T_e . At the same time, the warm exergy of the hot air could be better used to heat another flow of matter at or above T_e . From an exergy viewpoint, it is inefficient to use air above environmental temperature to heat air below environmental temperature, even if the heat exchanger has high exchanger heat transfer effectiveness.

The insight gained from this analysis can be useful when designing a heat exchange system, for example when deciding between using a heat exchanger of higher exchanger heat transfer effectiveness or pre-heating the outside air (e.g. by using a sunspace or the underground). In practice, such a decision would also have to consider the additional pressure drop usually associated with higher exchanger heat transfer effectiveness versus the possibility of using passive means to pre-heat environmental air. The analysis proposed in this paper can also be extended to other applications (e.g. in the food processing industry) involving heat exchange very near to environmental temperature.

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Nomenclature

A	Area [m^2]
c_p	Isobaric heat capacity [$\text{m}^2\text{s}^{-2}\text{K}^{-1}$; $\text{Jkg}^{-1}\text{K}^{-1}$]
C	Heat capacity [$\text{m}^2\text{kgs}^{-2}\text{K}^{-1}$; JK^{-1}]
Ex	Exergy [$\text{m}^2\text{kgs}^{-2}$; J]
$\dot{Ex}$	Exergy per second [$\text{m}^2\text{kgs}^{-3}$; Js^{-1}]
ΔEx	Exergy difference [$\text{m}^2\text{kgs}^{-2}$; J]
H	Enthalpy [$\text{m}^2\text{kgs}^{-2}$; J]
$\dot{m}$	Mass flow rate [kgs^{-1}]
Q	Thermal energy [$\text{m}^2\text{kgs}^{-2}$; J]
$\dot{Q}$	Thermal energy per second [$\text{m}^2\text{kgs}^{-3}$; Js^{-1}]
S	Entropy [$\text{m}^2\text{kgs}^{-2}\text{K}^{-1}$; JK^{-1}]
$\dot{S}$	Entropy per second [$\text{m}^2\text{kgs}^{-3}\text{K}^{-1}$; $\text{Js}^{-1}\text{K}^{-1}$]
T	Air temperature [K; °C with notation]
T'	Dimensionless temperature in heating mode [-]
T''	Dimensionless temperature in cooling mode [-]
U	Overall heat transfer coefficient [$\text{kgs}^{-3}\text{K}^{-1}$; $\text{Js}^{-1}\text{m}^{-2}\text{K}^{-1}$]

Greek letters

ε	Exchanger heat transfer effectiveness [-]
η	Exergy efficiency [-]

Subscripts

0	Reference environment
air	Air
c	Cold air
ch	Chemical
e	Reference environment state; dead state
eq	Equivalent
f	Functional
gen	Generated

h	Hot air
i	Indoor; room
in	Inlet
loss	Loss
me	Mechanical
min	Minimum
out	Outlet
product	Product
source	Source
th	Thermal
u	Universal

Abbreviations

HVAC	Heating, Ventilation and Air-Conditioning
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Sensitivity of exergy efficiencies of a vapour-compression heat pump for space cooling applications

Abstract: This paper critically analyses the sensitivity of exergy efficiencies for a simple vapour-compression heat pump cycle for space cooling applications, in their operating window at near-environmental temperatures. There are mainly two types of exergy efficiency definitions: the universal ones in which gross exergy inputs and outputs are considered, and the functional ones in which net exergy flows are considered respectively. This work focuses on exergy analysis of space cooling by application of a vapour-compression heat pump using air cooling and heating. This system is modelled as simple as possible. In this case there are four independent parameters which determine the behaviour of the system: the temperatures of the air streams entering the heat exchangers, the environmental temperature and the second law efficiency of the heat pump. The results show that the functional exergy efficiency is more sensitive to parameter changes than the universal exergy efficiency. These efficiencies show not much difference when the exergy value of the hot air stream can be used in a meaningful way in the building. There are four contributions to the overall exergy losses in the case of space cooling. Two determined by heat exchange in both heat exchangers, the amount of exergy of the hot air stream, and the irreversibility's within the heat pump itself, expressed in its second law efficiency.

Keywords: exergy; exergy efficiency; vapour-compression heat pump; space cooling; all-air system.

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Brief biographical notes about authors

P. Sakulpipatsin is a Doctoral Researcher in Climate Design Group, at the Faculty of Architecture, the Delft University of Technology, The Netherlands. He obtained two master degrees: in 2003 MSc in Energy Conversion and Management from the University

of Applied Sciences Offenburg, Germany; and in 2001 MSc in Computer Science from Chulalongkorn University, Thailand.

Dr. Ir. H.J. van der Kooi is Assistant Professor in the Field of Applied Thermodynamics and Chemical Engineering at the Delft University of Technology, The Netherlands.

Dr. E.C. Boelman has a background in Adsorption Cooling, Building Engineering and Management, and has been working on renewable energy and exergy analysis applied to buildings for several years.

Dr. L.C.M. Itard is a Researcher in the field of sustainable buildings and HVAC equipment and leads the group sustainable and healthy building of the Research Institute OTB at the Delft University of Technology, The Netherlands.

1. Introduction

Exergy provides a common basis for comparison between the performance of systems associated with buildings and building services (Schmidt, 2004; Annex37, 2002). For example, exergy analysis allows a designer to compare, on the same basis, between thermal energy supplied by electricity (e.g. to drive a heat pump) and by solar heat (e.g. through a window). It also allows comparison on the same basis between e.g. the electricity required to drive the heat pump and the thermal energy involved in the heat pumping process.

Exergy analyses of heat pumps have been proposed by a number of researchers. Szargut (2002) proposed component efficiencies of a vapour-compression heat pump to indicate the possibilities for improvement of heat pump installations. Bilgen and Takahashi (2002) carried out exergy analysis of a domestic heat pump, and presented analysis results in terms of *COP* and second-law efficiency. On the larger scale of energy supply systems, several authors have provided some different definitions of exergy efficiency (Sorin and Brodyansky, 1992; Tsatsaronis, 1993; Kotas 2001). Exergy efficiency is often defined considering the intended application of a given system under specific conditions, and therefore the definitions frequently lack uniformity. Woudstra (2002) distinguished two different types of exergy efficiency definitions, i.e. universal and functional ones. When compared to the universal exergy efficiency definition, the functional exergy efficiency definition has the advantage of being more sensitive to parameter changes because it deals with useful exergy transfer and therefore ‘net’ efficiency values and excludes the unused exergy discarded with outgoing flows.

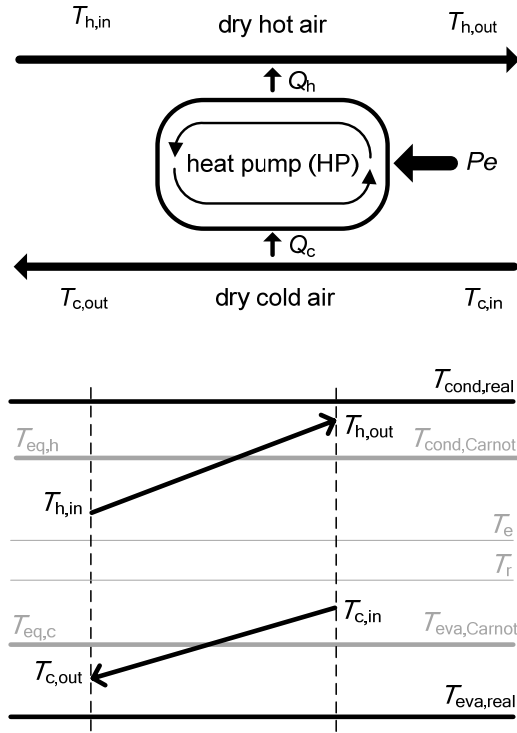
In order to be able to use appropriate exergy efficiencies as a performance criterion for HVAC components and systems in buildings, it is important to better understand the extent to which the use of different exergy efficiency definitions may lead to significantly different valuations of a same system, and also how changes in environmental temperature may lead to different exergy efficiency values for a particular system, notwithstanding unchanged operating temperatures. An analysis of exergy efficiency definitions applied for a simple counter-flow heat exchanger has shown the extent to which the use of output or product-based exergy efficiency definitions can lead to substantially different efficiency values (Boelman and Sakulpipatsin, 2005). This is particularly the case in situations involving small temperature differences and high exchanger heat transfer effectiveness. The next step is to systematically investigate which exergy efficiencies are the most appropriate ones for heat pump applications at near ambient conditions in buildings.

The main objective of this study is to better understand the value and sensitivity of exergy efficiency definitions applied to the system composed of a vapour-compression heat pump cycle in combination with all-air heat exchange and operating at near environmental temperatures.

2. Simplified vapour-compression heat pump model

For the purpose of this study, a simplified black-box heat pump model for space cooling applications is adopted, and shown in Figure 1.

Figure 1 Diagram of the simplified vapour-compression heat pump model (above) and air temperature profiles (below)



The model lumps the effect of internal heat pump parameters (e.g. working fluid, compressor characteristics) into a single efficiency parameter, namely the second-law efficiency η_{II} . The model is based on the use of air as heat source and as heat sink. Its operating conditions are characterised mainly by the temperatures $T_{c,in}$, $T_{c,out}$, $T_{h,in}$ and $T_{h,out}$. It is assumed that there is no energy loss occurring in the heat pump process and the heat exchange process. The energy conversion process in the electricity production is not considered in this study. The equivalent temperatures of the cold air $T_{eq,c}$ and the hot air $T_{eq,h}$ are assumed to be equal to the Carnot temperatures of the air streams flowing through the evaporator $T_{eva,Carnot}$ and the condenser $T_{cond,Carnot}$ respectively.

The equivalent temperatures (Çengel and Robert, 2001) of the cold air $T_{eq,c}$ and the hot air $T_{eq,h}$ are calculated from the air temperatures at the heat exchanger inlets T_{in} and outlets T_{out} by using equation 1.

$$T_{eq} = \frac{T_{out} - T_{in}}{\ln\left(\frac{T_{out}}{T_{in}}\right)} \quad (1)$$

The heat pump model is considered as simple as possible, in order to allow the analysis to be based on as few parameters as possible. The assumptions used for the exergy analysis of the heat pump are given in Table 1.

Table 1 Simplifying assumptions for the vapour-compression heat pump model

<i>Simplifying assumptions</i>
- the heat pump for space cooling applications in all-air systems - thermal energy losses in the heat exchangers ignored - the heat pump transferring heat from/to two dry air streams - pressure drop in pipes ignored - only thermal exergy (of the dry air) considered - specific heat of dry air constant: $1.005 \text{ kJkg}^{-1}\text{K}^{-1}$ - same mass flows for the hot air and the cold air: 1 kgs^{-1} $T_{eq,c}$ equal to $T_{eva,Carnot}$ and $T_{eq,h}$ equal to $T_{cond,Carnot}$ - counter-flow heat exchange in both the evaporator and the condenser - energy conversion losses in electricity production and transmission losses out of the system boundaries - the power input for air transport through the heat exchangers negligible compared to the power input of the compressor - effect of internal heat pump parameters lumped into η_{hp}, considered at 10%, 40% and 100%

Temperature domains of the environment and the dry air streams are defined based on typical ranges applicable to HVAC systems in space cooling applications, and are given in Table 2.

Table 2 Temperature domains

<i>Temperature domains</i>
$273.15 \text{ K} \leq T_e \leq 323.15 \text{ K}$ $10 \text{ K} \leq T_{h,in} - T_{c,in} \leq 50 \text{ K}$ $0 \leq T' \leq 1$

The operating temperatures of the dry air streams are taken over a somewhat broader range than in usual HVAC applications, in order to enable the temperature combinations required to obtain dimensionless temperatures T' covering the relatively broad range of

$0 \leq T' \leq 1$, where the temperature of the environment is between the inlet temperatures of the dry air streams $T_{h,in}$ and $T_{c,in}$. The dimensionless temperature T' is defined as in equation 21 and explained in item 5.1.

3. Practical application of heat pumps in HVAC systems

This item only gives an impression of considerations of importance for the choice of an appropriate air conditioning heat pump based system and possibilities to use (waste) thermal exergy of the hot air stream. In the next items, the basis for an as simple as possible exergetic evaluation of this type of system will be given and applied.

At temperatures close to the temperature of the environmental air, thermal energy of a substance has a low exergy value. This means that in principle a low exergy source must be preferably chosen to increase the exergy of the room air to be cooled. Such a source can for example be water at a temperature low enough to cool the air. The problem in this case is that the source must be available and that is certainly not always the case. Eventually storage of a cooler medium could be a compensable solution. Nevertheless storage needs space, cooling at appropriate moments, maintenance etc. This can lead to larger costs than for other possible systems. Much research has been done on cooling systems and these systems are often used in practice. A simple cooling system makes use of a pure substance applied in a cooling cycle operating at two different pressure levels. At the low pressure, and thus temperature, evaporation takes place in an evaporator to form the substance in a (superheated) gas phase. This gas is compressed to the high pressure and then cooled by an appropriate medium to a (sub cooled) liquid phase in a condenser. The liquid is expanded to the lower pressure and the mixture of liquid and gas is led to the evaporator to complete the cycle. This cooling machine is driven by power input for the compressor. At a low temperature, thermal energy is transferred from a medium (i.e. now air to be cooled) to a substance in the evaporator, and a higher temperature thermal energy is transferred from the substance in the condenser to another appropriate medium for example air from the environment. In an ideal (Carnot) cooling machine the exergy input (power) is completely used to increase the exergy of both air streams. In air conditioning applications, the exergy increases of both streams can be relatively much smaller than the thermal energy of both streams. In principle this is a good option for air conditioning. This type of the heat pump (cooling machine) for air conditioning in buildings will be used and described in an as simple as possible way in this paper. Next, important points for a designer, related to the use of a heat pump for space cooling applications, will be discussed in somewhat more details.

3.1. Practical application of heat pumps for space cooling in buildings

To choose a proper heat pump to be used for space cooling in buildings, the minimal temperature of the room air to be cooled first has to be determined. Some factors related to assigning the value in order to fulfil desired indoor air quality requirements are, for example, room air temperature and humidity, ventilation airflow rate and thermal properties of the building envelope. The temperature of the refrigerant (which usually is a pure substance) in the evaporator is normally chosen from 5K to 10K lower than this minimal temperature to assure a large enough driving force for transfer of thermal energy to the refrigerant in the evaporator. Also, the maximum cooling load determines the mass flow rate of the refrigerant. For the condenser, thermal energy from the condensing

refrigerant could be transferred to the environmental air. The condensing refrigerant temperature is normally determined by the maximum temperature of the environmental air and a reasonable temperature difference between the condensing refrigerant and the environmental air. The temperature difference is usually chosen from 5K to 10K. The temperatures of the refrigerant in the evaporator and the condenser are to determine the operating temperatures in the heat pump. This information should be considered to select a high-efficient and not too expensive heat pump.

Also, the condenser must be preferably placed on a high level of the building (e.g. on the roof) to reduce fouling of this heat exchanger, and on the cold side of the building. The refrigerant must preferably be non-toxic, incombustible, detectable, non-corrosive, and ecological friendly. It often has a high density but not too high operating pressure. Due to the negative effects of many halogenated refrigerants and their out-phasing, it is nearly impossible to find a refrigerant that fulfils all these requirements. A compromise will be then necessary. Normally the heat pump is used with direct thermal energy transfer from and to two different air streams. The air streams must be transported through the heat exchangers by ventilators which must use as little as possible power, and to realize this, the pressure drop in the air ducts must be as small as possible. A problem in the design of these heat exchangers is the large difference in volume flows of the air streams and the refrigerant. One possible, but complicated, solution is to use an intermediate medium for staged thermal energy transfer. Another possibility is to use a non condensing refrigerant, although this would require a more voluminous heat pump and lead to deceleration of the transfer rate of thermal energy from the refrigerant to the air streams.

3.2. Possibilities to use the exergy (increase) of the hot air stream

The exergy (increase) per second of the hot air stream ($\dot{E}x_{th}$) has a relation to the airflow rate ($\dot{m}$; kilogram per second), the specific heat of the air $c_{p,air}$ (assumed constant at 1.005 kJkg⁻¹K⁻¹), the equivalent temperature of the hot air stream $T_{eq,h}$ and the environmental air temperature T_e , as shown in equation 2.

$$\dot{E}x_{th} = \dot{m}c_{p,air} \left((T_{eq,h} - T_e) - T_e \ln \left(\frac{T_{eq,h}}{T_e} \right) \right) \quad (2)$$

Possibilities to use the exergy (increase) of the hot air stream depend on the magnitude of the airflow rate and the temperature levels of the hot air stream and the environment. By assuming that there is no thermal energy loss in heat transfer process in the condenser, the equivalent temperature of the hot air stream is then ideally similar to the condensing refrigerant temperature. In this case, the exergy of the hot air stream only depends on the airflow rate, and also the mass flow rate of the refrigerant. This shows that operating the hot air stream and the refrigerant at a higher airflow rate gives more opportunities of heating a matter that needs more exergy.

Further, considering possible temperature levels of the indoor air T_r and environment T_e for operating a vapour-compression heat pump in space cooling could be done in Table 3. The numbers (1-5) in Figure 2 indicate possible temperatures of T_e and T_r for operating the heat pump in space cooling, particularly in summer. In the summer, it is assumed that environmental air temperature T_e is in maximum 31°C (based on climate data of De Bilt, NL, from the TMY2 weather data source) and room air temperature is assumed at 24°C.

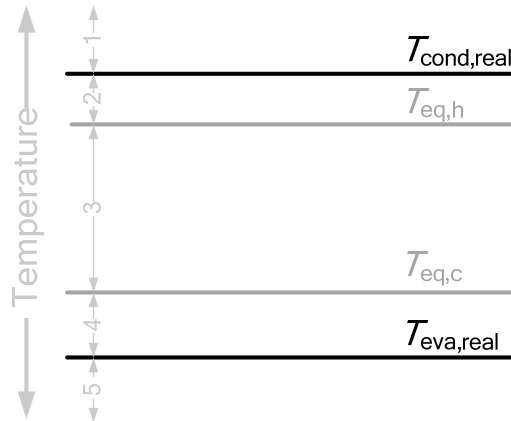
The heat pump is generally utilised when $T_e > T_r$ in the summer and therefore the temperature combinations between T_e and T_r in the dark gray area ($T_e < T_r$) are not applicable for the heat pump operation. Also T_r is always above $T_{eq,c}$ for the heat pump operation, according to the 2nd law of thermodynamics that thermal energy is transferred from a higher-temperature substance to a lower-temperature substance. This results that the temperature combinations between T_e and T_r in the light gray area ($T_r \leq T_{eq,c}$) are also not applicable for the heat pump operation.

Table 3 Possible temperature combinations (T_e and T_r) for operating a vapour-compression heat pump in space cooling

$T_e \backslash T_r$	1 ¹ ($T_r > T_{cond,real}$)	2 ¹ ($T_{cond,real} > T_r > T_{eq,h}$)	3 ¹ ($T_{eq,h} > T_r > T_{eq,c}$)	4 ¹ ($T_{eq,c} > T_r > T_{eva,real}$)	5 ¹ ($T_{eva,real} > T_r$)
1 ¹ ($T_e > T_{cond,real}$)	$T_e > T_r$ $T_e < T_r$	$T_e > T_r$ $T_e < T_r$	$T_e > T_r$ $T_e < T_r$		
2 ¹ ($T_{cond,real} > T_e > T_{eq,h}$)		$T_e > T_r$ $T_e < T_r$			
3 ¹ ($T_{eq,h} > T_e > T_{eq,c}$)			$T_e > T_r$ $T_e < T_r$		
4 ¹ ($T_{eq,c} > T_e > T_{eva,real}$)				$T_e > T_r$ $T_e < T_r$	
5 ¹ ($T_{eva,real} > T_r$)					$T_e > T_r$ $T_e < T_r$

¹ The numbers indicate in which temperature region T_e (or T_r) is. The temperature regions are illustrated in Figure 2.

Figure 2 Possible temperatures of T_e and T_r for operating a vapour-compression heat pump in space cooling



The temperature combinations between T_e and T_r in the vertical-striped area ($T_e > T_r$ and $T_r \geq T_{eq,h} \geq T_{eq,c}$) are possible for the heat pump operation. However, these combinations are not recommended because both hot and cold air streams can be directly used to cold the room air.

For the heat pump operating where the temperature combinations between T_e and T_r are in the horizontal-striped area, the equivalent temperature of the hot air stream $T_{eq,h}$ is between T_e and T_r , f.e. $24^\circ\text{C} < T_{eq,h} < 31^\circ\text{C}$. Also, for the heat pump operating where the temperature combinations between T_e and T_r are in the diagonal-striped area, $T_{eq,h}$ could ideally reach T_e , f.e. $T_{eq,h} = 31^\circ\text{C}$. The hot air stream operating in these conditions still has cool exergy because of $T_{eq,h} \leq T_e$, and could be used for cooling a medium at a higher temperature level (than $T_{eq,h}$). Some possibilities to use the exergy (increase) of the hot air stream are, for example, (pre)heating domestic hot water and heating a medium in a (seasonal) thermal storage.

For the heat pump operating where the temperature combinations between T_e and T_r are in the white area ($T_{eq,h} \geq T_e$ and $T_{eq,c} \leq T_r$), the hot air stream has warm exergy. In ideal, the hot air stream could have the maximum warm exergy if $T_{eq,h}$ reach the condensing refrigerant temperature (e.g. ca. 100°C for R-134a and R-22 and ca. 120°C for NH_3 in maximum). This might be high enough to heat a medium at a high temperature level, such as domestic hot water. However, it also depends on an amount of thermal energy transferred from the hot air stream, obtained from the heat pump.

4. Relevant definitions

4.1. Exergy

The thermal exergy values per second of the hot air stream and the cold air stream at the heat exchanger inlets and outlets $\dot{E}x_{th,c,in}$, $\dot{E}x_{th,h,in}$, $\dot{E}x_{th,c,out}$ and $\dot{E}x_{th,h,out}$ are calculated by using equation 3.

$$\dot{E}x_{th} = \dot{m}c_{p,air} \left((T - T_e) - T_e \ln \left(\frac{T}{T_e} \right) \right) \quad (3)$$

In this study, the exergy values (per second) of the air streams account only for the thermal contribution $\dot{E}x = \dot{E}x_{th}$, because the chemical and mechanical contributions are ignored. The thermal exergy depends not only on its temperature T , but also on the temperature of the environment T_e . When $T > T_e$ the thermal exergy can be called “warm exergy”, and on the other hand when $T < T_e$ the thermal exergy can be called “cool exergy” (Shukuya, 1996).

The exergy value of the electricity, for driving the compressor, is equal to the electricity input to the compressor Pe , according to the theory that electric energy can be totally converted into mechanical work (Moran and Shapiro, 1998). By this case, the exergy value and the energy value of the electricity are assumed same.

4.2. Functional exergy efficiencies

The functional exergy efficiency of the heat pump η_f is best defined as the exergy increase of the cold air stream, which is the desired output, divided by the exergy input values of all relevant streams to realise this desired output. In this case, electricity input of the heat pump Pe pays for the exergy increase of both cold and hot air streams and is

thus equal to the exergy value of the input to realise the desired output. This functional exergy efficiency $\eta_{f,I}$ can be defined as given in equation 4.

$$\eta_{f,I} = \frac{\Delta Ex_c}{Pe} = \frac{Ex_{c,out} - Ex_{c,in}}{Pe} \quad (4)$$

When the exergy increase of the hot air is also accounted as the desired output, the functional exergy efficiency $\eta_{f,II}$ can be defined as given in equation 5.

$$\eta_{f,II} = \frac{\Delta Ex_c + \Delta Ex_h}{Pe} = \frac{(Ex_{c,out} - Ex_{c,in}) + (Ex_{h,out} - Ex_{h,in})}{Pe} \quad (5)$$

Items 4.2.1 and 4.2.2 present derivations of the functional exergy efficiencies $\eta_{f,I}$ and $\eta_{f,II}$ where they are in the Carnot case (ideal case) and in the real case respectively.

4.2.1. The “Carnot” case

The Carnot heat pump is assumed to operate between two constant temperatures $T_{eq,c}$ and $T_{eq,h}$, where $T_{eq,c} = T_{eva,Carnot}$ and $T_{eq,h} = T_{cond,Carnot}$. In this case, exergy losses due to irreversibilities are ignored. From equation 4, where only the exergy increase of the cold air stream is considered as the desired output, the exergy increase of the cold air stream is simply determined by multiplying Q_c with the appropriate Carnot factor, and further by making use of equation 4 to express the electricity input in the Carnot case Pe_{Carnot} in the characteristic equivalent temperatures, where $Pe_{Carnot} = Q_h - Q_c$. The functional exergy efficiency in the Carnot case $\eta_{f,I,Carnot}$ is calculated by using equation 6:

$$\eta_{f,I,Carnot} = \frac{Q_c \left| 1 - \frac{T_e}{T_{eq,c}} \right|}{Q_c \left(\frac{T_{eq,h}}{T_{eq,c}} - 1 \right)} \quad (6)$$

In the cooling case, normally $T_e > T_{eq,c}$ and then the absolute value of $1 - (T_e/T_{eq,c})$ is equal to $(T_e/T_{eq,c}) - 1$. Using this assumption, the previous equation can be simplified as shown in equation 7:

$$\eta_{f,I,Carnot} = \left(\frac{T_e - T_{eq,c}}{T_{eq,h} - T_{eq,c}} \right) \quad (7)$$

This functional exergy efficiency $\eta_{f,I,Carnot}$ is of course smaller than 1 where T_e is between $T_{eq,c}$ and $T_{eq,h}$, as can also be seen directly from equation 7, because the exergy increase of the hot air stream is not used. This is the only loss occurring in this case. A challenge is to find a possibility to make use of this exergy increase of the hot air stream too. When $T_e < T_{eq,c}$ the absolute value of $1 - (T_e/T_{eq,c})$ is equal to that value of the term itself. In this case the appropriate form of the expression of the functional exergy efficiency is shown in equation 8:

$$\eta_{f,I,\text{Carnot: } T_e < T_{eq,c}} = \left(\frac{T_{eq,c} - T_e}{T_{eq,h} - T_{eq,c}} \right) \quad (8)$$

When accounting the exergy increase of the hot air stream ΔEx_h as the desired output for calculating the functional exergy efficiency $\eta_{f,II,\text{Carnot}}$, the calculation uses equation 9.

$$\eta_{f,II,\text{Carnot}} = \frac{Q_c \left| 1 - \frac{T_e}{T_{eq,c}} \right| + Q_h \left(1 - \frac{T_e}{T_{eq,h}} \right)}{Q_c \left(\frac{T_{eq,h}}{T_{eq,c}} - 1 \right)} \quad (9)$$

For the Carnot case, the functional exergy efficiency $\eta_{f,II,\text{Carnot}}$ is always equal to 1 where environmental air temperature is between the temperatures at the heat exchanger inlets of the hot air stream $T_{h,in}$ and the cold air stream $T_{c,in}$. This is because the electricity input Pe_{Carnot} is completely used to increase the exergy of the cold air stream as well as the exergy of the hot air stream, and there is no exergy loss due to heat transfer.

In reality, $T_{h,in}$ and $T_{c,in}$ change and are different from the condensing and evaporating refrigerant temperatures $T_{\text{cond,real}}$ and $T_{\text{eva,real}}$ respectively: where $T_{h,in} < T_{\text{cond,real}}$ and $T_{c,in} > T_{\text{eva,real}}$. Furthermore, a real heat pump will be considered where it has internal irreversibilities, in the next section in the “real” case.

4.2.2. The “real” case

In this real case, three sources of irreversibility are considered. For two first sources of irreversibility, in both heat exchangers there must be a driving force for heat transfer, that means large enough temperature differences between the hot and the cold air streams must be chosen in order to achieve a reasonable transfer rate of thermal energy from the cold air stream and to the hot air stream. The other source is irreversibility is internal in the heat pump cycle process. These irreversibilities are expressed in terms of temperature differences of the refrigerant in the condenser and the evaporator between the Carnot (ideal) case and the real case, and in term of additional electricity needed by the heat pump process to reach the ideal case.

