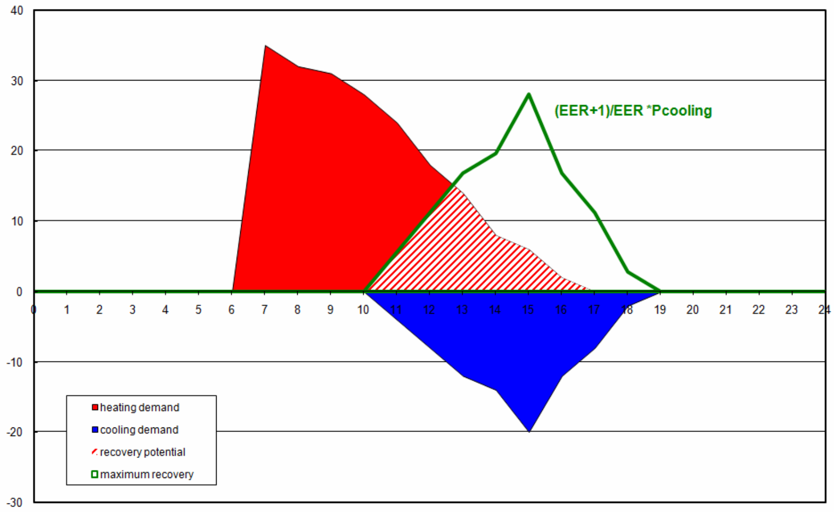
IEA-ECBCS ANNEX 48

Reversible Air-Conditioning

International Energy Agency
Energy Conservation in Buildings and Community Systems Programme

Analysis of building heating and cooling demands in the purpose of assessing the reversibility and heat recovery potentials



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Foreword

This document reports on a piece of work carried out in Subtask 1 "***Analysis of building heating and cooling demands and of equipment performances***" of IEA Annex 48 and is based upon the contribution of the participating countries.

This publication is an official Annex Report. It focuses on the issue of determining the heating and cooling demands of buildings in order to analyse the potential of energy savings obtained by the selection of a reversibility or recovery-based heat pumping solution.

It is aimed at building and HVAC designers as well as at researchers in the field.

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Editor

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Preface

International Energy Agency

The International Energy Agency (IEA) was established in 1974 within the framework of the Organization for Economic Co-operation and Development (OECD) to implement an international energy program. A basic aim of the IEA is to foster cooperation among the twenty-five IEA participating countries and to increase energy security through energy conservation, development of alternative energy sources and energy research, development and demonstration (RD&D).

Energy Conservation in Buildings and Community Systems

The IEA sponsors research and development in a number of areas related to energy. The mission of one of those areas, the ECBCS - Energy Conservation for Building and Community Systems Program, is to facilitate and accelerate the introduction of energy conservation, and environmentally sustainable technologies into healthy buildings and community systems, through innovation and research in decision-making, building assemblies and systems, and commercialization. The objectives of collaborative work within the ECBCS R&D program are directly derived from the on-going energy and environmental challenges facing IEA countries in the area of construction, energy market and research.
ECBCS addresses major challenges and takes advantage of opportunities in the following areas:

· exploitation of innovation and information technology;
· impact of energy measures on indoor health and usability;
· integration of building energy measures and tools to changes in lifestyles, work environment alternatives, and business environment.

The Executive Committee

Overall control of the program is maintained by an Executive Committee, which not only monitors existing projects but also identifies new areas where collaborative effort may be beneficial. To date the following projects have been initiated by the executive committee on Energy Conservation in Buildings and Community ((*) indicates work is completed):

Annex 1: Load Energy Determination of Buildings (*)
Annex 2: Ekistics and Advanced Community Energy Systems (*)
Annex 3: Energy Conservation in Residential Buildings (*)
Annex 4: Glasgow Commercial Building Monitoring (*)
Annex 5: Air Infiltration and Ventilation Centre
Annex 6: Energy Systems and Design of Communities (*)
Annex 7: Local Government Energy Planning (*)
Annex 8: Inhabitants Behaviour with Regard to Ventilation (*)
Annex 9: Minimum Ventilation Rates (*)
Annex 10: Building HVAC System Simulation (*)
Annex 11: Energy Auditing (*)
Annex 12: Windows and Fenestration (*)
Annex 13: Energy Management in Hospitals (*)
Annex 14: Condensation and Energy (*)
Annex 15: Energy Efficiency in Schools (*)
Annex 16: BEMS 1- User Interfaces and System Integration (*)
Annex 17: BEMS 2- Evaluation and Emulation Techniques (*)
Annex 18: Demand Controlled Ventilation Systems (*)
Annex 19: Low Slope Roof Systems (*)
Annex 20: Air Flow Patterns within Buildings (*)
Annex 21: Thermal Modelling (*)

Annex 22: Energy Efficient Communities (*)
Annex 23: Multi Zone Air Flow Modelling (COMIS) (*)
Annex 24: Heat, Air and Moisture Transfer in Envelopes (*)
Annex 25: Real time HEVAC Simulation (*)
Annex 26: Energy Efficient Ventilation of Large Enclosures (*)
Annex 27: Evaluation and Demonstration of Domestic Ventilation Systems (*)
Annex 28: Low Energy Cooling Systems (*)
Annex 29: Daylight in Buildings (*)
Annex 30: Bringing Simulation to Application (*)
Annex 31: Energy-Related Environmental Impact of Buildings (*)
Annex 32: Integral Building Envelope Performance Assessment (*)
Annex 33: Advanced Local Energy Planning (*)
Annex 34: Computer-Aided Evaluation of HVAC System Performance (*)
Annex 35: Design of Energy Efficient Hybrid Ventilation (HYBVENT) (*)
Annex 36: Retrofitting of Educational Buildings (*)
Annex 37: Low Exergy Systems for Heating and Cooling of Buildings (LowEx) (*)
Annex 38: Solar Sustainable Housing (*)
Annex 39: High Performance Insulation Systems (*)
Annex 40: Building Commissioning to Improve Energy Performance (*)
Annex 41: Whole Building Heat, Air and Moisture Response (MOIST-ENG) (*)
Annex 42: The Simulation of Building-Integrated Fuel Cell and Other Cogeneration Systems (FC+COGEN-SIM) (*)
Annex 43: Testing and Validation of Building Energy Simulation Tools (*)
Annex 44: Integrating Environmentally Responsive Elements in Buildings
Annex 45: Energy Efficient Electric Lighting for Buildings
Annex 46: Holistic Assessment Tool-kit on Energy Efficient Retrofit Measures for Government Buildings (EnERGo)
Annex 47: Cost Effective Commissioning of Existing and Low Energy Buildings
Annex 48: Heat Pumping and Reversible Air Conditioning
Annex 49: Low Exergy Systems for High Performance Buildings and Communities
Annex 50: Prefabricated Systems for Low Energy Renovation of Residential Buildings
Annex 51: Energy Efficient Communities
Annex 52: Towards Net Zero Energy Solar Buildings
Annex 53: Total Energy Use in Buildings: Analysis & Evaluation Methods
Annex 54: Analysis of Micro-Generation & Related Energy Technologies in Buildings
Working Group - Energy Efficiency in Educational Buildings (*)
Working Group - Indicators of Energy Efficiency in Cold Climate Buildings (*)
Working Group - Annex 36 Extension: The Energy Concept Adviser (*)

Participating countries in ECBCS:
Australia, Austria, Belgium, Canada, P.R. China, Czech Republic, Denmark, Finland, France, Germany, Greece, Italy, Japan, Republic of Korea, the Netherlands, New Zealand, Norway, Poland, Portugal, Spain, Sweden, Switzerland, Turkey, United Kingdom and the United States of America.

What is Annex 48?

Environmental concerns and the recent increase of energy costs open the door to innovative techniques to provide heating and cooling in buildings. Among these techniques, heat pumps represent an area of growing interest. Heat pumping is probably today one of the quickest and safest solutions to save energy and to reduce CO_2 emissions. Substituting a heat pump to a boiler may save more than 50% of primary energy, if electricity is produced by a modern gas-steam power plant.

The heat pump market was, till now, concentrated on residential buildings. A growing attention is now given to new and existing non-residential buildings where heating and cooling demands co-exist. In many non-residential buildings, an attractive energy saving opportunity consists in using the refrigeration machine for heat production. This can be done by condenser heat recovery whenever there is some simultaneity between heating and cooling demands. When there is no simultaneity, reversibility has to be looked for.

This is the matter considered in the frame of the International Energy Agency project: IEA-ECBCS Annex 48 "Heat pumping and reversible air conditioning".

What are the main aims of Annex 48 ?

The aim of the project is to promote the most efficient combinations of heating and cooling techniques in air-conditioned buildings, thanks to heat recovery and reversible systems. The main goals are:

- To allow a quick identification of heat pumping potentials in existing buildings;
- To help designers in preserving the future possibilities and in considering "heat pumping" solutions;
- To document the technological possibilities and heat pumping solutions;
- To improve commissioning and operation of buildings equipped with heat pump systems;
- To make available a set of reference case studies.

Which tasks are covered by Annex 48 ?

Five tasks are being performed :

Subtask 1 : Analysis of building heating and cooling demands and of equipment performances.

- Classification and characterization of existing building stock;
- Characterization of existing HVAC systems;
- Evaluation of the potential of heat recovery and heat pumping systems, in order to save energy and reduce CO_2 emissions;
- Development and use of simulation models to identify the heating and cooling demands and the best heat pumping potentials.

Subtask 2 : Design

- Development of a design handbook for heat pump systems.
- Development of innovative design tools addressed to architects, consulting engineers and installers, in such a way to reach a global optimisation of the whole HVAC system.

Subtask 3 : Global performances evaluation and commissioning methods

- Development of evaluation methods devoted to heat pump solutions
- Tests with synthetic data and with measured data
- Development of computer-based tool for heat pump system operation

Subtask 4 : Case studies and demonstration

- Documentation of reference case studies
- Use of case studies to test the methods and tools developed in the annex
- Conversion of most successful case studies into demonstration projects.

Subtask 5: Dissemination

- website
- paper work (leaflet, handbooks),
- workshops, seminars and conferences.

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GLOSSARY

ABBREVIATONS

AHU	Air Handling Unit
CAV	Constant Air Volume
CHF	Cool-Heating Floor
CONSOCLIM	Building Energy simulation tool
COP	Coefficient Of Performance
DF	Double Flux (Mechanical balanced mixing ventilation)
EER	Energy Efficient Ratio
FCU	Fan Coil Unit
DHW	Hot Domestic Water
HP	Heat Pump
HVAC	Heating Ventilating, Air-conditioning
LCP	Liquid Chilling package
PE	Primary Energy
SF	Single Flux (Mechanical Extract ventilation) Solar Factor
TFA	Treated Floor Area

I INTRODUCTION

The aim of the heating and cooling demand analysis is to assess the potential of heat pumping either by recovering the heat on chiller condenser, either by reconverting a chiller in a reversible heat pump. The first deliverable presented here aims to identify the most interesting cases in terms of energy saving potential among different building types, different air conditioning systems, different building loads and different climates. The second deliverable proposes an assessment of the CO_2 and Primary Energy savings of heating pumping in office buildings and health care institutions in Europe-15.

The scope of the study is here limited to office buildings and health care institutions in Europe.

The proposed methodology is as follows:
In a first step, typical cooling and heating demands in office buildings and health care institutions are defined. In this aim, building types which are representative of the air conditioned building stock are proposed : five office buildings and two health care buildings. Then, the main building parameters which can influence the heating, cooling demands are set. After, typical European climates are selected. Finally, all the building cases with the different building loads and climates have been simulated in order to get typical heating, cooling and hot domestic water demand profiles.

Based on these profiles, an analysis of heating, cooling and hot domestic water demands is carried out in order to make a first estimate of condenser heat recovery and reversibility potentials.

In a second step, classical HVAC systems are defined. An analysis of the system performances of HVAC equipment available on the market is carried out in order to define typical values.

In a third step, few configurations among all the studied building cases are selected as representative of the typical demand profiles. These selected building cases are modelled with the most representative air conditioning systems on the market. This coupled approach is necessary since the air conditioning systems interact with the building inducing extra loads such as dehumidification loads. Moreover the air conditioning systems have an importance on the heat recovery potential and the potential of reversible heat pump use. The simulation results for the building including the air conditioning systems are used to assess more finely the condenser heat recovery and reversibility potentials.

At last, an assessment of CO_2 reduction and primary energy savings in Europe-15 is achieved based on air-conditioned building stock data.

II BUILDING TYPOLOGY

1 Building geometry

The scope of the study is limited to office buildings and health care institutions. The typology of office buildings has been chosen based on a study of French building stock [FIL 2006a]:

1. The first type corresponds to buildings of huge areas mainly glazed. It is subdivided into three branches listed below with their estimated repartition in percentage of total office buildings:
   - Broad open space offices 14%
   - Broad partitioned offices 20%
   - Thin partitioned offices 33%

2. The second type concerns the buildings subjected to renovation. Its surface area is medium and this type is less glazed than the first one 8%

3. The third type concerns the small buildings existing in the industrial suburban zones 25%

Table 1: General characteristics of the office building types

Building type	**1a**	**1b**	**1c**	**2**	**3**
description	buildings of huge areas mainly glazed			Medium size retrofitted buildings	Small buildings of industrial suburban zones
	Broad open space offices	Broad partitioned offices	Thin geometry - glazed meeting room		
Stock share in % of surface	**14%**	**20%**	**33%**	**8%**	**25%**
Total surface area[1]	15 000 m^2			5 000 m^2	1 000 m^2
Floors (including ground floor)	12			4	2
Height under ceiling	3 m			3 m	2.7 m
% of surface area by type of use (with respect to useful total surface area)					
Offices	78%	55%	60%	55%	58%
Meeting rooms	16%	22%	21%	22%	18%
WC	3%	3%	3%	3%	3%
Circulations	3%	20%	16%	20%	21%
% of outside walls surface area (with respect to useful total surface area)					
Total	45%	50%	66%	67%	104%
Vertical walls (opaque and glazed)	37%	42%	58%	42%	54%
Roof	8%	8%	8%	25%	50%
Glazed surfaces (vertical)	13%	17%	26%	9%	21%
	50% of vertical surfaces with window[2]			27.5% of vertical surfaces with window[2]	34% of vertical surfaces with window[2]

[1] Area net: Sum of all areas between the vertical building components (walls, partitions,…), i.e. gross floor area reduced by the area for structural components

[2] This ratio is defined for the main facades (E/W or N/S), the other facades are assumed without any windows.

The typology of health care buildings has been chosen based on French stock:

1. The first type corresponds to a large hospital with a surface of 30 353 m² 40%

2. The second type concerns the rest-homes (care institutions including retirement homes) with a surface of 3 900 m² 60%

Table 2: General characteristics of the health care building types

Building type	**1**	**2**
description	Large hospital	Rest home
Stock share in % of surface	**40%**	**60%**
Total surface area	30 300 m²	3 900 m²
Floors (including ground floor)	5	4
Height under ceiling	3 m	3 m
% of surface area by type of use (with respect to useful total surface area)		
Offices, consultation rooms, emergency rooms	12.2%	-
Labo, restaurants, technical annexes	15.8%	-
Operation rooms	5.0%	-
Common room	-	14.7%
restaurant	-	6.1%
Technical premises	-	4.9%
Nursing room	-	4.9%
Bedrooms	22.3%	45.8%
Circulations	38.4%	17.1%
Toilets	6.2%	6.5%
% of outside walls surface area (with respect to useful total surface area)		
Total	50%	82%
Vertical walls (opaque and glazed)	30%	57%
Roof	20%	25%
Glazed surfaces (vertical)	8%	16%
	27.5% of vertical surfaces with window	~30% of vertical surfaces with window

More information about buildings is given in Annex1 of the ANNEXES to this report[3].
This typology is assumed to be similar to all European countries.

2 Building envelope

The main characteristics of the building envelope are its thermal insulation (thermal transfer coefficient), its thermal inertia, its solar heat transmission and its orientation.

2.1 Thermal insulation

The data on the European building stock are rare. Some data provided by [EPA 2006] are reported in the table below. However, the comparison between these data is not easy since the sources are not identical from one country to another.

[3] Annexes of this document are reported in a separate document called "Analysis of building heating and cooling demands in the purpose of assessing the reversibility and heat recovery potentials- ANNEXES" which can be found on http://www.ecbcs-48.org

Table 3: Thermal insulation of non residential building in Europe

Building type		Office buildings		Health care buildings	
		Existing buildings	New buildings	Existing buildings	New buildings
U_{wall} (in W/m²/K)	France[4]	1.15 to 1.55	0.4	1.05 to 1.35	0.4
	Belgium[5]		0.6		0.6
	Italy[6]	1.19	0.55 – 0.8	1.19	0.55 – 0.8
	Spain[7]	0.65			
	Germany[8]	1.3			
	Austria[9]	0.6	0.15-0.25	0.6	0.15-0.25
	The Netherlands[10]	2.4	0.4	2.4	0.4
U_{window} (in W/m²/K)	France	3.1 to 4.7	2.4	1.6 to 3	2.4
	Belgium	3.5		3.5	
	Spain	2.75-3.25		2.75-3.25	
	Austria	2-2.8	0.7-1.4	1.5-2.5	0.7-1.2
U_{roof} (in W/m²/K)	France	0.6 to 1	0.23	0.5 to 0.8	0.23
	Belgium		0.4		0.4
	Italy	1.4	0.55 – 0.8	1.4	0.55 – 0.8
	Spain	0.65			
	Germany	0.8			
	Austria	0.5	0.15-0.3	0.5	0.15-0.3
	The Netherlands	2.4	0.4		
Thermal bridge (in W/m/K)	France[2]		0.5-0.9		0.5-0.9

When separate data between office buildings and health care institutions are given, the average thermal insulation of health care institutions is better than those of office buildings.
Moreover, a study by EURIMA [LIM 2001] shows that insulation levels in countries are predominately climate driven. Insulation levels in Sweden and Finland for instance are higher than Spain, Italy and Greece. In order to take into account this difference, two levels of insulation will be considered according to the climatic zones[11].
In order to simplify, the thermal bridges are not considered; they are included in an overall thermal transfer coefficient of walls.
The chosen average U values are better than the average values in the building stock in order to include the building retrofit and the new buildings. The values are compiled in the following Table.

Table 4: Average U values of building envelope

Building type	Office buildings		Health care buildings	
Climatic zones[12]	Zone 1-4	Zone 5	Zone 1-4	Zone 5
U_{wall} (in W/m²/K)	0.8	0.6	0.6	0.4
U_{window} (in W/m²/K)	3	2	3	2
U_{roof} (in W/m²/K)	0.4	0.3	0.3	0.2

No sensitivity study on this parameter is done. This parameter has a large influence on heating energy demand and cooling energy demand but it is assumed to have a lower influence on simultaneity of heating and cooling demand.

4 France : Existing buildings refer to thermal regulation of 1976 and new buildings refer to thermal regulation of 2000 (reference U values)
5 Belgium: Values for all non residential buildings from Walloon Region, Law 1986. Similar values for Flanders Region and Brussels capital Region.
6 Italy: Values for all non residential buildings from 1961 to 1996
7 Spain: Values for all non residential buildings
8 Germany : Values for all non residential buildings (80% - 20% better)
9 Austria : Existing buildings refer to values from 1981 to 1995; New buildings refer to data after 1995
10 The Netherlands: Values for all non residential buildings . Existing buildings built between 1950 and 1975; New buildings built after 1992
11 see 4
12 see 4

2.2 *Thermal inertia*

The thermal inertia is assumed to have an impact on the simultaneity of heating and cooling demand. Based on average values on the building stock, the inertia of each building case is proposed in Table 5.

Table 5: Average thermal inertias for each building case

Building type	Office buildings					Health care buildings	
	1a	**1b**	**1c**	**2**	**3**	**1**	**2**
Thermal inertia[13]	low	low	low	medium	medium	high	medium

No sensitivity study on this parameter is done since this parameter has not a large influence on the energy demand [FIL, 2006a].

2.3 *Solar transmission*

The solar factor of windows and the solar protections have a large influence on the building energy demand.