In the Carnot case where there is no irreversibility occurring in the heat pump, $T_{eq,c} = T_{\text{eva,Carnot}}$ and $T_{eq,h} = T_{\text{cond,Carnot}}$. Therefore the condensing and evaporating refrigerant temperatures can be expressed in equation 10 and 11 respectively.

$$T_{\text{cond,real}} = T_{\text{cond,Carnot}} + \Delta T_{\text{cond}} = T_{h,eq} + \Delta T_{\text{cond}} \quad (10)$$

$$T_{\text{eva,real}} = T_{\text{eva,Carnot}} - \Delta T_{\text{eva}} = T_{c,eq} - \Delta T_{\text{eva}} \quad (11)$$

where ΔT_{cond} and ΔT_{eva} are the temperature differences of the refrigerant in the condenser and the evaporator between the Carnot case and the real case.

The functional exergy efficiency $\eta_{f,I,\text{real}}$ can now be determined from equation 12.

$$\eta_{f,I,real} = \frac{Q_{c,real} \left| 1 - \frac{T_e}{T_{eq,c}} \right|}{Pe_{real}} \quad (12)$$

where Pe_{real} is the electricity input of the heat pump. The electricity input of the heat pump includes the effects of the temperature differences ΔT_{cond} and ΔT_{eva} , but not the effects of the entropy change in the compressor. The process in the compressor is assumed adiabatic.

By assuming that there is no energy loss in the heat pump, the electricity power for the heat pump $\dot{P}_{e,real}$ is equal to the difference between the amounts of heat per second that is released from the heat pump and that is removed from the cold stream ($\dot{Q}_{h,real}$ and $\dot{Q}_{c,real}$ respectively), and calculated by using equation 13.

$$\dot{P}_{e,real} = \dot{Q}_{h,real} - \dot{Q}_{c,real} \quad (13)$$

where

$$\dot{Q}_{c,real} = \dot{m}_c < c_{p,air} > (T_{c,in} - T_{c,out}) \quad (14)$$

$$\begin{aligned} \dot{Q}_{h,real} &= \dot{Q}_{c,real} \left(\frac{T_{cond,real}}{T_{eva,real}} \right) \\ &= \dot{m}_c < c_{p,air} > (T_{c,in} - T_{c,out}) \left(\frac{T_{h,eq} + \Delta T_{cond}}{T_{c,eq} - \Delta T_{eva}} \right) \end{aligned} \quad (15)$$

The functional exergy efficiency in the real case $\eta_{f,I,real}$ is always smaller than the functional exergy efficiency in the Carnot case $\eta_{f,I,Carnot}$, when considering the same thermal energy of the cold air stream ($Q_{c,real}$), because the electricity input in the real case (Pe_{real}) is bigger than the electricity input in the Carnot (ideal) case (Pe_{Carnot}) due to the bigger temperature differences of the refrigerant in the condenser and the evaporator $T_{cond,real} - T_{eva,real} > T_{cond,Carnot} - T_{eva,Carnot}$ (see Figure 1).

Also the ratio between $T_{cond,real}$ and $T_{eva,real}$ can be simplified in term of COP_{real} from equation 23 and by assumption in equation 13, shown as in equation 16.

$$COP_{real} = \frac{Q_{c,real}}{Pe_{real}} = \frac{Q_{c,real}}{Q_{h,real} - Q_{c,real}} = \frac{1}{\frac{Q_{h,real}}{Q_{c,real}} - 1}$$

Since $\frac{Q_{h,real}}{Q_{c,real}} = \frac{T_{cond,real}}{T_{eva,real}}$ therefore,

$$\frac{T_{\text{cond,real}}}{T_{\text{eva,real}}} = \frac{Q_{\text{h,real}}}{Q_{\text{c,real}}} = \frac{T_{\text{eq,h,real}}}{T_{\text{eq,c,real}}} \frac{1}{COP_{\text{real}}} + 1 \quad (16)$$

When accounting the exergy increase of the hot air stream ΔEx_h as the desired output for calculating the functional exergy efficiency $\eta_{f,II,\text{real}}$, the calculation uses equation 17.

$$\eta_{f,II,\text{real}} = \frac{Q_{\text{c,real}} \left| 1 - \frac{T_e}{T_{\text{eq,c}}} \right| + Q_{\text{h,real}} \left(1 - \frac{T_e}{T_{\text{eq,h}}} \right)}{Pe_{\text{real}}} \quad (17)$$

Considering where environmental air temperature is between the inlet temperatures of the hot air stream $T_{h,\text{in}}$ and the cold air stream $T_{c,\text{in}}$, the functional exergy efficiency $\eta_{f,II,\text{real}}$ is always bigger than $\eta_{f,I,\text{real}}$ since it also includes ΔEx_h in the numerator, but always less than one since Pe_{real} also accounts internal irreversibilities in occurring the heat pump.

4.3. Universal exergy efficiencies

So far only relevant changes of the exergy values of the cold air stream and eventually also of the hot air stream were taken into account in the numerator of the efficiencies considered, and only the electricity input in the denominator. In a more general form, only gross exergy inputs and outputs are considered for an exergy efficiency definition. The disadvantage of the exergy efficiency definition for the system investigated in this paper is a loss in sensitivity for changes in the relevant parts of the system, because both numerator and denominator become larger, and thus less sensitive for characteristic changes in the system studied.

The overall universal exergy efficiency of the heat pump system $\eta_{u,I}$ is defined as the total exergy value of the air outlet streams divided by the total exergy value of the air inlet streams including the electricity input to the heat pump Pe , as shown in equation 18 (Woudstra, 2002).

$$\eta_{u,I} = \frac{\Sigma Ex_{\text{out}}}{\Sigma Ex_{\text{in}}} = \frac{Ex_{\text{c,out}} + Ex_{\text{h,out}}}{Ex_{\text{c,in}} + Ex_{\text{h,in}} + Pe} \quad (18)$$

A simpler expression can be obtained from which the behaviour of the universal exergy efficiency $\eta_{u,I}$ can be seen more directly. In this case the exergy values of the exiting streams are calculated from the sum of the exergy value of each of the entering streams plus the exergy increase of each stream. After substitution of these expressions in equation 18 and after division of both numerator and denominator by the sum of the exergy values of the entering streams, equation 19 is then obtained.

$$\eta_{u,I} = \frac{1 + \frac{\Delta Ex_c + \Delta Ex_h}{Ex_{\text{c,in}} + Ex_{\text{h,in}}}}{1 + \frac{Pe}{Ex_{\text{c,in}} + Ex_{\text{h,in}}}} \quad (19)$$

When the sum of the exergy values of the entering streams is relatively small compared to the electricity input and the sum of the exergy increases of both streams, the right hand terms are large compared to 1. From equation 19 it can be seen that this efficiency then goes to the one shown in equation 20. In the other limit, when the exergy values of the entering streams are large compared to electricity input and the exergy increases of both streams, this universal exergy efficiency $\eta_{u,I}$ goes to 1 and is not sensitive to small changes in these exergy values.

For other universal exergy efficiencies different from equation 18, if some of the outputs are considered as losses then these outputs do not contribute to the numerator of equation 18. For example, when the exergy value of the hot air stream is considered as a loss, the universal exergy efficiency $\eta_{u,II}$ is calculated by using equation 20.

$$\eta_{u,II} \equiv \frac{Ex_{c,out}}{Ex_{c,in} + Ex_{h,in} + Pe} \quad (20)$$

5. Approach

The simplified vapour-compression heat pump model (Figure 1) is used to calculate the values of the exergy efficiencies, in order to analyse the sensitivity of universal and functional exergy efficiencies to variations in operating temperatures and second-law efficiency of the heat pump model. Exergy analysis results are presented as a function of a defined dimensionless temperature. The following parameters are allowed to vary: environmental air temperature T_e ; inlet temperatures of the hot and cold air streams $T_{c,in}$ and $T_{h,in}$; and heat pump's second-law efficiency η_{II} . Outlet temperatures of the hot and cold air streams $T_{c,out}$ and $T_{h,out}$ are determined as simple functions of inlet temperatures $T_{c,in}$ and $T_{h,in}$ and second-law efficiencies η_{II} .

Heat pump operating temperatures are determined as follows. On the cold side, $T_{c,out}$ is assumed to be always 5K below $T_{c,in}$. On the hot side, $T_{h,out}$ values are calculated by using equations 23 to 25 in a trial and error method. On the first trial, $T_{h,out}$ is estimated as ca. 5K above $T_{h,in}$ and this value is used to calculate a tentative value for η_{II} . In each subsequent trial, $T_{h,out}$ is increased by 0.05K (or less for $\eta_{II} > 50\%$). The $T_{h,out}$ calculations are repeated until a $T_{h,out}$ value is obtained that meets the targeted second-law efficiency (defined as $\eta_{II} = 10\%$, 50% or 90%). This $T_{h,out}$ value is then used with the other T_h and T_c values to calculate the functional and universal exergy efficiencies (explained in item 5.3 and 5.4).

The chemical contributions to the exergy values of the cold air stream at the evaporator inlet $Ex_{ch,c,in}$ and outlet $Ex_{ch,c,out}$ are ignored, because the cold air is assumed to be dry and to have a constant composition (see Table 1). Also there is no difference in the mechanical contribution to the exergy values of the cold air at the inlet $Ex_{me,c,in}$ and the outlet $Ex_{me,c,out}$, because pressure drop in the evaporator is neglected. Therefore the exergy values of the cold air at the inlet and outlet points are equal to their thermal contributions. This is also assumed for the hot air. In this study, the exergy analysis thus considers only the thermal contribution to the exergy values.

The dimensionless temperature is introduced in item 5.1. The calculations of the second-law efficiency, the functional exergy efficiency and the universal exergy efficiency, of the heat pump model are explained in items 5.2, 5.3 and 5.4, respectively.

5.1. Dimensionless temperature

A dimensionless temperature T' is defined (Boelman and Sakulpipatsin, 2005) in order to present and discuss exergy calculation results as a function of environmental air temperature T_e in relation to inlet air temperatures $T_{c,in}$ and $T_{h,in}$. The dimensionless temperature expresses a distance between a cold inlet air temperature $T_{c,in}$ and an environmental air temperature T_e , relative to the inlet air temperature difference $T_{h,in} - T_{c,in}$. T' can be formulated as equation 21.

$$T' = \frac{T_e - T_{c,in}}{T_{h,in} - T_{c,in}} \quad (21)$$

For $T' < 0$, $T_{c,in}$ is above environmental air temperature T_e . This could be the case of heat pumps in industry or other applications where waste-heat is upgraded.

For $T' = 0$, $T_{c,in}$ is equal to T_e . In practice, this could correspond e.g. to an air-source heat pump used to produce hot sanitary water.

For $0 < T' < 1$, $T_{c,in}$ is below T_e and $T_{h,in}$ is above T_e . This could be the case of simultaneous heating (e.g. of a swimming pool) and cooling (e.g. an office, but also an ice rink).

For $T' = 1$, $T_{h,in}$ is equal to T_e , as in the case of air-cooled refrigerating machines.

For $T' > 1$, both $T_{h,in}$ and $T_{c,in}$ are below T_e , as in the case for water-cooled air-conditioning and refrigeration cycles.

Since the study focuses on a heat pump used for cooling purposes in buildings, only exergy calculation results at $T' \geq 0$ is discussed, in the next chapter.

5.2. Second-law efficiency

The second-law efficiency η_{II} is defined by a ratio of the actual performance COP to the maximum theoretical performance COP_{Carnot} . The second-law efficiency provides an indication of how closely the performance of an actual heat pump cycle approaches the maximum theoretical limit defined by the operating temperatures.

$$\eta_{II} = \frac{COP}{COP_{Carnot}} \quad (22)$$

The COP is defined as the ratio of the thermal energy provided by the heat pump Q_c and the electricity supplied to the compressor Pe .

$$COP = \frac{Q_c}{Pe} \quad (23)$$

The amount of thermal energy per second removed from the cold air stream $\dot{Q}_c$ can be calculated by using equation 24.

$$\dot{Q}_c = \dot{m}_c < c_{p,air} > (T_{c,out} - T_{c,in}) \quad (24)$$

In Equation 24 $\dot{m}_c$ is the air flow rate of the cold air (kilogram per second), and $\langle c_{p,air} \rangle$ is the mean specific heat capacity of this stream. This mean specific heat capacity is defined in equation 25.

$$\langle c_{p,air} \rangle \equiv \frac{\int_{T_{c,out}}^{T_{c,in}} c_{p,air}(T) dT}{T_{c,in} - T_{c,out}} \quad (25)$$

The thermal energy per second released by the heat pump $\dot{Q}_h$ is calculated in the same way as shown in equation 24. The thermal energy released by the heat pump Q_h corresponds in theory, when there is no thermal energy loss, to the sum of the thermal energy removed from the cold stream Q_c and the electricity supplied to the compressor Pe at the unit of time. According to the assumptions in Table 1, Pe can be written as the difference between the amounts of heat that is released from the heat pump and that is removed from the cold stream: $Q_h - Q_c$.

The COP_{Carnot} is a function of the equivalent temperatures of the cold air $T_{eq,c}$ and the hot air $T_{eq,h}$. By assuming that the Carnot equivalent temperatures of the cold air $T_{eq,c}$ and the hot air $T_{eq,h}$ are equal to the temperatures of the evaporator $T_{eva,Carnot}$ and the condenser $T_{cond,Carnot}$ respectively, COP_{Carnot} can be expressed as shown in Equation 26.

$$COP_{Carnot} = \frac{T_{eq,c}}{T_{eq,h} - T_{eq,c}} \quad (26)$$

The COP and η_{II} express the performance of the heat pump, and depend on the machine characteristics (e.g. working fluid, compressor and heat exchanger behaviour) as well as on some operating conditions (e.g. heat transfer fluid temperatures, flow rates and specific heats). They have the advantage of being independent from environmental conditions. For example, the COP and η_{II} of a ground-coupled heat pump are not affected by daily or seasonal fluctuations in atmospheric temperature. These efficiencies focus on machine characteristics and operating conditions, and provide a well-defined basis for expressing and mutually comparing the performance of similar machines.

5.3. Functional exergy efficiencies

For the calculations of the functional exergy efficiency in item 6, equation 33 is used. The functional exergy efficiency $\eta_{f,I,real}$ can be rewritten as a function of the functional exergy efficiency in the Carnot case $\eta_{f,I,Carnot}$ and the second-law efficiency η_{II} , as shown in equation 12. The derivation of the functional exergy efficiency $\eta_{f,I,real}$ is following.

$$\eta_{f,I,real} = \frac{Q_{c,real} \left| 1 - \frac{T_e}{T_{eq,c}} \right|}{Pe_{real}} = \frac{Q_{c,real} \left| 1 - \frac{T_e}{T_{eq,c}} \right|}{\frac{Q_{c,real}}{COP_{real}}} = COP_{real} \left| 1 - \frac{T_e}{T_{eq,c}} \right| \quad (27)$$

Substitution of the expressions from equation 7 and equation 22 in equation 27 leads to the relations shown in equation 28, when $T_e > T_{eq,c}$.

$$\eta_{f,I,real} = \left| 1 - \frac{T_e}{T_{eq,c}} \right| \eta_{II} COP_{Carnot} = \eta_{II} \eta_{f,I,Carnot} \quad (28)$$

The functional exergy efficiency $\eta_{f,I,real}$ is equal to the product of the functional exergy efficiency in the Carnot case $\eta_{f,I,Carnot}$ with the second law efficiency of the heat pump η_{II} and is thus directly influenced by the irreversibilities in the heat pump itself and by the irreversibilities related to heat transfer in the heat exchangers, all these contributions are lumped in one parameter, the second law efficiency η_{II} .

The equivalent temperatures of the hot and cold air streams $T_{eq,c}$ and $T_{eq,h}$ are parameters of $\eta_{f,I,Carnot}$ that are shown in the equation 7. In the following, the derivations of the parameters in terms of $T_{c,in}$ and $T_{h,in}$ are shown.

As described above, $T_{c,out}$ is assumed to be always 5K below $T_{c,in}$. With the assumption and equation 1, the equivalent temperature of the cold air stream is written in term of the temperature of the cold air stream at the heat exchanger inlet $T_{c,in}$ as shown in equation 29.

$$T_{eq,c} = \frac{-5}{\ln \left(1 - \frac{5}{T_{c,in}} \right)} \quad (29)$$

Usually, $T_{h,in}$ will be measured and is normally close to the temperature of the environmental air. In order to be able to calculate both $T_{h,out}$ and $T_{eq,h}$, two expressions for the thermal energy per second that is released from the heat pump ($\dot{Q}_h$) are given in equation 30 and equation 31. Both expressions are assumed that the airflow rates for the hot air and the cold air streams ($\dot{m}_h$ and $\dot{m}_c$) are equal to 1 kgs⁻¹ (Table 1). From these expressions it is then possible to relate both unknown temperatures to each other.

$$\dot{Q}_h = 1 < c_{p,air} > (T_{h,out} - T_{h,in}) \quad (30)$$

$$\dot{Q}_h = \dot{Q}_c \frac{T_{eq,h}}{T_{eq,c}} = 1 < c_{p,air} > (T_{c,in} - T_{c,out}) \left(\frac{T_{eq,h}}{T_{eq,c}} \right) = 5 < c_{p,air} > \left(\frac{T_{eq,h}}{T_{eq,c}} \right) \quad (31)$$

Then, $T_{eq,h}$ can be determined in terms of $T_{c,in}$ and $T_{h,in}$, as shown in Equation 32:

$$T_{eq,h} = \frac{-T_{h,in}}{\ln \left(1 - \frac{5}{T_{c,in}} \right)} \frac{\frac{5}{T_{c,in}}}{1 - \frac{5}{T_{c,in}}} \quad (32)$$

From equation 28, the functional exergy efficiency $\eta_{f,I,real}$ is presented in relation to the inlet temperatures of the hot and cold air streams $T_{c,in}$ and $T_{h,in}$ (equations 29 and 32 respectively) and the second-law efficiency η_{II} as shown in equation 33.

$$\eta_{f,I,real} = \eta_{II} \frac{\left(-\frac{T_e}{5} \ln \left(1 - \frac{5}{T_{c,in}} \right) - 1 \right)}{\left(\frac{T_{h,in}}{5} \left(\frac{1}{1 - \frac{5}{T_{c,in}}} - 1 \right) - 1 \right)} \quad (33)$$

5.4. Universal exergy efficiencies

For the calculations of the universal exergy efficiency in item 6, equation 18 is used. The universal exergy efficiency $\eta_{u,I,real}$ is a function of the exergy values of the hot and the cold air streams at the heat exchanger inlets and outlets, and of the electricity input of the heat pump.

This item presents $\eta_{u,I}$ as a function of $T_{h,in}$, $T_{c,in}$ and η_{II} . $Ex_{c,in}$ and $Ex_{h,in}$ are already in terms of the inlet temperatures. The followings present derivations of $Ex_{c,out}$ and $Ex_{h,out}$ in terms of the inlet temperatures, and Pe_{real} in terms of the inlet temperatures and the second-law efficiency.

The exergy value per second of the cold air stream at the heat exchanger outlet $\dot{Ex}_{c,out}$ is calculated by using equation 34 (with the assumption $\dot{m}_c = 1$, Table 1), by substituting $T_{c,out}$ in the exergy calculation in equation 3. $T_{c,out}$ is assumed to be always 5K below $T_{c,in}$.

$$\dot{Ex}_{c,out} = 1 < c_{p,air} > \left(T_{c,in} - 5 - T_e - T_e \ln \left(\frac{T_{c,in} - 5}{T_e} \right) \right) \quad (34)$$

To calculate the exergy value per second of the hot air stream at the heat exchanger outlet $\dot{Ex}_{h,out}$, first the outlet temperature of the hot air stream in the real case $T_{h,out,real}$ must be given. The temperature $T_{h,out,real}$ is solved from these relations in equation 16, equation 35 and equation 36. In equation 35 and equation 36, it is assumed that $\dot{m}_h$ and $\dot{m}_c$ are equal to 1 kgs⁻¹ (Table 1).

$$\dot{Q}_{h,real} = 1 < c_{p,air} > (T_{h,out,real} - T_{h,in}) \quad (35)$$

$$\dot{Q}_{h,real} = \dot{Q}_{c,real} \frac{T_{eq,h,real}}{T_{eq,c,real}} = 1 < c_{p,air} > (T_{c,in} - T_{c,out}) \left(\frac{T_{eq,h,real}}{T_{eq,c,real}} \right) = 5 < c_{p,air} > \left(\frac{T_{eq,h,real}}{T_{eq,c,real}} \right) \quad (36)$$

Then $T_{h,out,real}$ can be determined in equation 37.

$$T_{h,out,real} = T_{h,in,real} + 5 \left(\frac{T_{eq,h} - T_{eq,c}}{\eta_{II} T_{eq,c}} + 1 \right) \quad (37)$$

$\dot{E}x_{h,out}$ can be determined in equation 38.

$$\dot{E}x_{h,out} = < c_{p,air} > \left(T_{h,in,real} + 5 \left(\frac{T_{eq,h} - T_{eq,c}}{\eta_{II} T_{eq,c}} + 1 \right) - T_e - T_e \ln \left(\frac{T_{h,in,real} + 5 \left(\frac{T_{eq,h} - T_{eq,c}}{\eta_{II} T_{eq,c}} + 1 \right)}{T_e} \right) \right) \quad (38)$$

The electricity per second input to the heat pump $\dot{P}_{e,real}$ is obtained from the equations of η_{II} (equation 22) and COP_{real} (equation 23). After expressing the thermal energy per second removed from the cold air stream $\dot{Q}_c$ in the product of the airflow rate $\dot{m}_c$, the mean specific heat capacity $< c_{p,air} >$, and the temperature change of the cold air stream ($T_{c,in} - T_{c,out}$), this relation is given in equation 39 with the assumptions: $T_{c,in} - T_{c,out} = 5K$ (item 5) and $\dot{m}_c = 1$ (Table 1). $T_{eq,c}$ and $T_{eq,h}$ are calculated in by using equation 29 and equation 32.

$$\dot{P}_{e,real} = \frac{5 < c_{p,air} > T_{eq,c}}{\eta_{II} (T_{eq,h} - T_{eq,c})} \quad (39)$$

After substitution of the equations of $\dot{E}x_{c,in}$, $\dot{E}x_{h,in}$, $\dot{E}x_{c,out}$, $\dot{E}x_{h,out}$ and $\dot{P}_{e,real}$ in Equation 18 the universal exergy efficiency $\eta_{u,I,real}$ can be determined. What can nicely been seen is that the mass stream, and eventually also the mean heat capacities, although these can be different from each other in principle but not in the application considered in this paper, drop out from numerator and denominator. From this it may be clear that also the absolute value of the electricity power input and the exergy losses are of importance too. This universal exergy efficiency $\eta_{u,I,real}$ in equation 18 is much more complicated to determine and gives not much added value due to the relatively low exergy values of all air streams and, as remarked before, is less sensitive than the first universal exergy efficiency.

6. Analysis results and discussions

This item presents and discusses sensitivities of the exergy efficiency definitions to temperature changes within the temperature domains and the temperature combinations given in Table 2. In item 6.1, sensitivities of the universal exergy efficiency and the functional exergy efficiency, where the exergy efficiency definitions are applied for 30% of the second-law efficiency and a same set of heat pump operation parameter values (in terms of temperatures and mass flow rate), are compared and discussed. In item 6.2, sensitivities of the universal exergy efficiency and the functional exergy efficiency are studied separately, where they are applied for some different values of the second-law efficiency and the inlet temperatures of the air streams $T_{h,in}$ and $T_{c,in}$.

6.1. Comparison of the universal and the functional exergy efficiencies at the same second-law efficiency

As outlined in the previous item, the second-law efficiency η_{II} is independent from environmental conditions, while the universal exergy efficiency η_u and the functional exergy efficiency η_f are intrinsically linked to the environmental conditions. Figure 3 presents some calculation results of these exergy efficiencies to illustrate this difference.

The plots in Figure 3 display sensitivities of the exergy efficiency definitions (η_u and η_f) to environmental air temperatures T_e where T_e is between $T_{c,in}$ and $T_{h,in}$. All other operating conditions are kept constant, at the values shown in Table 4. The horizontal axis shows dimensionless temperatures T' (equation 21) and the corresponding environmental air temperatures T_e , for the specific case of $T_{h,in}=50^\circ\text{C}$ and $T_{c,in}=0^\circ\text{C}$.

Figure 3 An example of how the exergy efficiencies can vary with environmental air temperatures

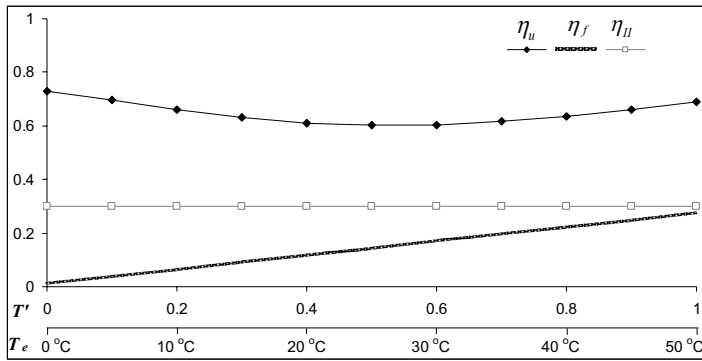


Table 4 Operating conditions for this example

Operating conditions			
• $T_{h,in}=50^\circ\text{C}$	• $T_{h,out}=58.5^\circ\text{C}$	• $0^\circ\text{C}\leq T_e\leq 50^\circ\text{C}$	• $COP=1.43$
• $T_{c,in}=0^\circ\text{C}$	• $T_{c,out}=-5^\circ\text{C}$	• $\dot{P}_{e,real}=3.5\text{kW}$	• $\eta_{II}=30\%$

Figure 3 shows a constant line of the second-law efficiency values $\eta_{II}=30\%$ (equation 22) along with other lines of the universal and functional exergy efficiency values (equations 18 and 33). The lower line presents the functional exergy efficiency values and the upper line presents the universal exergy efficiency values. The lower line likes linear and varies from 0.013 to 0.277. In this range, the functional exergy efficiency depends on the dimensionless temperature T' and increases when T' increases. The functional exergy efficiency accounts only for exergy change of the cold air stream, but not for exergy change of the hot air stream. The upper line likes polynomial with order 2. The upper line varies from 0.730 to 0.601 and has the lowest point around $T'=0.55$. The lowest point appears in the upper line because the universal exergy efficiency definition accounts all exergy inputs and all exergy outputs. An adapted universal exergy efficiency (e.g. the one in equation 20) could be more sensitive within the considered range of T' .

This example shows that among the efficiency definitions (η_u , η_f and η_{II}), the functional exergy efficiency η_f is most sensitive and most appropriate for exergy performance assessment of the heat pump operating in a cooling mode and where T_e is between $T_{c,in}$ and $T_{h,in}$.

6.2. Universal and functional exergy efficiencies

This item discusses sensitivities of the universal and functional exergy efficiencies that are influenced by changes of operating temperatures and the second-law efficiency.

The discussion is based on some calculation results of the universal and functional exergy efficiencies (η_u and η_f), in conditions that are applicable for space cooling in buildings. The second-law efficiency η_{II} is kept constant at 10% and 40%, as well as 100% to show how far the exergy efficiencies are from the ideal ones.

The calculations are carried out at a constant environmental air temperature ($T_e=30^\circ\text{C}$), and for temperatures of the cold air stream at the heat exchanger inlet $T_{c,in}$ at 25°C (comparable to a temperature of air leaving an office building in summer) and 0°C (comparable to the same for a cooled room). Temperatures of the hot air stream at the heat exchanger inlet $T_{h,in}$ are calculated according to the dimensionless temperature T' (equation 21) considered where $0 \leq T' \leq 1$.

Figure 4 shows the calculation results of the universal and functional exergy efficiencies, where assuming environmental air temperature T_e constant at 30°C and the second-law efficiency η_{II} constant at 40%. The calculation results are made for $T_{c,in} = 25^\circ\text{C}$ and 0°C .