Table 6: Average values of window solar Factor with or without solar protection in office buildings

	Office buildings		
	SF	SF with protection	Solar protection use
Offices	0.6	0.2	Figure 1
Conference room			Figure 1
Circulations			No solar protection
toilets			No solar protection

Table 7: Average values of window solar Factor with or without solar protection in health care buildings

	Health care buildings		
	Solar factor	Solar factor with protection	Solar protection use
Offices, consultation rooms, emergency rooms (type 1)	0.6	0.2	Figure 1
Labo, restaurants, technical annexes (type 1)			Figure 1
Operation rooms (type 1)			No solar protection
Common room (type 2)			As in bedrooms
Restaurant (type 2)			As in bedrooms
Technical premises (type 2)			No solar protection
Nursing room (type 2)			As in bedrooms
Bedrooms (types 1&2)			Fully used in summer, partially used in mid season
Circulations (types 1&2)			No solar protection
Toilets (types 1&2)			No solar protection

Based on a study of the solar protections in offices [ALE 2006], the rate of use versus outside lighting is given on the figure below.

[13] according French thermal regulation RT 2005

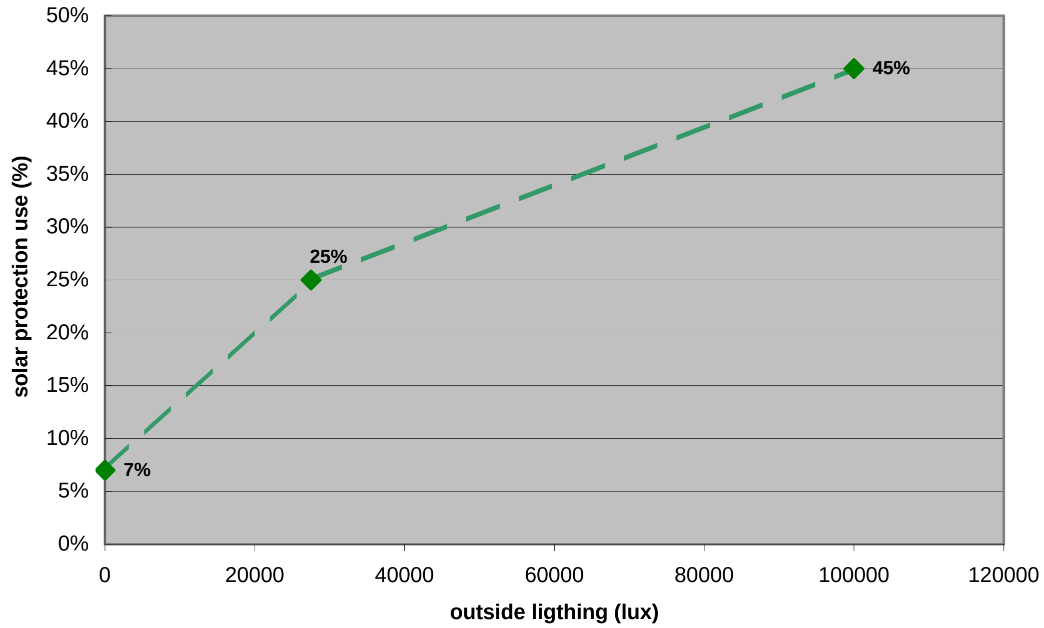


Figure 1: Average use of solar protections in offices

In order to take into account the solar heat gains, two cases are studied based on solar factor in case of solar protection : external blind SF =0.2 and internal blind SF = 0.4.

2.4 Orientation

The main orientation of the building has generally an influence on the simultaneous demand of heating and cooling. The N/S orientation tends to increase the simultaneous demand of heating in the North and cooling in the South. Moreover, no privileged direction of building can be easily defined in the building stock.

Two orientation cases are studied: East/West or North/South.

3 Building loads

The main characteristics of the building loads are the occupancy (sensible and latent loads), the artificial lighting and the electrical appliances. The set point temperatures and humidity should also be considered. The heterogeneity of building loads has a large influence on the simultaneity of heating and cooling demands.
The average sizing parameters are defined in the following Table based on [AIC 1993a] and [AIC 1993b].

Table 8: Average sizing parameters for office buildings and health care institutions

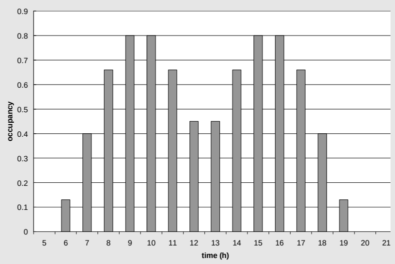
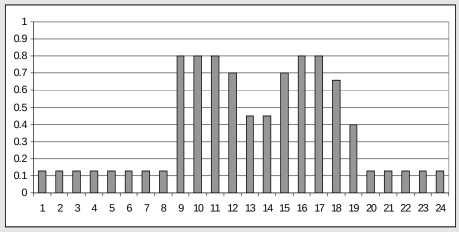
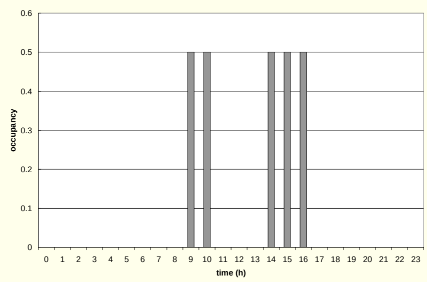
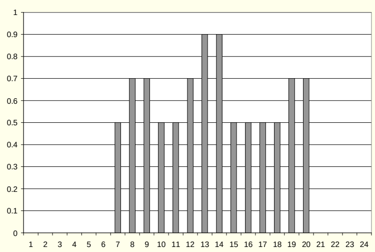
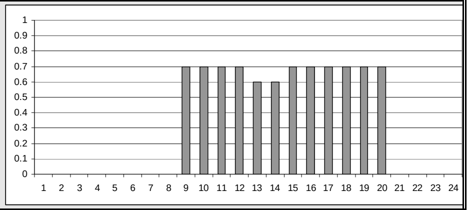
Office buildings	Person rate (m²/pers)	Lighting power (W/m²)	Appliances (W/m²)	Health care institutions	Person rate (m²/pers)	Lighting power (W/m²)	Appliances (W/m²)
Offices	12	18	15	Offices, consultation rooms, emergency rooms (type 1)	26	10	7.5
Conference room	3.5	18	0	Labo, restaurants, technical annexes (type 1)	20	10	20
Circulations	0	12	0	Operation rooms (type 1)	10	50	30
toilets	0	6	0	Common room (type 2)	6	12	2
				Restaurant (type 2)	5	12	2
				Technical premises (type 2)	0	0	0
				Nursing room (type 2)	30	10	20
				Bedrooms (types 1&2)	8(1) 20(2)	10(1) 10(2)	7 (1) 2.8(2)
				Circulations (types 1&2)	0	12(1&2)	0
				Toilets (types 1&2)	0	6(1&2)	0

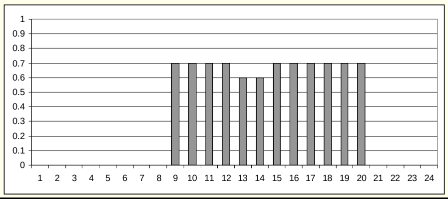
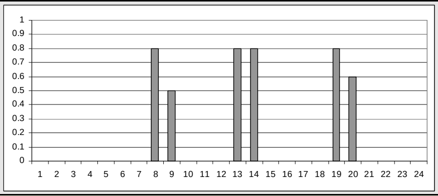
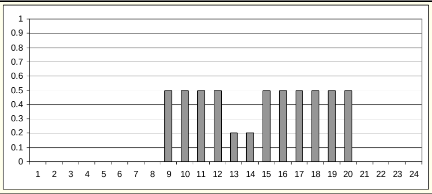
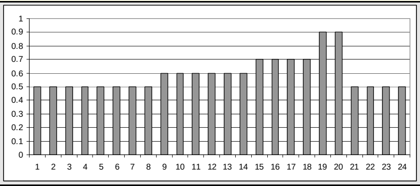
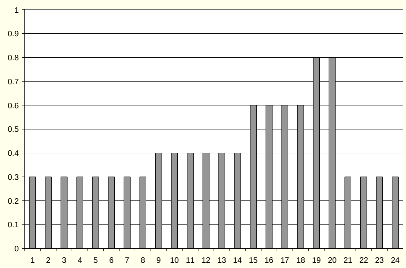
In office buildings, since there is a large disparity in internal building loads, a sensitivity study is carried out. A "low" load building is taken into account, with an appliances power of 7.5 W/m² in offices instead of 15W/m² and a lighting power of 10 W/m² in conference rooms and offices instead of 18 W/m².

3.1 *Occupancy*

The occupancy profiles are defined for each room type as follows :

Table 9: Occupation profiles – ratio of occupancy hour by hour compared to sizing value

Office buildings		Health care institutions	
Occupancy from Monday to Friday		Occupancy all the week	
Offices		Offices, consultation rooms, emergency rooms (type 1)	
Conference room		Labo, restaurants, technical annexes (type 1)	
		Operation rooms (type 1)	

	Common room (type 2)	
	Restaurant (type 2)	
	Nursing room (type 2)	
	Bedrooms (type 2)	
	Bedrooms (type 1)	

No sensitivity study on this parameter is done. The occupancy profiles and the loads due to persons have a large influence on heating and cooling energy demands but low differences between buildings are assumed.

3.2 *Artificial lighting*

Artificial lighting is an important source of internal load which differ with the use of the premises. The use profiles are described below:

Table 10: Artificial lighting use

Office buildings	Use of artificial lighting	Lighting profile	Health care institutions	Use of artificial lighting	Lighting profile
Offices	Figure 2	Office	Offices, consultation rooms, emergency rooms (type 1)	Figure 3	Offices of health care institutions
Conference room	Figure 2	Conference	Labo, restaurants, technical annexes (type 1)	100%	from 9h to 20h
Circulations	Figure 2	Office	Operation rooms (type 1)	100%	from 9h to 20h
toilets	Figure 2	office	Common room (type 2)	Same ratio as occupation profile	
			Restaurant (type 2)	Same ratio as occupation profile	
			Technical premises (type 2)	0%	-
			Nursing room (type 2)	100%	from 9h to 20h
			Bedrooms (types 1&2)	Figure 4	
			Circulations (type 2)		

	Circulation (type 1)	Figure 5	
	Toilets (type 1)	10%	All the day
	Toilets (type 2)		

A study [ALE 2006] has shown that the use of artificial lighting in office buildings can be described as:

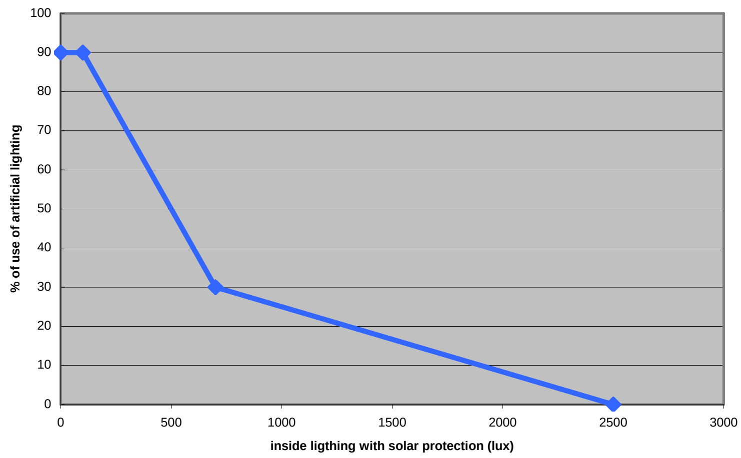


Figure 2 : Average use of artificial lighting in office buildings (without presence detector)

In office buildings, the artificial lighting is supposed switch off during non occupancy.

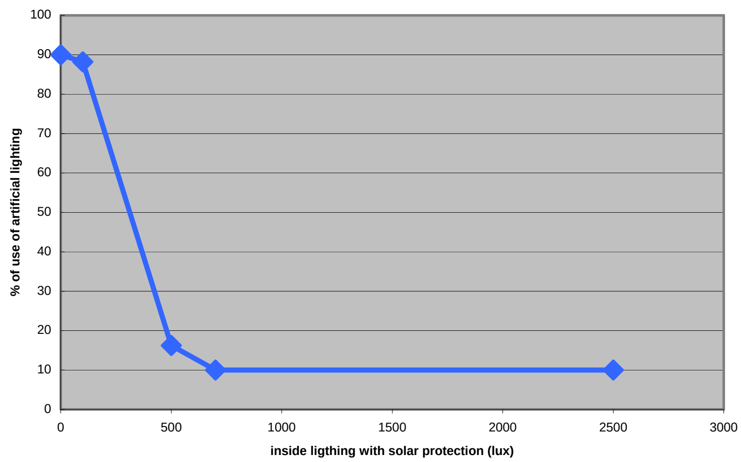


Figure 3 : Average use of artificial lighting in office buildings (with a dimmer and without presence detector)

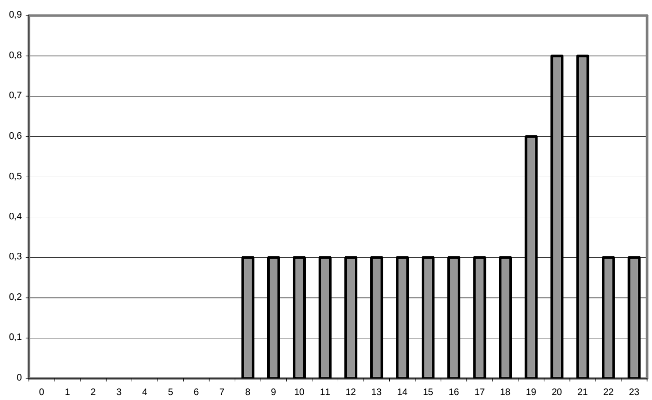


Figure 4: Ratio of use of artificial lighting in bedrooms (Health care Type 1 & 2) and circulations (Health care Type 2)

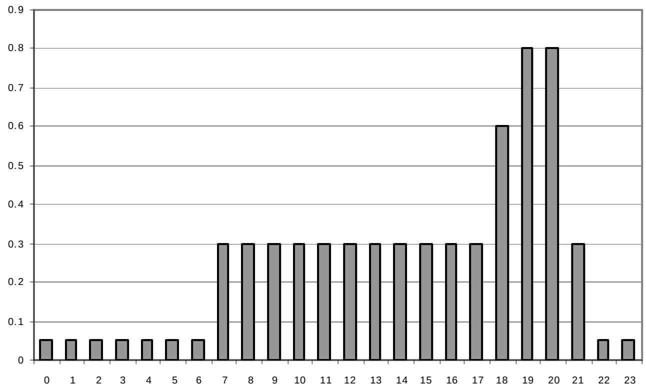


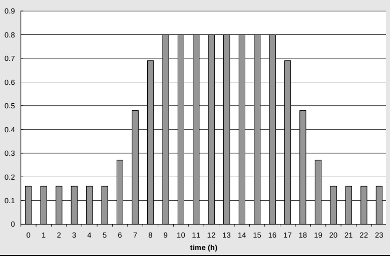
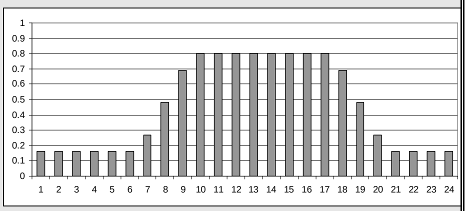
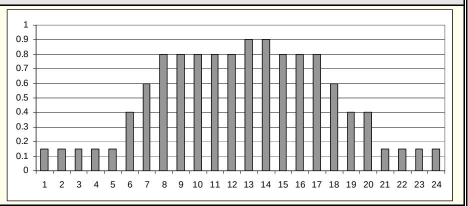
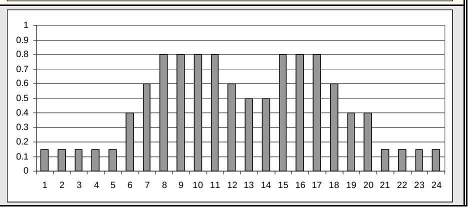
Figure 5: Ratio of use of artificial lighting in circulations of large hospitals

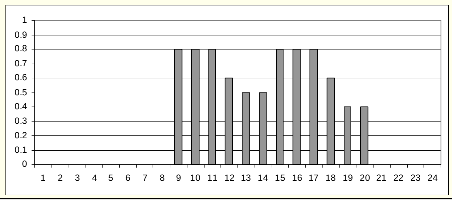
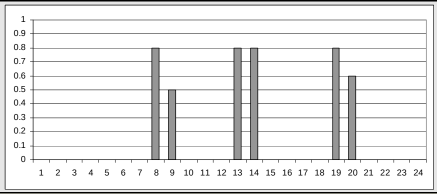
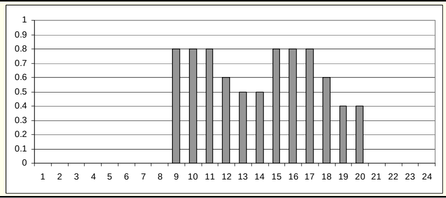
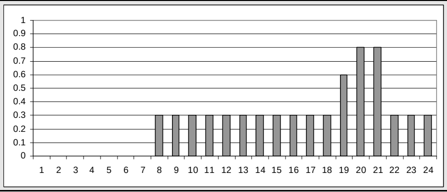
No sensitivity study on this parameter is done.

3.3 *Appliance loads*

The appliance use profiles are defined for each room type as follows :

Table 11: Appliance profiles – ratio of use hour by hour compared to sizing value

<table>
<tr><th colspan="2">Office buildings</th><th colspan="2">Health care institutions</th></tr>
<tr><td colspan="2">from Monday to Friday,
Week End as night during week</td><td colspan="2">Occupancy all the week</td></tr>
<tr><td>Offices</td><td></td><td>Offices, consultation rooms, emergency rooms (type 1)</td><td></td></tr>
<tr><td colspan="2" rowspan="2"></td><td>Labo, restaurants, technical annexes (type 1)</td><td></td></tr>
<tr><td>Operation rooms (type 1)</td><td></td></tr>
</table>

	Common room (type 2)	
	Restaurant (type 2)	
	Nursing room (type 2)	
	Bedrooms (types 1&2)	

No sensitivity study on this parameter is done. The appliance use profiles and the installed power have a large influence on heating and cooling demands but low differences between buildings are assumed.

3.4 Ventilation and infiltration

The Table 12 shows a comparison of ventilation rate in different European countries [LIM 2001]. Large differences are observed between countries. Another study [TIP 2001] shows similar results (Figure 6).

Table 12: Minimum ventilation rate criteria for offices in different European countries [LIM 2001]

Country	Whole building ventilation rate criteria
Belgium (Walloon region)	Single Office : 2.9 $m^3/h/m^2$ (so with 12 m^2/pers, 35 m^3/h/pers) Landscape Office : 2.5 $m^3/h/m^2$ (so with 12 m^2/pers, 30 m^3/h/pers) Meeting room : 8.6 $m^3/h/m^2$ (so with 3.5 m^2/pers, 30 m^3/h/pers)
France	Offices : 25 m^3/h/pers Meeting room: 18 m^3/h/pers or 30 m^3/h/pers (smoking)
Italy	Offices : 40 m^3/h/pers Meeting room: 35 m^3/h/pers
Finland	15 -36 m^3/h/pers
Germany	Offices : 20 - 30 m^3/h/pers
Switzerland	Offices : 25 - 30 m^3/h/pers
United Kingdom	Offices : 29 m^3/h/pers (recommended)
Greece	Offices : 25.5 – 42.5 m^3/h/pers Meeting room: 51 - 68 m^3/h/pers

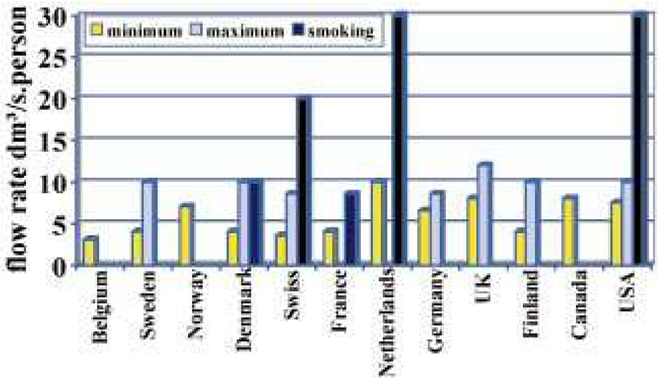


Figure 6: Required flow rates per person for offices in standards [TIP 2001]

Concerning health care institutions, some reference ventilation rates are taken from [AIC 1993b].

Table 13: Ventilation rates according to space type

	low	high
Offices	25 $m^3/h/pers$	40 $m^3/h/pers$
Conference room	30 $m^3/h/pers$	40 $m^3/h/pers$
Offices, consultation rooms, emergency rooms (type 1)	25 $m^3/h/pers$	40 $m^3/h/pers$
Labo, restaurants, technical annexes (type 1)	6 vol/h	
Operation rooms (type 1)	15 vol/h	
Common room (type 2)	18 $m^3/h/pers$	40 $m^3/h/pers$
Restaurant (type 2)	18 $m^3/h/pers$	40 $m^3/h/pers$
Nursing room (type 2)	4 vol/h	
Bedrooms (types 1&2)	25 $m^3/h/pers$	40 $m^3/h/pers$

To enable a comparison between whole buildings in air changes per hour (ach) the given requirements for each country should be expressed at the same reference pressure. [LIM, 2001] has taken 50Pa in order to compare the airtightness requirements in each country. In order to compare, a typical building volume of 300 m^2 and a surface area of 250m^2 air leakage rates has been assumed.

Table 14: Whole building airtightness requirements in different European countries [LIM, 2001]

Country	Whole building airtightness	Normalised air change rate to 50Pa
Belgium	Only recommendations: <3 ach at 50Pa for dwellings with balanced mechanical ventilation <1 ach when heat recovery devices are used	1.0 to 3 ach
France	According to RT2005, the reference values vary from 0.8 to 2.5 $m^3/(h.m^2)$ at 4 Pa depending on the type of construction 1.2 for offices buildings and health care institutions	5.3 ach
Italy	According to 18.12.75, Recommendation for an envelope air leakage value for schools : 10 $m^3/(h.m^2)$ at 98 Pa It also gives prescribed air change rates between 1.5 to 5.0 ach for rooms in office buildings	1.0 to 3.2 ach (for schools)
Netherlands	According to NEN 2687, Class 1 : min 0.4 – 0.72 ach at 10 Pa Class &: max 1.4 2.24 ach at 10 Pa Class 2 : max 0.72 – 1.15 ach at 10 Pa	3.3 ach to 6.5 ach
Norway	Buildings other than houses with two storeys or less : 3 ach at 50Pa Buildings other than houses with more than two storeys: 1.5 ach at 50Pa	1.5 to 3.0 ach
Switzerland	New buildings : upper limit 0.75 $m^3/(h.m^2)$ at 4 Pa Refurbished buildings : upper limit 1.5 $m^3/(h.m^2)$ at 4 Pa	3.3 ach to 6.6 ach
United Kingdom	According to CIBSE TM23 2000, air leakage index in offices with mechanical ventilation of 5.0 $m^3/(h.m^2)$ at 50 Pa in good practice and 2.5 $m^3/(h.m^2)$ at 50 Pa in best practice	2.0 to 4.1 ach

In Germany, the air tightness requirements are according to DIN 13829-2001: 3 ach/h in naturally ventilated buildings and 1.5 ach/h in buildings with mechanical ventilation.

Since airtightness differences between countries are quite low, no sensitivity study on air infiltration will be achieved. The air permeability of health care buildings under 4Pa will be fixed to 1.2 $m^3/h/m^2$. In office buildings, the air permeability will be defined as 1.7× (heat loss surface)/(surface area).

Large differences on ventilation rate are found in Europe. In order to take into account these differences, two cases of ventilation rate are studied (Table 13).

3.5 Set points

	Cooling set point	Heating set point	Night setback
Offices	24	21	No air conditioning during inoccupancy. 15°C set point during heating period
Conference room			
Offices, consultation rooms, emergency rooms (type 1)			
Labo, restaurants, technical annexes (type 1)	26	20	
Operation rooms (type 1)	26	24	
Common room (type 2)	26	21	
Restaurant (type 2)			
Nursing room (type 2)			
Bedrooms (types 1&2)			

Humidity control is only provided in operation rooms. The set point is fixed to 55%±10% relative humidity.

Two set point temperatures in office buildings are studied in order to account for differences between buildings. The values of 21°C/24°C and 20°C/25°C are assumed to be representative of all countries of Europe-15[14].

3.6 Domestic Hot Water

In the following, Domestic Hot Water in office buildings is not considered since its consumption represents less than 3% of energy consumption in office buildings in France in 2001. Moreover, the DHW is rarely centralized and heat recovery on local tanks is not easy.

On the other hand, the Domestic Hot Water represents more than 14% of energy consumption in Hospitals in France in 2001. It is taken into account for the large hospital and the rest home. Based on [AIC 1991], the hot water consumption is simulated by using the following equation:

$$V_h = \frac{V_{pj}}{24}, \quad (V_h)_i = \alpha_i . V_h$$

Vpj : Daily volume consumption (in L)
Vh : Average hourly volume consumption (in L)
Vhi : Hourly volume consumption (in L)
with α_i a dimensionless coefficient defined hour by hour (Figure 7 and Figure 8).

[14] French decrees set a heating temperature at 19°C and a cooling temperature not below 26°C. In fact, these temperatures are not used in buildings.

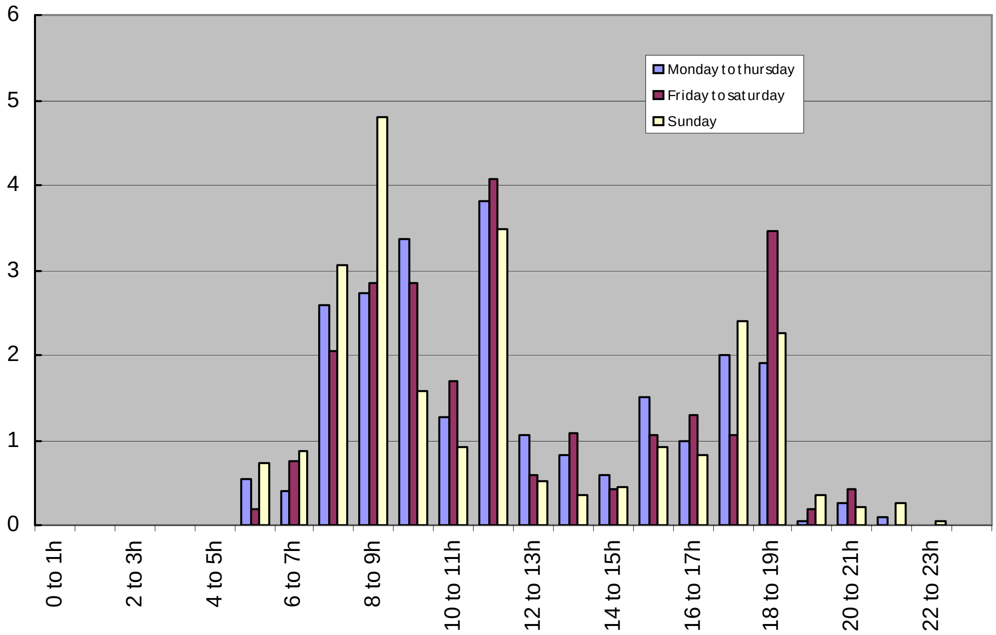


Figure 7 : Hourly share of hot water consumption in large hospitals

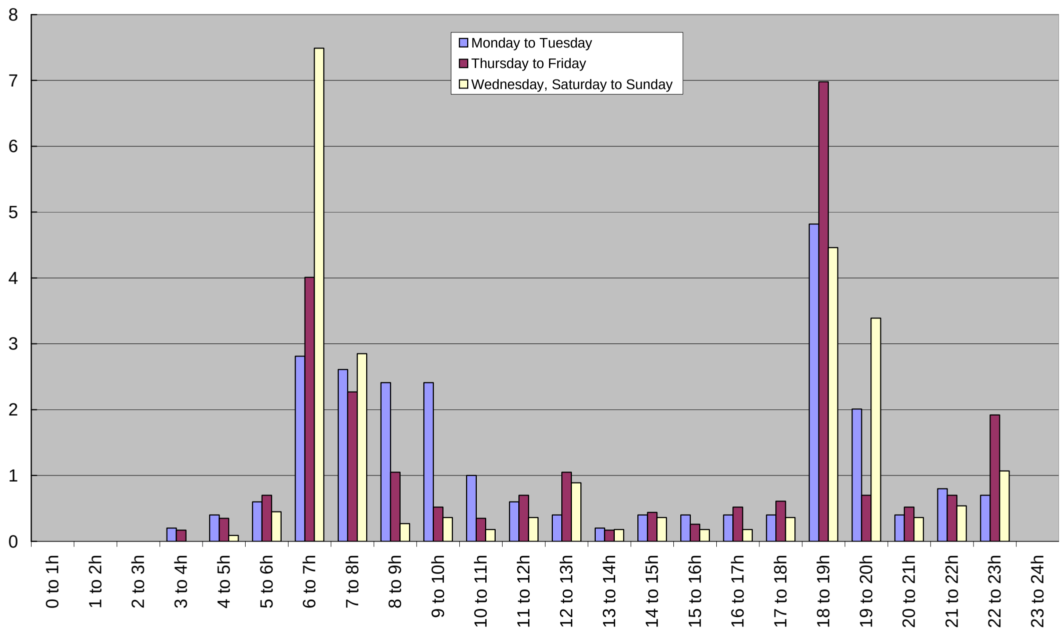


Figure 8 : Hourly share of hot water consumption in rest homes

Based on [AIC 1991], the daily volume consumption is defined in the following Table :

Table 15: Hot Water consumption on health care institutions

	Large hospital		Rest home	
	Ratio	L/day	Ratio	L/day
bedroom	60 L/bed/day	18 000	40 L/bed/day	3 600
Restaurant	10 L/meal	9 000	5 L/meal	1 350

3.7 *Heterogeneity of loads in buildings*

Large heterogeneity of loads in Hospitals is considered through the high loads in operation rooms and the other rooms. The office buildings have been defined without taking into account the heterogeneity of loads between rooms. Indeed, the offices are often not uniformly occupied what can have an impact in the case of thin partitioned offices. Moreover, it is common to find sever rooms or computer rooms in office buildings where the internal loads are very high and cooling is required all the time. Server rooms can be studied independently as a separate homogenous thermal zone. Their cooling demand can be assessed easily since server rooms are generally not occupied and often without any window and their internal loads are usually constant all along the day.

3.8 *Summary of sensitivity study on building types*

Building parameters	
Building geometry	7 types: 5 office buildings, 2 health care institutions
Thermal insulation	-
Thermal inertia	-
Solar gains	Two solar factor levels with solar protection : 0.2 and 0.4
Orientation	Two cases : N/S or E/W in office buildings only
Person loads	-
Ventilation	Two cases : low ventilation rate, high ventilation rate
Set points	Two cases: 21-24°C and 20-25°C in office buildings
Internal loads	Two cases: low appliances and lighting loads, high appliances and lighting loads in office buildings No change on the load profiles

4 Climates

The European climates are split in few climates based on the study of heating and cooling degree days.
The heating degree days are calculated as follows :

$$HDD = \sum_{j=1}^{8760}\left(15 - \frac{T_{max}^{j} + T_{min}^{j}}{2}\right)^{+}$$

The basis of heating degree days is taken to 15°C. This temperature is considered as the outside temperature for which no heating is necessary on the basis of a 21°C comfort temperature and 6°K of internal gains.

Considering the cooling degree days, one must consider not only temperature but also the dehumidification loads. The cooling degree days are so based on "sensible degree days" and "latent degree days" weighted on an enthalpy basis. The reference temperature is taken to 15°C and the humidity ratio is taken at 15°C and 100% relative humidity. The dehumidification is not often controlled and it is just a consequence of the cold distribution temperature in the building. The chosen value is so quite arbitrary and corresponds to typical values at the exit of a cooling coil in an Air handling Unit.

$$CDD = \sum_{j=1}^{8760}\left(\frac{T_{max}^{j} + T_{min}^{j}}{2} - 15\right)^{+} + \frac{hfg}{cp_{air}}\left(\frac{w_{max}^{j} + w_{min}^{j}}{2} - w(100\%, 15°C)\right)^{+}$$

where hfg is the latent heat of vaporization at 0°C (2501 kJ/kg) and cp_{air} is the specific heat of air (1008 J/kg/K), w the humidity ratio in (kg/kg) and T^{j}_{max} and T^{j}_{min} the maximum and minimum temperatures of the day j, respectively.

Based on similarity of heating and cooling degree days, climatic zones are defined (Figure 9) :

- Zone 1 : Low heating demand, high cooling demand
  - Corresponds to south of Spain, south of Italy, French Mediterranean cost, Greece
- Zone 2 : Low heating demand, medium cooling demand
  - Corresponds to Portugal, North West of Spain
- Zone 3: Medium heating demand, medium cooling demand
  - Corresponds to North of Italy, South of France
- Zone 4: Medium heating demand, low cooling demand
  - Corresponds to North of France, Belgium, The Netherlands, South of UK, West of Germany
- Zone 5: High heating demand, low cooling demand
  - Corresponds to the east of EU-15 and North East of EU-15

Five climates will be simulated in order to have an overview of the different European climates. Furthermore, one could consider some other parameters such as solar radiation, daily temperature difference and ratio between sensible and latent cooling demand. In order to limit the simulations to few climatic zones, these parameters are not considered.

The meteorological data from Athens, Lisboa, Torino, Paris and Munchen are taken into account for simulation of each zone.

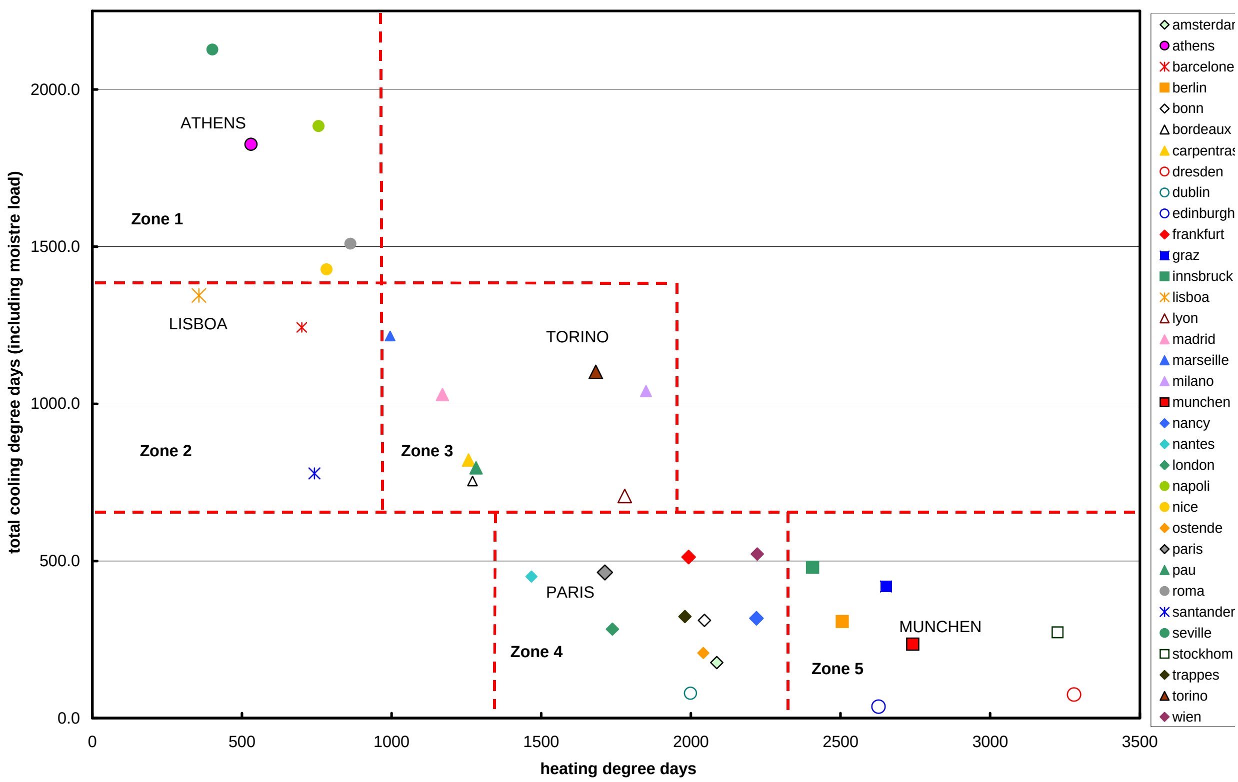


Figure 9: Climatic zones based on cooling and heating degree days

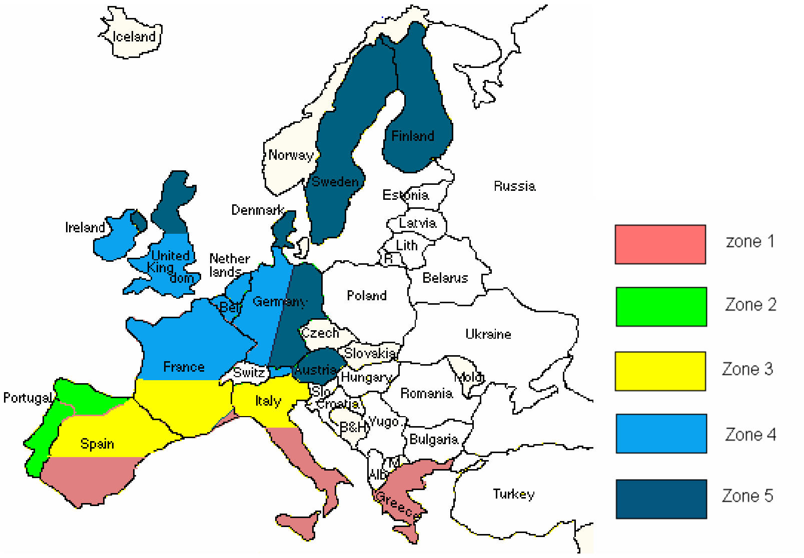


Figure 10: European climatic zones

III ANALYSIS OF HEATING AND COOLING DEMANDS

The simulations are carried out for the seven building types described hereinbefore, the five climatic zones, two solar factor levels of windows and two levels of ventilation rate and only in office buildings, two orientations, two levels of internal loads and two set point levels. Thus, 840 heating and cooling demand profiles are set including 40 cases for the health care institutions.