Figure 4 (a) shows the results of the universal exergy efficiency η_u where T' is between 0 and 1. As T' increases from 0 to 1, η_u decreases from 1 to 0.70 (for the plot for $T_{c,in} = 0^\circ\text{C}$) and from 1 to 0.46 (for the plot for $T_{c,in} = 25^\circ\text{C}$). The plot for $T_{c,in} = 0^\circ\text{C}$ is above the plot for $T_{c,in} = 25^\circ\text{C}$. This is because, at a given T' , $T_{h,in}$ and $T_{h,out} - T_{h,in}$ from the plot for $T_{c,in} = 0^\circ\text{C}$ are always higher than (or equal to) those from the plot for $T_{c,in} = 25^\circ\text{C}$ (for example at $T'=0.6$: $T_{h,in}=50^\circ\text{C}$ and $T_{h,out}-T_{h,in}=7.6\text{K}$ for $T_{c,in}=0^\circ\text{C}$; and $T_{h,in}=33^\circ\text{C}$ and $T_{h,out}-T_{h,in}=5.6\text{K}$ for $T_{c,in}=25^\circ\text{C}$. At $T'=0.9$, $T_{h,in}=33^\circ\text{C}$ and $T_{h,out}-T_{h,in}=6.8\text{K}$ for $T_{c,in}=0^\circ\text{C}$; and $T_{h,in}=30.56^\circ\text{C}$ and $T_{h,out}-T_{h,in}=5.45\text{K}$ for $T_{c,in}=25^\circ\text{C}$). These make $Ex_{h,in}$, $Ex_{h,out}$ and $Ex_{h,out} - Ex_{h,in}$, from the plot for $T_{c,in} = 0^\circ\text{C}$ are larger than those from the plot for $T_{c,in} = 25^\circ\text{C}$, while both plots have the same values of $Ex_{c,in}$ and $Ex_{c,out}$, and therefore η_u from the plot for $T_{c,in} = 0^\circ\text{C}$ is higher than η_u from the plot for $T_{c,in} = 25^\circ\text{C}$ at a same T' point.

As T' increases from 0 to 0.6, the universal exergy efficiencies η_u change in the almost entire range of the η_u decreases (from 1 to 0.70 for the plot for $T_{c,in} = 0^\circ\text{C}$ and from 1 to 0.46 for the plot for $T_{c,in} = 25^\circ\text{C}$). This is because $T_{h,in}$ decreases exponentially according to the change of T' , and $T_{h,out} - T_{h,in}$ also decreases exponentially (but in much more narrow range comparing to the $T_{h,in}$ decreases) from 7.37K to 5.57K (for $T_{c,in} = 25^\circ\text{C}$) and from 19.42K to 7.60K (for $T_{c,in}=0^\circ\text{C}$). The changes of $T_{h,out} - T_{h,in}$ are to maintain the constant value of the second-law efficiency ($\eta_{II}=40\%$). When T' approaches 0, $T_{h,in}$ is very far from T_e and $T_{h,out}-T_{h,in}$ becomes much smaller relative to $T_{h,in}-T_e$. This makes $Ex_{h,in}$ and $Ex_{h,out}$ are large, but the exergy difference is very relatively small, and therefore makes η_{II} approach 1.

Figure 4a Sensitivities of η_u to variations in $T_{c,in}$

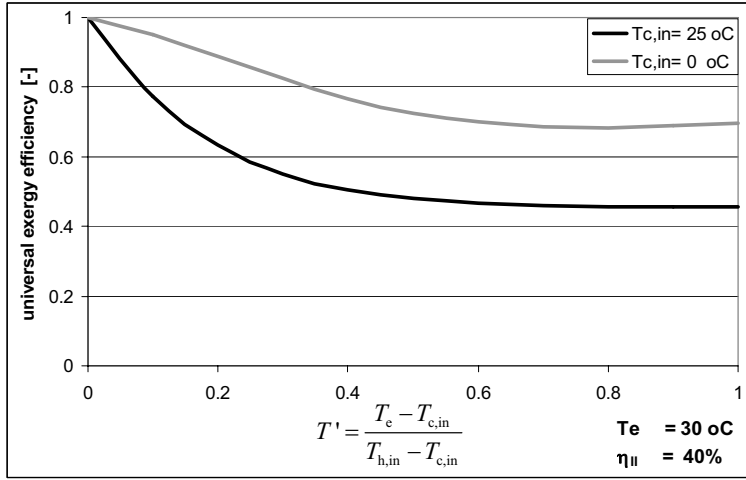
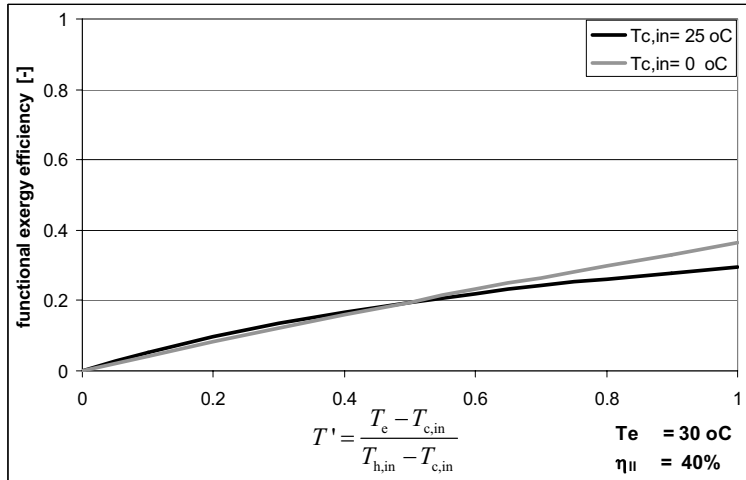


Figure 4b Sensitivities of η_f to variations in $T_{c,in}$



As T' increases from 0.6 to 1, the universal exergy efficiencies η_u change less than 0.01. In the period of T' , $T_{h,out}$ and $T_{h,in}$ change not much and this makes insignificant changes of $Ex_{h,in}$ and $Ex_{h,out}$. The universal exergy efficiencies η_u for $T_{c,in} = 25^\circ\text{C}$ and $T_{c,in} = 0^\circ\text{C}$ are not sensitive to the increase of T' from 0.6 to 1.

Figure 4 (b) shows the results of the functional exergy efficiency η_f where T' is between 0 and 1. The plot for $T_{c,in} = 0^\circ\text{C}$ has a similar trend as the plot for $T_{c,in} = 25^\circ\text{C}$ has. The maximum difference between the η_f values from both plots is 0.07 at $T'=1$. The changes of η_f are more sensitive than the changes of η_u where T' is between 0.6 and 1.0.

Figure 5 shows the calculation results of the universal and functional exergy efficiencies, where $T_c=30^\circ\text{C}$ and $T_{c,in} = 25^\circ\text{C}$. The calculation results are made for $\eta_{II} = 10\%$, 40% and 100% .

Figure 5a Sensitivities of η_u to variations in η_{II}

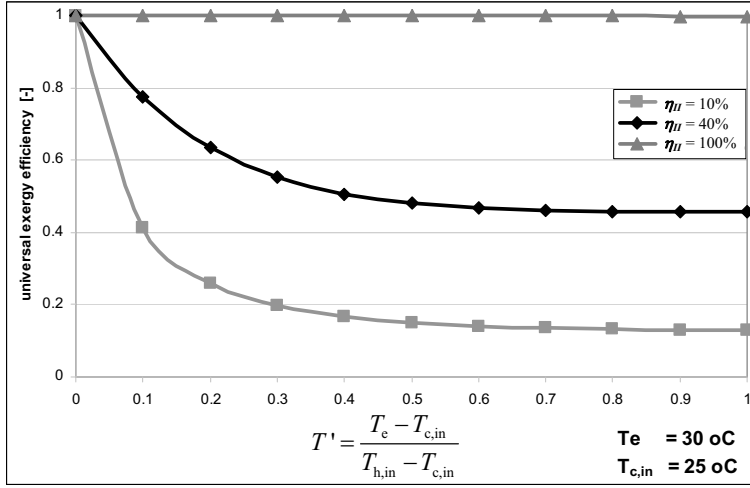


Figure 5b Sensitivities of η_f to variations in η_{II}

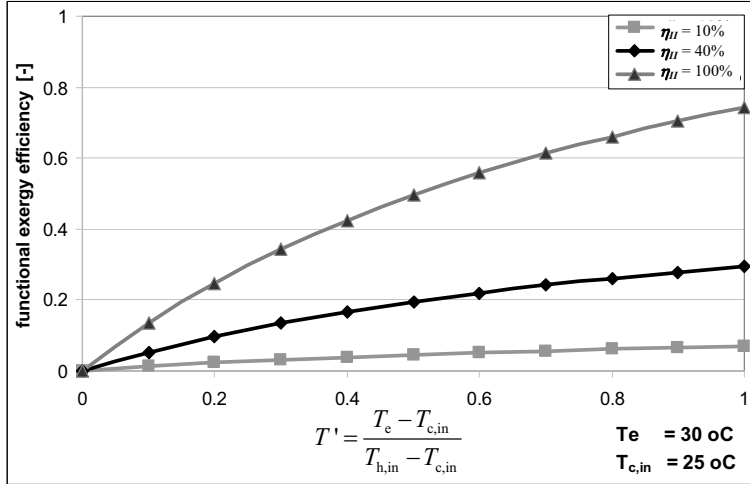


Figure 5 (a) shows the results of the universal exergy efficiency η_u where T' is between 0 and 1. The plot for $\eta_{II}=100\%$ shows that all of the results are around 1. This is because $T_{h,out} - T_{h,in}$ changes in a very small range, and makes $Ex_{h,in}$ and $Ex_{h,out}$ quite similar for at a T' point (where $0 \leq T' \leq 1$). The other plots are different from that plot, according to $T_{h,out} - T_{h,in}$ changes to maintain constant values of η_{II} for each plot. Also all of the plot are not sensitive where $0.6 \leq T' \leq 1$.

Comparing the plots in Figure 5 (a) with the plots in Figure 4 (a), the difference between the sensitivities of the universal exergy efficiency η_u affected by the change of η_{II} and by the change of $T_{c,in}$ is not so obvious. The maximum difference between the η_u values is 0.86 at $T'=1$ from the plots (the plot for $\eta_{II}=10\%$ and the plot for $\eta_{II}=100\%$) in Figure 5

(a), and is 0.24 at $T'=1$ from the plots (the plot for $T_{c,in} = 0^\circ\text{C}$ and the plot for $T_{c,in} = 25^\circ\text{C}$) in Figure 4 (a).

Figure 5 (b) shows the results of the functional exergy efficiency η_f where T' is between 0 and 1. The plots for $\eta_{II} = 10\%$ and for $\eta_{II} = 100\%$ have similar trends as the plot for $\eta_{II} = 40\%$ has. In the ideal case, where $\eta_{II} = 100\%$, η_f could reach in maximum 0.74 at $T'=1$. For the other plots at $T'=1$, η_f is 0.07 for the plot for $\eta_{II} = 10\%$ and 0.29 for the plot for $\eta_{II} = 40\%$. These η_f values are proportional to each other, according to their η_{II} values. These relations are shown as in equation 28, where the functional exergy efficiency in the real case $\eta_{f,I,real}$ varies on the second-law efficiency η_{II} and the functional exergy efficiency in the Carnot case $\eta_{f,I,Carnot}$. For the other T' points, these η_f values also behave in such the relation.

Comparing the plots in Figure 5 (b) with the plots in Figure 4 (b), the sensitivity of the functional exergy efficiency η_f is more influenced by the change of η_{II} than by the change of $T_{c,in}$. The maximum difference between the η_f values is 0.68 at $T'=1$ from the plots (the plot for $\eta_{II} = 10\%$ and the plot for $\eta_{II} = 100\%$) in Figure 5 (a), and is 0.07 at $T'=1$ from the plots (the plot for $T_{c,in} = 0^\circ\text{C}$ and the plot for $T_{c,in} = 25^\circ\text{C}$) in Figure 4 (a).

7. Conclusions

This paper critically analyses exergy efficiency definitions for a simple vapour-compression heat pump cycle, in order to better understand what and how the exergy efficiencies can be applied for operation of HVAC systems for space cooling application at near-environmental temperatures. A simplified vapour-compression heat pump model is taken as an example for analysing the sensitivity of universal and functional exergy efficiency definitions to variations in operating temperatures, environment temperature and heat pump machine efficiency. This study only focuses on the heat pump model used for cooling purposes in all-air systems. A dimensionless temperature is used to illustrate the analysis results, and to discuss the sensitivity of the exergy efficiency definitions to variations on temperature and the second-law efficiency.

The functional exergy efficiency definition for the heat pump model is explained in the Carnot (ideal) case and the real case respectively, followed by the explanation of the universal exergy efficiency definition for the heat pump model in the real case. These definitions are derived as a function of inlet temperature of the hot and cold air streams that exchange heat with the heat pump ($T_{h,in}$ and $T_{c,in}$) and the second-law efficiency η_{II} .

The results indicate that, for a same set of the heat pump operating conditions (which are inlet temperatures and second-law efficiency), changes in environmental air temperature can lead to significant variations in exergy efficiency of the heat pump. This is because exergy recognizes that the energy content of a substance or flow of matter is usable only down to the environmental conditions. Hence, the exergy efficiency of a heat pump cycle will be higher when it operates further from the environmental temperature.

The results also show how exergy efficiency values can differ, depending on whether the efficiency definition considers gross exergy inputs and outputs (universal exergy efficiency) or neglects discarded exergy flows (functional exergy efficiency). Because the functional exergy efficiency considers only net exergy flows, it can be higher and is more sensitive to temperature changes than the universal exergy efficiency. The changes of η_f are more sensitive than the changes of η_u where the dimensionless T' is between 0.6 and

1.0. The difference between the sensitivities of the universal exergy efficiency η_u affected by the change of η_{II} and by the change of $T_{c,in}$ is not so obvious. But in the case that considering the sensitivity of the functional exergy efficiency η_f , the change of η_{II} have much more influence than the change of $T_{c,in}$.

The functional exergy efficiency η_f is recommended to be used as a performance criterion for a heat pump for space cooling application, especially where it operates T_c between $T_{h,in}$ and $T_{c,in}$ and close to $T_{h,in}$. In this temperature range, $T_{h,out}$ could be not so high, as seen from the example in the paper. This is good in environmental friendly aspect, due to the low temperature level, that makes less thermal pollution to the environment.

For future work, the sensitivity of the exergy efficiency definitions should be studied in the case that the heat pump is used for other application, e.g. space heating application.

Acknowledgements

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Nomenclature

c_p	Isobaric heat capacity [$\text{m}^2\text{s}^{-2}\text{K}^{-1}$; $\text{Jkg}^{-1}\text{K}^{-1}$]
COP	Coefficient of Performance [-]
Ex	Exergy [$\text{m}^2\text{kgs}^{-2}$; J]
$\dot{Ex}$	Exergy per second [$\text{m}^2\text{kgs}^{-3}$; Js^{-1}]
$\dot{m}$	Airflow rate [kgs^{-1}]
Pe	Electricity [$\text{m}^2\text{kgs}^{-2}$; J]
$\dot{Pe}$	Electricity power [$\text{m}^2\text{kgs}^{-3}$; Js^{-1}]
Q	Thermal energy [$\text{m}^2\text{kgs}^{-2}$; J]
$\dot{Q}$	Thermal energy per second [$\text{m}^2\text{kgs}^{-3}$; Js^{-1}]
T	Air temperature [K; °C with notation]
T'	Dimensionless temperature [-]

Greek letters

η	Exergy efficiency [-]
η_{II}	Second-law efficiency [-]
η_f	Functional exergy efficiency [-]
η_u	Universal exergy efficiency [-]

Subscripts

air	Air
c	Cold air
Carnot	Carnot
ch	Chemical
cond	Condensing
e	Environmental
eq	Equivalent

eva	Evaporating
h	Hot air
in	Inlet; input
me	Mechanical
out	Outlet; output
r	Room air; indoor air
real	Real
th	Thermal

Abbreviations

HP	Heat pump
HVAC	Heating, Ventilation and Air Conditioning
NL	The Netherlands
TMY	Typical Meteorological Year

Exergy analysis as an assessment tool of heat recovery of dwelling ventilation systems

P. Sakulpipatsin, E.C. Boelman and J.J.M. Cauberg

Climate Design Group, Building Technology, Faculty of Architecture, Delft University of Technology,
PO Box 5043, 2600 GA Delft, the Netherlands

Abstract

This paper presents steady-state energy and exergy analyses for dwelling ventilation with and without air-to-air heat recovery, and discusses the relative influence of heat and electricity on the exergy demand by ventilation airflows. Energy and exergy analysis results for De Bilt, NL, are presented in terms of heat and electricity, on an instantaneous and a daily basis. The amount of electricity input to fans and the heat recovery unit (HRU) is much more significant in terms of exergy than of energy, due to the higher exergy value of electricity. From an exergy viewpoint, it could make sense to use the HRU only when environmental air temperature is low enough to compensate the additional need for electricity. When the air temperature is not too low, electricity input could be decreased by letting ventilation air bypass the HRU or by operating the HRU at low ventilation airflow rate, depending on outdoor temperatures and indoor occupancy conditions.

Key words: exergy, heat recovery, ventilation, winter, buildings.

1. Introduction

In cold and moderate climates, improvements in building shell insulation and air-tightness imply a shift in heating loads from transmission and infiltration towards ventilation. Heat recovery from ventilation airflow plays an increasingly important role in minimising energy needs. Such heat recovery systems rely on the input of electric power (to drive fans, heat pumps, etc.) in order to recover thermal energy. Since electricity input is relatively small compared to the amounts of thermal energy recovered, such systems are efficient from an energy viewpoint. One important yet often overlooked aspect, however, is the difference in 'quality' between the high-grade electricity input and the lower grade thermal energy recovered. This paper analyzes the effectiveness of heat recovery from ventilation airflows from the viewpoint of exergy. The results provide a common basis for evaluating different forms of energy, considering their different abilities to produce work in relation to a given environment.

Exergy analysis provides a common basis for evaluating systems using heat and electricity, considering their different abilities to produce work in relation to a given environment (Boelman 2002; Wall 1990; Rosen and Dincer 2001; Ala-Juusela

(ed.) 2004). Unlike energy, exergy is not subject to a conservation law.

The paper begins with an explanation of the system approach of the energy and exergy analyses and with descriptions of the dwelling ventilation systems. Then it describes the energy and exergy calculation methods and presents some characteristic climate data for De Bilt, the Netherlands. After that, analysis results are presented and discussed on an instantaneous and a daily basis. Finally, recommendations for operation of the dwelling ventilation systems are given in the conclusion.

2. Approach

Energy and exergy analyses are performed for dwelling ventilation, using mechanical exhaust with natural air supply without heat recovery as well as different systems of balanced ventilation with heat recovery. The aim of the developed calculation method is to assess the different systems of air to air heat recovery from exhaust air in terms of exergy. Infiltration air volumes, which can differ as a function of the ventilation system, are not considered.

The analysis assumes steady state operation and considers only dry air, thus neglecting latent heat exchange between ventilation air flows. Pressure difference between air entering and leaving the dwelling is ignored for the analysis. General calculation values, used for the exergy analysis in this paper, are given in Table 1.

Table 1. General calculation values (CEN 2000, 2005).

Parameters	Values
air density (ρ_{air})	1.23 kg m ⁻³
spec. heat capacity of air ($c_{p,\text{air}}$)	1.008 kJ kg ⁻¹ K ⁻¹
environmental air temperature (T_e)	from -13°C to 19°C
room air temperature (T_r)	21°C
ventilation airflow rates ($\dot{Q}$)	0.028, 0.042, 0.063 m ³ /s

The mechanical exhaust ventilation with natural air supply (hereunder “mechanical exhaust ventilation”) uses an AC fan (Model: CVE 166 form Itho bv. 2005a) or a DC fan (Model: CVE ECO-fan 2 from Itho bv. 2005b). Environmental air enters the dwelling at temperature $T_{r,\text{in}}$ and goes through the exhaust fans at room air temperature $T_{r,\text{out}}$, as shown in Figure 1a.

The balanced ventilation with heat recovery case (Figure 1b) assumes a DC Heat Recovery Unit (Model: HRU ECO-fan 3 S B from Itho bv. 2005c), containing two DC fans and a counter-flow heat exchanger with high thermal effectiveness ε . The effectiveness and fan electricity input are calculated by interpolating from manufacturer’s data (Itho bv. 2005c) in relation to ventilation airflow rates (see Table 2). The environmental air gains heat from exhaust air entering the heat exchanger in the HRU

at room air temperature $T_{r,\text{out}}$ and leaving the dwelling at temperature $T_{e,\text{out}}$.

Table 2. Thermal effectiveness ε versus airflow rates $\dot{Q}$.

$\dot{Q}$ [m ³ /s]	ε [-]
0.063	0.94
0.042	0.96
0.028	0.97

3. Exergy analysis of dwelling ventilation

Different ventilation systems are compared based on calculations of thermal energy and thermal exergy demands by ventilation airflows in relation to environmental air temperature and electricity input to a ventilation unit. The exergy analysis is carried out for the heating season. The calculations steps from equations 1 to 4 (ASHRAE 2000) and equations 5 and 6 (Wall 1990) apply to both mechanical exhaust ventilation and balanced ventilation. Equation 7 is used to calculate supply air temperatures at the HRU (ASHRAE 1993).

3.1. Electricity input to ventilation unit

The relation between the pressure drop of the system P and the ventilation airflow rate $\dot{Q}$ is obtained according to equation 1, where C is a coefficient. Average C values can be calculated from manufacturers’ data (here we used CVE ECO-fan from Itho bv. 2005b and HRU ECO-fan 3 S B from Itho bv. 2005c).

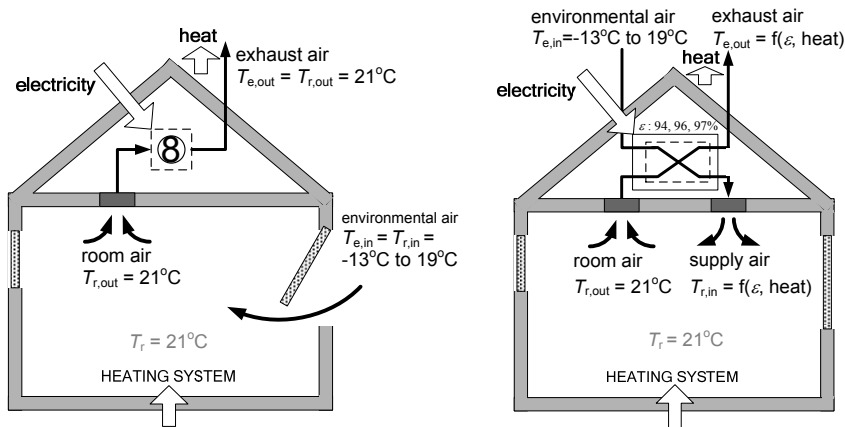


Figure 1: (a, left, reference case) Dwelling ventilation: mechanical exhaust ventilation with natural air supply; (b, right) Dwelling ventilation: balanced ventilation with heat recovery.

$$P = P_{f, \text{st}} = C\dot{Q}^2 \quad (1)$$

$$P_{f, \text{ke}} = \frac{1}{2} \rho_{\text{air}} \left(\frac{\dot{Q}}{A_{\text{duct}}} \right)^2 \quad (2)$$

$$P_{f, \text{tot}} = P_{f, \text{st}} + P_{f, \text{ke}} \quad (3)$$

$$\dot{P}e_1 = \dot{P}e_2 \left(\frac{D_2}{D_1} \right)^4 \left(\frac{\dot{Q}_1}{\dot{Q}_2} \right)^3 \left(\frac{\rho_1}{\rho_2} \right) \quad (4)$$

The fan total pressures $P_{f, \text{tot}}$ for the AC and DC fans are calculated using equations 1 to 3, where $P_{f, \text{st}}$ is the fan static pressure, $P_{f, \text{ke}}$ is the fan kinetic pressure, ρ_{air} is the air density (assumed constant over the temperature range in Table 1), A_{duct} is the inner duct cross-section area and $\dot{Q}$ is the airflow rate given in Table 1.

The electricity $\dot{P}e$ input to the AC and DC fans is calculated using the fan law in equation 4 (ASHRAE 2000), where D is the fan impeller diameter. The fan law is used to predict performance of the fan when test data (data with subscript 2 in equation 4) are available. The test data for equation 4 come from data at the intersection points where the system line (plotted by using equation 1) intercepts the working line of the fans in the graph of fan static pressure versus airflow rate given by the fan producer (used Itho bv. 2005a, 2005b).

3.2. Thermal energy and exergy demands by ventilation airflows

Thermal energy demand $\dot{E}n_{\text{th}}$ and thermal exergy demand $\dot{E}x_{\text{th}}$ by ventilation airflows are calculated by using equations 5 and 6.

$$\dot{E}n_{\text{th}} = \rho_{\text{air}} \dot{Q} c_{p, \text{air}} (T_{r, \text{out}} - T_{r, \text{in}}) \quad (5)$$

$$\dot{E}x_{\text{th}} = \rho_{\text{air}} \dot{Q} c_{p, \text{air}} \left(T_{r, \text{out}} - T_{r, \text{in}} - T_e \ln \left(\frac{T_{r, \text{out}}}{T_{r, \text{in}}} \right) \right) \quad (6)$$

where $T_{r, \text{out}}$ is the temperature of the air leaving the room, equal to room air temperature ($T_r=21^\circ\text{C}$), $T_{r, \text{in}}$ is the temperature of air supplied to the room and T_e is environmental air temperature.

For the mechanical exhaust ventilation, $T_{r, \text{in}}=T_e$, between -13°C and 19°C . For the balanced ventilation with heat recovery, the supply air temperature $T_{r, \text{in}}$ is calculated by using heat exchanger thermal effectiveness ε (ASHRAE 1993) and heat balance equations, assuming the same airflow rates though the HRU for the supply air and the exhaust air.

$$T_{r, \text{in}} = T_e + \varepsilon(T_r - T_e) \quad (7)$$

where T_e is environmental air temperature and T_r is room air temperature.

3.3. Total energy and exergy demands by ventilation airflows

The total energy demand by ventilation airflow $\dot{E}n$, is determined by adding thermal energy demand by ventilation airflow $\dot{E}n_{\text{th}}$ (equation 5) and electricity input to the ventilation unit $\dot{P}e$ (equation 4). The total exergy demand by ventilation airflow $\dot{E}x$, is determined by adding thermal exergy demand by ventilation airflow $\dot{E}x_{\text{th}}$ (equation 6) and electricity input to ventilation unit $\dot{P}e$ (equation 4).

Table 3 shows electricity inputs in relation to the ventilation airflow rates of the applied fans and the HRU, calculated according to equations 1 to 4.

Table 3. Electricity inputs to the fans and the HRU $\dot{P}e$ versus ventilation airflow rates $\dot{Q}$.

	$\dot{Q}$ [m ³ /s]	$\dot{P}e$ [W]
AC fan	0.028	14
	0.042	35
	0.063	45
DC fan	0.028	6
	0.042	8
	0.063	22
DC HRU	0.028	28
	0.042	47
	0.063	110

The electricity input to the fans and the HRU depends on the ventilation airflow rate and on the pressure loss, but is not directly affected by environmental air temperature. The inputs of exergy and energy for electricity are identical because electric energy can in theory be totally converted into mechanical work.

3.4. Daily operation profiles

Daily energy and exergy demands by ventilation airflows for representative days in the heating season are calculated and discussed. In this paper, Figure 2 shows three hourly environmental air temperature profiles, for winter days of maximum, minimum and intermediate mean daily environmental air temperatures. Climate data for De Bilt, the Netherlands are taken from the TMY2 weather data (NREL 1995).

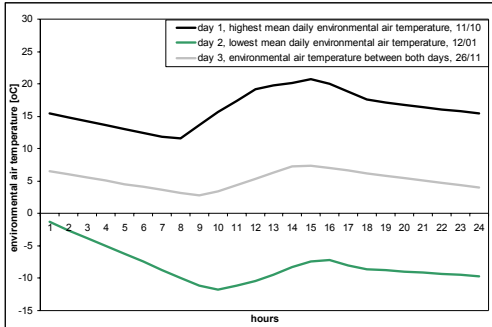
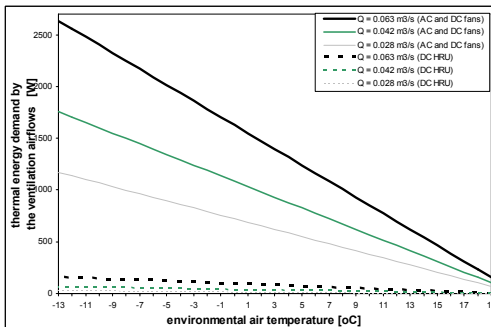


Figure 2. Hourly environmental air temperature profiles in 3 winter days in De Bilt, the Netherlands.