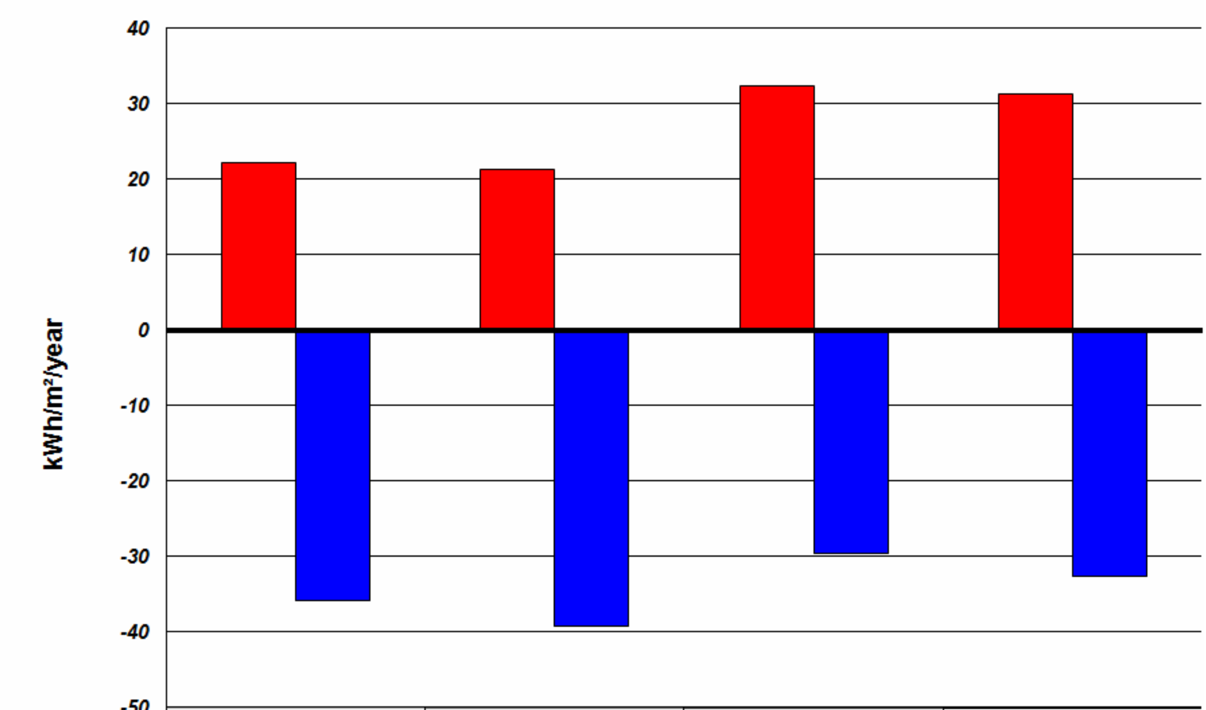
1 Office Buildings results

1.1 Heating and cooling demands

The results for each simulation case are presented in a Table in Annex 2 of the ANNEXES to this report [STA, 2008]. One can notice that :

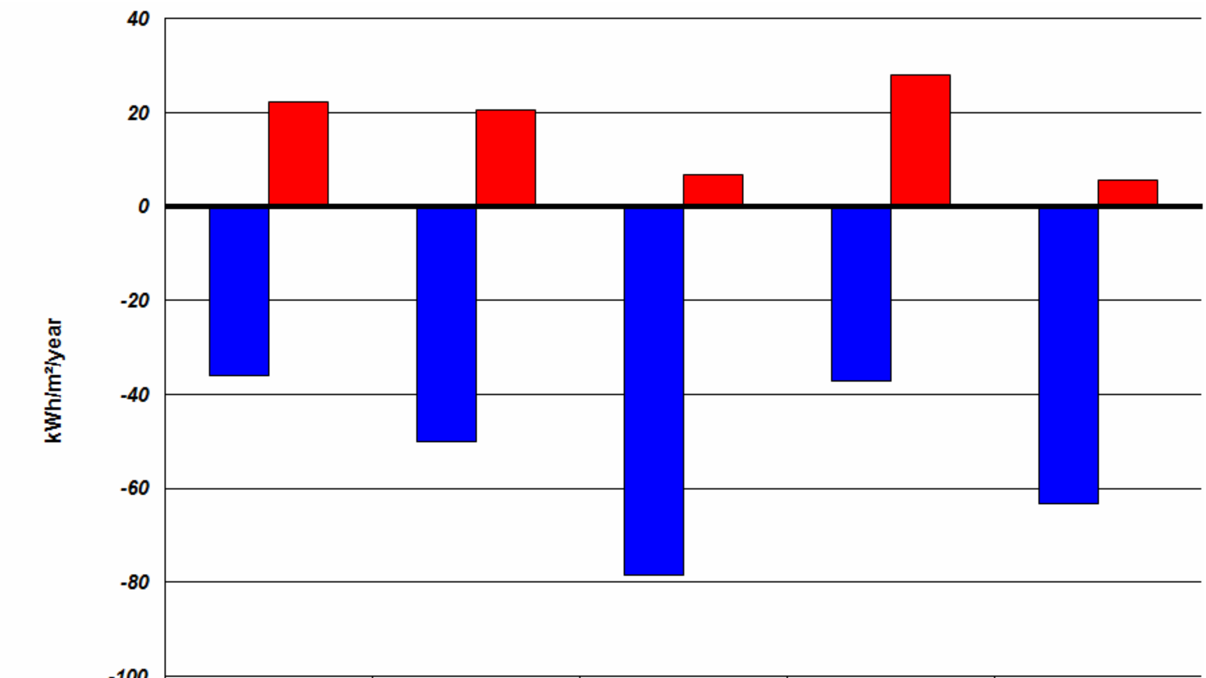
- A lower solar factor (from 0.4 to 0.2 when solar protections are used) reduces the cooling demand by about 9% (Figure 11) but increases the heating demand by about 4% in the case of building type 1a in Paris. The impact of this parameter depends also on the window surface (building type) and climate ;
- A higher ventilation rate decreases the cooling demand by about 17% (Figure 11) but increases the heating demand by about 46% in the case of building type 1a in Paris. The impact of this parameter depends strongly on the climate ;
- The climate has a large influence on heating and cooling demands (Figure 12). The heating demand for the building of type 1a is five times higher in Munich than in Lisbon. The cooling demand is two times higher in Athens than in Paris.
- The building geometry (window surface related to floor area and compactness) has a large impact on energy demand (Figure 13). The building type 1a requires almost three times less heating than type 3. On the contrary, it requires once and half more cooling than type 3.
- The orientation does not have a large influence (Figure 14) on the energy demand but it is supposed to have an impact on the concomitant cooling and heating demand. This point will be discussed further.
- Figure 14 shows that the heating and cooling demands decrease of 15% in the building type 1a when the heating/cooling set points pass from 21/24°C to 20/25°C. This impact depends strongly on climate but also in less importance on building geometry.
- The reduction of internal loads (from 15 W/m² to 7.5 W/m² for appliances and from 18 to 10 W/m² for lighting) implies an increase of heating demand of about 53% and a decrease of about 46% of the cooling demand on building type 1a in Paris with 21/24°C set point temperatures. According to the building type, the climate, the ventilation rate and the solar factor, the heating demand can increase from 16% to 57% and the cooling demand can decrease from 14% to 48%.

All the selected parameters have a large influence on the energy demand except for building orientation. Notice that the buildings are considered without any humidity control. Detailed results are compiled in Annex 1 of the ANNEXES to this report [STA, 2008].



	Type: 1a ; Orient.: NS ; Climate: PARIS ; Solar F: LOW ; Ventil.: LOW	Type: 1a ; Orient.: NS ; Climate: PARIS ; Solar F: HIGH ; Ventil.: LOW	Type: 1a ; Orient.: NS ; Climate: PARIS ; Solar F: LOW ; Ventil.: HIGH	Type: 1a ; Orient.: NS ; Climate: PARIS ; Solar F: HIGH ; Ventil.: HIGH
heating demand	22,2	21,3	32,4	31,3
cooling demand	-36,0	-39,2	-29,6	-32,6

Figure 11: Impact of ventilation rate and solar factor on the heating and cooling demands (example: building of type 1a in Paris, with a North/south orientation)



	Type: 1a ; Orient.: NS ; Climate: PARIS ; Solar F: LOW ; Ventil.: LOW	Type: 1a ; Orient.: NS ; Climate: TORINO ; Solar F: LOW ; Ventil.: LOW	Type: 1a ; Orient.: NS ; Climate: ATHENES ; Solar F: LOW ; Ventil.: LOW	Type: 1a ; Orient.: NS ; Climate: MUNICH ; Solar F: LOW ; Ventil.: LOW	Type: 1a ; Orient.: NS ; Climate: LISBOA ; Solar F: LOW ; Ventil.: LOW
cooling demand	-36,0	-50,2	-78,5	-37,1	-63,2
heating demand	22,2	20,5	6,7	27,9	5,6

Figure 12: Impact of climatic zone on the heating and cooling demands (example: building of type 1a, with a North/south orientation, low solar factor and low ventilation rate)

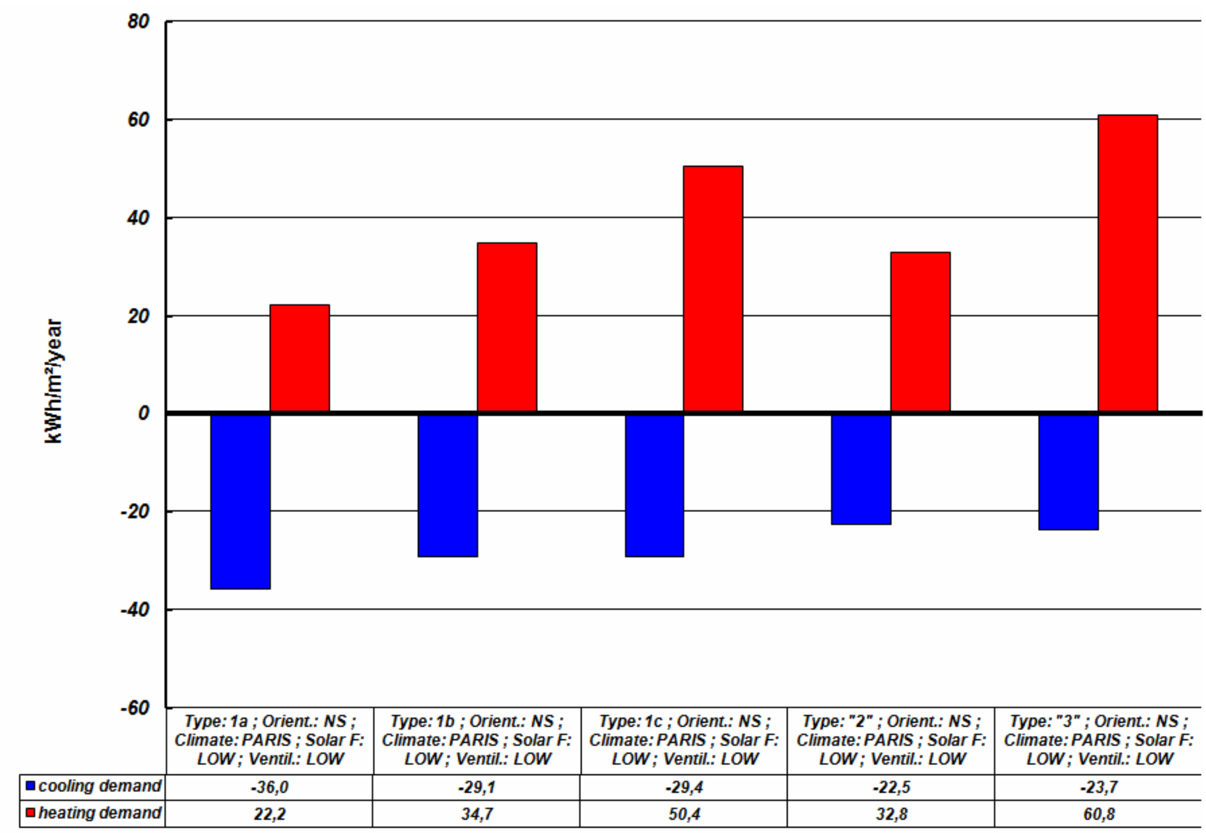


Figure 13: Impact of <u>building type</u> on the heating and cooling demands (example: in Paris, with a North/south orientation, low solar factor and low ventilation rate)

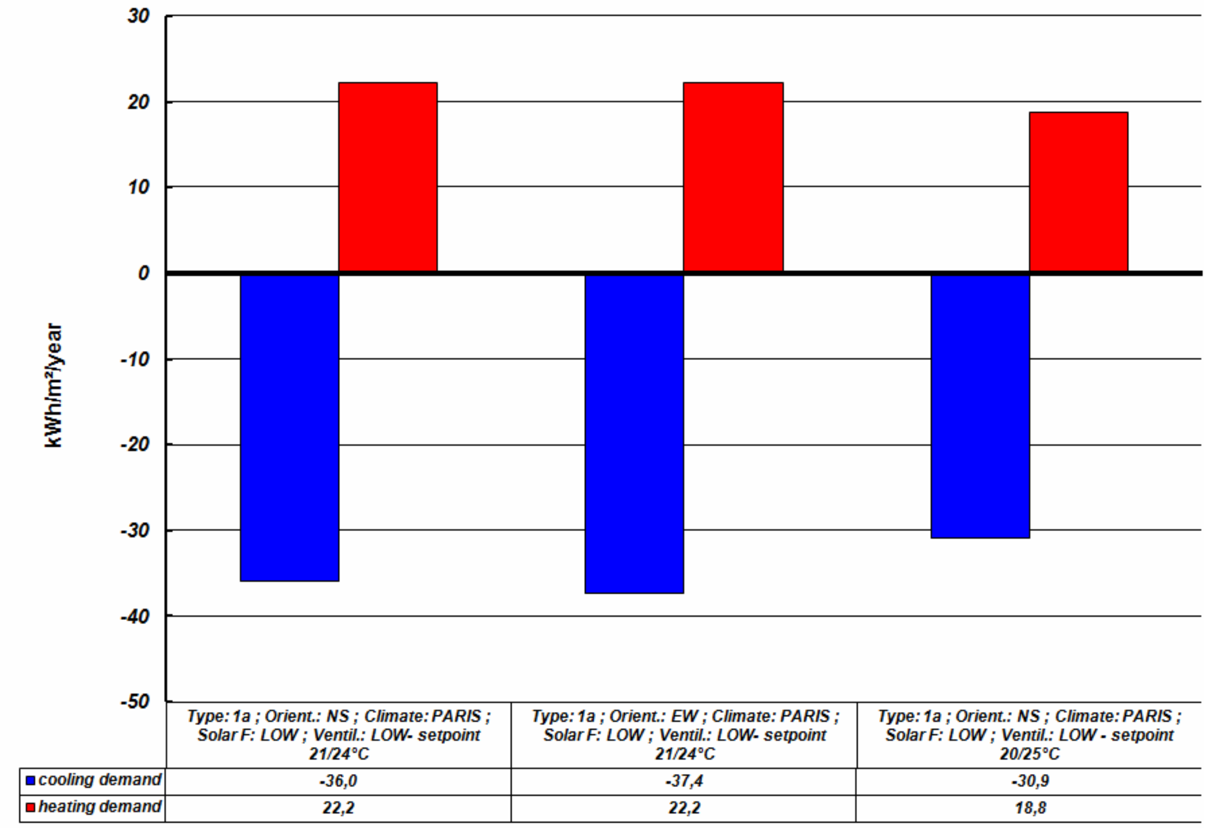


Figure 14: Impact of <u>orientation and set points</u> on the heating and cooling demands (example: building of type 1a, in Paris, with low solar factor and low ventilation rate)

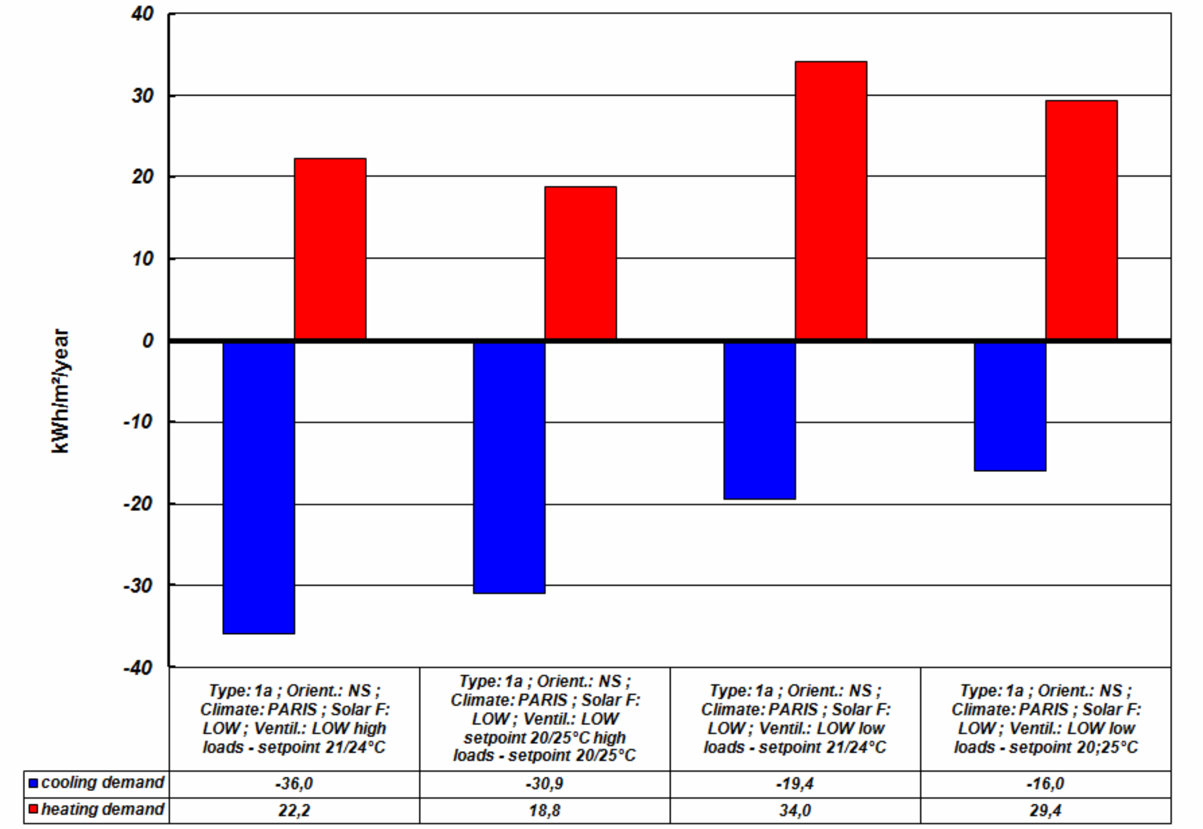


Figure 15: Impact of internal loads and set points on the heating and cooling demands (example: building of type 1a, in Paris, with low solar factor and low ventilation rate)

1.2 Reversibility potential

The reversibility potential depends on the heating power capacity which can be reached by adapting a chiller in a heat pump[15] and on the non simultaneous demand of cooling and heating.
The reversibility potential is calculated hour by hour (Figure 16) as the percentage of heating demand which could be provided by a chiller operating in heat pump under the following conditions :

- The "chiller" operates in priority in cooling mode; so reversibility is possible only when no cooling is required;
- The maximum heating power available is assessed to 0.8 × maximum cooling power of the chiller[16], the supplementary demand is assumed to be covered by the boiler.

No consideration on the emitter temperature levels required is done here. The % of reversibility is calculated as the ratio between the total heat provided by heat pumping and the total space heating demand.

[15] Assuming the heat pump is sized on the cooling capacity

[16] This value is based on data from manufacturer's catalogues. Even if some differences between chillers exist, one can notice that at rating conditions the cooling power is the range of magnitude of 10% lower than the heating power. However, the rating heating conditions are at 7°C outside air temperature which do not correspond to the worst conditions whereas the rating cooling conditions are imposed to 35°C that is close to the worst conditions. If we considered the heating power at –5°C, the heating power would be about 20% lower than cooling power at rating conditions.

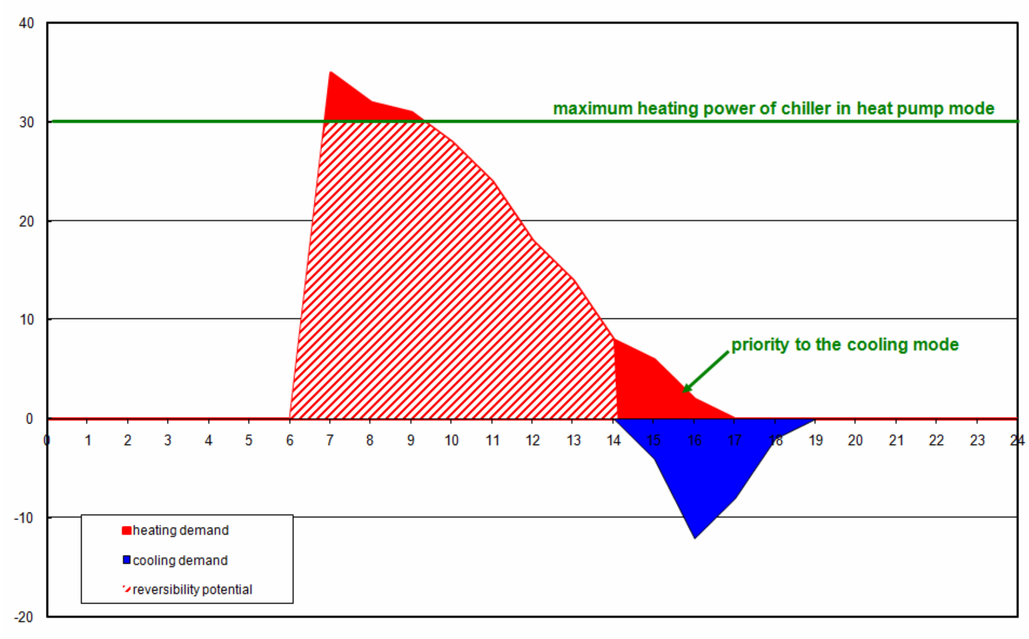


Figure 16: Estimate of the space heating potential of the chiller used in heat mode

Since one assumes the "reversible chiller" is designed based on peak cooling load, the reversibility potential depends on the ratio between peak cooling and peak heating loads. The peak cooling load is based on sensible cooling load and does not take into account the dehumidification load. The chiller will be sized to account for the dehumidification loads (except in case of chilled ceilings) according to the sizing rules. Thus, the installed cooling power will be higher than the peak cooling load. This fact is not taken into account in this chapter. On Figure 17, one can see for the different simulation cases that the peak cooling load is still higher than peak heating load in Athens climate. In Lisbon climate, the peak cooling load is almost always superior to the peak heating load. In Torino and in a lesser extent in Paris climate, the ratio between peak cooling load and peak is close to 1. In Munich, the ratio ranges between 0.33 and 1.36 according mainly on Building type.

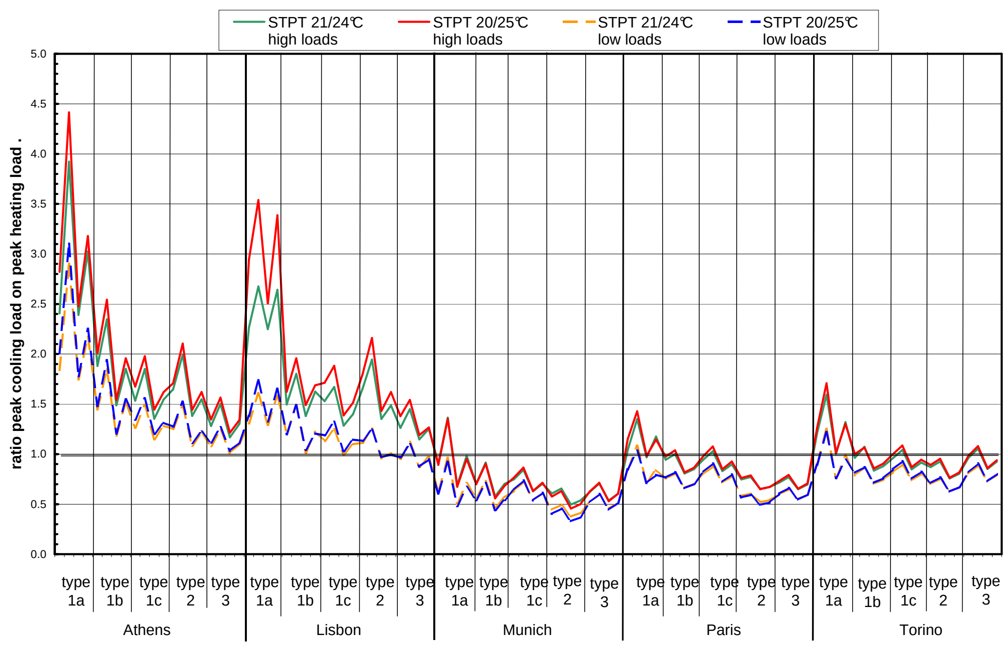


Figure 17: Ratio of the peak cooling load on the peak heating load for the different climates and building types in a North/south main orientation (other subdivisions correspond to: (low solar factor/low ventilation), (high solar factor, low ventilation), (low solar factor/high ventilation) and (high solar factor, high ventilation))

For all the cases, the reversibility potential ranges between 47% and 99%. The solar factor and the ventilation rate do not have a high impact on the reversibility potential (Figure 18). On the other hand, the climate has an impact on the reversibility potential which is larger in Paris, Munich and Turin than in Athens and Lisbon (Figure 19). These differences on the reversibility potential are mainly due to the simultaneous heating and cooling demand which is more important in Athens and Lisbon than in the other climate zones. In fact, in Mediterranean climates, the cooling demand is far from negligible in winter season in high load office buildings. The reversibility is not possible if cooling and heating are required simultaneously. Largest differences on the reversibility potential are found between the type 3 building and type 1 b building (Figure 20). If orientation does not have a high influence on the reversibility potential, the change of set point from 21/24°C to 20/25°C induces up to 21% increase of the reversibility potential among the simulation cases (Figure 21). The reversibility potential is also higher when the internal building loads are lower since there are fewer hours of simultaneous cooling and heating in Lisbon and Athens. In the other climates, lower internal loads induces from one hand lower simultaneous hours of cooling and heating, and from the other hand, the ratio between the maximum heating demand and the maximum cooling demand is increased so that the chiller in heating mode cannot provide anymore all the heating demand in some cases. Indeed, Figure 22 shows that the influence of internal loads on the reversibility potential is much lower in Paris than in Lisbon due to these opposite effects.

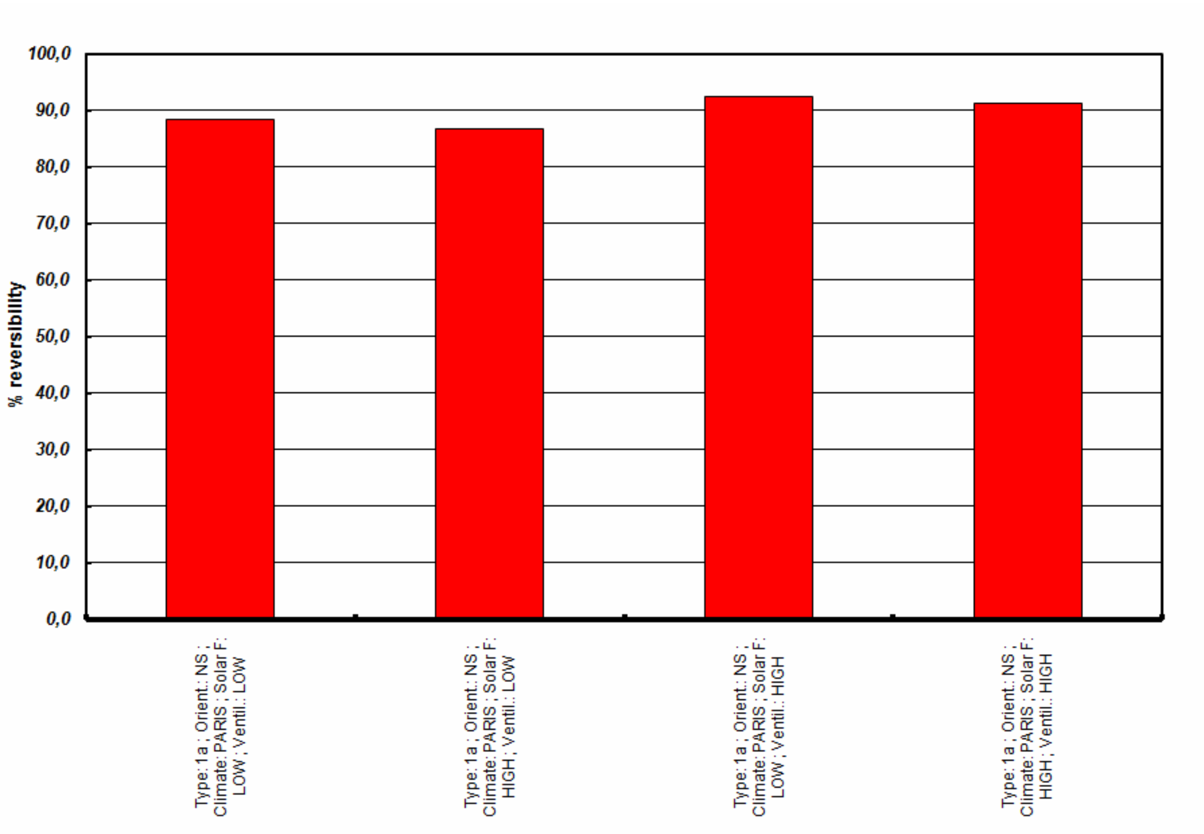


Figure 18: Reversibility potential in % for building type 1a in Paris for <u>two levels of ventilation rate and solar factor</u>

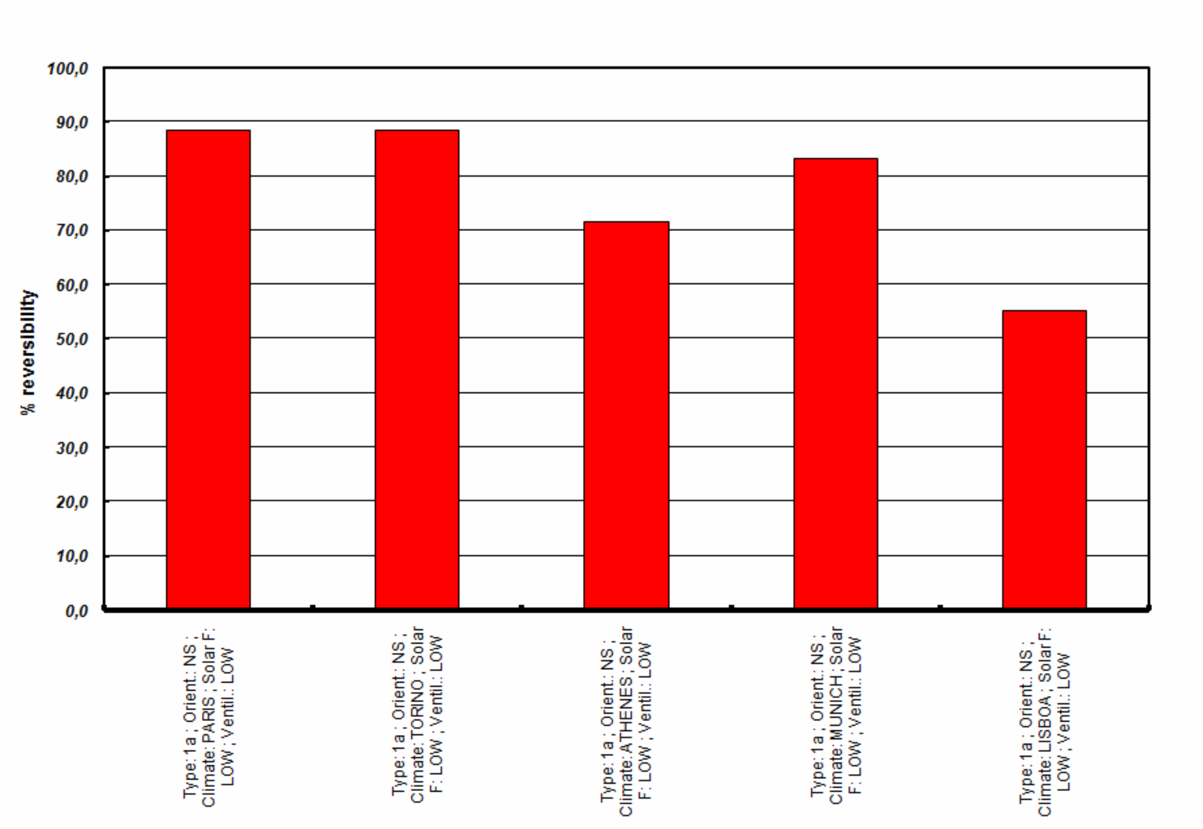


Figure 19: Reversibility potential in % for building type 1a in <u>different climates</u>

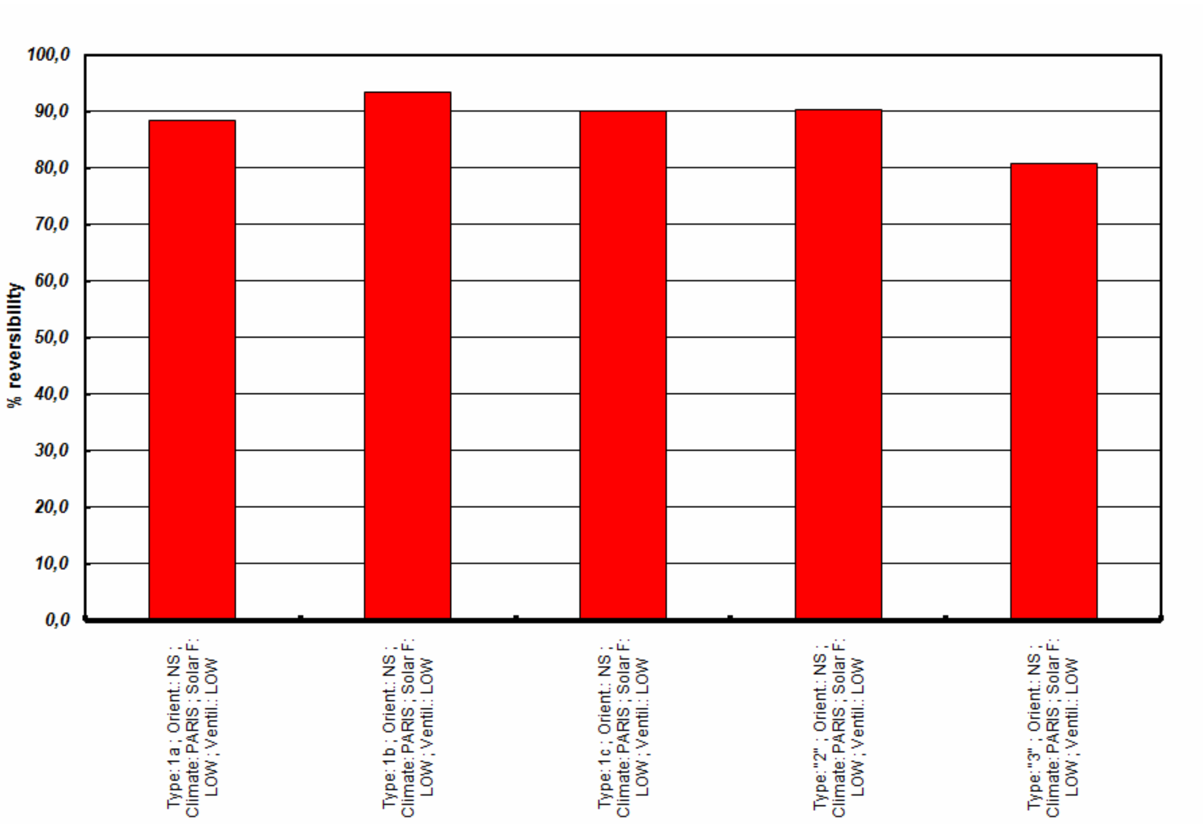


Figure 20: Reversibility potential in % for <u>different building types</u> in Paris

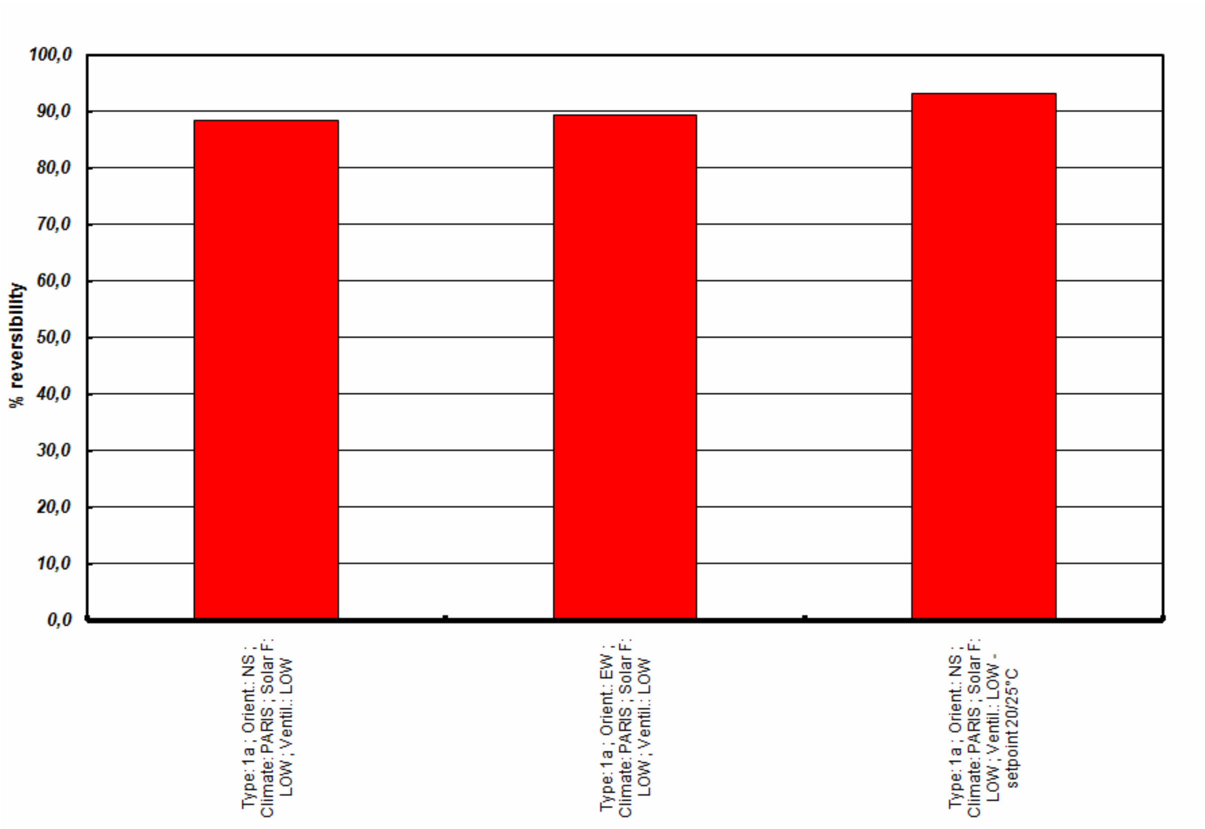


Figure 21: Reversibility potential in % for building type 1a in Paris for <u>two levels of set point temperatures and two orientations</u>

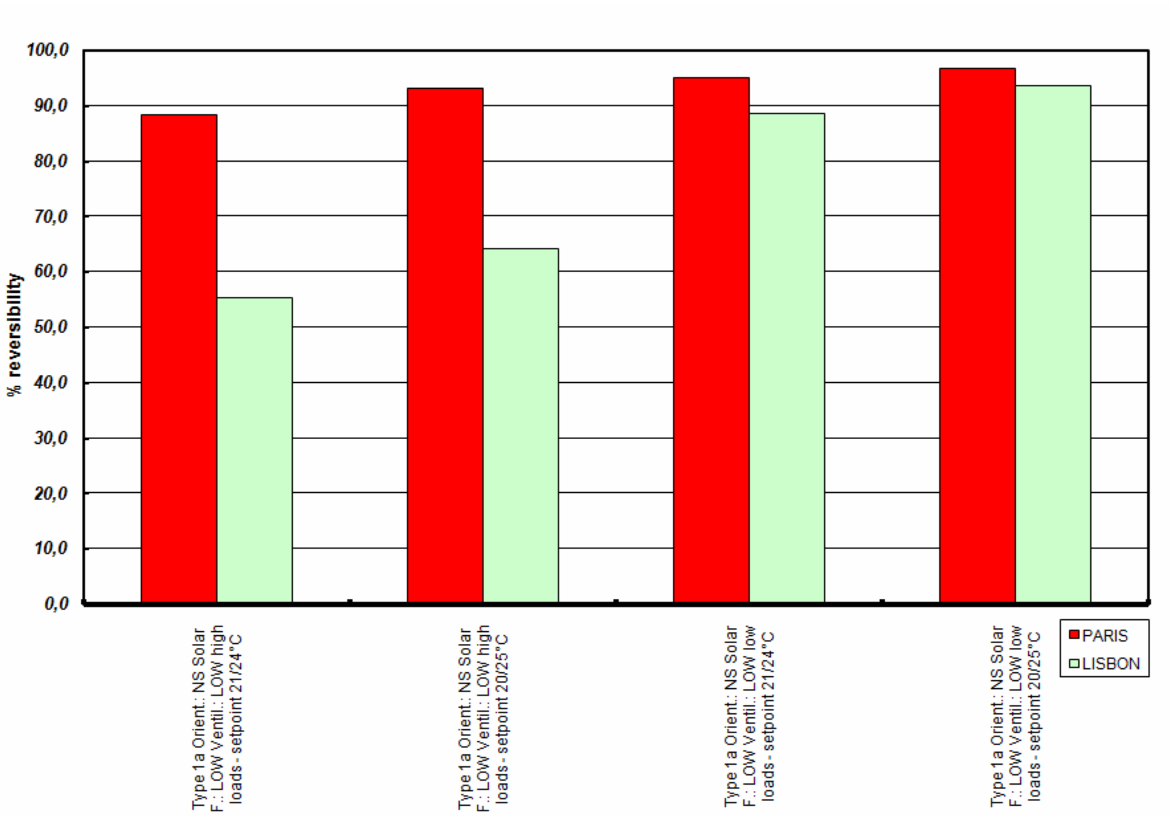


Figure 22: Reversibility potential in % for building type 1a for <u>two levels of set point temperatures and two internal load levels</u>

1.3 *Recovery potential*

The recovery potential depends on the simultaneous heating and cooling demand and on the heat power available on chiller condenser (only consideration on energy recovery for space heating is taken into account here in office buildings, but heat recovery could also possible for Domestic Hot Water and/or humidification).
The recovery potential is calculated hour by hour (Figure 23) as the percentage of heating demand which could be provided by a chiller condenser under the following conditions :

- the "chiller" is in operation in order to provide the cooling demand;
- the "chiller" is designed to meet the peak cooling load;
- The maximum heating power available on the condenser is calculated based on energy conservation principle such as (EER+1)/EER × cooling power provided by the chiller at the step time, the supplementary heating demand is assumed to be covered by the boiler.