The hourly ventilation unit operation plan for these three days is given in Table 4.

Table 4. Hourly operation plan of the dwelling ventilations.

hour	ventilation airflow rate [m ³ /s]
0:00-8:00	0.028
8:00-9:00	0.042
9:00-17:00	0.063
17:00-18:00	0.042
18:00-24:00	0.028



4. Results and Discussion

This chapter presents energy and exergy analysis results for the applied fans and the HRU in terms of heat and electricity, on an instantaneous and on a daily basis.

4.1. Thermal energy and thermal exergy demands by ventilation airflows

Figure 3 shows thermal energy $\dot{E}_{n_{th}}$ and thermal exergy $\dot{E}_{x_{th}}$ demands by the ventilation airflows, as a function of environmental air temperature and ventilation airflow rate. Using the AC fan or DC fan in the mechanical exhaust ventilation has no effect on the demand for thermal energy or exergy as long as the fans operate between the same supply and exhaust air temperatures and at the same airflow rate. Ventilating at higher airflow rate increases thermal energy and exergy demands in all cases, because the ventilation heat demand depends on the ventilation airflow rate.

In absolute terms, the thermal exergy demand is much lower than the thermal energy demand because the temperature differences involved are relatively small (note that the graphs have different scales). However, the thermal energy demand varies linearly with environmental air temperature, while the thermal exergy demand does not. Hence, in relative terms the thermal exergy demand increases more strongly than the thermal energy demand for lower environmental air temperatures.

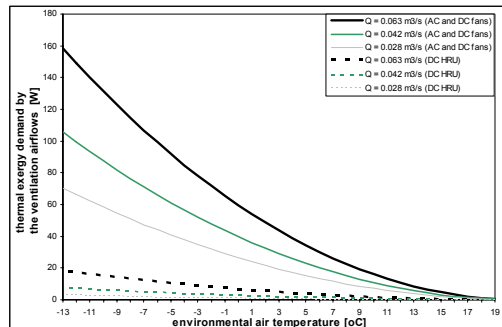


Figure 3. (a, left) Thermal energy demand by the ventilation airflows $\dot{E}_{n_{th}}$ versus T_e ; (b, right) Thermal exergy demand by the ventilation airflows $\dot{E}_{x_{th}}$ versus T_e .

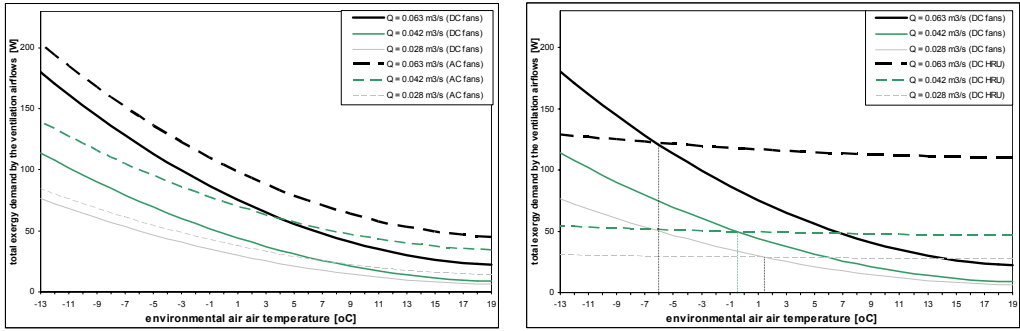


Figure 4. Total exergy demand by ventilation airflow ($\dot{E}_x$) of the ventilation systems versus T_e :
(a, left) mechanical exhaust ventilation with natural air supply;
(b, right) balanced ventilation with heat recovery.

Figure 4 illustrates the total exergy demand by ventilation airflow $\dot{E}_x$ of the ventilation systems as a function of environmental air temperature and ventilation airflow rate. The smooth lines represent the total exergy demand of the mechanical exhaust ventilation using the DC fan. The dashed lines represent the total exergy demand of the mechanical exhaust ventilation using the AC fan (Figure 4a) and the total exergy demand of the balanced ventilation with heat recovery (Figure 4b).

As environmental air temperature T_e increases from -13°C to 19°C , the total exergy demand decreases for a given ventilation airflow rate. The sharpest decrease is for the DC fan, followed by the AC fan and lastly by the DC HRU.

The DC HRU lines in Figure 4b are less sensitive to T_e variations in the range of -13°C to 19°C because the DC HRU requires mainly electricity, while the AC and DC fans require relatively more heat. The DC fan requires less exergy than the AC fan for the entire range of T_e . Compared to the DC HRU, the DC fan requires more exergy at lower T_e , and relatively less exergy as T_e increases towards the room air temperature T_r .

The total exergy demand lines ($\dot{E}_x$) for the DC fan and the DC HRU intersect at a given environmental air temperature ($T_{e,\text{intersect}}$) for a given ventilation airflow rate (Figure 4b). $T_{e,\text{intersect}}$ shows at which environmental air temperature the total exergy demands of two different ventilation systems are equal. These $\dot{E}_x$ lines could be applied for evaluating the total exergy demands of different ventilation systems at different environmental air temperatures. For example, at $T_e = -10^\circ\text{C}$, the balanced ventilation with the DC HRU at $0.063 \text{ m}^3/\text{s}$

ventilation airflow rate uses ca. 125 W, while the mechanical exhaust ventilation with the DC fan at the same airflow rate uses ca. 150 W. Nevertheless, at $T_e = -3^\circ\text{C}$ the balanced ventilation with the DC HRU at the same airflow rate uses ca. 120 W, while the mechanical exhaust ventilation with the DC fan uses around 100 W. At $T_e > T_{e,\text{intersect}}$, using the mechanical exhaust ventilation with the DC fan results in less total exergy demand than using the balanced ventilation with the DC HRU at the same airflow rate. Moreover, $T_{e,\text{intersect}}$ increases when operating the ventilation units at lower airflow rate, since electricity input to the DC HRU is less. For lower airflow rates, exergy demand for the balanced ventilation unit with the DC HRU is less than for the mechanical exhaust ventilation with the DC fan.

4.2. Ratio between the total exergy and the total energy demands by ventilation airflows

Figure 5 illustrates ratios between total exergy $\dot{E}_x$ and total energy $\dot{E}_n$ demands by ventilation airflows of the dwelling ventilation systems, for the range of environmental air temperature T_e from -13°C to 19°C .

In Figure 5a, the $\dot{E}_x/\dot{E}_n$ ratio decreases with increasing environmental air temperature T_e , on the left part of the graph (heat dominated part). This reflects the fact that exergy demand due to ventilation and heat exchange $\dot{E}_{x,\text{th}}$ decreases with rising T_e . On the right part of the graph, the $\dot{E}_x/\dot{E}_n$ ratio increases steeply as T_e rises.

This is because of a significant increase in the relative share of electricity input: although the

thermal exergy demand $\dot{E}_{x_{th}}$ decreases sharply, as shown in Figure 3b, the electric power $\dot{P}_e$ required for driving the fans remains unchanged. Because exergy assigns a higher value to electricity than to low grade heat, when T_e increases, the total exergy demand decreases at a much slower rate than the total energy demand.

In Figure 5b, the electricity input to drive the balanced ventilation with heat recovery always exceeds the thermal exergy required in ventilation and heat transfer, for the entire range of environmental air temperatures T_e between -13°C and 19°C .

Moreover the $\dot{E}_x/\dot{E}_n$ ratios in Figure 5b are bigger than in Figure 5a, for the same environmental air temperatures and ventilation airflow rates (note that the graphs have different scales). This is because the electricity input to the balanced ventilation is significantly bigger than the thermal exergy required in the balanced ventilation.

4.3. The total energy and the total exergy demands in a winter day

The previous items focused on instantaneous energy and exergy values in relation to environmental air temperatures. This item presents examples of cumulative energy and exergy demands by dwelling ventilations, in the course of three representative winter days in De Bilt, the Netherlands (Figure 2).

Daily energy and exergy demands by the dwelling ventilation units for the hourly ventilation plan (Table 4) are given in Table 5.

Thermal energy and exergy demands is highest on the coldest day (day 2) and lowest on the warmest

day (day 1). Electricity for the AC fan, the DC fan, and the HRU are the same for all three days, because the hourly ventilation operation is the same in all three cases.

Table 5. Daily energy and exergy demands (heat and electricity).

		$\dot{E}_{n_{th}}$ [kWh]	$\dot{E}_{x_{th}}$ [kWh]	$\dot{P}_e$ [kWh]	Total $\dot{E}_n$	Total $\dot{E}_x$
day 1 warm	AC	4.891	0.054	0.565	5.456	0.618
	DC	4.891	0.054	0.193	5.084	0.247
	HRU	0.174	0.004	0.985	1.159	0.989
day 2 cold	AC	30.823	1.603	0.565	31.388	2.168
	DC	30.823	1.603	0.193	31.016	1.796
	HRU	1.126	0.112	0.985	2.111	1.097
day 3 avg.	AC	16.571	0.456	0.565	17.136	1.021
	DC	16.571	0.456	0.193	16.764	0.649
	HRU	0.601	0.032	0.985	1.586	1.017

Within each day, the thermal exergy demand is the same for the AC and DC fans, because they operate between the same temperatures. In all three days, the total energy demand is lowest for the HRU, due to the significantly lower thermal energy demand. The total exergy demand, on the other hand, is lower for the HRU only on the coldest day (day 2). On the warmer (day 1) and average (day 3) days, the total exergy demand is lower for the DC fan, which uses the least electricity.

In terms of total exergy demand, these results indicate that it is worthwhile to reduce the thermal exergy demand on the colder day by using heat recovery. On milder days, however, the high-valued electricity required by the HRU exceeds the savings in low-grade heat obtained by using a HRU to preheat ventilation air.

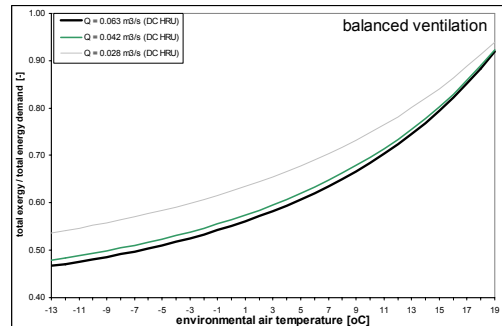
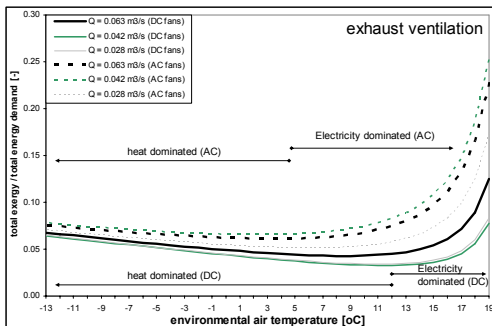


Figure 5. Ratios between the total exergy and the total energy demands $\dot{E}_x/\dot{E}_n$ by ventilation airflows versus T_e :
(a, left) mechanical exhaust ventilation with natural air supply;
(b, right) balanced ventilation with heat recovery.

These results consider the total exergy demand, which is different from the total exergy consumption. While demand is annotated to final use, consumption is considered as a flow from primary energy transformation to final energy delivery for end use. Because energy conversion flows could have different configurations for heat and electricity production, these forms of energy could thus have different impact on irreversibilities throughout the whole energy flows.

5. Conclusions

Exergy analysis assigns values to thermal energy according to its temperature level: systems closer to environmental air temperature have lower thermal exergy. Electricity has a high exergy value because it can directly be converted into work. In terms of exergy the amount of electricity input to fans and heat recovery unit (HRU) is much more significant than in terms of energy, because electricity has a higher exergy value than thermal energy.

This paper presented steady-state energy and exergy analyses for dwelling ventilation with and without air to air heat recovery from ventilation airflow, for dwelling ventilation systems using exhaust ventilation with and without air to air heat recovery in winter conditions in the Netherlands.

At lower environmental air temperatures T_e , mechanical exhaust ventilation with natural air supply has higher total exergy demand than balanced ventilation with heat recovery. In this range, there is relatively more demand for thermal exergy than for electricity. This trend reverses as T_e increases towards the room air temperature, as the heat demand decreases. The balanced ventilation with heat recovery is less sensitive to T_e , since it requires mainly electric exergy.

From the viewpoint of total exergy consumption at room level, it could make sense to use the HRU only when T_e is low enough to compensate the additional need for electricity and let ventilation air bypass the HRU, or if possible to operate the HRU at low ventilation airflow rate in order to decrease the electricity input when T_e is not too low. Nevertheless, the ventilation airflow rate must be qualified to guarantee the indoor occupancy conditions.

In terms of energy demand, the balanced ventilation system with HRU is a better alternative for dwelling ventilation in the heating season in the Netherlands,

since the system needs less energy. However, in terms of exergy demand, the HRU requires more exergy because of the additional electricity input. The thermal exergy recovered from the exhaust air is relatively small, since the exhaust air is relatively close to the environmental air temperature, and thus has low thermal exergy.

This analysis considered the final exergy demand at room level, without focusing on the efficiency of energy conversion and delivery processes (electricity, heating system). This conclusion may change if instead of the exergy demand the exergy consumption is considered, including the exergy efficiencies of electricity and heat production.

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r Room air
th Thermal

Abbreviations

AC Alternating Current
air Air
DC Direct Current
HRU Heat Recovery Unit
NL The Netherlands

Nomenclature

A Area [m^2]
 c_p Specific heat capacity [$\text{m}^2\text{s}^{-2}\text{K}^{-1}$; $\text{Jkg}^{-1}\text{K}^{-1}$]
 C Coefficient [-]
 D Fan impeller diameter [m]
 $\dot{E}_n$ Energy [$\text{m}^2\text{kg}\cdot\text{s}^{-3}$; W]
 $\dot{E}_x$ Exergy [$\text{m}^2\text{kg}\cdot\text{s}^{-3}$; W]
 P Pressure drop of the system [$\text{m}^{-1}\text{kg}\cdot\text{s}^{-2}$; Pa]
 $P_{f,\text{tot}}$ fan total pressures [$\text{m}^{-1}\text{kg}\cdot\text{s}^{-2}$; Pa]
 $P_{f,\text{st}}$ fan static pressure [$\text{m}^{-1}\text{kg}\cdot\text{s}^{-2}$; Pa]
 $P_{f,\text{ke}}$ fan kinetic pressure [$\text{m}^{-1}\text{kg}\cdot\text{s}^{-2}$; Pa]
 $\dot{P}_e$ Electricity [$\text{m}^2\text{kg}\cdot\text{s}^{-3}$; W]
 $\dot{Q}$ Ventilation airflow rates [m^3/s]
 T Air temperature [K; °C with notation]

Greek letters

ε Thermal effectiveness [-]
 ρ Density [kgm^{-3}]

Subscripts

air Air
duct Duct
e Reference environment state; dead state
in Inlet
out Outlet

How useful is exergy analysis of buildings and building services?

Abstract

The paper introduces an extended method for energy and exergy analysis of buildings and building services, according to an energy demand build-up model from the building side to the energy supply side. The method is intended to enable building designers (and building engineers) to compare between the impact of improvements in the building envelope and in building services. Some dynamic energy and exergy calculations are used, as examples, to demonstrate the method and to study the effects of changing some building parametric values and the effects of supplying thermal energy at low and high temperature levels. The calculation results are applicable for buildings in a cold climate.

Keywords: exergy; built environment; building; HVAC system.

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Brief biographical notes: P. Sakulpipatsin is a Doctoral Researcher in Climate Design Group, at the Faculty of Architecture, the Delft University of Technology, The Netherlands. He obtained two Master Degrees: in 2003 MSc in Energy Conversion and Management from the University of Applied Sciences Offenburg, Germany; and in 2001 MSc in Computer Science from Chulalongkorn University, Thailand.

L.C.M. Itard is a Researcher in the field of sustainable buildings and HVAC equipment and leads the group sustainable and healthy building of the Research Institute OTB at the Delft University of Technology, The Netherlands.

H.J. van der Kooi is an Assistant Professor in the Field of Applied Thermodynamics and Chemical Engineering at the Delft University of Technology, The Netherlands.

E.C. Boelman has a background in Adsorption Cooling, Building Engineering and Management, and has been working on renewable energy and exergy analysis applied to buildings for several years.

1. Introduction

Buildings account for more than one third of the world's primary energy demand (ECBCS, 2007), and a substantial share of this energy is used to maintain room air temperatures at around 20°C. Because of the low temperature level, the actual demand for exergy in space heating and cooling applications is low. In most cases, however, this demand is met by high grade energy sources, such as fossil fuels or electricity. The building sector therefore has a high potential for improving the quality match between energy supply and demand.

Exergy analysis has been utilized in the optimization of thermal processes in power plants and in industry (Kotas, 1985; Szargut et al., 1988; Ahern 1980). The exergy concept has been applied to the built environment as well (Shukuya, 1996; Dincer, 2002; Schmidt and Shukuya, 2003; Annex37, 2002; Schmidt, 2004; Asada and Boelman, 2004; Itard, 2005; Arslan and Kose, 2006; Sakulpipatsin et al., 2007a; Hepbasli, 2007; Gunerhan and Hepbasli, 2007; Öztürk, 2004). Research projects in the past have mostly focused on exergy analysis applied to HVAC systems, but only a very limited amount on physical aspects of buildings, building processes (e.g. building design, construction, use, demolition etc.) and building wastes. Additionally, there have been only a few research studies on the relevance of the exergy concept for design of buildings and building services. As a consequence, exergy analysis is only used by a small group of people in the building profession at this moment. Rosen (2002) collected some reasons why it is not yet widely accepted by the building industry at present. Exergy methods might seem cumbersome or complex (e.g. choosing a suitable reference environment), and the results might seem difficult to interpret and to understand.

This paper proposes a detailed analysis of the usefulness of applying exergy analysis to buildings and building services. The analysis is based on a build-up model from the energy demand of the building to the energy supply side which accounts for both heating and cooling demands of the building. The build-up model was developed by Sakulpipatsin et al. (2006) and Bezuijen (2006), and based on a model developed by Schmidt (2004) and in accordance with the European Standard EN ISO 13790 (2004). It has been further developed in this work. Different kinds of improvements in the building envelope and in building services are compared for a building in a cold climate. The thermal exergy and thermal energy demands of the building and thermal energy and thermal exergy losses in the building services when parametric values of the building and the building services are changed are investigated in order to draw conclusions about the usefulness of exergy analysis related to buildings.

This paper first describes the framework for the energy and exergy analysis in item 2. The energy flow model from demand to supply side is explained in item 2.1. The methods for energy and exergy analysis of a building and building services are described in items 2.2 and 2.3 respectively. The methods are applied for a reference building and building services, where the details of the analysis are given in item 3.1. Description of the reference building is given in item 3.2. Analysis results of the building are shown in item 4.1. These results are used for comparison with other analysis results of the building when building parametric values of the building are changed. Results of the comparison are shown in items 4.2 to 4.7. Item 5 presents exergy losses in the building services when they have different temperature levels of the supplied thermal energy. Item 5.4 summarises the uses of the exergy concept in the building and the built environment with a design example for building services. Item 6 concludes the usability and usefulness of the exergy analysis when the exergy concept is applied for building and building services design.

2. Basic framework for the energy and exergy analysis

2.1. *The energy flow model from demand to supply side*

The basic framework is used for the energy and exergy analysis of buildings and building services. The model is adapted from the energy flow model for building services equipment, from primary energy supply systems to buildings, developed by Schmidt (2004). The build-up model was further developed by Sakulpipatsin et al. (2006) and Bezuijen (2006), and has been extended in this work. Figure 1 illustrates the build-up of the energy demands within the boundaries of the building to the external energy supply system, via the building services. The build-up model consists of two parts: the local part and the external part. Energy from the external part is supplied to the local part by different energy carriers:

heat gains and therefore influences the thermal demand. In the services part, only the electricity distribution related to electricity needed for the working of the thermal energy emission, control and distribution will be accounted for.

Table 1 Local energy conversion processes

<i>External part</i>	<i>Local part</i>	<i>Energy conversion processes/machines</i>
Thermal energy carriers	Thermal energy	District heating/cooling, active solar heating
	Electricity	Thermocouple
Chemical substances	Thermal energy	Fossil burning, fuel cell, combined heat and power generation
	Electricity	Fuel cell, combined heat and power generation
Electricity/ natural forces	Thermal energy	Heat pump, refrigerator, electromagnetic heat
	Electricity	Electricity transformer, wind turbine, photovoltaic

2.2. General mathematical description of the building thermal energy model

The building model used in this paper is simulated by using the simulation tool TRNSYS version 16 (TRNSYS16, 2004). The model is based on a multi-zone building component and uses energy balances presented in TRNSYS. This component is a non-geometrical balance model with one air node per zone, representing the thermal capacity of the zone air volume. In the non-geometrical balance model any air point in the zone has the same properties (e.g. temperature and humidity). In this section only the principles that are important to understand the results of analysis are presented. More details of the mathematical description of the building thermal energy model could be found in the TRNSYS user manual (TRNSYS16, 2004).

The building model is considered as a one-zone model and all the walls are attached to environment. In this item, first the energy balance for a zone is presented in order to calculate the building thermal energy demand. After that, the energy balance for the building envelope surfaces is presented in order to calculate internal thermal energy change of the surfaces at a time. Then calculation of thermal energy from infiltration and ventilation is presented. These energy balances and calculations are presented as a baseline of the exergy balances and calculations that are described in item 2.3.

2.2.1. Energy balance for a zone

This energy balance refers to the TRNSYS output in the multi-zone building component “NTYPE 904”. The system boundary for this energy balance includes the inside surface nodes of all surfaces of a zone, and therefore the balance also includes all radiative energy flows. The system boundary does not include the inside of any wall, so the energy from an active layer and the stored energy in walls are not part of this balance, but of the detailed balance for surfaces described in item 2.2.2. The energy balance for a zone is shown as equation 1.

$$DQ_{\text{air}}/dt = Q_{\text{heating}} - Q_{\text{cooling}} + Q_{\text{inf}} + Q_{\text{vent}} + Q_{\text{tran}} + Q_{\text{gain}} + Q_{\text{sol}} \quad (1)$$

DQ_{air}/dt is the change of the internal thermal energy of the zone. Q_{heating} and Q_{cooling} are the thermal energy supplied by heating and cooling equipments respectively. Q_{inf} and Q_{vent} are the thermal energy gains from infiltration and ventilation respectively. Q_{tran} is the thermal energy gain by thermal transmission between the inner surface nodes and environment, via the walls. Q_{tran} could be, for example, energy stored in the walls, going to a slab cooling, and directly transmitted. Therefore Q_{tran} is not only the total transmitted part, but contains also the part that has been accumulated in the wall. Q_{gain} is the (convective and radiative) internal thermal energy gain from occupants and appliances. Q_{sol} represents the solar gains, including both direct and diffuse parts, absorbed on the boundary surfaces of the zone. The absorbed solar gains of the inside surfaces of all windows are also taken into account. These absorbed gains may go inside or outside. Following the conventions used in TRNSYS, Q_{heating} and Q_{cooling} are always positive (or zero), whereas Q_{inf} , Q_{vent} and Q_{tran} may be negative or positive, depending on the balances described in items 2.2.2 and 2.2.3. TRNSYS version 16 uses the transfer function method for describing the thermal behaviour of a building (Mitalas, 1972), considering the thermal mass effects of the building envelope. Q_{heating} and Q_{cooling} are calculated as a function of the zone temperature. The zone temperature is free floating in the comfort region where Q_{heating} and Q_{cooling} are zero.

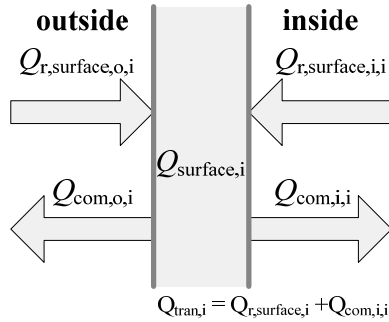
2.2.2. Energy balance for surfaces

This energy balance refers to the TRNSYS output in the multi-zone building component “NTYPE 906”. The balance shows the detailed energy balance for surfaces, as equation 2.

$$DQ_{\text{surface}}/dt = -Q_{\text{com},i} + Q_{\text{com},o} + Q_{r,\text{wall},i} + Q_{r,\text{wall},o} \quad (2)$$

DQ_{surface}/dt is the change of the internal thermal energy of the surfaces. The thermal energy represents heat accumulated in the walls. $Q_{\text{com},i}$ and $Q_{\text{com},o}$ are the combined (convective and radiative) thermal energy flows into the zone (going into zone +; going into building envelope -), and from outside to the building envelope (going to outside -; going into building envelope +) respectively. $Q_{r,\text{surface},i}$ and $Q_{r,\text{surface},o}$ are the total radiative gains for the inside and outside nodes on building envelope surfaces respectively. The summation of $Q_{\text{com},i}$ and $Q_{r,\text{surface},i}$ is equal to Q_{tran} in equation 1. Therefore DQ_{surface} is the amount of energy that is absorbed in the surfaces in the time interval dt , and the total amount of $Q_{\text{com},o}$ and $Q_{r,\text{surface},o}$ is released from the outer surfaces to environment. Figure 2 illustrates the energy balance for a surface.

Figure 2 Energy balance for a surface



2.2.3. Thermal energy from infiltration and ventilation

Thermal energy values per second from infiltration and ventilation airflows ($\dot{Q}_{\text{inf}}$ and $\dot{Q}_{\text{vent}}$ respectively) are calculated by using equations 3 and 4.

$$\dot{Q}_{\text{inf}} = \dot{m}_{\text{inf}} c_p (T_i - T_o) \quad (3)$$

$$\dot{Q}_{\text{vent}} = \dot{m}_{\text{vent}} c_p (T_{\text{vent,out}} - T_{\text{vent,in}}) \quad (4)$$

where $\dot{m}_{\text{inf}}$ and $\dot{m}_{\text{vent}}$ are the infiltration and the ventilation airflow rates (kilogram per second), c_p is the heat capacity of air (assumed constant at $1.008 \text{ kJkg}^{-1}\text{K}^{-1}$), T_o and T_i are air temperatures outside and inside the building, $T_{\text{vent,out}}$ and $T_{\text{vent,in}}$ are air temperatures at the ventilation outlet and inlet.

2.3. Thermal exergy calculation method for the building thermal energy model

In this study, thermal exergy values are derived from physical exergy using equations 5 and 6. It is assumed that there is no chemical reaction occurring and no pressure change. The validity of these assumptions for buildings and building services were discussed in detail in (Sakulpipatsin et. al., 2007a). When temperature of the thermal energy source T is constant equation 7a is used. When T is not constant, equation 7b is used. This method is applicable for thermal exergy calculations in a steady state process, assumed that contributions such as from kinetic energy and potential energy can be neglected or do not play a role at all to exergy values (Sakulpipatsin et. al., 2007a).

$$dE_{ph} = c_p dT - \frac{c_p T_o}{T} dT + \frac{RT_o}{P} dP \quad (5)$$

When $dP = 0$,

$$dE_{ph} = dE_{th} = c_p dT - \frac{c_p T_o}{T} dT \quad (6)$$

For a source at T_i , the change in thermal exergy can be related to δQ_{rev}

$$E_{th} = Q_{rev} \left(1 - \frac{T_o}{T} \right) \quad (7a)$$

For a change in thermal conditions from T_1 to T_2 , the change in thermal exergy is:

$$\Delta E_{th} = c_p \left((T_2 - T_1) - T_o \ln \left(\frac{T_2}{T_1} \right) \right) \quad (7b)$$

According to the energy balance for a zone in item 2.2.1, thermal exergy values of the thermal energy are calculated as given in equations 8 to 15.