No consideration on the emitter temperature levels required is done here. The % of recovery is calculated as the ratio between the total heat recovery for space heating and the total space heating demand.

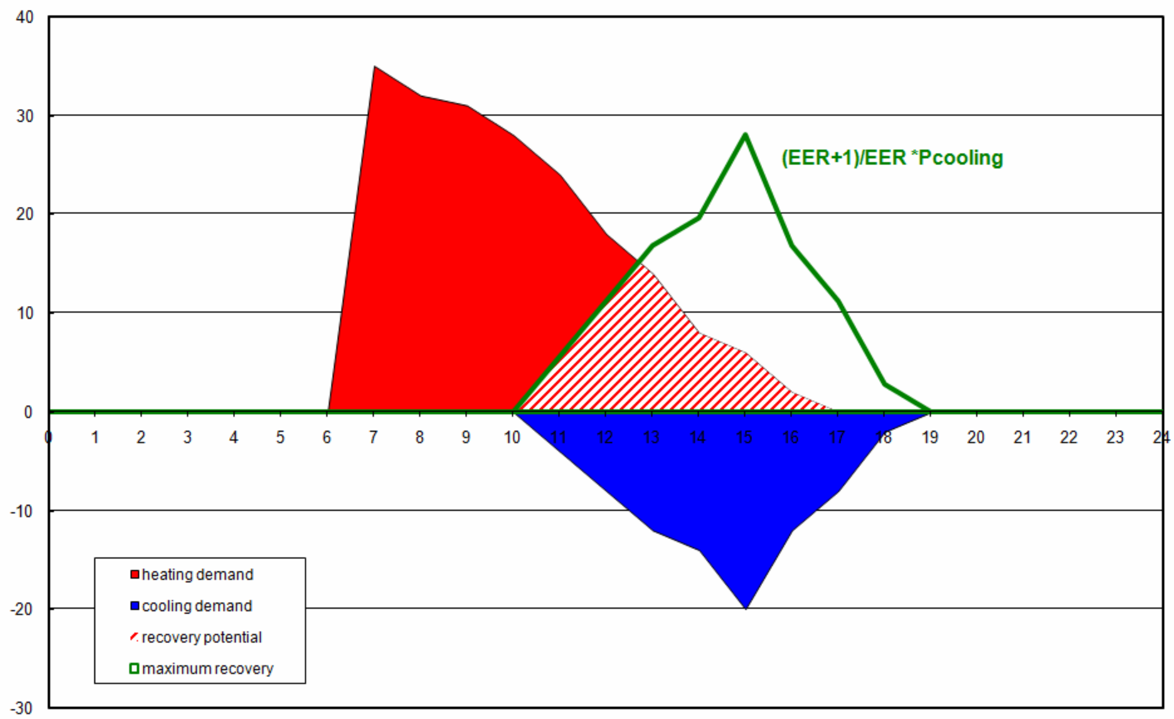


Figure 23: Estimate of heating energy recovery on chiller condenser

For all the cases, the recovery potential ranges between 0% and 45%. The solar factor and the ventilation rate have a clear impact on the recovery potential (Figure 24). Furthermore, the recovery potential is higher in climates (Athens and Lisbon) where the reversibility potential is the lowest (Figure 25). Largest differences on the recovery potential are found between the type 3 building (lowest potential) and the type 1 a building (highest potential) (Figure 26). If orientation does not have a high influence on the recovery potential, the change of set point from 21/24°C to 20/25°C induces about 40% decrease of the recovery potential (Figure 27). Higher set points and lower internal building loads can reduce the recovery potential to almost zero as in Paris (Figure 28).

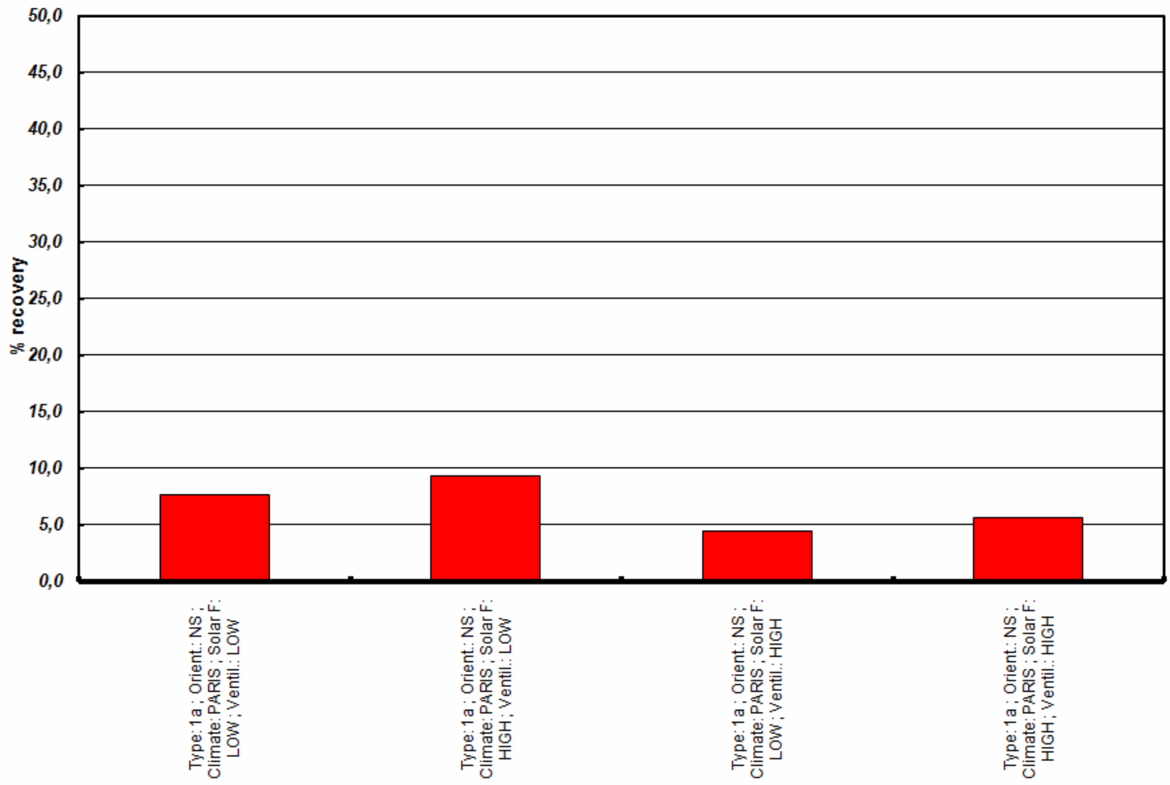


Figure 24: Recovery potential in % for building type 1a in Paris for <u>two levels of ventilation rate and solar factor</u>

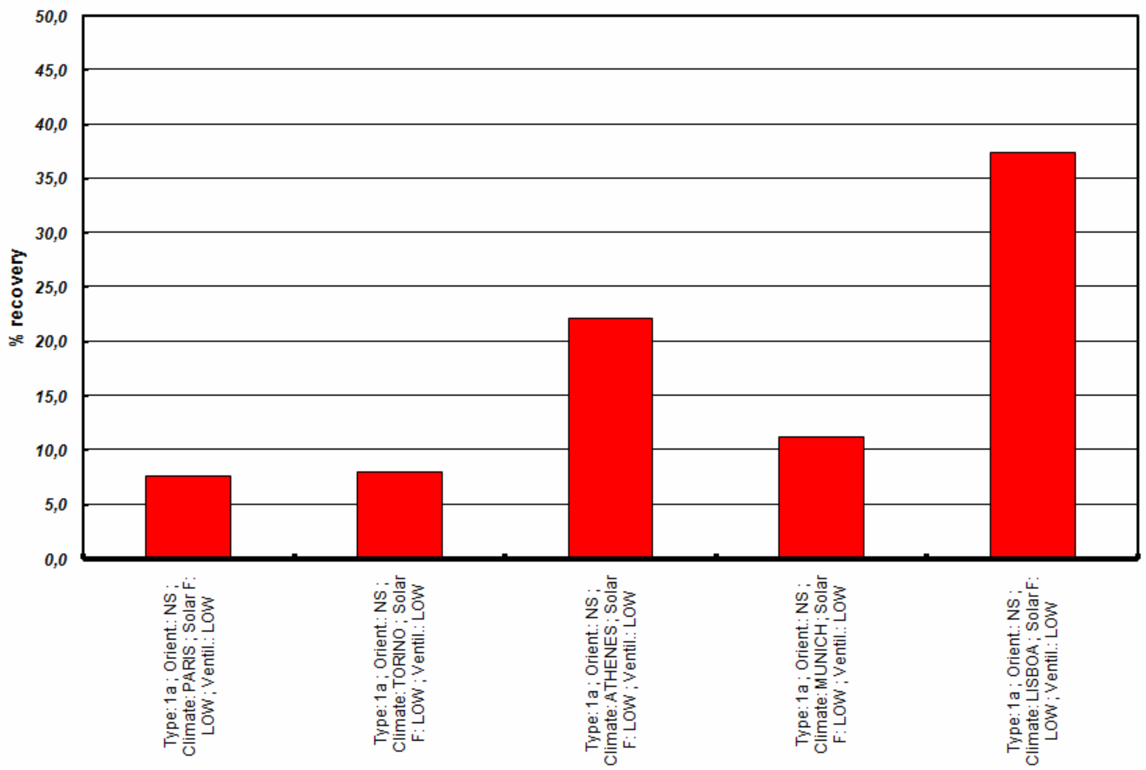


Figure 25: Recovery potential in % for building type 1a in <u>different climates</u>

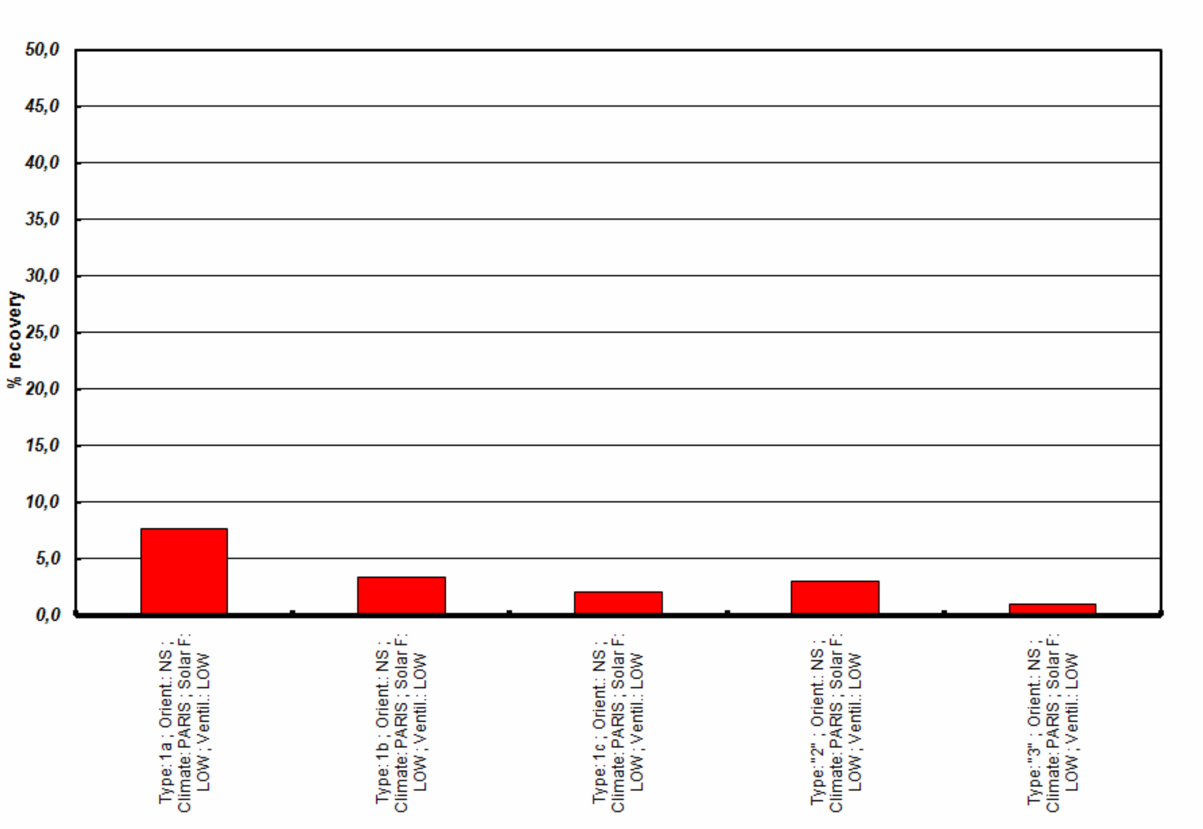


Figure 26: Recovery potential in % for <u>different building types</u> in Paris

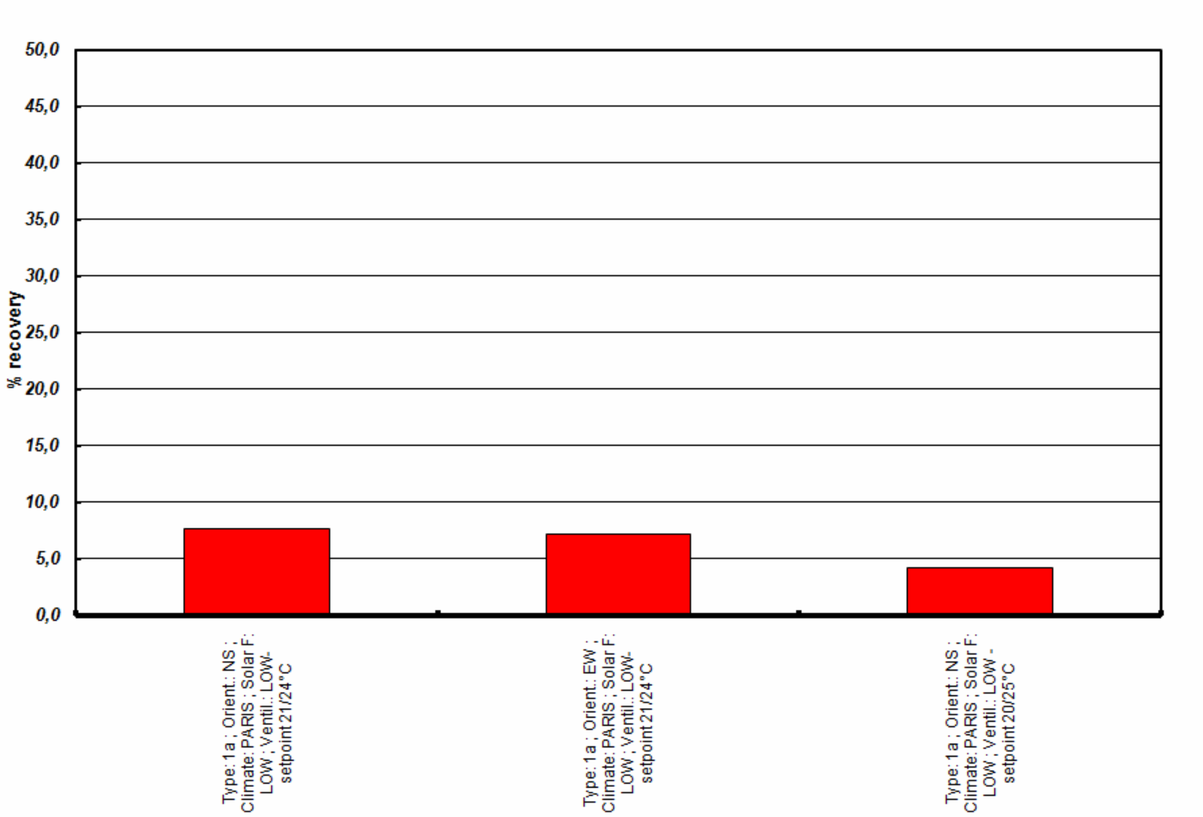


Figure 27: Recovery potential in % for building type 1a in Paris for <u>two levels of set point temperatures and two orientations</u>

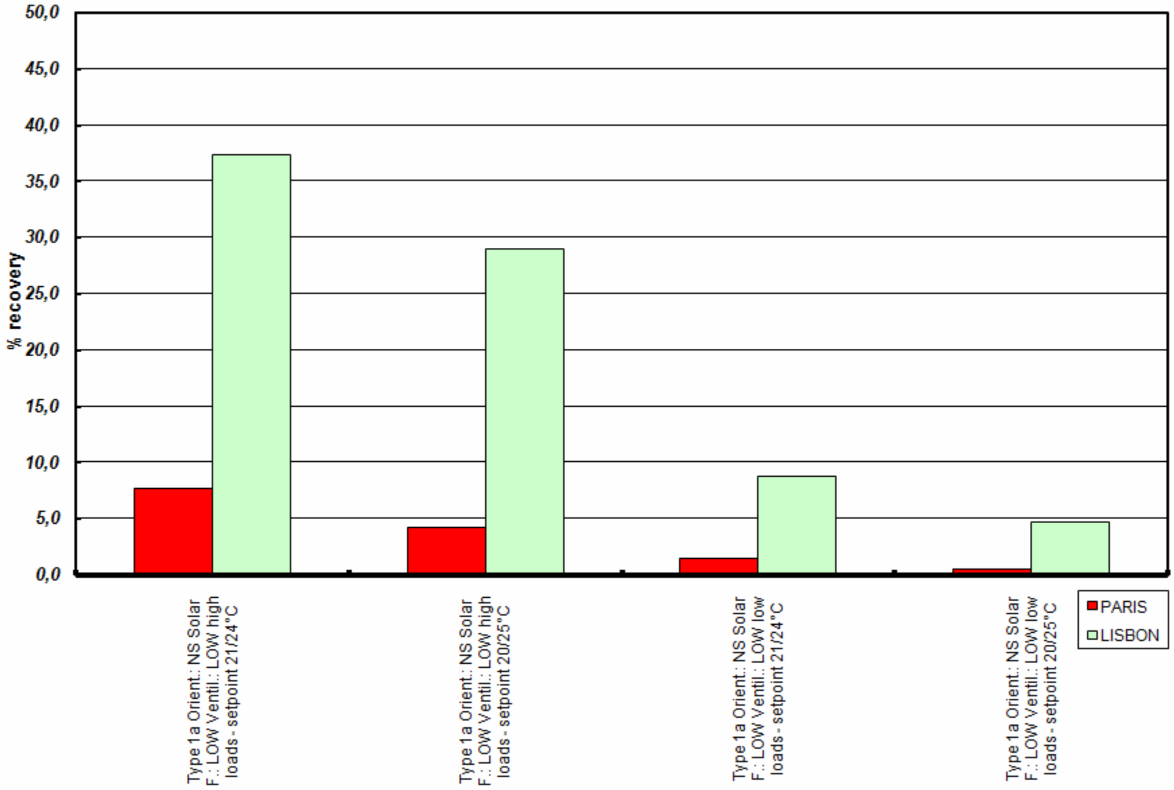


Figure 28: Recovery potential in % for building type 1a for <u>two levels of set point temperatures and two internal load levels</u>

2 Health care institutions

2.1 *Heating and cooling demands*

The results for each simulation case are presented in a Table in Annex 3 of the ANNEXES to this report [STA, 2008]. One can notice that :

- The ventilation rate has a very high impact on the heating demand, with an increase of the heating demand by about 60% to more than 100% in type 1 buildings and about 50% in type 2 buildings (Figure 29) when the ventilation rate is multiplied by 2;
- Except in the climate of Athens, a higher ventilation rate allows one to reduce the cooling demand. The reduction of the cooling demand in the climate of Paris, in the building of type 1, is about 10%when ventilation rate is doubled (Figure 29). If a twice better solar protection reduces the cooling demand by about 7% in building of type 1. The impact of solar factor is much more higher in type 2 building (the cooling demand is reduced from 15% to 56% by using better solar protection) since the solar gains are preponderant in the cooling demand ;
- The climate has a large influence on the heating and cooling demands (Figure 30). The heating demand for the building of type 1 is eight times higher in Munich than in Lisbon. The cooling demand is four times higher in Athens than in Munich.
- The buildings of type 1 and 2 are very different in terms of cooling demand (Figure 31). The cooling demand is higher in building of type 1 as well as the heating demand. Notice that in Rest home buildings, the air conditioned space is often limited to restaurant and common rooms, so that a type 2 bis has been introduced to account for this.
- Domestic Hot Water uses a lot of energy in health care institutions all along the year (Figure 32). Calculations are made by considering 55°C set point temperature and main water temperature from METEONORM vers. 5.0. In large hospitals, control humidity is required in some zones, which implies large energy demands for steam humidification (Figure 33).

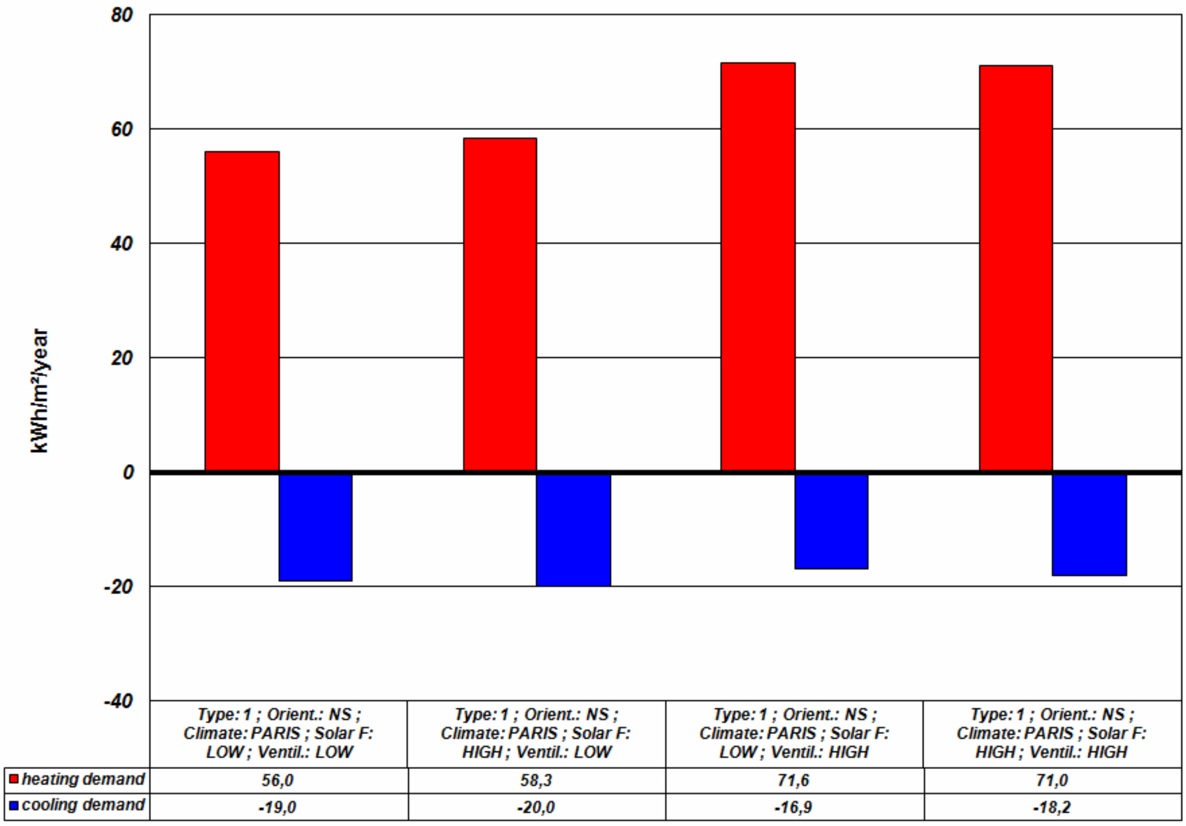


Figure 29: Impact of <u>ventilation rate and solar factor</u> on the heating and cooling demands (example: Large Hospital in Paris)

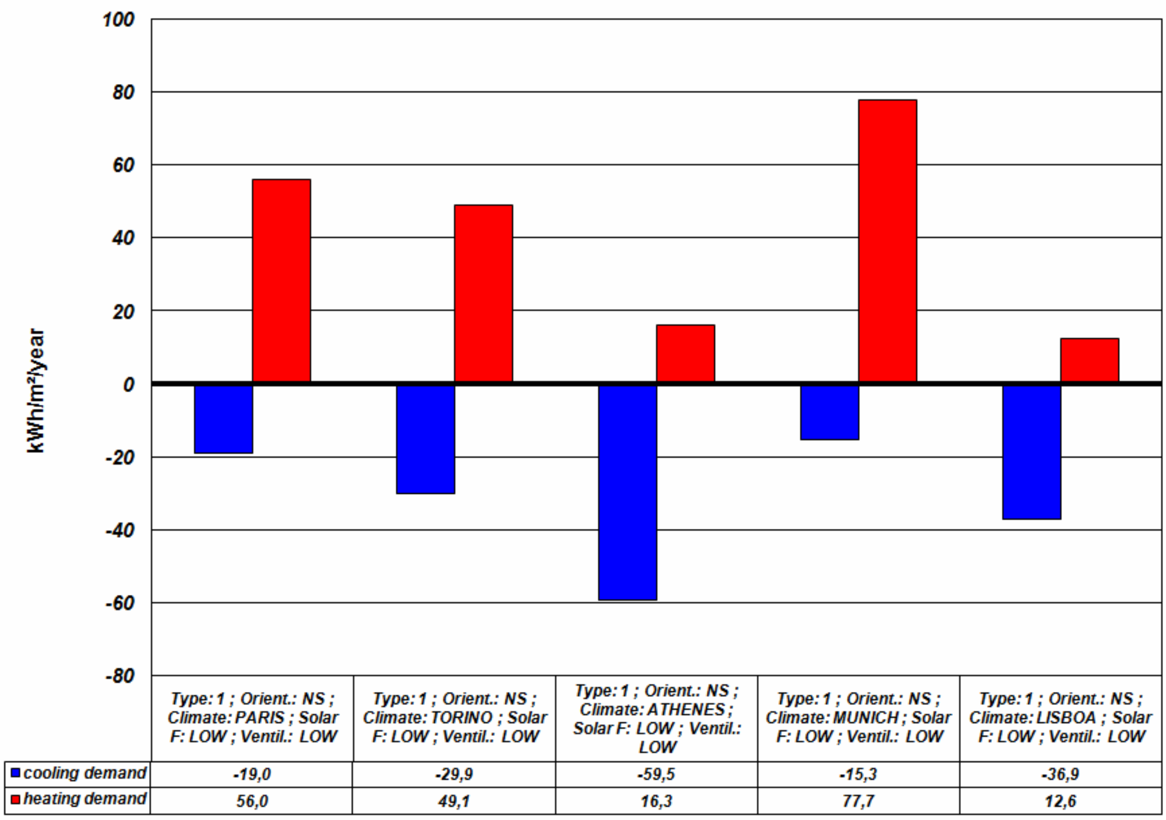


Figure 30: Impact of <u>climatic zone</u> on the heating and cooling demands (example: Large Hospital, Low solar factor, Low ventilation rate)

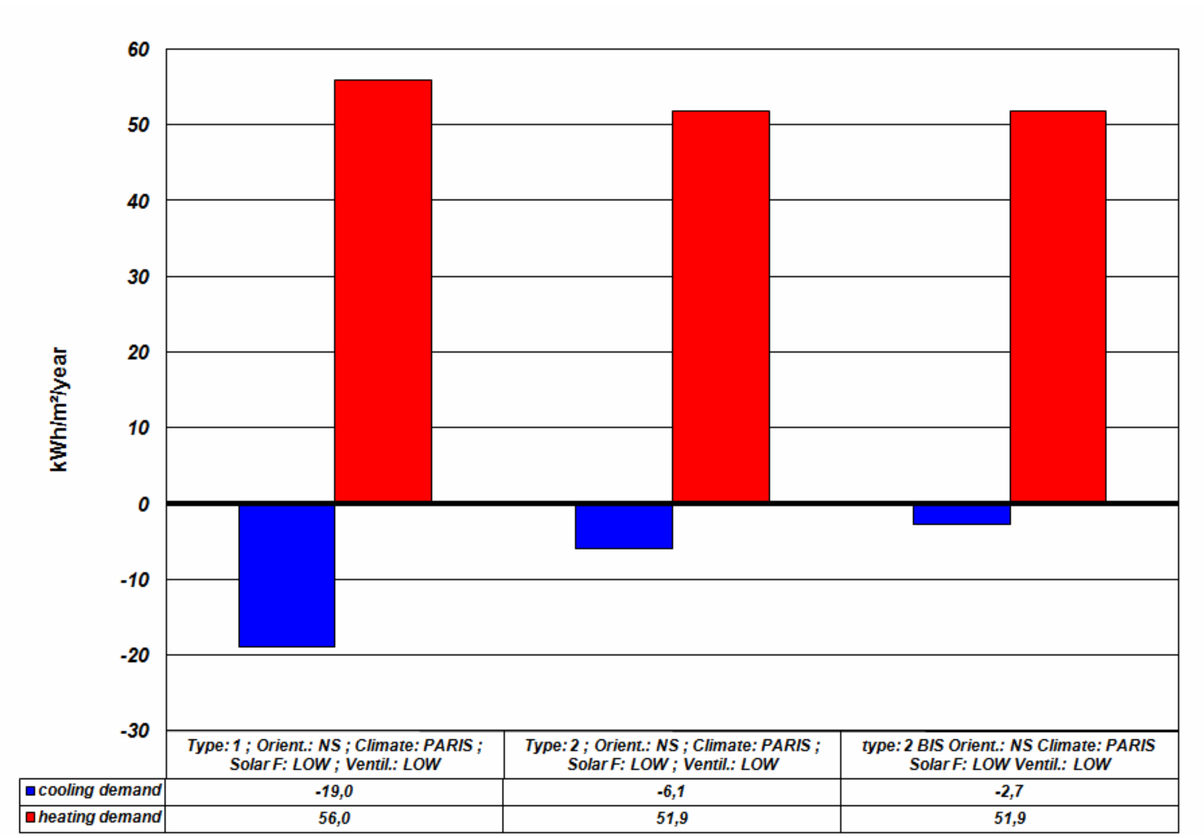


Figure 31: Impact of building type on the heating and cooling demands (example: in Paris, with a North/south orientation, low solar factor and low ventilation rate)

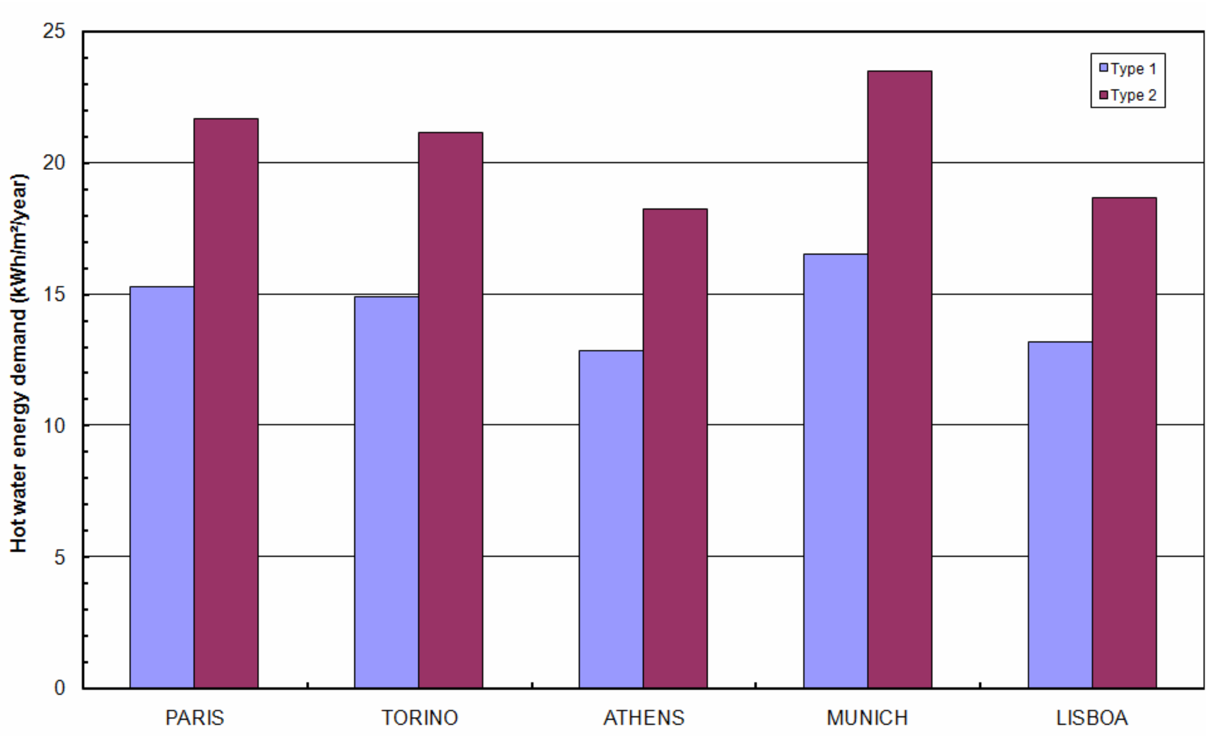


Figure 32: Energy demand for Domestic Hot Water in health care institutions in different climates

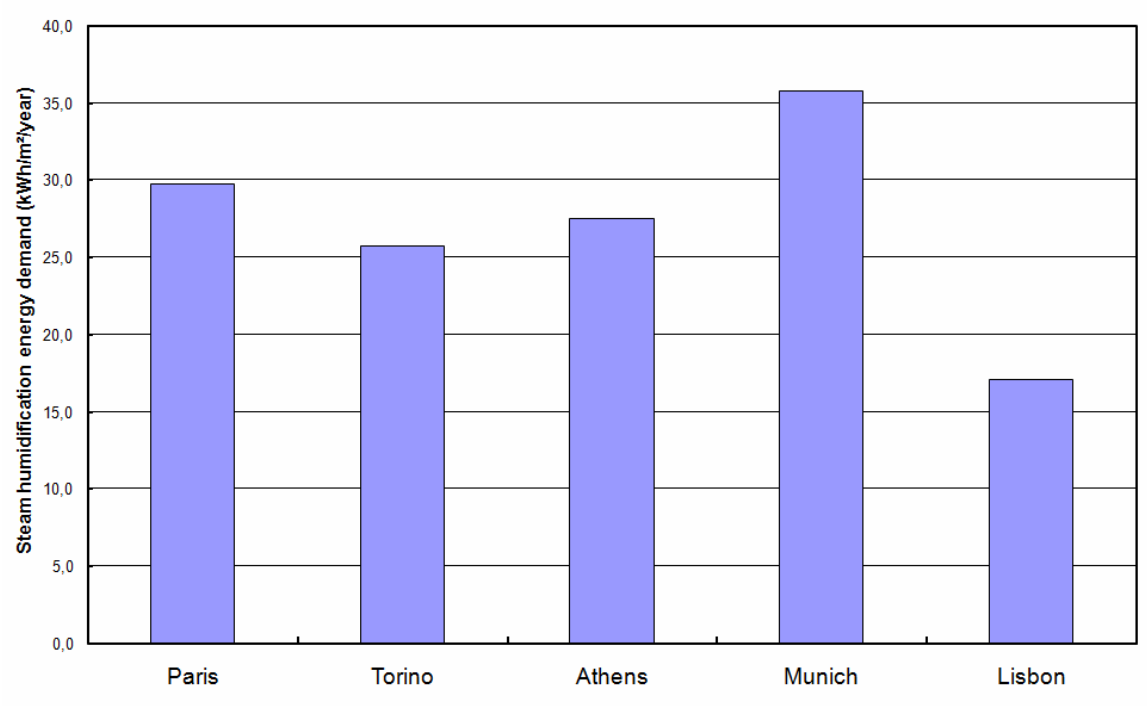


Figure 33: Energy demand for steam humidification in large hospitals in different climates

2.2 *Reversibility potential*

The reversibility potential is calculated hour by hour as in the Figure 16.

For all the cases, the reversibility potential ranges between 22.6% and 99.8%. The best reversibility potentials are in fully air conditioned Rest homes (Figures 34, 35 and 36). It appears very adequate whatever the climatic zone. In large hospitals, the best potentials for reversibility are in Paris, Torino, and Munich. On the other hand, Athens and Lisbon are more adequate climates for reversibility in partially air conditioned rest homes. This is due to the fact in partially air conditioned rest homes, the maximum cooling power is lower than in fully air conditioned heath care institutions and so the maximum heating power available by heat pumping is lower. In the hottest climates, the ratio between heating and cooling power is much favourable. The solar factor and the ventilation rate do not have a high impact on the reversibility potential (Figure 37).