For DQ_{air} :

$$E_{th,air} = DQ_{air} \left(1 - \frac{T_o}{T_i} \right) \quad (8)$$

For $Q_{heating}$ and $Q_{cooling}$:

$$E_{th,heating} = Q_{heating} \left(1 - \frac{T_o}{T_i} \right) \quad (9)$$

$$E_{th,cooling} = \left| Q_{cooling} \left(1 - \frac{T_o}{T_i} \right) \right| \quad (10)$$

In equations 8 to 10, T_o and T_i are assumed constant during the time interval considered.

For Q_{inf} and Q_{vent} :

$$E_{inf} = Q_{inf} \left(1 - \frac{T_o}{(T_i - T_o)} \ln \left(\frac{T_i}{T_o} \right) \right) \quad (11)$$

$$E_{vent} = Q_{vent} \left(1 - \frac{T_o}{(T_{vent,out} - T_{vent,in})} \ln \left(\frac{T_{vent,out}}{T_{vent,in}} \right) \right) \quad (12)$$

For Q_{tran} , because the boundary of the energy balance for a zone in equation 1 does not include the walls (but only the inside wall surface nodes) E_{tran} is calculated as the summation of the thermal exergy value of $Q_{tran,i}$ at $T_{surface,i}$ on all the walls:

$$E_{th,tran} = \sum_{i=1}^n E_{th,tran,i} = \sum_{i=1}^n \left(Q_{tran,i} \left(1 - \frac{T_o}{T_{surface,i}} \right) \right) \quad (13)$$

For Q_{gain} , thermal exergy values of Q_{gain} depend on temperatures of the thermal energy sources $T_{source,i}$ that are assumed constant:

$$E_{th,gain} = \sum_{i=1}^n E_{th,gain,i} = \sum_{i=1}^n \left(Q_{gain,i} \left(1 - \frac{T_o}{T_{source,i}} \right) \right) \quad (14)$$

For Q_{sol} , thermal exergy value of Q_{sol} is calculated in the same way as calculation of $E_{th,gain}$ (Jeter, 1981):

$$E_{th,sol} = Q_{sol} \left(1 - \frac{T_o}{T_{sun}} \right) \quad (15)$$

The exergy losses of the interaction of solar radiation and the atmosphere are considered to be negligible in equation 15. Alternatively, various approaches to the exergy calculation of solar radiation have been developed. These approaches are reported by Petela (2003) and Candau (2003). From these literatures, researchers focus mainly on the three radiation exergy relations, as derived by Petela (1961), Spanner (1964) and Jeter (1981). Petela (1961) derived the formulae for the exergy of thermal radiation. Spanner (1964) proposed his approximate formula for the direct solar radiation exergy. Jeter (1981) based on the analysis of heat engine, came to the result that the exergy of heat radiation is determined by the result that the exergy of heat radiation is determined by the Carnot efficiency. According to other researchers, no new concepts on the heat radiation exergy formula have been brought to date except developing the existing results with some specifics for solar radiation (Petela, 2003). These three approaches give approximately the same result of exergy value of solar radiation, calculated where $T_o=293.15K$ and $T_{sun}=6000K$. The difference between the results is in maximum 2% of that calculated by using the Jeter approach. So, for the sake of simplicity, the Jeter approach is used in this work (equation 15).

According to the energy balance for surfaces in item 2.2.2, thermal exergy value in the wall i ($E_{th,surface,i}$), thermal exergy value of thermal energy flow from the inside surface i to the zone air ($E_{th,com,i,i}$) and thermal exergy value of thermal energy flow from the outside surface i to environment ($E_{th,com,o,i}$), are calculated by using equations 16 to 18. The thermal exergy values $E_{th,com,i,i}$ and $E_{th,com,o,i}$ are calculated for the convective and radiative parts (Q_{conv} and Q_{rad}) separately. Equation 7a and the equation of Petela (2003) are used for the calculations in the convective and radiative parts respectively.

$$E_{th,surface,i} = DQ_{surface,i} \left(1 - \frac{T_o}{(T_{surface,i,i} - T_{surface,o,i})} \ln \left(\frac{T_{surface,i,i}}{T_{surface,o,i}} \right) \right) \quad (16)$$

$$E_{th,com,i,i} = Q_{conv,i,i} \left(1 - \frac{T_o}{(T_{surface,i,i} - T_{star})} \ln \left(\frac{T_{surface,i,i}}{T_{star}} \right) \right) + Q_{rad,i,i} \left(1 - \frac{4}{3} \frac{T_o}{T_{surface,i,i}} + \frac{1}{3} \left(\frac{T_o}{T_{surface,i,i}} \right)^4 \right) \quad (17)$$

where T_{star} is the star temperature. The star temperature T_{star} is an artificial temperature. It is defined by Seem (1987) to calculate an energy value of the long-wave radiation exchange between the surfaces within the zone and the convective thermal energy flows from the inside surfaces to the zone air. This method considers the parallel energy flow from a wall surface by convection to the air node and by radiation to other wall and window elements.

$$E_{th,com,o,i} = Q_{conv,o,i} \left(1 - \frac{T_o}{(T_{surface,o,i} - T_o)} \ln \left(\frac{T_{surface,o,i}}{T_o} \right) \right) + Q_{rad,o,i} \left(1 - \frac{4}{3} \frac{T_o}{T_{surface,o,i}} + \frac{1}{3} \left(\frac{T_o}{T_{surface,o,i}} \right)^4 \right) \quad (18)$$

$DQ_{surface,i}$ is the accumulated thermal energy in the wall i , $T_{surface,i,i}$ and $T_{surface,o,i}$ are the inside and the outside surface temperatures respectively. $Q_{com,i,i}$ is the (convective and radiative) thermal energy flow from the inside surface to the zone air and $Q_{com,o,i}$ is the (convective and radiative) thermal energy flow to the outside surface from environment.

In addition, thermal exergy value in the window is zero, since thermal energy is assumed not to be accumulated in the window for the study.

3. Calculation method and description of the reference building

3.1. Calculation method

Item 4 focuses on the energy and exergy demands in the reference building (item 3.2), and item 5 focuses on the demands in the emission, control, distribution and local energy conversion parts of building services. The analysis in item 5 is based on calculation on year-basis for several types of building services including low and high temperature emission & control systems for heating and cooling, losses in the electricity distribution system and in the local energy conversion system. Finally an example of optimization will be given.

The analysis in item 4 is based on energy and exergy calculation made for two hours (one in a winter day and another one in a summer day) in a cold sea climate in the Netherlands. The calculations are carried out by using the simulation tool TRNSYS and the TMY2 weather data source (NREL, 1995). Furthermore, calculations of energy and exergy values for heating and cooling the reference building are carried out for a complete TMY. The results are then accumulated for one year and presented in terms of energy and exergy demand, in order to investigate the changes of the yearly values of the energy and the exergy values in relation to the changes of the building parametric values. Magnitudes and locations of thermal energy and thermal exergy flows through the building are pinpointed, in order to minimise thermal exergy and thermal energy losses. The differences between the thermal energy and thermal exergy values in the building, when changing some building parametric values, are also studied. This is because thermal energy and thermal exergy losses of a building could be extremely minimized when having a good building operation plan (e.g. for ventilation and for sun protection). The building operation could be in a dynamic way, but the building envelope construction is fixed in the building construction phase and can not be changed in the building operation phase, unless the building would be renovated in the building operation phase. Having different building envelope affects the thermal demands for the entire building lifetime.

The parameters studied are:

- Wall and floor thickness at a constant overall U coefficient
- Insulation thickness at a constant wall thermal mass
- U-value and g-value of windows
- Size of windows

The effects in building geometry and orientation are not considered in this work.

The thermal energy and thermal exergy values presented in this item are based on the thermal energies in the energy balance for a zone (item 2.2.1), and therefore thermal exergy values $E_{th,surface,i}$, $E_{th,com,i,i}$ and $E_{th,com,o,i}$ (equations 16 to 18) are not mentioned in item 4. The equations are applied to a reference building. Because of the parametric study, the results can be generalized however.

3.2. Description of the reference building

For this study, a simple reference building is chosen as a cubic box that is 10 m height and has 100 m² of gross ground floor area. The net floor area of the whole building is 300 m². The cubic-boxed form is

selected for this study, in order to eliminate the effects in building geometry and orientation. The physical properties of the building envelope components are presented in Table 2 and Table 3.

Table 2 Physical properties of the building envelope components

No.	Function	Orientation	Area [m^2]	Thickness [m]	U-value [$Wm^{-2}K^{-1}$]	g-value[-]
1	wall	north	100.00	0.170	0.511	-
2	wall	south	100.00	0.170	0.511	-
3	wall	east	100.00	0.170	0.511	-
4	wall	west	100.00	0.170	0.511	-
5	ceiling	horizontal	100.00	0.141	0.316	-
6	floor	horizontal	100.00	0.080	0.040	-
7	Window (double glazing)	south	42.50 (glazing) 7.50 (frame)	-	1.300 (glazing) 2.269 (frame)	0.591

Table 3 Physical properties of the building envelope components (continued)

No.	Solar absorptance [-]		Convection heat transfer coefficient [$Wm^{-2}K^{-1}$]		F_{sky} [-]
	Inside	Outside	Inside	Outside	
1	0.6	0.6	3.160	24.670	0.5
2	0.6	0.6	3.160	24.670	0.5
3	0.6	0.6	3.160	24.670	0.5
4	0.6	0.6	3.160	24.670	0.5
5	0.6	0.6	3.160	24.670	1.0
6	0.6	0.6	3.160	0.000	0.0
7	0.6	0.6	3.160	24.670	0.5

Figures 3 to 5 present the configurations used for the building envelope components (wall, ceiling and floor). These configurations are based on data provided by TRNSYS (TRNSYS16, 2004).

The building is used from Monday to Friday during 8:00-18:00. It is occupied by 30 people. One person has one own appliance unit, consisting of one computer and one screen. The sensible thermal energy rate emitted by the people is 75 Watt per person. The thermal energy rate emitted by the appliance units is 230 Watt per appliance unit. The total thermal energy gain from artificial lightings is 400 Watt. These thermal energy values are assumed constant during the opening times of the building. When the building is closed these internal gains are assumed to be zero.

The building is naturally ventilated with a rate of 4 (m^3/hr)/ m^3 during the opening time and a rate of 0 during closing time. The infiltration airflow rate is assumed constant at 0.6 (m^3/hr)/ m^3 all over the day. There is no mechanical ventilation and no (de)humidification in the building.

The indoor air temperature is maintained between 20°C and 26°C during the occupied time and between 10°C and 35°C during the unoccupied time. The following temperatures are assumed constant for the calculations: the body skin temperature is 306 K; the temperatures of sun and artificial lighting are 6000 K; and the appliance surface temperature is 313 K. The initial temperature of the indoor air is assumed at 20°C, and used for the first calculation. The initial temperatures at all inside and outside surface nodes are assumed equal to the initial temperature of the indoor air.

Figure 3 Wall configuration

Outside	Thickness [m]	Conductivity [$W m^{-1} K^{-1}$]	Capacity [$J kg^{-1} K^{-1}$]	Density [$kg m^{-3}$]
wood	0.009	0.140	900	530
foam insulation	0.061	0.040	1400	10
concrete block	0.10	0.510	1000	1400
Inside				

Figure 4 Ceiling configuration

Outside	Thickness [m]	Conductivity [W m ⁻¹ K ⁻¹]	Capacity [J kg ⁻¹ K ⁻¹]	Density [kg m ⁻³]
wood	0.019	0.140	900	530
fiber glass	0.112	0.040	840	12
plaster board	0.010	0.160	840	950
Inside				

Figure 5 Floor configuration

Outside	Thickness [m]	Conductivity [W m ⁻¹ K ⁻¹]	Capacity [J kg ⁻¹ K ⁻¹]	Density [kg m ⁻³]
insulation	N/A	massless material ¹ : resistance = 25.075 m ² K W ⁻¹		
concrete slab	0.080	1.130	1000	1400
Inside				

4. Thermal energy and thermal exergy demands of the reference building

4.1. Energy and exergy analysis of the reference building

Table 4 shows the values of thermal energy and thermal exergy (sorted by input and output) in the reference building based on the system boundary of the energy balance for a zone in item 2.2.1. These values are accounted for an hour in a winter day (when the calculated hourly thermal energy demand for heating the building is the highest one; the 274th hour of the year) and a summer day (when the calculated hourly thermal energy demand for cooling the building is the highest one; the 4096th hour of the year).

Table 4 Thermal energy and thermal exergy (Q_{th} and E_{th}) in the reference building for the 274th hour of the TMY ($T_i=20^\circ\text{C}$, $T_o=-11.77^\circ\text{C}$; in a winter day) and for the 4096th hour of TMY ($T_i=26^\circ\text{C}$, $T_o=27.36^\circ\text{C}$; in a summer day)

items	Q_{th} [kJ]				E_{th} [kJ]			
	274th hour		4096th hour		274th hour		4096th hour	
	input	output	input	output	input	output	input	output
heating/cooling	205200	0	0	49930	22239	0	0	228
infiltration	0	23230	997	0	0	1307	2	0
ventilation	0	154800	6647	0	0	8709	15	0
transmission	0	61300	0	12110	0	6140	0	146
internal gain	34380	0	34380	0	6655	0	2504	0
solar gain	253	0	21050	0	242	0	19996	0
total	239833	239330	63074	62040	29136	16156	22517	373

In the 274th hour of the year (in winter), the total thermal energy output is dominated by the thermal energy output by ventilation (154.8 MJ). The thermal energy output by ventilation is a significant thermal energy source for a (pre)heating process such as preheating domestic hot water and preheating air entering the building. The thermal energy output by transmission consists of 2 parts: one stored in the building envelope (31.7 MJ; DQ_{surface}/dt in equation 2) and another gone to outside (29.6 MJ; $Q_{\text{com,o}} + Q_{\text{r,surface,o}}$ in equation 2). The thermal energy stored in the building envelope is calculated by using the energy balance equation for surfaces in equation 2. The total thermal energy input is dominated by the thermal energy for heating the building (205.2 MJ; or 86% of the total thermal energy input). The total

¹ Massless: this material type is only used when TRNSYS is not able to create the transfer functions of a wall with only massive layers. In that case this layer type is used for very thin layers where the thermal mass can be neglected.

energy input and output are not equal due to the calculations by TRNSYS, but the differences are small (0.21% for the 274th hour and 1.64% for the 4096th hour).

In the 4096th hour of the year (in summer), the total thermal energy input is dominated by the thermal energy input by internal gain (34.4 MJ) and solar gain (21.1 MJ). The thermal energy output by transmission is 12.1 MJ. However, the thermal energy stored in the building envelope (15.3 MJ) is larger than the thermal energy output by transmission. The total thermal energy output is dominated by the thermal energy for cooling the building (49.93 MJ; or 80% of the total thermal energy output).

From Table 4, the total thermal exergy inputs and the total thermal exergy outputs are not equal for the hours. This means that there is thermal exergy loss occurring in the building. The thermal exergy loss in the hot summer hour (22.14 MJ) is much bigger than the thermal exergy loss in the cold winter hour (12.98 MJ).

In the 274th hour of the year (in winter), the total thermal exergy output is still dominated by the thermal exergy output by ventilation (8.7 MJ; 54% of the total). The thermal exergy outputs by transmission and by infiltrations are 38% and 8% of the total respectively. The total thermal exergy input is dominated by the thermal exergy for heating the building (20.24 MJ; 76% of the total). The relative exergy contains of internal gains is very low. Both the energy and exergy analysis shows that reducing ventilation and transmission losses are key issues.

In the 4096th hour of the year (in summer), the total thermal exergy input is mainly dominated by the thermal exergy input by solar gain (20 MJ; 89% of the total). The total thermal exergy output is much smaller than the total thermal exergy input. From the energy analysis it could be concluded that reducing internal gains is even important as reducing solar gains. However, the exergy analysis shows that there is not much potential present in the internal gains, whereas the solar gains have a high exergetic potential.

From the analysis for the cold and hot hours, it shows that reducing Q_{heating} remains a main concern in the cold days. Whereas the internal gains are important in the energy analysis, the exergy analysis shows that the solar gain creates the main exergy losses when cooling is needed. This means that these solar gains should be minimized, or better captured to be useful somewhere else (for instance for domestic hot water production or electricity generation; Sakulpipatsin et. al., 2007b).

Table 5 shows thermal energy and thermal exergy values for heating and cooling the reference building during the whole year (TMY). These values are summations of thermal energy or thermal exergy values that are calculated for every hour in the TMY. For cooling the results are presented both in the case where cooling is needed whereas the outdoor temperature is still lower than the indoor temperature (internal gains are predominant) and in the case where the outdoor temperature is higher than the indoor temperature.

Table 5 Thermal energy and thermal exergy demands of the reference building in the TMY

		Energy [MJ/a]	Exergy [MJ/a]
heating		114510	6109
cooling	$T_i < T_o$	988	4
	$T_i > T_o$	2217	16

The results in Table 5 show that, in the TMY, the thermal energy demand for heating is much bigger than the thermal energy demands for cooling. The ratio between the thermal exergy demand and thermal energy demand for heating is ca. 0.053. It is much bigger than the ratios for cooling ($4/988=0.004$ when $T_i < T_o$; $16/2217=0.007$ when $T_i > T_o$), because the ratios $Q_{\text{heating}}/E_{\text{th,heating}}$ and $Q_{\text{cooling}}/E_{\text{th,cooling}}$ solely depend on the ratio between T_o and T_i (see equations 9 and 10). In addition, the temperature difference between T_e and T_i (for each hour) in the cooling case is usually smaller than the temperature difference in the heating case. The thermal energy demand for cooling when $T_i > T_o$ is bigger than the demand for cooling when $T_i < T_o$. This means that environmental air is a considerable source to cool the indoor air for the whole TMY. The ratio between the thermal energy demand and the thermal exergy demand for cooling when $T_i > T_o$ is about 1.8 times the ratio for cooling when $T_i < T_o$.

The thermal energy and the thermal exergy values of the exhaust ventilation air ($Q_{\text{th,vent,exhaust}}$ and $E_{\text{th,vent,exhaust}}$) from the reference building in the TMY are also calculated, as summations of $Q_{\text{th,vent,exhaust}}$ or $E_{\text{th,vent,exhaust}}$ values that are calculated for every hour in the TMY. $Q_{\text{th,vent,exhaust}}$ and $E_{\text{th,vent,exhaust}}$ values are calculated by using equations 4 and 12 respectively. For the reference building, $Q_{\text{th,vent,exhaust}}$ and

$E_{th,vent,exhaust}$ in the TMY are 126410 MJ/year and 2835 MJ/year respectively. $Q_{th,vent,exhaust}$ is larger than the thermal energy demand for heating (114510 MJ/year), because $Q_{th,vent,exhaust}$ occurs from temperature difference between inside and outside of the building, and sometimes $Q_{th,vent,exhaust}$ is also produced when the building is cooled (when $T_i > T_o$). When the building is heated, $Q_{th,vent,exhaust}$ and $E_{th,vent,exhaust}$ for the TMY are 106327 MJ/year (84% of the total) and 2720 MJ/year (96% of the total) respectively.

Exhaust ventilation air has been used to preheat inlet ventilation air in balanced ventilation systems and as a thermal energy source in other heating systems, since the exergy value of the exhaust ventilation air is sometimes higher than the exergy values of the inlet ventilation air and of the thermal energy source. Results of exergy assessment of heat recovery in dwelling ventilation systems at a cold climate (De Bilt, NL) could be found in (Sakulpipatsin et. al., 2007c). The assessment presents relative influence of thermal energy and electricity on the exergy demand by the ventilation airflow, on an instantaneous and a daily basis. Furthermore, exergy analysis is applied for selecting a design option for combination between dwelling ventilation and domestic hot water production in winter days (Sakulpipatsin et. al., 2007b). In that work, the exhaust ventilation air is used to preheat domestic hot water, using a heat exchanger or a heat pump. The energy and the exergy demands in winter days for De Bilt, the Netherlands, are shown at the component level, in terms of thermal energy and electricity, for the systems. The results indicates that for these winter days the total energy demands, but not the total exergy demands, of balanced ventilation and domestic hot water production with preheat by the exhaust ventilation air using a heat pump are lowest. The total exergy demands are dominated by exergy of electricity input to the heat recovery unit and the heat pump.

4.2. Changing the thicknesses of the walls

This study compares thermal energy and thermal exergy values in the reference building by changing the thicknesses of the walls, while the U-values of the walls are kept constant (at ca. $0.511 \text{ Wm}^{-2}\text{K}^{-1}$). For each change, the thicknesses of all the walls will be changed at the same values. The cases considered are as following.

- Case A: 0.10 m of concrete block, 0.061 m of foam insulation, and 0.009 m of wood (the reference case; the results are presented in Table 4)
- Case B: 0.20 m of concrete block, 0.053 m of foam insulation, and 0.009 m of wood
- Case C: 0.30 m of concrete block, 0.045 m of foam insulation, and 0.009 m of wood
- Case D: 0.40 m of concrete block, 0.038 m of foam insulation, and 0.009 m of wood

For all the cases, the energy and exergy analyses for the 274th hour and the 4096th hour of the TMY show that the thermal energy values and the thermal exergy values by infiltration and ventilation are not sensitive to the changes of the wall thicknesses. This is because the values depend on the airflow rates by infiltration and ventilation that are assumed constant and the indoor air temperatures at the hours are constant at 20°C and 26°C respectively. The thermal energy values by internal and solar gains are constant at the hours as well, because the temperatures of the thermal energy values are assumed constant at the hours. These make the thermal exergy values by the gains constant too. The thermal exergy values for heating and cooling the building depend on their thermal energy values, due to the constant indoor air temperatures at the hours. Therefore, there are only the thermal energy values and the thermal exergy values by transmission and for heating and cooling that are not constant for all the cases. The thermal energy values and the thermal exergy values by transmission and for heating and cooling of all cases are presented in Table 6.

Comparing case D to case A (reference case), the differences (in %) in the thermal energy values for heating and in the thermal exergy values for heating are about 1.41% of the values in the reference case. In addition, the differences (in %) in the thermal energy values for cooling and in the thermal exergy values for cooling are ca. 18% of the values in the reference case. The percentage in the cooling case is larger than the percentage in the heating case, because in the reference case the thermal energy and thermal exergy values for cooling are much smaller than the values for heating.

The thermal energy and thermal exergy demands for heating and cooling of all cases for the whole year TMY are given in Table 7. Comparing case D to case A, all thermal energy and thermal exergy demands for heating and cooling ($Q_{heating}$, $E_{heating}$, $Q_{cooling}$ and $E_{cooling}$) of case D are smaller than the demands in case A.

Table 6 Thermal energy and thermal exergy values (Q_{th} and E_{th}) by transmission and for heating and cooling of all cases for the 274th and 4096th hour of the TMY

Items	Q_{th} [kJ]				E_{th} [kJ]			
	274th hour		4096th hour		274th hour		4096th hour	
	Input	Output	Input	Output	Input	Output	Input	Output
Case A: : 0.10 m of concrete block, 0.061 m of foam insulation, and 0.009 m of wood (ref. case)								
heating/cooling	205200	0	0	49930	22239	0	0	228
transmission	0	61300	0	12110	0	6140	0	146
Case B: : 0.20 m of concrete block, 0.053 m of foam insulation, and 0.009 m of wood								
heating/cooling	202400	0	0	45200	21935	0	0	206
transmission	0	58540	0	17120	0	5889	0	219
Case C: 0.30 m of concrete block, 0.045 m of foam insulation, and 0.009 m of wood								
heating/cooling	202500	0	0	42610	21946	0	0	194
transmission	0	58670	0	19760	0	5901	0	262
Case D: 0.40 m of concrete block, 0.038 m of foam insulation, and 0.009 m of wood								
heating/cooling	202300	0	0	40940	21925	0	0	187
transmission	0	58400	0	21370	0	5876	0	289

Table 7 Thermal energy and thermal exergy demands for heating ($Q_{heating}$ and $E_{heating}$) and cooling ($Q_{cooling}$ and $E_{cooling}$) of all cases for TMY

	$Q_{heating}$ [MJ/a]		$Q_{cooling}$ [MJ/a]		$E_{heating}$ [MJ/a]		$E_{cooling}$ [MJ/a]	
	$T_i > T_o$	$T_i < T_o$	$T_i < T_o$	$T_i > T_o$	$T_i > T_o$	$T_i < T_o$	$T_i < T_o$	$T_i > T_o$
Case A (ref. case)	114510	0	989	2218	6109	0	4	16
Case B	112361	0	891	2035	6029	0	4	16
Case C	111420	0	856	1965	5988	0	3	15
Case D	110542	0	836	1923	5947	0	3	15

4.3. Changing the thickness of the floor

This study compares thermal energy and thermal exergy values in the reference building by changing the thickness of the floor, while the U-value of the floor is kept constant (at ca. $0.040 \text{ Wm}^{-2}\text{K}^{-1}$). The thickness of the floor will be changed by increasing the thickness of the concrete slab from 8 cm to 38 cm at 10 cm intervals, since the thickness of the insulation in the floor is assumed too small and does not have any thermal mass effect. The cases considered are as following.

- Case A: 0.08 m of concrete slab (the reference case; the results are presented in Table 4)
- Case B: 0.18 m of concrete slab
- Case C: 0.28 m of concrete slab
- Case D: 0.38 m of concrete slab

For the 274th hour and the 4096th hour of the TMY, the thermal energy values and the thermal exergy values by infiltration, ventilation, internal and solar gains, are constant for all cases. The thermal exergy values for heating and for cooling the building depend on their thermal energy values. These are according to the same reasons mentioned in the previous sub-item. The thermal energy values and the thermal exergy values by transmission and for heating and cooling of all cases are presented in Table 8.

Comparing case D to case A (reference case), the differences (in %) in the thermal energy values for heating and in the thermal exergy values for heating are very small ca. 0.49% of the values in the reference case, and the differences (in %) in the thermal energy values for cooling and in the thermal exergy values for cooling are around 9% of the values in the reference case. The percentage in the cooling case is larger than the percentage in the heating case, due to the same reason discussed in the previous sub-item.

Table 8 Thermal energy and thermal exergy values (Q_{th} and E_{th}) by transmission and for heating and cooling of all cases for the 274th and 4096th hour of the TMY

Items	Q_{th} [kJ]				E_{th} [kJ]			
	274th hour		4096th hour		274th hour		4096th hour	
	Input	Output	Input	Output	Input	Output	Input	Output
Case A: : 0.08 m of concrete slab (ref. case)								
heating/cooling	205200	0	0	49930	22239	0	0	228
transmission	0	61300	0	12110	0	6140	0	146
Case B: 0.18 m of concrete slab								
heating/cooling	203800	0	0	47930	22087	0	0	219
transmission	0	59970	0	14550	0	6020	0	180
Case C: 0.28 m of concrete slab								
heating/cooling	203900	0	0	46630	22098	0	0	213
transmission	0	60050	0	15930	0	6027	0	200
Case D: 0.38 m of concrete slab								
heating/cooling	204200	0	0	45610	22130	0	0	208
transmission	0	60290	0	16720	0	6049	0	213

The thermal energy and thermal exergy demands for heating ($Q_{heating}$ and $E_{heating}$) and cooling ($Q_{cooling}$ and $E_{cooling}$) of all cases for the whole year TMY are given in Table 9. By changing the thickness of the floor, $E_{heating}$ and $E_{cooling}$ vary on $Q_{heating}$ and $Q_{cooling}$ respectively. All the demands of case D are smaller than the demands of case A.

Table 9 Thermal energy and thermal exergy demands for heating ($Q_{heating}$ and $E_{heating}$) and cooling ($Q_{cooling}$ and $E_{cooling}$) of all cases for TMY

	$Q_{heating}$ [MJ/a]		$Q_{cooling}$ [MJ/a]		$E_{heating}$ [MJ/a]		$E_{cooling}$ [MJ/a]	
	$T_i > T_o$	$T_i < T_o$	$T_i < T_o$	$T_i > T_o$	$T_i > T_o$	$T_i < T_o$	$T_i < T_o$	$T_i > T_o$
Case A (ref. case)	114510	0	989	2218	6109	0	4	16
Case B	113972	0	944	2091	6101	0	4	16
Case C	113393	0	929	2060	6084	0	4	15
Case D	113125	0	919	2014	6076	0	4	15

4.4. Changing the thicknesses of the insulations in the walls

This study compares thermal energy and thermal exergy values in the reference building by changing the insulation thicknesses (which are the 2nd wall layers) in the walls. The effect by building mass to the thermal values should be very small and not affect to this study because density of the insulations is very small. For each change, the thicknesses of all the layers will be changed at the same values: from 6.1 cm to 12.1 cm at 1 cm intervals. The building where the thicknesses of the insulation layers are 6.1 cm is the reference case. The U-values of the walls where the thicknesses of the insulation layers are 6.1 cm are $0.511 \text{ Wm}^{-2}\text{K}^{-1}$. The cases considered are as following.

- Case A: 0.061 m of insulation (the reference case; the results are presented in Table 4)
- Case B: 0.071 m of insulation (U-values is $0.453 \text{ Wm}^{-2}\text{K}^{-1}$)
- Case C: 0.081 m of insulation (U-values is $0.407 \text{ Wm}^{-2}\text{K}^{-1}$)
- Case D: 0.091 m of insulation (U-values is $0.370 \text{ Wm}^{-2}\text{K}^{-1}$)
- Case E: 0.101 m of insulation (U-values is $0.338 \text{ Wm}^{-2}\text{K}^{-1}$)
- Case F: 0.111 m of insulation (U-values is $0.312 \text{ Wm}^{-2}\text{K}^{-1}$)
- Case G: 0.121 m of insulation (U-values is $0.289 \text{ Wm}^{-2}\text{K}^{-1}$)

For the 274th hour and the 4096th hour of the TMY, the thermal energy values and the thermal exergy values by infiltration, ventilation, internal and solar gains, are constant for all cases. The thermal exergy values for heating and for cooling the building depend on their thermal energy values. These are according to the same reasons mentioned in item 4.2. The thermal energy values and the thermal exergy values by transmission and for heating and cooling of all cases are presented in Table 10.

Table 10 Thermal energy and thermal exergy values (Q_{th} and E_{th}) by transmission and for heating and cooling of all cases for the 274th and 4096th hour of the TMY

Items	Q_{th} [kJ]				E_{th} [kJ]			
	274th hour		4096th hour		274th hour		4096th hour	
	Input	Output	Input	Output	Input	Output	Input	Output
Case A: 0.061 m of insulation (ref. case)								
heating/cooling	205200	0	0	49930	22239	0	0	228
transmission	0	61300	0	12110	0	6140	0	146
Case B: 0.071 m of insulation								
heating/cooling	203600	0	0	50410	22065	0	0	230
transmission	0	59750	0	11870	0	5999	0	142
Case C: 0.081 m of insulation								
heating/cooling	202300	0	0	50690	21925	0	0	231
transmission	0	58470	0	11620	0	5883	0	138
Case D: 0.091 m of insulation								
heating/cooling	201200	0	0	50690	21805	0	0	231
transmission	0	57300	0	11330	0	5775	0	135
Case E: 0.101 m of insulation								
heating/cooling	200300	0	0	51070	21708	0	0	233
transmission	0	56410	0	11200	0	5693	0	132
Case F: 0.111 m of insulation								
heating/cooling	199500	0	0	51060	21621	0	0	233
transmission	0	55700	0	10970	0	5628	0	130
Case G: 0.121 m of insulation								
heating/cooling	198800	0	0	51340	21545	0	0	234
transmission	0	55000	0	10870	0	5563	0	128

In table 10, comparing case G to case A (reference case), the differences (in %) in the thermal energy values for heating, in the thermal exergy values for heating, in the thermal energy values for cooling and in the thermal exergy values for cooling are approximately 3% of the values in the reference case. The percentages in the cooling case are not so different from the percentages in the heating case. The thermal energy and the thermal exergy values for heating in case G are smaller than the values in case A, but the thermal energy and the thermal exergy values for cooling in case G are larger than the values for cooling in case A.

The thermal energy and thermal exergy demands for ($Q_{heating}$ and $E_{heating}$) and cooling ($Q_{cooling}$ and $E_{cooling}$) of all cases for the whole year TMY are given in Table 11. By changing the thickness of the insulations in the walls, $E_{heating}$ and $E_{cooling}$ also vary on $Q_{heating}$ and $Q_{cooling}$ respectively. $Q_{heating}$ and $E_{heating}$ of case D are smaller than $Q_{heating}$ and $E_{heating}$ of case A, but $Q_{cooling}$ and $E_{cooling}$ of case D are larger than $Q_{cooling}$ and $E_{cooling}$ of case A.

Table 11 Thermal energy and thermal exergy demands for heating ($Q_{heating}$ and $E_{heating}$) and cooling ($Q_{cooling}$ and $E_{cooling}$) of all cases for TMY

	$Q_{heating} [MJ/a]$		$Q_{cooling} [MJ/a]$		$E_{heating} [MJ/a]$		$E_{cooling} [MJ/a]$	
	$T_i > T_o$	$T_i < T_o$	$T_i < T_o$	$T_i > T_o$	$T_i > T_o$	$T_i < T_o$	$T_i < T_o$	$T_i > T_o$
Case A (ref. case)	114510	0	989	2218	6109	0	4	16
Case B	111642	0	1000	2332	5990	0	4	18
Case C	109330	0	1008	2438	5893	0	4	19
Case D	107505	0	1012	2501	5815	0	4	20
Case E	105916	0	1021	2596	5748	0	4	21
Case F	104621	0	1024	2665	5693	0	4	22
Case G	103478	0	1031	2722	5748	0	4	23

4.5. Changing the window types

This study focuses on variation on the window types. U-values and g-values of the window types considered are as following.

- Case A: $1.300 \text{ Wm}^{-2}\text{K}^{-1}$ of U-value, 0.591 of g-value (the reference case; the results are presented in Table4)
- Case B: $2.800 \text{ Wm}^{-2}\text{K}^{-1}$ of U-value, 0.755 of g-value
- Case C: $1.300 \text{ Wm}^{-2}\text{K}^{-1}$ of U-value, 0.298 of g-value

For all the cases and for the 274th hour and the 4096th hour of the TMY, the thermal energy values and the thermal exergy values by infiltration, ventilation and internal gain, are constant. The thermal exergy values for heating and for cooling the building depend on their thermal energy values. These are according to the same reasons mentioned in sub-item 4.2. But thermal energy values by solar gain are different due to changing properties of the window. The thermal exergy values by the solar gain depend on their thermal energy values because the temperatures of the sun and the indoor air are assumed constant. Therefore, the thermal energy values and the thermal exergy values by transmission, by solar gain and for heating and cooling, are not constant for all the cases. The thermal energy values and the thermal exergy values by transmission, by solar gain and for heating and cooling, of all cases are presented in Table 12.

Table 12 Thermal energy and thermal exergy values (Q_{th} and E_{th}) by transmission, by solar gain and for heating and cooling, of all cases for the 274th and 4096th hour of the TMY

Items	$Q_{th} [kJ]$				$E_{th} [kJ]$			
	274th hour		4096th hour		274th hour		4096th hour	
	Input	Output	Input	Output	Input	Output	Input	Output
Case A: $1.300 \text{ Wm}^{-2}\text{K}^{-1}$ of U-value, 0.591 of g-value (ref. case)								
heating/cooling	205200	0	0	49930	22239	0	0	228
transmission	0	61300	0	12110	0	6140	0	146
solar gain	253	0	21050	0	242	0	19996	0
Case B: $2.800 \text{ Wm}^{-2}\text{K}^{-1}$ of U-value, 0.755 of g-value								
heating/cooling	209700	0	0	52610	22727	0	0	240
transmission	0	65840	0	14010	0	6548	0	162
solar gain	313	0	25490	0	299	0	24213	0
Case C: $1.300 \text{ Wm}^{-2}\text{K}^{-1}$ of U-value, 0.298 of g-value								
heating/cooling	205800	0	0	39300	22304	0	0	179
transmission	0	61750	0	12640	0	6179	0	175
solar gain	129	0	10290	0	123	0	9775	0

Comparing case B to case A (reference case), the differences (in %) in the thermal energy values for heating and in the thermal exergy values for heating are more or less 2.19% of the values in the

reference case, and the differences (in %) in the thermal energy values for cooling and in the thermal exergy values for cooling are ca. 5% of the values in the reference case. All thermal energy and thermal exergy values for heating and cooling in case B are larger than the values in case A

Comparing case C to case A (reference case), the differences (in%) in the thermal energy values for heating and in the thermal exergy values for heating are very small ca. 0.29% of the values in the reference case, and the differences (in%) in the thermal energy values for cooling and in the thermal exergy values for cooling are around 21% of the values in the reference case. The thermal energy and thermal exergy values for cooling in case C are smaller than the values in case A, due to the low g-value of the window in the case C. Also, with the rapid decrease of the thermal exergy by solar gain, the total thermal exergy loss in case C is much smaller than the total thermal exergy loss in case A.

The thermal energy and thermal exergy demands for heating and cooling of all cases for the whole year TMY are given in Table 13. All the demands of case B are larger than the demands of case A. Q_{heating} and E_{heating} of case C are larger than Q_{heating} and E_{heating} of case A, but Q_{cooling} and E_{cooling} of case D are smaller than Q_{cooling} and E_{cooling} of case A.

Table 13 Thermal energy and thermal exergy demands for heating (Q_{heating} and E_{heating}) and cooling (Q_{cooling} and E_{cooling}) of all cases for TMY

	$Q_{\text{heating}} [\text{MJ/a}]$		$Q_{\text{cooling}} [\text{MJ/a}]$		$E_{\text{heating}} [\text{MJ/a}]$		$E_{\text{cooling}} [\text{MJ/a}]$	
	$T_i > T_o$	$T_i < T_o$	$T_i < T_o$	$T_i > T_o$	$T_i > T_o$	$T_i < T_o$	$T_i < T_o$	$T_i > T_o$
Case A (ref. case)	114510	0	989	2218	6109	0	4	16
Case B	119238	0	1046	2420	6324	0	4	19
Case C	126119	0	717	929	6530	0	3	5

4.6. Changing the window sizes

This study focuses on variation to sizes of the window placed in the south façade (i.e. 25%, 50% and 75% of the façade). The building that has the window of 50% of the south façade is the reference case for the study. The cases considered are as following.

- Case A: 50% of the south façade (the reference case; the results are presented in Table 4)
- Case B: 25% of the south façade
- Case C: 75% of the south façade

For all the cases and for the 274th hour and the 4096th hour of the TMY, the thermal energy values and the thermal exergy values by infiltration, ventilation and internal gain, are constant. The thermal exergy values for heating and for cooling the building and by solar gain depend on their thermal energy values. These are according to the same reasons mentioned in the sub-item 4.5. The thermal energy values and the thermal exergy values by transmission, by solar gain and for heating and cooling, of all cases are presented in Table 14.

Sizing the window influences the amount of solar gain of the building, and then on the thermal energy and thermal exergy values for both heating and cooling. When increasing the window size, the thermal energy and thermal exergy values for heating and cooling increase as well. In addition, the total exergy losses in case B and case C change very much from case A, due to the high sensitivities of the thermal exergy by solar gain to the window size changes.

The thermal energy and thermal exergy demands for heating (Q_{heating} and E_{heating}) and cooling (Q_{cooling} and E_{cooling}) of all cases for the whole year TMY are given in Table 15. When increasing the window size, Q_{cooling} and E_{cooling} increase, but the Q_{heating} and E_{heating} decrease

Table 14 Thermal energy and thermal exergy values (Q_{th} and E_{th}) by transmission, by solar gain and for heating and cooling, of all cases for the 274th and 4096th hour of the TMY

Items	Q_{th} [kJ]				E_{th} [kJ]			
	274th hour		4096th hour		274th hour		4096th hour	
	Input	Output	Input	Output	Input	Output	Input	Output
Case A: 50% of the south façade (ref. case)								
heating/cooling	205200	0	0	49930	22239	0	0	228
transmission	0	61300	0	12110	0	6140	0	146
solar gain	253	0	21050	0	242	0	19996	0
Case B: 25% of the south façade								
heating/cooling	203400	0	0	39760	22044	0	0	181
transmission	0	59800	0	12840	0	6007	0	177
solar gain	129	0	10750	0	124	0	10212	0
Case C: 75% of the south façade								
heating/cooling	206900	0	0	59700	22423	0	0	272
transmission	0	62570	0	11750	0	6248	0	122
solar gain	371	0	30930	0	355	0	29381	0

Table 15 Thermal energy and thermal exergy demands for heating ($Q_{heating}$ and $E_{heating}$) and cooling ($Q_{cooling}$ and $E_{cooling}$) of all cases for TMY

	$Q_{heating}$ [MJ/a]		$Q_{cooling}$ [MJ/a]		$E_{heating}$ [MJ/a]		$E_{cooling}$ [MJ/a]	
	$T_i > T_o$	$T_i < T_o$	$T_i < T_o$	$T_i > T_o$	$T_i > T_o$	$T_i < T_o$	$T_i < T_o$	$T_i > T_o$
Case A (ref. case)	114510	0	989	2218	6109	0	4	16
Case B	122077	0	722	989	6375	0	3	5
Case C	108412	0	1238	4235	5878	0	5	42

4.7. Analysis and comparison of the results

The energy and exergy analyses of the reference building for the 274th hour and the 4096th hour of the TMY shows that the changes of the thicknesses of the walls and the floor, as well as the insulation thickness in the walls, do not make any difference to the thermal energy and thermal exergy values caused by infiltration and ventilation airflows, solar gain and internal gains. This is because the thermal values depend on the infiltration and the ventilation airflow rates that are assumed constant, and the indoor air temperatures at the hours are also constant at 20°C and 26°C respectively. The maximum differences in the thermal energy values and in thermal exergy values for heating are 3.12% less than the values in the reference case, when increasing the insulation thickness in the walls to 0.121 m. The maximum differences in the thermal energy values and in thermal exergy values for cooling are 18.01% less than the values in the reference case, when increasing the thicknesses of the walls to ca. 0.40 m. When considering the changes of window sizes or type, the solar gain becomes important, especially in terms of exergy, to the total thermal exergy demand input to the building. The larger window provides higher value of the total thermal exergy demand input to the building, and also the small window provides a lower value. The window that has a low g-value provides much less value of the total thermal exergy demand input to the building. The changes of the solar gains have huge influences on the total exergy loss of the building.

Table 16 presents the savings of the total thermal energy and the total thermal exergy for the whole year TMY in percentage of the values in the reference case. The total amounts are accounted by the thermal values in both the heating and cooling cases. In the reference case, the total thermal energy and the total thermal exergy values are 117715 MJ and 6129 MJ respectively. The ratios between the savings of the total thermal exergy and the total thermal energy vary from 0.26 (=0.16%/0.60%) to 0.84 (=6.70%/7.99%).

Table 16 Savings of the total thermal energy and the total thermal exergy for the whole TMY in percentage of the values in the reference case by changing some building parametric values

	Total thermal energy [%]	Total thermal exergy [%]
Increasing wall thickness		
0.20 m of concrete block, 0.053 m of foam insulation	2.06%	1.31%
0.30 m of concrete block, 0.045 m of foam insulation	2.95%	2.01%
0.40 m of concrete block, 0.038 m of foam insulation	3.75%	2.68%
Increasing floor thickness		
0.18 m of concrete block	0.60%	0.13%
0.28 m of concrete block	1.13%	0.42%
0.38 m of concrete block	1.41%	0.55%
Increasing insulation thickness in the walls		
0.071 m of insulation	2.33%	1.91%
0.081 m of insulation	4.20%	3.48%
0.091 m of insulation	5.69%	4.73%
0.101 m of insulation	6.95%	5.81%
0.111 m of insulation	7.99%	6.69%
0.121 m of insulation	8.91%	5.78%
Changing the window types		
2.800 Wm ⁻² K ⁻¹ of U-value, 0.755 of g-value	-4.24%	-3.56%
1.300 Wm ⁻² K ⁻¹ of U-value, 0.298 of g-value	-8.54%	-6.67%
Changing the window sizes		
25% of the south façade	-5.16%	-4.14%
75% of the south façade	3.26%	3.33%

5. Exergy losses in building services

This item presents exergy losses that could occur in the building services part (Figure 1). Exergy loss of a system in the building services part ($E_{\text{loss},i}$) is simply defined as the difference between the exergy input and the exergy output of the system ($E_{\text{in},i} - E_{\text{out},i}$). Detailed calculation of exergy loss in each system is presented in sub-items 5.1 to 5.3. In addition, the systems in the building services part are also assessed in terms of the system energy efficiency ($\eta_{Q,i}$) and the system exergy efficiency ($\eta_{E,i}$). These measures are used for investigation of what the relevance is of exergy analysis of the systems and for identification of the location of the main exergy loss. For each system, the system energy efficiency is defined as the ratio between the energy output and the energy input of the system ($Q_{\text{out},i}/Q_{\text{in},i}$), and the system exergy efficiency is defined as the ratio between the exergy output and the exergy input of the system ($E_{\text{out},i}/E_{\text{in},i}$).

Item 5.1 presents exergy losses in the thermal energy emission & control and thermal distribution systems, where heating and cooling equipments in different temperature levels of the thermal energy supply are applied. The equipments are described in detail in the item, separately for heating and cooling cases. Item 5.2 explains how exergy losses in the electricity distribution system occur and presents calculation methods of the system energy efficiency and the system exergy efficiency. Item 5.3 describes exergy input and exergy output of the (local) energy conversion and energy storage systems. Item 5.4 gives a design example for building services (where they contain thermal energy emission & control and thermal distribution systems, electricity distribution system and (local) energy conversion system), and presents energy and exergy analysis results of the building services.

5.1. Exergy losses in the thermal energy emission & control system and the thermal distribution system

The thermal energy emission & control system means a system (including its accessory) that transfers thermal energy from a local energy conversion system (or from an energy storage system) into a heat

emission system, via a thermal distribution system. In this item, the thermal energy emission & control system and the thermal distribution system are a radiator and air ducts respectively. The details of these systems are given in this item, as following in the “heating case” and “cooling case” sessions.

Exergy losses in the thermal energy emission & control system and the thermal distribution system consist of two parts: a thermal and an electric one. The thermal exergy losses occur in heat transfer processes, and the electric exergy losses are the auxiliary energy of the systems. Therefore the exergy input and the exergy output of the thermal energy emission & control system are the summation of the thermal exergy input of the system and the auxiliary energy of the system ($E_{th,i} + E_{aux,i}$) and the thermal exergy input of the previous system ($E_{th,i-1}$) respectively. In this case, the exergy output of the system is the total exergy demand of a building ($E_{th,i-1}$). The exergy loss ($E_{loss,i}$), the system energy efficiency ($\eta_{Q,i}$) and the system exergy efficiency ($\eta_{E,i}$), of the system are calculated by using equations 19-21, respectively.

$$E_{loss,i} = (E_{th,i} + E_{aux,i}) - E_{th,i-1} \quad (19)$$

$$\eta_{Q,i} = \frac{Q_{th,i-1}}{Q_{th,i} + Q_{aux,i}} \quad (20)$$

$$\eta_{E,i} = \frac{E_{th,i-1}}{E_{th,i} + E_{aux,i}} \quad (21)$$

For the thermal distribution system, these measures are applied in a similar way where they are applied for the thermal energy emission & control system as explained above.

The following examples present exergy losses in the thermal energy emission & control system and the thermal distribution system. These examples illustrate the effects of supplying thermal energy with low- or high-temperature thermal energy emission & control system to the reference building (item 3.2). The exergy losses are calculated on a hourly basis. The calculations use the weather data of De Bilt, the Netherlands, in the TMY. These exergy loss calculation results are accumulated and presented for the year, in heating and cooling cases separately.

For the study, the following energy supply equipments used in the thermal energy emission & control system and in the thermal distribution system are described.

- heating case
 - thermal energy emission & control system
 - low ($25^{\circ}\text{C } T_{\text{surface}}$) or high temperature ($50^{\circ}\text{C } T_{\text{surface}}$) radiator
 - system thermal loss efficiency of 0.95
 - no auxiliary energy required
 - thermal distribution system
 - system thermal loss efficiency of 0.92; temperature drop of 5K
 - auxiliary energy of $3.37 \text{ W/kW}_{Q_{th}}$ for pumps
- cooling case
 - thermal energy emission & control system
 - low ($10^{\circ}\text{C } T_{\text{surface}}$) or high temperature ($15^{\circ}\text{C } T_{\text{surface}}$) cooling element
 - system thermal loss efficiency of 0.95
 - no auxiliary energy required
 - thermal distribution system
 - system thermal loss efficiency of 0.92; temperature increase of 5K
 - auxiliary energy of $3.37 \text{ W/kW}_{Q_{th}}$

Tables 17 and 18 present the energy and exergy calculation results of the reference building (item 3.2), the thermal energy emission & control system and the thermal distribution system, for the TMY separately in heating and cooling cases respectively. The indoor air temperatures are calculated by TRNSYS, with the condition mentioned in item 3.2.

Table 17 Energy and exergy demands in heating case [MJ/a]

Items	Energy [MJ/a]			Exergy [MJ/a]		
	Q_{th}	Q_{aux}	$\eta_Q [-]$	E_{th}	E_{aux}	$\eta_E [-]$
Case A: low temperature radiator (25°C $T_{surface}$)						
Building	114653.43	0.00	-	6118.64	0.00	-
Thermal energy emission & control	120687.82	0.00	0.95	8812.78	0.00	0.69
Thermal distribution	131182.41	442.08	0.92	11584.77	442.08	0.73
Case B: high temperature radiator (50°C $T_{surface}$)						
Building	114653.43	0.00	-	6118.64	0.00	-
Thermal energy emission & control	120687.82	0.00	0.95	17467.82	0.00	0.35
Thermal distribution	131182.41	442.08	0.92	20696.28	442.08	0.83

Table 18 Energy and exergy demands in cooling case [MJ/a]

Items	Energy [MJ/a]			Exergy [MJ/a]		
	Q_{th}	Q_{aux}	$\eta_Q [-]$	E_{th}	E_{aux}	$\eta_E [-]$
Case A: low temperature cooling element (10°C $T_{surface}$)						
Building	3206.10	0.00	-	20.48	0.00	-
Thermal energy emission & control	3374.84	0.00	0.95	176.73	0.00	0.12
Thermal distribution	3668.30	12.36	0.92	261.49	12.36	0.65
Case B: high temperature cooling element (15°C $T_{surface}$)						
Building	3206.10	0.00	-	20.48	0.00	-
Thermal energy emission & control	3374.84	0.00	0.95	115.10	0.00	0.18
Thermal distribution	3668.30	12.36	0.92	192.10	12.36	0.56

In Table 17, case A and case B present the energy supply system in which the thermal energy emission & control system is a low temperature radiator (25°C $T_{surface}$) and a high temperature radiator (50°C $T_{surface}$) respectively. The rest of the system is described above in the heating case.

In Table 18, case A and case B present the energy supply system in which the thermal energy emission & control system is a low temperature radiator (10°C $T_{surface}$) and a high temperature radiator (15°C $T_{surface}$) respectively. The rest of the system is described above in the cooling case.

Table 17 shows that the energy analysis results are not sensitive to the operations of the systems at the different temperature levels of the thermal energy supply. The auxiliary energy of the thermal distribution system in both cases is relatively small to the thermal energy input of the system. This causes the system energy efficiencies of the thermal distribution system in case A and case B are around the thermal loss efficiencies of the system.

In terms of exergy, the energy analysis results are sensitive to the operations of the systems at the different temperature levels of the thermal energy supply. The system exergy efficiencies η_E are always less than the system energy efficiencies η_Q . The system exergy efficiency of the thermal energy emission & control system in case A is about 2 times the system exergy efficiency in case B, because there are many exergy losses occurring in the system in case B. However, the system exergy efficiency of the thermal distribution system in case A is less than the system exergy efficiency in case B, because the temperature drop in the system in case B occurs at the higher temperature level (55-50°C) and the auxiliary exergy of the system in case B becomes relatively smaller when comparing the value with the thermal exergy of the system. Some recommendations from the exergy analysis are supplying thermal energy at low temperature and minimising temperature drop occurring in the thermal distribution system to reduce thermal exergy losses, as well as minimising auxiliary energy of the thermal distribution system to reduce electric exergy losses.

In cooling case, the system energy efficiencies of the thermal distribution system in case A and case B are around the thermal loss efficiencies of the system, because the auxiliary energy of the thermal distribution system in both cases is still relatively small to the thermal energy input of the system.

In terms of exergy, the system exergy efficiencies of the thermal energy emission & control system in case A and case B in cooling case are much smaller than the system exergy efficiencies of the system in case A and case B in heating case, because the operating temperature of the system in cooling case is closer to the temperature of the environment when comparing that with the temperature difference in heating case. Also in the cooling case the system exergy efficiency of the thermal energy emission & control system in case A is smaller than the system exergy efficiency in case B, because in case A the system operates at higher temperature and closer to the environment than the system operation in case B. The system exergy efficiency of the thermal distribution system in the cooling case in both cases is lower than the system exergy efficiency of the system in heating case, because in cooling case the temperature difference between the operating temperature of the system and the environment is smaller. However, in the cooling case the system exergy efficiency of the thermal distribution system in case A is higher than the system exergy efficiency in case B, because the temperature drop in the system in case A occurs at the lower temperature level (5-10°C) and the auxiliary exergy of the system in case A becomes relatively smaller when comparing the value with the thermal exergy of the system. Some recommendations from the exergy analysis in the cooling case are related to the temperature level of the thermal energy supply and auxiliary energy of the thermal distribution system: supply of thermal energy at a temperature close to environment; and minimising temperature increase and auxiliary energy of the thermal distribution system.

5.2. Exergy losses in the electricity distribution system

Exergy and energy losses in the electricity distribution system are electric ones. These losses could be further converted into thermal losses and then totally dissipate in the environment. The parts of the energy conversion from electricity to thermal energy are not considered in this study. Therefore the exergy input and the exergy output of the system are the electric exergy input of the system ($E_{pe,i}$) and the electric exergy input of the previous system ($E_{pe,i-1}$) respectively. The exergy loss ($E_{loss,i}$), the system energy efficiency ($\eta_{Q,i}$) and the system exergy efficiency ($\eta_{E,i}$), of the system are calculated by using equations 22-24 respectively.

$$E_{loss,i} = E_{pe,i} - E_{pe,i-1} \quad (22)$$

$$\eta_{Q,i} = \frac{Q_{pe,i-1}}{Q_{pe,i}} \quad (23)$$

$$\eta_{E,i} = \frac{E_{pe,i-1}}{E_{pe,i}} \quad (24)$$

The exergy and energy of electricity are identical because electric energy can in theory be totally converted into mechanical work (Moran and Shapiro, 1998). In this study, the exergy of electricity is equal to its electric energy. Therefore the exergy losses in the system are equal to their electric energy losses, and the system energy efficiency and the system exergy efficiency of the system are identical.

5.3. Exergy losses in the (local) energy conversion and the energy storage systems

Exergy losses in the energy storage system are affected by physical material properties of the storage. This topic is not studied in this work.

Exergy losses in the (local) energy conversion system depend not only on physical properties of the boundary of the system, but also on processes in the system itself. Possible processes in the system are listed in Table 1. The exergy input and output of the system are from the external part and to the local part. Depending on processes in the system, the exergy input and output are, for example, given in Table 19.

Table 19 Exergy output and exergy input of the local energy conversion processes

<i>External part</i>	<i>Local part</i>	<i>Exergy output and exergy input</i>
Thermal energy carriers	Thermal energy	Exergy input: thermal exergy of the energy carrier in the external part
		Exergy output: thermal exergy of the energy carrier in the local part
	Electricity	Exergy input: thermal exergy of the energy carrier in the external part
		Exergy output: electric exergy (a product between voltage and current at a time unit)
Chemical substances	Thermal energy	Exergy input: chemical exergy of the chemical substance (Gibb's energy)
		Exergy output: thermal exergy of the energy carrier in the local part
	Electricity	Exergy input: chemical exergy of the chemical substance (Gibb's energy)
		Exergy output: electric exergy (a product between voltage and current at a time unit)
Electricity/ Natural Forces	Thermal energy	Exergy input: electric exergy (a product between voltage and current at a time unit) mechanical work
		Exergy output: thermal exergy of the energy carrier in the local part
	Electricity	Exergy input: electric exergy (a product between voltage and current at a time unit) mechanical work
		Exergy output: electric exergy (a product between voltage and current at a time unit)

5.4. Design example for building services

This item presents results of energy and exergy analysis of the reference building (item 3.2) and building services, to summarise the uses of the exergy concept in a building and the built environment. The building services consist of thermal energy emission & control system, thermal distribution system, electricity distribution system and local energy conversion system. The building services for this example are selected and operated close to indoor air temperature, as they have high system exergy efficiency ($\eta_{E,i}$). The thermal energy emission & control system consists of a low-temperature heating panel and a high-temperature cooling panel. In addition, the energy storage system is not considered. Details of the building services are as following, for heating and cooling cases separately.

- heating case (only thermal heat from the district heating is input to the building services)
 - thermal energy emission & control system
 - low-temperature heating panel ($35^{\circ}\text{C } T_{in}/28^{\circ}\text{C } T_{out}$)
 - system thermal loss efficiency of 0.95
 - no auxiliary energy required
 - thermal distribution system
 - system thermal loss efficiency of 0.92; temperature drop of 10K
 - auxiliary energy of $3.37 \text{ W/kW}_{Q_{th}}$ for pumps
 - electricity distribution system
 - system thermal loss efficiency of 0.97
 - local energy conversion system
 - heat exchanger
 - district heating at temperature of 90°C
 - system thermal loss efficiency of 0.89
 - auxiliary energy of $0.01 \text{ W/kW}_{Q_{th}}$
- cooling case (only electricity is input to the building services)
 - thermal energy emission & control system
 - high-temperature cooling panel ($10^{\circ}\text{C } T_{in}/23^{\circ}\text{C } T_{out}$)
 - system thermal loss efficiency of 0.95
 - no auxiliary energy required
 - thermal distribution system
 - system thermal loss efficiency of 0.92; temperature drop of 10K
 - auxiliary energy of $3.37 \text{ W/kW}_{Q_{th}}$
 - electricity distribution system
 - system thermal loss efficiency of 0.97
 - local energy conversion system

- a cooling machine of $COP=1.50^2$
- system thermal loss efficiency of 0.95
- auxiliary energy of 35 W/kW_{Qth}
- mean temperature of hot air from the cooling machine assumed 35°C

The energy and exergy analyses are carried out on hourly basis. The analysis results are accumulated and presented in annual values separately for heating and cooling cases, for the TMY of De Bilt, the Netherlands. The weather data used for the calculations are taken from the TMY2 data source (NREL, 1995).

The amount of electricity used in the building is not presented in Table 20 and Table 21, but it is used for calculation of thermal energy and thermal exergy demands of the building. Therefore for the electricity distribution system (Figure 1), the exergy output of the system is accounted only for the auxiliary exergy of the thermal energy emission & control system and of the thermal distribution system, but not for the electricity used in the building. The exergy losses accounted from the electricity used in the building could be treated separately and it is not studied in this work.

Table 20 shows thermal energy and thermal exergy demands of the reference building, thermal energy and thermal exergy inputs of the building services, auxiliary energy of the building services, and system energy and system exergy efficiencies of the building services, for the TMY in heating case.

Table 20 Energy and exergy values of the reference building and services for heating case [MJ/a]

Items	Energy [MJ/a]			Exergy [MJ/a]		
	Q_{th}	Q_{aux}	$\eta_Q [-]$	E_{th}	E_{aux}	$\eta_E [-]$
Building	114653.43	0.00	-	6118.64	0.00	-
Thermal energy emission & control	120687.82	0.00	0.95	17467.82	0.00	0.35
Thermal distribution	131182.41	442.08	0.92	20696.28	442.08	0.83
Electricity distribution	0.00	455.76	0.97	0.00	455.76	0.97
Energy conversion	147395.97	457.23	0.89	35218.89	457.23	0.59

In Table 20, the thermal energy demand of the building is 95 % of the thermal energy input of the thermal energy emission & control system, corresponding to the thermal loss efficiency of the system. But the thermal exergy demand of the building is much less than the thermal exergy input of the system. The exergy efficiency of the system is 35%. For other systems in the building services, energy efficiencies of the systems are more or less the same as their thermal loss efficiencies. This is because the auxiliary energy (Q_{aux}) of the systems is relatively small when comparing to the thermal energy input (Q_{th}) of the systems. The overall energy efficiency of the building and the services is $(114653.43)/(147395.97+457.23) = 77.55\%$.

The overall exergy efficiency of the building and the services is $(6118.64)/(35218.89+457.23) = 17.15\%$. This efficiency is higher than the overall exergy efficiency of a typical way to supply thermal energy by burning fossil, ca. 3.3% (Nieuwlaar and Dijk, 1993). The thermal energy supply system consists of a heat-exchanger to heat a circulating flow of water from 40°C to 60°C. The hot water (at 60°C) is cooled down to 40°C in a radiator to maintain 20°C of indoor air temperature of a building. This shows that the system in this example is better than the system in the typical way that uses burning fossils to get thermal energy, while both systems could reach the same system energy efficiency. Selecting a right energy source could improve the overall exergy efficiency, as can be seen in this case.

Auxiliary exergy of the building services are equal to their energy values, by the assumption that the auxiliary energy is electricity.

Table 21 presents thermal energy and thermal exergy demands of the reference building, thermal energy and thermal exergy outputs of the building services, auxiliary energy of the building services, and system energy and system exergy performances (or efficiencies) of the building services, for the TMY in the cooling case.

² The machine is a small chiller with relatively low COP .

Table 21 Energy and exergy values of the reference building and services for the cooling case [MJ/a]

Items	Energy [MJ/a]			Exergy [MJ/a]		
	Q_{th}	Q_{aux}	$\eta_Q [-]$	E_{th}	E_{aux}	$\eta_E [-]$
Building	3206.10	0.00	-	20.48	0.00	-
Thermal energy emission & control	3374.84	0.00	0.95	115.10	0.00	0.18
Thermal distribution	3668.30	12.36	0.92	192.10	12.36	0.56
Electricity distribution	0.00	12.74	0.97	0.00	12.74	0.97
Energy conversion	6435.61	2683.52	1.37	162.27	2683.52	0.13

In Table 21, the thermal energy demand of the building (3206.10 MJ/a) is also close to the thermal energy output of the thermal energy emission & control system (3374.84 MJ/a). The difference of the values is due to the thermal gain of the system. The energy efficiencies of the systems in the building services are more or less same as their thermal loss efficiencies as well, because the auxiliary energies (Q_{aux}) of the systems are still much relatively small when comparing to the thermal energy outputs of the systems (Q_{th}).

The thermal energy output (Q_{th}) of the local energy conversion system (for the cooling machine) could be regarded as thermal energy emitted to the nature. The amount of the thermal energy output is calculated from the system thermal loss efficiency (0.95), COP (1.5) and the thermal energy output of the thermal distribution system (3668.30 MJ/a): $(3668.30 \times 2.5 / 1.5) / 0.95 = 6435.61$ MJ/a. The auxiliary energy of the local energy conversion system including electricity input to the cooling machine (Q_{aux}) is calculated from the thermal energy outputs of the thermal distribution system and the local energy conversion system (3668.30 MJ/a and 6435.61 MJ/a respectively), the COP of the cooling machine (1.5), the auxiliary energy of the electricity distribution system and the local energy conversion system (12.74 MJ/a and 35 W/kW $_{Q_{th}}$ respectively): $(3668.30 / 1.5) + (6435.61 \times 35) / 1000 + 12.74 = 2683.52$ MJ/a. Therefore the system energy performance (or efficiency) of the building services in the cooling case is $3206.10 / 2683.52 = 1.19$; less than the COP of the cooling machine.

The thermal exergy value of the thermal energy output (E_{th}) of the local energy conversion system (for the cooling machine) is calculated from the thermal energy output (Q_{th}) of the system at mean temperature of the hot exhaust air from the cooling machine (35°C). The calculations are made for every hour in the TMY year by using the hourly values of the environmental air temperature. The thermal exergy value of the thermal energy output (E_{th} , 162.27 MJ/a) is the summed values of the calculation results. The electric exergy output of the system (E_{aux} , 2683.52 MJ/a) is identical to the electric energy output (Q_{aux}) of the same system, according to the assumption mentions in item 5.2. The exergy efficiency (η_E) of the building services is $(20.48 + 162.27) / (2683.52) = 6.81\%$, and when neglecting the thermal exergy output of the local energy conversion system (E_{th}), η_E is $(20.48) / (2683.52) = 0.76\%$. These efficiencies are usually low, since electricity (which has high exergy) is used to produce thermal energy (which has low exergy).

6. Conclusions and recommendations

This paper presents exploratory work to study the relevance of exergy analysis of buildings and building services. The work makes use of an extended built-up model in which the energy balance is considered from the demand side to the supply side, developed by Sakulpipatsin et. al. (2006) and Bezuijen (2006), and exergy analysis based on the results obtained is added too. This allows to perform a complex exergy analysis both including the building and its services. The method is intended to enable building designers (and building engineers) to compare between the impact of improvements in the building envelope and in building services.

The proposed method for energy and exergy analysis of a building and services is examined by using the building simulation tool, TRNSYS, for a reference building and its services. The building to be studied is chosen as a cubic box in order to eliminate the effects in building geometry and orientation. The building operation plans and the occupancy requirements are given according to those in a typical office building in the Netherlands. The details of the building envelope construction, the building operation plans and the occupancy requirements are given in item 3.2. The building and the building services with some changes of their parametric values are investigated, to learn what the relevance is of exergy analysis of the building and the building services. Some interesting remarks that are found from the energy and exergy analysis are given, as following:

- For the reference building, reducing the thermal energy supplied by heating equipments remains a key concern in the cold days. While the internal thermal energy gains are crucial in the energy analysis, the exergy analysis shows that the solar exergy gain creates the main exergy losses when cooling is needed. These solar exergy gains should be minimized, or better captured to be useful somewhere else (for instance for domestic hot water production or electricity generation; Sakulpipatsin et. al., 2007b).
- In the TMY year, the annual demands of the thermal energy and the thermal exergy for heating the building are much bigger than the annual demands of the thermal energy and the thermal exergy for cooling the building respectively (Table 5). The ratio between the thermal exergy demand and the thermal energy demand for heating is around 5% and also much bigger than the ratios between the thermal demands for cooling, which are less than 0.07% for both cases where indoor air temperatures are higher and lower environmental air temperatures.
- The energy and exergy analyses of the reference building for the 274th hour and the 4096th hour of the TMY shows that the changes of the thicknesses of the walls and the floor, as well as the insulation thickness in the walls, do not make any difference of the thermal energy and thermal exergy values caused by infiltration and ventilation airflows, solar gain and internal gains. This is because the thermal values depend on the infiltration and the ventilation airflow rates that are assumed constant, and the indoor air temperatures at the hours are also constant at 20°C and 26°C respectively. The maximum differences in the thermal energy values and in thermal exergy values for heating are 3.22% less than the values in the reference case, when increasing the insulation thickness in the walls to 0.121 m (item 4.4). The maximum differences in the thermal energy values and in thermal exergy values for cooling are 21.96% less than the values in the reference case, when increasing the thicknesses of the walls to ca. 0.40 m (item 4.2). When considering the changes of window sizes or type, the solar gain becomes important, especially in terms of exergy, to the total thermal exergy input. The larger window provides higher value of the total thermal exergy input, and also the small window provides lower value. The window that has a low g-value provides much less value of the total thermal exergy input. The changes of the solar gains have huge influences on the total exergy loss of the building. By changing the building envelope parametric values mentioned in item 4.2-4.7, the total thermal energy and the total thermal exergy necessary for the building compared to the reference building shows savings up to 8.91% and 6.69% respectively (Table 16).
- By changing the building services parametric values, the exergy losses in the building services depend on the temperature level of the thermal energy supply and (electric) auxiliary energy required by the building services. This is applicable for both heating and cooling cases.
- Energy analysis shows that energy distribution and energy emission systems are quite efficient and it is worth to try reducing the energy demand of the building. However, the exergy analysis shows that to increase the rational use of energy, the building envelope offers little possibilities because it is already near-environmental, but there is a high potential for improvements in the thermal energy emission and thermal energy generation systems. In addition, the exergy analysis shows that solar gain could play a vital role in this improvement.

An ideal building and building services combinations in a cold country would, on one hand, operate as closely as possible to the required temperature (e.g. indoor temperature) and uses the lowest amount of electricity, so as to minimize exergy demands input from the external part. On the other hand, it is necessary to supply energy that has exergy of a matched level required by the local part, to minimize exergy losses in the local energy conversion: energy that has higher exergy value is more appropriate to be supplied to other processes that need this higher exergy value. In real systems, a compromise needs to be sought between system size and costs on the one hand and effective utilization of energy resources on the other hand.

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Nomenclature

COP	Coefficient of Performance [-]
c_p	Heat capacity [$m^2s^{-2}K^{-1}$; $Jkg^{-1}K^{-1}$]
E	Exergy [m^2kgs^{-2} ; J]
F_{sky}	Fraction of the sky in the total hemisphere seen by a specified surface, used as a weighting factor between ambient temperature and fictive sky temperature, for calculation of long-wave radiative exchange between the external building surface and ambient [-]
$g-value$	Solar thermal gain coefficient [-]
$\dot{m}$	Airflow rate [kgs^{-1}]
P	Pressure [$m^{-1}kgs^{-2}$; Pa]
Pe	Electricity; auxiliary energy [m^2kgs^{-2} ; J]
Q	Thermal energy [m^2kgs^{-2} ; J]
$\dot{Q}$	Thermal energy per second [m^2kgs^{-3} ; Js^{-1}]
t	Time [s]
T	Air temperature [K; °C with notation]
$u-value$	Thermal transmission value [$kgs^{-3}K^{-1}$; $Wm^{-2}K^{-1}$]
V	Building volume [m^3]
<i>Greek letters</i>	
η	System efficiency [-]

Subscripts

a	Annual
air	Air
aux	Auxiliary
com	Combined convective and radiative part
conv	Convective
cooling	Cooling
E	Exergy
exhaust	Exhaust air
gain	Internal heat gain
heating	Heating
i	Indoor; inside
in	Inlet
inf	Infiltration
loss	Loss
o	Outdoor; Reference environment state; outside
out	Outlet
Pe	Electricity
ph	Physical
Q	Thermal energy
r	Radiative
rad	Radiative
rev	Reversible
sol	Solar gain
source	Source
star	Star
sun	Sun
surface	Surface
th	Thermal
tran	Thermal transmission
vent	ventilation
wall	Wall

Abbreviations

a	Annual
HVAC	Heating Ventilation Air Conditioning
NL	Netherlands
TMY	Typical meteorological year

7. Closure

The work presented in this dissertation is fundamental and needed for a new field in building research and practice. The applicability of existing exergy-related definitions is systematically investigated in built-environment conditions (e.g. smaller temperature differences between a system and the environment), incorporated to existing exergy calculation models, and presented in this dissertation. Knowledge that is obtained from this work is essential to future developments of design instruments and guidelines for exergy-efficient building and building services design.

This final chapter presents relationships of this research with past and current research in item 7.1, compiles some remarks acquired from this research work and answers the research questions (which are mentioned in chapter 1) in item 7.2, and gives some future research recommendations and outlook in item 7.3.

7.1. Multiplying effect of this research

The research is related to past and current research. It has been carried out in close collaboration with the international LowExNet network of exergy researchers. LowExNet is a follow-up of an annex on low-exergy systems for heating and cooling of buildings supported by the International Energy Agency (IEA Annex 37). Some publications have been co-authored with one of the key researchers in LowExNet.

The PhD research work is a follow-up of previous research projects carried in the Faculty of Architecture at the Delft University of Technology by Dr. Asada (Visiting Researcher grant from NWO¹ in 2002) and Mr. Sakulpipatsin (DUT² Junior Research fellow in 2004). The PhD research is in turn being followed up by the EOS-LT³ project. This project is entirely financed by SenterNovem (an agency of the Dutch Ministry of Economic Affairs) and coordinated by the Faculty of Architecture of the TUD. It runs for four years with a total budget of one million Euros for three technical universities in the Netherlands (the universities of Delft, Eindhoven and Twente). The project aims at developing knowledge for design decisions and investments which contribute to the sustainability of energy supply in the built environment, in relation to human comfort, energy resources and economy aspects. The PhD research carried out by Mr. Sakulpipatsin, was a catalyst for the EOS project, and feeds it with fundamental knowledge and instruments for exergy analysis of building and HVAC systems. This setup allows continuity of the exergy research in the Faculty and in the Netherlands.

¹ NWO is the national organisation for science research of the Netherlands.

² DUT stands for Delft University of Technology.

³ EOS-LT stands for Energy Research Strategy Long Term.

During the course of the PhD research, the COSTeXergy project (COST Action 24) was initiated and has been running. The project has been supported by the European cooperation in the field of scientific and technical research and implemented by exergy researchers from the LowExNet network. In addition, research outputs of the PhD research have served as inputs to the formulation of the annex on low exergy systems for high-performance buildings and communities (IEA Annex 49). This annex is a follow-up of the IEA Annex 37.

7.2. Conclusions of this dissertation and answers to research questions

The general problem addressed in this dissertation is that exergy is often perceived as a highly-complex concept to be used in the building and building services design process. Systematic analysis is required to establish the applicability of the concept to the built environment. This dissertation presents calculation methods for exergy analysis of HVAC components and systems as well as building systems, and provides some concrete examples of exergy analyses on the levels of HVAC systems and building systems in order to make the exergy concept more familiar and usable to the building profession.

The goal of the research presented in this dissertation is to develop knowledge into the applicable domains and potential added value of exergy analysis in the built environment. Specifically, this work is concerned with the potential added value of exergy analysis for buildings and building services, in particular HVAC systems. The study is carried out in the levels of HVAC components and systems and of building systems, and provides metrics that can be used to quantify and express exergy values in buildings and HVAC systems.

The work presented in this dissertation consists of four main tasks:

- critical analysis of the influence of possible definitions of the standard state of air, to determine the exergy of air in buildings (chapter 2);
- critical analysis of exergy efficiency definitions for HVAC components operating at near-environmental temperatures: air-to-air heat exchangers (chapter 3) and vapour-compression heat pumps (chapter 4);
- exergy analysis as an assessment tool of heat recovery of dwelling ventilation systems (chapter 5);
- example of investigated calculation of energy and exergy demands in buildings and building services (chapter 6).

This item recapitulates and answers the research questions from the research introduction and summarises the main findings from chapters 2-6.

7.2.1. Reference environments to determine the exergy of air in buildings

The definition of exergy cannot be dissociated from that of the environment. In systems operating at steady state and relatively far from environmental conditions, fixed standard environmental conditions can be answered without significant loss of

accuracy. This is often not the case for buildings and building services, which operate relatively near environmental conditions. When considering exergy definitions applicable to buildings and building services, it is thus essential to carefully consider the effect of changes in environmental conditions.

Chapter 2 critically analyses the influence of possible definitions of the standard state of environmental air, to determine the exergy of air in buildings. The exergy value of air entails three contributions, a thermal one related to the air temperature, a mechanical one related to the air pressure, and a chemical one related to the humidity ratio of the air. In this work, a simple equation is derived that can be used by HVAC and building designers to calculate the exergy of humid air based on standard HVAC parameters like mass flow and humidity ratio. The novelty of this equation is that it allows the reference temperature to be varied. Expressions found in the literature required a fixed value of reference temperature, which is not convenient for dynamic calculations. This new equation is then used to calculate exergy values for three different climates throughout the year. The results are then used to investigate whether a characteristic temperature and humidity ratio of air can be used to calculate a reasonably accurate exergy value of air, without the need to perform detailed hourly calculations. It is also investigated under which conditions it is allowable to simplify calculations by assuming that air contains no water vapour at all.

The exergy of air in buildings is determined for the outdoor climate conditions of three climate zones (a temperate sea climate zone, Lisbon PT; a cold climate zone, De Bilt NL; and a hot and humid climate zone, Bangkok TH), with specific indoor climate conditions.

The results show that it is acceptable in some climates to consider a static reference environment only, instead of a dynamic reference environment, for calculating the exergy value of air in buildings for a year. In a cold climate, the exergy value of the air strongly depends on its thermal contribution. Accordingly, the outdoor air temperature might be sufficient as a reference environment for the exergy calculation. This is not acceptable for the exergy calculation in a hot and humid (or temperate sea) climate, where the chemical contribution to exergy due to moisture can be substantial.

Exergy calculation modules found in the literature are normally based on the thermal contribution only. While this approach has the advantage of simplicity, it can entail significant accuracy losses when non-negligible latent heat loads are involved.

It is recommended to pay attention to this accuracy issue when exergy chains are calculated in buildings because exergy losses throughout the exergy chain could change strongly depending on the chosen reference and indoor air properties. This will be the case in particular when latent heat loads are involved, e.g. in humidification and dehumidification processes.

7.2.2. Exergy application for HVAC components and systems

Exergy analysis can assist HVAC and building designers in selecting technologies most likely to minimise energy resource depletion in buildings and building services. The analysis ascribes quality levels to energy types (e.g., electricity, heat)

and potentials (e.g. temperature differences from the environment). It also allows energy resource utilisation systems to be rank-ordered as to thermodynamic effectiveness, thereby indicating thermodynamic improvement possibilities. The location and magnitude of energy quality level degradation, e.g. from heat transfer (temperature drop) or energy conversion (e.g. electricity into low-grade heat), can also be made explicit.

In this item, exergy analysis is carried out for HVAC components and systems. In the level of HVAC components, critical analyses of exergy efficiency definitions are carried out for air-to-air heat exchangers (chapter 3) and vapour-compression heat pumps (chapter 4). These analyses resulted in a better understanding of exergy values and of the sensitivity of exergy efficiency definitions applied to these equipments operating at near environmental temperatures. This work introduces a dimensionless temperature as a measure to express the sensitivity of the exergy efficiency definitions to temperature variations.

In the level of HVAC systems, energy and exergy analyses for dwelling ventilation with and without air-to-air heat recovery (chapter 5) are presented, and the relative influence of heat and electricity on the exergy demand by ventilation airflows is discussed.

7.2.2.1. Air-to-air heat exchangers

Chapter 3 presents a critical analysis of existing exergy efficiency definitions with a potential for use in the design of buildings and building services. It compares energy and exergy efficiencies for a simple air-to-air sensible heat exchanger, as a first step towards better understanding how different efficiency definitions can be applied to buildings and building services operating at near-environmental temperatures, and towards generating ideas on how to summarise and present this information to building services engineers and building designers.

Many existing exergy efficiency definitions were developed for use with relatively large temperature differences and comparatively far from environmental temperatures. However, many heat exchange processes in building systems take place relatively close to environmental temperature (Wall, 1990), and may involve rather small temperature differences. At such conditions exergy efficiency may become more sensitive to relatively small changes in environmental temperature, and to the relation between heat exchange temperatures and environmental temperature.

Exergy efficiency of a system (or equipment) could be defined in various ways, depending on the significance of various conditions such as sensitivity for changes in the system, applicability in practice and accuracy. Woudstra (2002) distinguishes two different sorts of exergy definitions, i.e. universal efficiencies and functional efficiencies. The universal efficiency has been criticised in the literature (Woudstra, 2002) as not being sufficiently sensitive to changes in a system. Also the universal efficiency makes no distinction regarding usability (exergy transferred inside the heat exchanger or discarded with the outgoing flows), or intended use in heating or cooling mode. Functional efficiency is more sensitive to exergy loss within the system, and yields 'net' efficiency values since it excludes the exergy discarded with outgoing flows.

Chapter 3 uses functional exergy efficiency and a dimensionless temperature to gain insight into the effect of varying temperatures in heat exchange at near-environmental temperatures, e.g. in buildings and building services for indoor climate control. The analysis is performed with an air-to-air sensible heat exchanger model for heating purposes, assuming constant values of the exchanger heat transfer effectiveness. An analysis of universal exergy efficiency for air-to-air heat exchangers operating at near-environmental temperatures was done by Boelman and Sakulpipatsin (2005).

The analysis results show that the functional exergy efficiency in combination with the dimensionless temperature introduced in this work can be used as a guide for selecting temperatures to operate heat exchangers near environmental temperature in an exergy efficient way. The dimensionless temperature expresses a distance between the hot inlet air temperature and the environmental air temperature, relative to the inlet air temperature difference. The functional exergy efficiency shows that not only heat exchanger performance (expressed in terms of exchanger heat transfer effectiveness), but also the relationship between temperatures (in the heat exchanger and of the environment) is important to operate the heat exchanger efficiently.

For exergy efficient operation, it is recommended to select temperature combinations corresponding to dimensionless temperatures not less than one. In this range, the equivalent temperatures of hot and cold air are above environmental temperature. Although the hot air loses warm exergy, the heat exchange is exergy efficient because there is sufficient gain of warm exergy by the cold air. In the range of dimensionless temperature less than one, the equivalent temperature of cold air is below the temperature of the environment and heating this cold air implies losing its cool exergy. The cool exergy of the cold air could be better used for cooling another flow of matter, at or below temperature of the environment. From an exergy viewpoint, it is inefficient to use air above environmental temperature to heat air below environmental temperature, even if the heat exchanger has a high exchanger heat transfer effectiveness.

The insight gained from this analysis can be useful when designing a heat exchange system, for example when deciding between using a heat exchanger of higher exchanger heat transfer effectiveness and pre-heating the outside air (e.g. by using a sunspace or the underground). In practice, such a decision would also have to consider the additional pressure drop usually associated with higher exchanger heat transfer effectiveness versus the possibility of using passive means to pre-heat environmental air.

7.2.2.2. Vapour-compression heat pumps

An analysis of exergy efficiency definitions applied for a simple air-to-air sensible heat exchanger has shown the extent to which the use of output or product-based exergy efficiency definitions can lead to substantially different efficiency values. The next step is to systematically investigate which exergy efficiencies are the most appropriate ones for heat pump applications at near ambient conditions in buildings.

In chapter 4, the universal and functional exergy efficiency definitions are critically analysed for a simple vapour-compression heat pump cycle where the cycle operates at

near-environmental temperatures and for space cooling application. The analysis is performed with the heat pump cycle, assuming constant values of the second-law efficiency. The analysis results are presented as a function of a dimensionless temperature. This dimensionless temperature is different from the one that is used in chapter 3, in order to be applicable for the heat pump operating in cooling mode.

The results indicate that, for a same set of heat pump operating conditions (which are inlet temperatures and second-law efficiency), changes in environmental air temperature can lead to significant variations in the universal and functional exergy efficiencies of the system.

The results also show how exergy efficiency values can differ, depending on whether the efficiency definition considers gross exergy inputs and outputs (universal exergy efficiency) or neglects discarded exergy flows (functional exergy efficiency). Because the functional exergy efficiency considers only net exergy flows, it can be more sensitive to temperature changes than the universal exergy efficiency.

The functional exergy efficiency is recommended to be used as a performance criterion for the heat pump for space cooling application, especially when the temperature of the environmental air is between the inlet temperatures of the hot and cold air streams and also close to the inlet temperature of the hot air stream.

7.2.2.3. Dwelling ventilation systems

Heat recovery from ventilation airflow plays an increasingly important role in minimising energy needs in buildings, especially in cold and moderate climates. Such heat recovery systems rely on the input of electric power (to drive fans, heat pumps, etc.) in order to recover thermal energy. Since electricity input is relatively small compared to the amounts of thermal energy recovered, such systems are efficient from an energy point of view. One important yet often overlooked aspect, however, is the difference in ‘quality’ between the high-grade electricity input and the lower grade thermal energy recovered.

Chapter 5 presents steady-state energy and exergy analyses for dwelling ventilation systems using exhaust ventilation with and without air to air heat recovery from ventilation airflow, in winter conditions in the Netherlands. Energy and exergy analysis results are presented in terms of heat and electricity, on an instantaneous and a daily basis.

In terms of energy demand, the balanced ventilation system with the heat recovery unit is the most efficient alternative for dwelling ventilation in the heating season in the Netherlands, since the system needs less energy. However, in terms of exergy demand, the heat recovery unit requires more exergy because of the additional electricity input needed to overcome the pressure drop of the heat exchanger. The thermal exergy recovered from the exhaust air is relatively small, since the exhaust air temperature is relatively close to the environmental air temperature, and thus has a low thermal exergy value.

From the viewpoint of total exergy consumption (thermal exergy by a ventilation airflow plus electricity exergy by a ventilation unit) at room level, it could make sense to use heat recovery only when the temperature of the environmental air is low

enough to compensate the additional need for electricity, when the temperature of the environmental air is not too low let ventilation air bypass the heat recovery unit, or if possible by operating the heat recovery unit at low ventilation airflow rate. Nevertheless, the ventilation airflow rate must be qualified to guarantee indoor occupancy conditions.

This analysis considered the final exergy demand at room level, without focusing on the efficiency of energy conversion and delivery processes (electricity, heating system). This conclusion may change if, instead of the exergy demand for ventilation only, the exergy consumptions for ventilation and heating are considered, including the exergy efficiencies of electricity and heat production.

7.2.3. Exergy application for building systems

Chapter 6 presents some examples of energy and exergy analysis of a building including its building services. The analysis is based on a build-up model from the energy demand of the building to the energy supply side. The build-up model is adapted from the energy flow model accounting through building services equipment, from primary energy supply systems to buildings, developed by Schmidt (2004). The analysis method used for the examples is intended to enable building designers (and building engineers) to compare, in terms of exergy, the impact of improvements in the building envelope and in building services.

The examples of the energy and exergy analysis of the building and its building services with some changes of their parametric values provided in chapter 6 are studied by using the building simulation tool TRNSYS. The analysis results show that:

- Reducing thermal energy supplied by heating equipments to a building in a cold climate remains a key concern in the cold days. While the internal thermal energy gains are crucial in the energy analysis, the exergy analysis shows that the solar exergy gain creates the main exergy losses when cooling is needed. These solar exergy gains should be minimized, or better captured to be useful somewhere else (for instance for domestic hot water production or electricity generation; Sakulpipatsin et. al., 2007b).
- Exergy losses in the building services depend on a temperature level of the thermal energy supply and (electric) auxiliary energy required by the building services. This is applicable to both heating and cooling cases.
- Energy analysis shows that energy distribution and energy emission systems are quite efficient and it is worth to try reducing the energy demand of the building. However, the exergy analysis shows that to increase the rational use of energy, the building envelope offers little possibilities because it is already near-environmental, but there is a high potential for improvements in the thermal energy emission and thermal energy generation systems. In addition, the exergy analysis shows that solar gain could play a vital role in this improvement.

An ideal building and building services combinations in a cold country would, on one hand, operate as closely as possible to the required temperature (e.g. indoor temperature) and use the lowest amount of electricity, so as to minimize exergy

inputs from the external energy supply system. On the other hand, it is necessary to supply energy that has exergy of a matched level required by the building and the building services, to minimize exergy losses in the local energy conversion: energy that has a higher exergy value is more appropriate to be supplied to other processes that need this higher exergy value. In real systems, a compromise needs to be sought between system size and costs on the one hand and effective utilization of energy resources on the other hand.

7.3. Future research recommendations and outlook

In this item, recommendations for future research and outlook are given. The recommendations for future research are based on each chapter of the dissertation. The outlook is related to dissemination of the knowledge obtained during the course of this research to people in the building profession.

- **Chapter 2:** work described in chapter 2 could be applied for development of an exergy psychometric chart of humid air, where the exergy value of the air is expressed in terms of humidity ratio and temperature of the air. The exergy psychometric chart could be somehow dynamic that reference values used for exergy calculation can be changed according to the actual environment.
- **Chapters 3 and 4:** as for further development of measures to express exergy values in HVAC components and systems, the sensitivity of the exergy efficiency definitions should be further investigated for more complex systems or for other applications in HVAC systems. For example, for a heat exchanger, the effects of different types of (air) flow should be taken into the account and should include the exergy loss by dissipation of mechanical energy in the heat exchanger. For a heat pump, the sensitivity of the exergy efficiency definitions should be also studied in other cases, e.g. the case that the heat pump is used for space heating application.
- **Chapter 5:** for building energy supply and HVAC systems, minimisation of exergy uses of a more complex system (e.g. ground cooling/heating system) should be studied to find out an optimal solution of energy supply to buildings in terms of exergy.
- **Chapter 6:** for building systems, study of building exergy demands should also consider dynamic occupancy (or building operation) requirements (e.g. for lighting, domestic hot water and air movement etc). The building exergy demands should be considered in various forms of the energy carrier (e.g. heat and electricity). This would make the calculated results of the exergy values more accurate.
- **Outlook:** to disseminate the knowledge obtained from this research, some learning tools should be developed and then used for teaching students and practitioners in a field related to building and HVAC system design. Development of a simple graphical user input interface for an exergy-analysis tool for building and building services design was initiated during the course of this PhD research. The tool is described in (Sakulpipatsin et. al., 2005). In addition, students gave their feedbacks and their main point of concern was to

improve the accessibility of this exergy-analysis tool mainly meant to be used by building and HVAC system designers.

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Summary

The concept of exergy is stated as the maximum work that can be obtained from an energy flow or produced by a system. The fraction of exergy content expresses the quality of an energy source or flow. This concept can be used to combine and compare all flows of energy according to their quantity and quality. Unlike energy, exergy is always destroyed during conversions because of the irreversible nature of energy conversion process. The exergy concept enables people to articulate what is consumed by all working systems (e.g. man-made systems like thermo-chemical engines and heat pumps, or biological systems including the human body) when energy and/or materials are transformed for human use.

Exergy analysis can give insight into the extent to which the quality levels of energy supply (e.g. high-temperature combustion) and energy demand (e.g. low-temperature heat) are matched. High-valued energy such as electricity and mechanical work consists of pure exergy. Energy which has a very limited convertible potential, such as heat close to room air temperature, is low-valued energy. Low exergy heating and cooling systems therefore allow the use of low-valued energy, which can be delivered by sustainable energy sources, as well. However, in most cases, the low-valued energy demand is met with high quality sources, such as fossil fuels or using electricity.

Many researchers and practicing engineers refer to exergy methods as powerful tools for developing and optimizing systems and processes. Exergy losses clearly pinpoint the locations, causes and sources of deviations from ideal circumstances in a system. Exergy efficiencies are measures of the approach to ideal. Nevertheless, exergy analysis is only used by a small group of people, because the analysis method might seem cumbersome or complex (e.g. choosing a suitable reference environment) to some people and the results might seem difficult to interpret and understand.

In building profession, the exergy concept has been applied to the built environment. Some researchers have also used the exergy concept in a context of sustainable development. In the last few years, a working group of the International Energy Agency has been formed within the Energy Conservation in Buildings and Community Systems programme: “Low Exergy Systems for Heating and Cooling of Buildings; IEA Annex 37”. The overall objective of the IEA Annex 37 was to promote the rational use of energy by means of low-valued and environmentally sustainable energy sources. This PhD research has been carried out in close collaboration with the international LowExNet network of exergy researchers, which is a follow-up of the annex. During the course of the PhD research, the COSTexergy project (COST Action 24) and the EOS-LT project (entirely financed by SenterNovem) were initiated and have been running. In addition, research outputs of the PhD research have served as inputs to the formulation of the annex on low exergy systems for high-performance buildings and communities (IEA Annex 49).

The objective of this PhD research is to develop knowledge into the applicable domains and potential added values of exergy analysis in the built environment, by

studying under what conditions exergy could function as a useful concept for the built environment. The research is carried out in the levels of HVAC components and systems and of building systems, and provides metrics that can be used to quantify and express exergy values in buildings and HVAC systems.

Firstly, the influence of possible definitions of the standard state of environmental air are critically analysed in order to determine the exergy of air in buildings. The exergy value of air entails three contributions, a thermal one related to the air temperature, a mechanical one related to the air pressure, and a chemical one related to the humidity ratio of the air. The possibility to calculate the exergy of air in buildings, based on only one or two of these contributions, for example expressed by a characteristic air temperature and/or air as dry air, is explored for three different locations on earth. These values are compared to those calculated using hourly statistical climate data during one year. The results show that it is acceptable in some climates to consider a static reference environment only, instead of a dynamic reference environment, for calculating the exergy value of air in buildings for a year. In a cold climate, the exergy value of the air strongly depends on its thermal contribution. Accordingly, the outdoor air temperature might be sufficient as a reference environment for the exergy calculation. This is not acceptable for the exergy calculation in a hot and humid (or temperate sea) climate, where the chemical contribution to exergy due to moisture can be substantial.

Secondly, exergy analysis is carried out for HVAC components and systems.

In the level of HVAC components, critical analyses of exergy efficiency definitions are carried out for air-to-air sensible heat exchangers and vapour compression heat pumps. The exergy efficiency definitions that were studied in the work are: the universal ones in which gross exergy inputs and outputs are considered, and the functional ones in which net exergy flows are considered respectively. A dimensionless temperature is defined and used to illustrate the analysis results. The dimensionless temperature expresses a distance between the hot (or cold) inlet air temperature and the environmental air temperature, relative to the inlet air temperature difference. These analyses resulted in a better understanding of exergy values and of the sensitivity of exergy efficiency definitions applied to these equipments operating at near environmental temperatures.

The functional exergy efficiency in combination with the dimensionless temperature can be used as a guide for selecting temperatures to operate heat exchangers near environmental temperature in an exergy efficient way. The functional exergy efficiency shows that not only heat exchanger performance (expressed in terms of exchanger heat exchanger effectiveness), but also the relationship between temperatures (in the heat exchanger and of the environment) is important to operate the heat exchanger efficiently. The analysis for the air-to-air sensible heat exchangers can be useful when designing a heat exchange system, for example when deciding between using a heat exchanger of higher exchanger heat transfer effectiveness and pre-heating the outside air (e.g. by using a sunspace or the underground).

The functional exergy efficiency is also recommended to be used as a performance criterion for the heat pump for space cooling application, especially when the

temperature of the environmental air is between the inlet temperatures of the hot and cold air streams and also close to the inlet temperature of the hot air stream.

In the level of HVAC systems, energy and exergy analyses for dwelling ventilation with and without air-to-air heat recovery, in winter conditions in the Netherlands, are presented. The analyses are carried out on an instantaneous and a daily basis. The analysis results show that, from the viewpoint of total exergy consumption (which is the summation of thermal exergy by a ventilation airflow and electricity exergy by a ventilation unit) at room level, it could make sense to use heat recovery only when the environmental air temperature is low enough to compensate the additional need for electricity, when the temperature of the environmental air is not too low let ventilation air bypass the heat recovery unit, or if possible by operating the heat recovery unit at low ventilation airflow rate. Nevertheless, the ventilation airflow rate must be qualified to guarantee the indoor occupancy conditions.

Lastly, a method for energy and exergy analysis of a building and building services is proposed. The analysis is based on a build-up model from the energy demand of the building side to the energy supply side. This method is intended to enable building designers (and building engineers) to compare, in terms of exergy, the impact of improvements in the building envelope and in building services. In addition, some examples of the energy and exergy analysis of the building and its building services with some changes of their parametric values are studied by using the building simulation tool TRNSYS. The analysis results show that, in terms of exergy, solar exergy gains in a cold day create the main exergy losses when cooling is needed. These solar exergy gains should be minimized, or better captured to be useful somewhere else e.g. for domestic hot water production or electricity generation. Exergy losses in the building services depend on a temperature level of the thermal energy supply and (electric) auxiliary energy required by the building services, and this is applicable for both heating and cooling cases.

This research provides knowledge that is essential to future development of design instruments and guidelines for exergy efficient building and building services design. Yet, the exergy analyses for the HVAC components and systems and for the building systems are carried out only with outdoor conditions of a cold climate. The exergy analyses for other climates are excluded from this study, since the standard states of environmental air in different climates for the analyses are not similar and should be carefully defined in a proper way. In addition, buildings in different climates are mostly designed in different ways. Exergy in buildings and building services, where they have other different and more complex types, is an interesting topic to study in the near future, and at the same time the knowledge obtained from the research should be disseminated to students and practitioners in a field related to building and HVAC system design.

Poppong Sakulpipatsin

Samenvatting

Het exergieconcept is gedefinieerd als de maximale hoeveelheid arbeid die uit een energiestroom of een energiesysteem verkregen kan worden. De exergiefractie drukt de kwaliteit van een energiebron of een stroom uit. Dit concept kan worden gebruikt om alle energiestromen volgens hun hoeveelheid en kwaliteit te combineren en te vergelijken. In tegenstelling tot energie, wordt exergie altijd vernietigd tijdens omzettingen wegens het onomkeerbare karakter van energieomzettingsprocessen. Het exergieconcept maakt het mogelijk om aan te geven wat er vernietigd wordt bij alle mogelijke werkende systemen (bv. kunstmatige, door de mens gemaakte, systemen zoals thermoschemische motoren en warmtepompen, of biologische systemen met inbegrip van het menselijke lichaam) waarbij energie en/of materialen voor menselijk gebruik worden omgezet.

Exergieanalyse kan inzicht geven in de mate waarin de kwaliteitsniveaus van energieaanbod (bv. verbranding bij hoge temperatuur) en energievraag (bv. lage temperatuur thermische energie) met elkaar in overeenstemming zijn. Hoogwaardige energie, zoals elektriciteit en mechanische arbeid, bestaat volledig uit exergie. Energie, die slechts voor een zeer beperkt aantal doelen ingezet kan worden, zoals thermische energie met een temperatuur dicht bij de omgevingstemperatuur, is laagwaardige energie. Verwarmings- en koelsystemen met een lage kwaliteit van energie kunnen gebruik maken van energie met een lage kwaliteit, die ook door duurzame energiebronnen geleverd kan worden. Echter wordt, in de meeste gevallen, aan de vraag naar energie met een lage kwaliteit voldaan door gebruik te maken van bronnen met een hoge kwaliteit, zoals fossiele brandstoffen of elektriciteit.

Veel onderzoekers en praktiserende ingenieurs geven aan dat op het exergieconcept gebaseerde methoden krachtige hulpmiddelen zijn om systemen en processen te ontwikkelen en te optimaliseren. Exergie verliezen verwijzen duidelijk naar de plaatsen, de oorzaken en de bronnen van afwijkingen van ideale thermodynamische omstandigheden in een systeem. De exergie-efficiëntie is een maat voor de afwijking van deze ideale omstandigheden. Niettemin wordt exergieanalyse slechts gebruikt door een kleine groep mensen omdat deze analysemethode als tijdrovend en moeilijk ervaren wordt (bv. het kiezen van een geschikte standaardomgeving) en omdat de resultaten als moeilijk te interpreteren en te begrijpen worden gezien.

In de bouwwereld is het exergieconcept toegepast op de gebouwde omgeving. Sommige onderzoekers hebben het exergieconcept ook gebruikt in samenhang met duurzame ontwikkeling. In de laatste jaren is er een werkgroep gevormd, onder auspiciën van het International Energy Agency, in het kader van het Energy Conservation in Buildings and Community Systems Program, met als naam: „Low Exergy Systems for Heating and Cooling of Buildings; IEA Annex 37“. De algemene doelstelling van IEA Annex 37 was om het rationele gebruik van energie door middel van energiebronnen van lage kwaliteit en die ecologisch gezien duurzaam zijn aan te moedigen. Dit promotieonderzoek is uitgevoerd in nauwe samenwerking met onderzoekers werkzaam binnen het internationale netwerk,

LowExNet, dat gezien kan worden als een vervolg van Annex 37. Tijdens het promotieonderzoek werden het COSTeXergy project (COST Action C24) en het EOS-LT project (dat volledig door SenterNovem wordt gefinancierd) ingediend en goedgekeurd, beide lopen nu. Bovendien zijn onderzoeksresultaten van het promotieonderzoek gebruikt bij de formulering van onderzoek binnen Annex 49 op het gebied van de toepassing van exergiesystemen met een lage kwaliteit voor goed functionerende gebouwen en gemeenschappen (IEA Annex 49).

De doelstelling van dit promotieonderzoek is om kennis te ontwikkelen over de gebieden waarbinnen exergieanalyse in de gebouwde omgeving toegepast kan worden en wat de toegevoegde waarde daarvan is, door te onderzoeken onder welke omstandigheden het gebruik van het exergieconcept in de gebouwde omgeving nuttig kan zijn. Het onderzoek is uitgevoerd op de niveaus van HVAC componenten en systemen en van systemen betreffende het gebouw en heeft geleid tot een aanpak waardoor het mogelijk is om de exergiewaarden van die systemen en van onderdelen daarvan te bepalen.

In de eerste plaats is kritisch nagegaan wat de invloed is van mogelijke definities van de standaardtoestand van omgevingslucht om de exergiewaarde van lucht in gebouwen te bepalen. De exergiewaarde van lucht is opgebouwd uit drie bijdragen: een thermische die samenhangt met de luchttemperatuur, een mechanische die samenhangt met de luchtdruk, en een chemische die samenhangt met de relatieve luchtvochtigheid. De mogelijkheid om de exergiewaarde van lucht in gebouwen te berekenen, die slechts op één of twee van deze bijdragen gebaseerd is, bij voorbeeld door een karakteristieke luchttemperatuur en/of lucht als droge lucht te beschouwen, is onderzocht voor drie verschillende plaatsen op de aarde. Deze exergiewaarden zijn vergeleken met die welke berekend zijn op basis van statistische klimaatgegevens per uur gedurende één jaar. De resultaten tonen aan dat het in sommige klimaten aanvaardbaar is om alleen van een statische referentieomgeving gebruik te maken, in plaats van de dynamische referentieomgeving, voor het berekenen van de exergiewaarde van lucht in gebouwen voor een jaar. In een koud klimaat, hangt de exergiewaarde van de lucht sterk af van zijn thermische bijdrage. Dienovereenkomstig kan de buitenluchttemperatuur volstaan om de exergiewaarde van binnenlucht te kunnen berekenen. Dit is niet aanvaardbaar voor de berekening van de exergiewaarde van lucht in een heet en vochtig (of gematigd zee) klimaat, waar de chemische bijdrage tot de exergie van lucht, wegens de luchtvochtigheid, aanzienlijk kan zijn.

Ten tweede is exergieanalyse toegepast op HVAC componenten en systemen.

Op het niveau van HVAC componenten is een kritische analyse uitgevoerd van mogelijke exergie-efficiëntie definities voor lucht-lucht warmtewisselaars voor de overdracht van voelbare thermische energie en voor compressie warmtepompen. De definities van exergie-efficiëntie die in dit proefschrift onderzocht werden, zijn: de universele efficiënties, waarin respectievelijk de bruto toegevoerde exergie en de bruto afgevoerde exergie, en de functionele efficiënties, waarin de netto exergiestromen gebruikt zijn. Een dimensieloze temperatuur is gedefinieerd en gebruikt om het gedrag van de exergie-efficiënties als functie daarvan weer te geven. De dimensieloze temperatuur is gedefinieerd als de verhouding van het verschil tussen de inlaat temperatuur van de warme (of de koude) lucht en de temperatuur van de buitenlucht ten opzichte van het verschil tussen de temperaturen van de beide

luchtinlaat stromen. Deze analyse resulteerde in een beter begrip voor de exergiewaarden en tot inzicht in de gevoeligheid van de genoemde exergie-efficiënties voor genoemde apparatuur die dicht bij omgevingstemperatuur werkt.

De functionele exergie-efficiëntie in combinatie met de dimensieloze temperatuur kan worden gebruikt voor het selecteren van temperaturen om warmtewisselaars die dichtbij omgevingstemperatuur toegepast worden exergie efficiënt te laten functioneren. De functionele exergie-efficiëntie laat zien dat niet alleen de karakteristiek van een warmtewisselaar (uitgedrukt in de effectiviteit van de warmtewisselaar), maar ook het verband tussen de temperaturen (in de warmtewisselaar en die van de omgeving) belangrijk is om de warmtewisselaar efficiënt te laten werken. De analyse voor lucht-lucht warmtewisselaars kan nuttig zijn wanneer een warmtewisselingssysteem ontworpen wordt, bijvoorbeeld om te beslissen tussen het gebruik van een warmtewisselaar met een hogere effectiviteit van warmteoverdracht en het voorverwarmen van de buitenlucht (bv. door warmte uit een door de zon verwarmde ruimte of bodemwarmte te gebruiken).

Het gebruik van de functionele exergie-efficiëntie wordt ook aanbevolen om als prestatie criterium voor een warmtepomp voor ruimtekoeling toegepast te worden, vooral wanneer de temperatuur van de omgevingslucht ligt tussen de ingangstemperaturen van de hete en koude luchtstromen en ook niet veel verschilt van de inlaattemperatuur van de hete luchtstroom.

Op het niveau van HVAC systemen, worden de energie- en de exergieanalyses voor woningventilatie met en zonder lucht-lucht warmteterugwinning, onder winterse omstandigheden in Nederland, gepresenteerd. De analyses zijn uitgevoerd op een instantane en op een dagelijkse basis. De analyseresultaten tonen aan dat, uit het oogpunt van totale exergieconsumptie (die de som is van de thermische exergie van de ventilatieluchtstroom en de elektrische exergie nodig voor het ventilatie systeem) op ruimteniveau, het verstandig kan zijn om warmteterugwinning slechts te gebruiken wanneer de omgevingsluchttemperatuur laag genoeg is om de extra behoefte aan elektriciteit te compenseren, en wanneer de temperatuur van de omgevingslucht niet te laag is om dan de ventilatielucht niet voor de warmteterugwinning te gebruiken, of, als dat mogelijk is, om de luchtstroom door de lucht-lucht warmte wisselaar te verminderen. Niettemin moet de ventilatieluchtstroom zodanig zijn dat de binnenluchtkwaliteit voor de aanwezigen voldoet aan de eisen.

Ten slotte wordt een methode voor het uitvoeren van een energie- en een exergieanalyse van een gebouw en van de bijbehorende bouwsystemen voorgesteld. De analyse is gebaseerd op een model waarbij begonnen wordt om de energievraag van het gebouw te bepalen en tenslotte nagegaan wordt hoe daarin van aanbodzijde voorzien kan worden. Deze methode is bedoeld om gebouwontwerpers (en bouwingenieurs) in staat te stellen om, op basis van exergie, het effect van verbeteringen van het gebouw en de bouwsystemen te kunnen vergelijken. Bovendien zijn een aantal voorbeelden van de energie- en de exergieanalyse van gebouw en de bouwsystemen nader uitgewerkt en bestudeerd, waarbij essentiële parameters veranderd zijn, hierbij werd gebruik gemaakt van het gebouwssimulatieprogramma TRNSYS. De analyseresultaten tonen aan dat, in termen van exergie, de bijdrage van ingestraalde zonne-exergie op een koude dag de oorzaak van relevante exergieverliezen is wanneer koeling wordt toegepast. Deze

opwarming ten gevolge van ingestraalde zonne-exergie moet geminimaliseerd worden of, beter, worden opgevangen om elders gebruikt te worden bv. om warm water te produceren of voor elektriciteitsopwekking. De exergieverliezen van gebouw- energiesystemen hangen af van het temperatuurniveau van de toegevoerde thermische energie en de daarvoor benodigde toegevoerde (elektrische) energie, dit geldt zowel voor verwarmen als voor koelen.

Dit onderzoek verstrekt kennis die essentieel is voor de toekomstige ontwikkeling van ontwerpinstrumenten en richtlijnen voor het exergie-efficiënt ontwerpen van gebouwen en van de gebouwssystemen. De exergieanalyses voor HVAC componenten en -systemen en voor het gebouw zijn alleen uitgevoerd voor de buitenluchtcondities van een koud klimaat. De exergieanalyses voor andere klimaten zijn geen onderwerp geweest van deze studie, aangezien de standaardtoestanden voor omgevingslucht in verschillende klimaten verschillend zijn en zorgvuldig en op een juiste manier gedefinieerd moeten worden. Bovendien worden de gebouwen in verschillende klimaten meestal op verschillende manieren ontworpen. Exergieanalyse toegepast op andere en complexere gebouwen en op energiesystemen in het gebouw, is een interessant onderwerp om in de nabije toekomst uit te voeren, en gelijktijdig zou de kennis die in dit onderzoek verkregen is moeten worden doorgegeven aan studenten en vaklieden die zich bezig houden met het ontwerp van gebouwen en van HVAC systemen.

Poppong Sakulpipatsin

Propositions/Stellingen

1. The exergy concept is useful to pinpoint magnitudes and locations of thermodynamic imperfections occurring through an energy supply chain, to realise improvements, and ultimately to reduce primary energy input.

Het exergie concept is bruikbaar om grootte en plaats van thermodynamische onvolkomenheden, die in een energie keten optreden vast te stellen, om verbeteringen te realiseren en uiteindelijk ook om de benodigde hoeveelheid primaire energie te verminderen.

2. A nice example of the strength of the exergy concept is the application of fossil primary energy for low temperature heating.

Een goed voorbeeld van de kracht van het exergie concept is de toepassing van fossiele primaire energie voor lage temperatuur verwarming.

3. The difference in ‘quality’ between the high-grade electricity input and the lower-grade thermal energy recovered should not be overlooked when designing dwelling ventilation systems.

Het verschil in „kwaliteit“ tussen de hoogwaardige elektriciteit's toevoer en de teruggewonnen laagwaardige thermische energie moeten niet veronachtzaamd worden bij het ontwerp van de ventilatie systemen van woningen.

4. Environment plays an overwhelming role in building and HVAC system design and operation. Optimal solutions for the design and the operation directly hinge on the actual indoor and outdoor climatic conditions.

De omgeving speelt een overheersende rol bij het ontwerp en het gebruik van gebouw en HVAC systemen. Optimale oplossingen voor het ontwerp en het gebruik van deze systemen worden bepaald door de actuele klimatologische condities binnen en buiten.

5. The exergy concept, based on the definition of a reference environment, is suited to be used to express, partly, the ecological component of sustainability.

Het exergie concept, gebaseerd op de definitie van een referentie omgeving, is geschikt om gebruikt te worden om er, gedeeltelijk, de ecologische component van duurzaamheid mee uit te drukken.

6. Using the exergy concept in applied sciences also leads to alleviation of actual problems in the present societies.

Het gebruik van het exergie concept in toegepaste wetenschappen leidt ook tot het oplossen van actuele problemen in de huidige samenlevingen.

7. Everything comes to equilibrium after all.

Alles komt uit eindelijk tot evenwicht.

8. People from different cultures and climates have diverse styles of working. The one is not necessarily better than the other.

Mensen uit verschillende culturen en klimaten hebben verschillende manieren van werken. De ene is niet noodzakelijker wijze beter dan de andere.

9. Cross-disciplinary education is necessary for good integral building design.

Een multi disciplinaire opleiding is nodig om een goed, integraal, gebouw ontwerp te maken.

These propositions are considered opposable and defensible and as such have been approved by the supervisor, Prof. Ir. P. G. Luscure.

Deze stellingen worden oponeerbaar en verdedigbaar geacht en zijn als zodanig goedgekeurd door de promotor, Prof. Ir. P. G. Luscure.

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Poppong Sakulpipatsin

Delft, January 2008

Curriculum Vitae

Name:

Popping Sakulpipatsin

Place and date of birth:

Suphan Buri, Thailand, 11 December 1977

Education:

- | | |
|---------|--|
| 10/2003 | MSc. in Energy Conversion & Management
University of Applied Sciences Offenburg, Offenburg, Germany |
| 10/2001 | MSc. in Computer Science
Chulalongkorn University, Bangkok, Thailand |
| 03/1999 | B.Eng. in Civil Engineering,
Chulalongkorn University, Bangkok, Thailand |

Publications:

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Nowadays, energy systems in buildings are designed based solely on the energy conservation principle. Nevertheless, this principle alone does not provide a full understanding of important aspects of energy use in buildings. From this viewpoint, exergy analysis can quantify the potential for improving the match between the quality levels of energy supply (e.g. high-temperature combustion) and energy demand (e.g. low-temperature heat), and the contribution of this match to better energy resource utilisation.

EXERGY EFFICIENT BUILDING DESIGN

The objective of this PhD research is to develop knowledge into the applicable domains and potential added values of exergy analysis in the built environment, by studying under what conditions exergy could function as a useful concept for the built environment. The research is carried out at the levels of HVAC components and systems and of building systems. It provides metrics that can be used to quantify and express exergy values in buildings and HVAC systems.