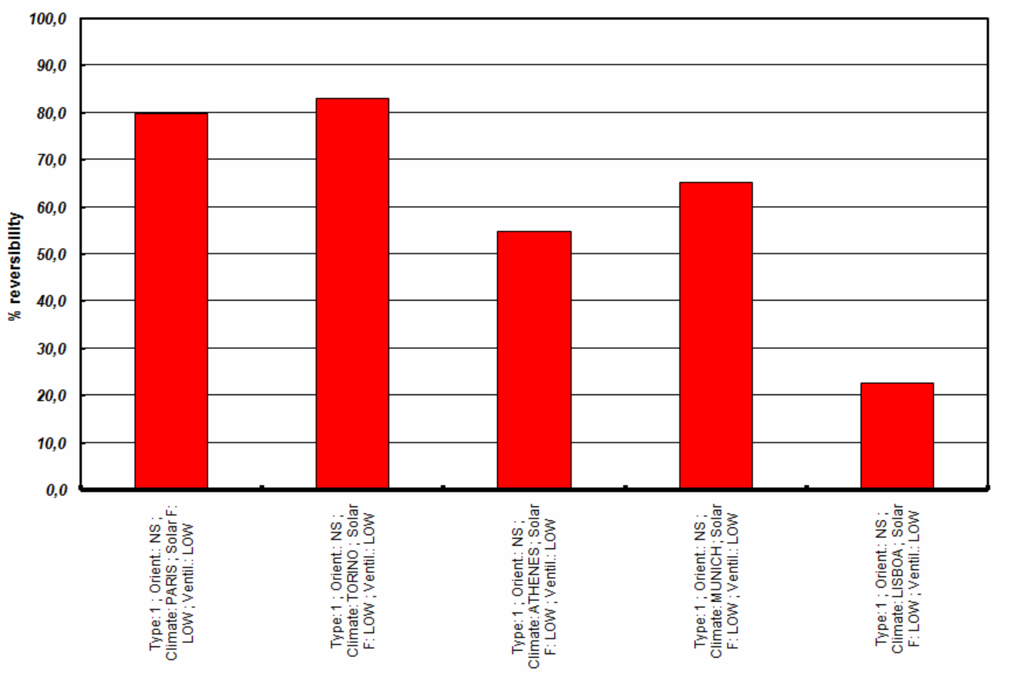


Figure 34: Reversibility potential in % for large hospitals in <u>different climates</u>

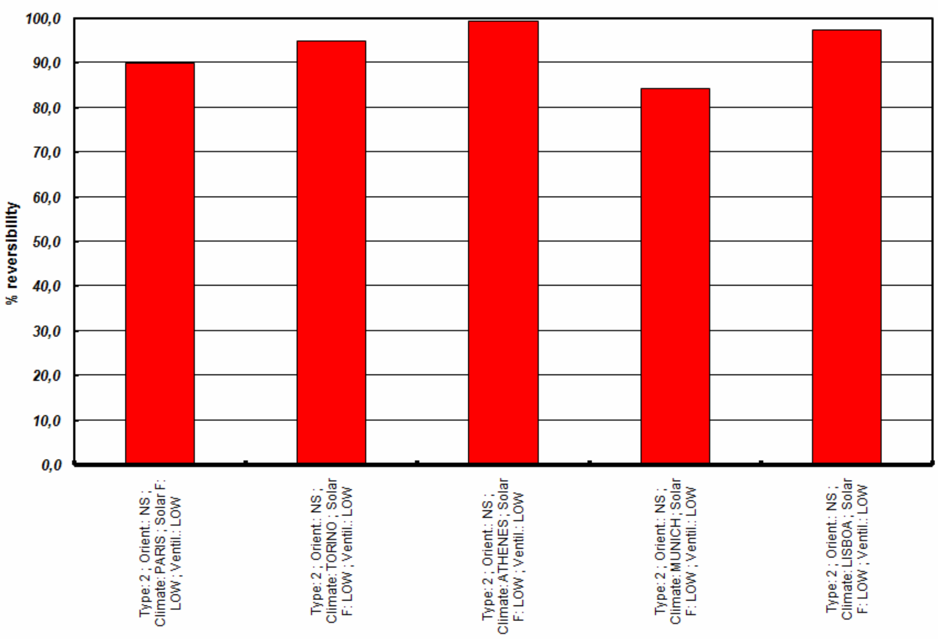


Figure 35: Reversibility potential in % in fully air conditioned rest homes in <u>different climates</u>

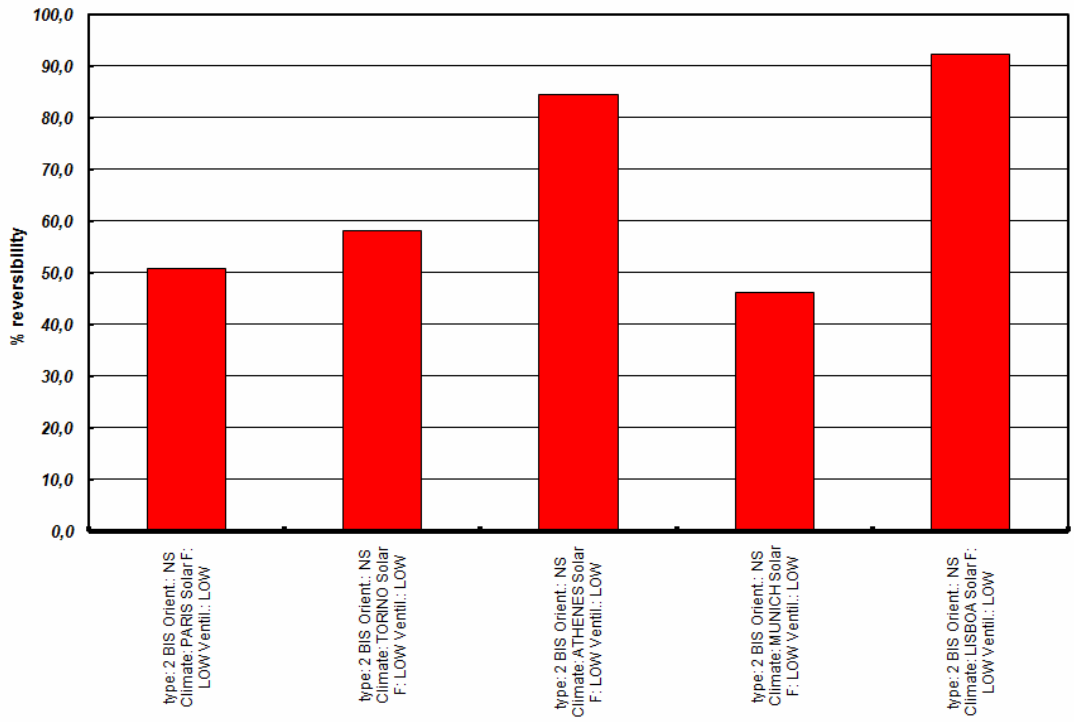


Figure 36: Reversibility potential in % for partially air-conditioned rest homes in <u>different climates</u>

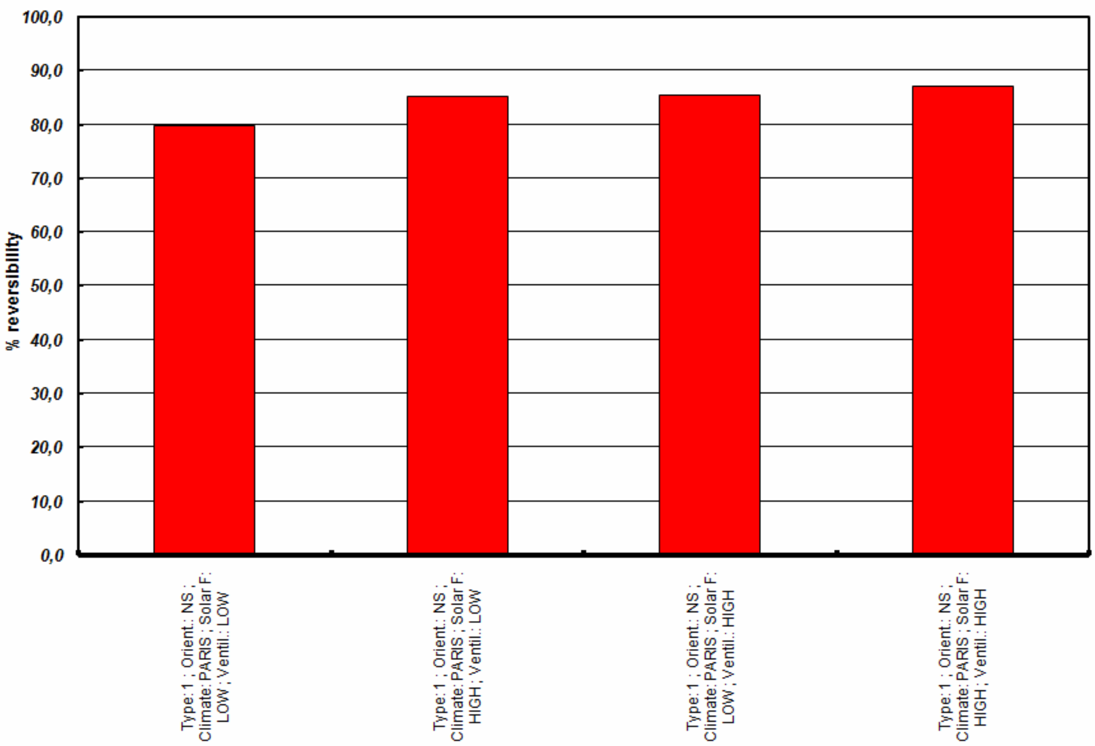


Figure 37: Reversibility potential in % for large hospitals in Paris for <u>two levels of ventilation rate and solar factor</u>

2.3 *Recovery potential*

The heat on the chiller condenser can be recovered for space heating, Domestic Hot Water and steam humidification. The potentials are calculated first separately according to space heating, steam humidification and domestic hot water:

- The heat recovery on space heating depends on the simultaneous heating and cooling demand and on the heat power available. The recovery potential is calculated hour by hour (Figure 23) as the percentage of heating demand which could be provided by the chiller condenser.
- The heat recovery by steam humidification is limited since the steam generation is at temperatures higher than the maximum temperature which can provide the chiller at the condenser[17]. The recovery potential on preheat of water to 55°C represents only about 5% of energy required for steam generation. So, it can represent at best 1.5kWh/m^2/year of energy savings. The recovery potential is calculated hour by hour on the same principle as for space heating but considering that only 5% of the energy required by steam generation can be recovered.
- The heat recovery on Hot Domestic Water is assessed on the same principle as for space heating. No consideration on the temperature level required is taken into account here.

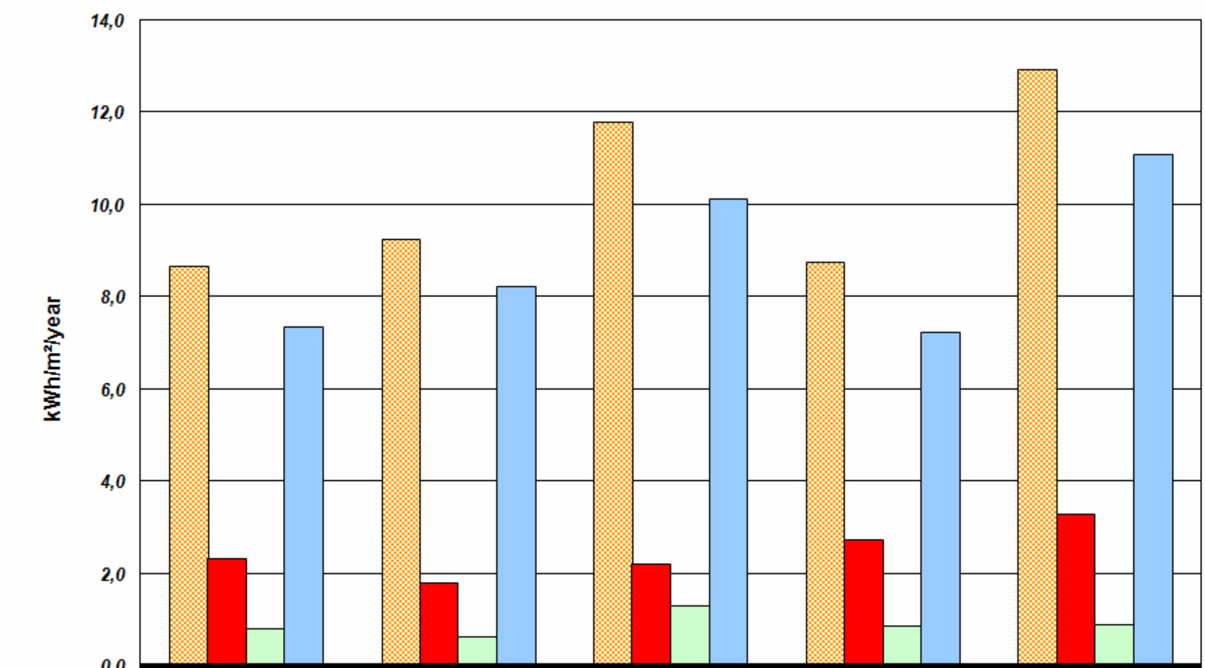
Then, the heat recovery potentials are calculated including the three applications (space heating, steam humidification and DHW) without considering any priority between them.

Figures 38 and 39 show the recovery potential in kWh/m^2/year in both types of health care institutions for different climates. The best potential of heat recovery is largely on Hot Domestic Water. The recovery on space heating is quite interesting in large hospitals but very low in rest homes. In fact, in large hospitals, the number of simultaneous heating and cooling hours is superior to 1000 whatever the case. On the contrary, in rest homes, the number of concomitant heating and cooling hours is very low in most of the cases. The recovery potential on steam generation is quite low. The total on Figure 38 and Figure 39 corresponds to a simultaneous recovery on space heating, DHW and steam generation with consideration on the limitation of heat on the condenser.

One can see on Figure 40 that the recovery potential on DHW ranges between 10% and 84% of the total heat required for DHW. The recovery potential on space heating ranges between 0% and 26%. In partially air-conditioned rest homes, the recovery potential for space heating is almost zero and for domestic hot water is about twice less than in fully air conditioned rest homes (not represented on the following figures).

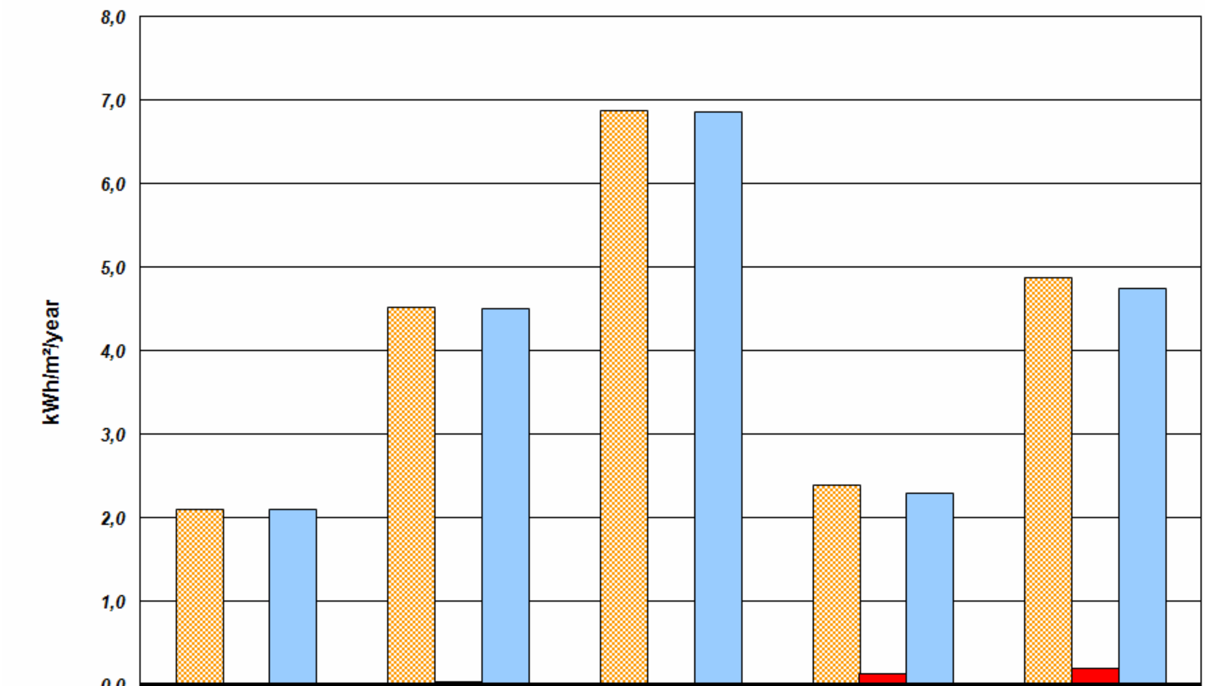
If the solar factor does not have important impact on the recovery potential, the impact of ventilation rate is slight (Figure 41).

[17] A much more promising solution would be to replace the steam humidifier by an adiabatic humidifier and a heating coil coupled to the heat pump; This solution is however less and less used in hospitals.



	Type: 1 ; Orient.: NS ; Climate: PARIS ; Solar F: LOW ; Ventil.: LOW	Type: 1 ; Orient.: NS ; Climate: TORINO ; Solar F: LOW ; Ventil.: LOW	Type: 1 ; Orient.: NS ; Climate: ATHENES ; Solar F: LOW ; Ventil.: LOW	Type: 1 ; Orient.: NS ; Climate: MUNICH ; Solar F: LOW ; Ventil.: LOW	Type: 1 ; Orient.: NS ; Climate: LISBOA ; Solar F: LOW ; Ventil.: LOW
total	8,7	9,2	11,8	8,8	12,9
space heating	2,3	1,8	2,2	2,7	3,3
steam	0,8	0,6	1,3	0,8	0,9
DHW	7,3	8,2	10,1	7,2	11,1

Figure 38: Recovery potential in kWh/m²/year in large hospitals for <u>different climates</u>



	Type: 2 ; Orient.: NS ; Climate: PARIS ; Solar F: LOW ; Ventil.: LOW	Type: 2 ; Orient.: NS ; Climate: TORINO ; Solar F: LOW ; Ventil.: LOW	Type: 2 ; Orient.: NS ; Climate: ATHENES ; Solar F: LOW ; Ventil.: LOW	Type: 2 ; Orient.: NS ; Climate: MUNICH ; Solar F: LOW ; Ventil.: LOW	Type: 2 ; Orient.: NS ; Climate: LISBOA ; Solar F: LOW ; Ventil.: LOW
total	2,1	4,5	6,9	2,4	4,9
space heating	0,0	0,0	0,0	0,1	0,2
DHW	2,1	4,5	6,9	2,3	4,7

Figure 39: Recovery potential in kWh/m²/year in rest homes for <u>different climates</u>

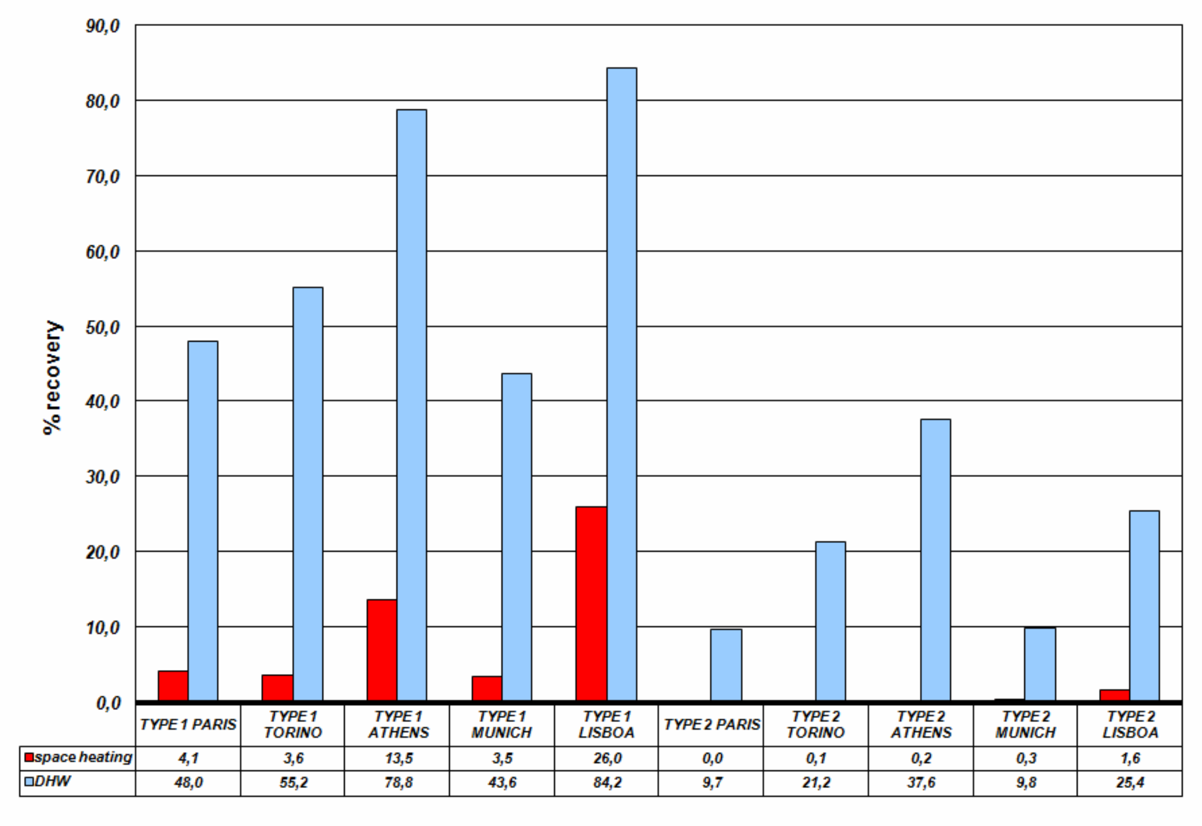


Figure 40: Recovery potential on space heating and Domestic Hot Water in % in large hospitals (type 1) and Rest homes (Type 2) for <u>different climates</u>

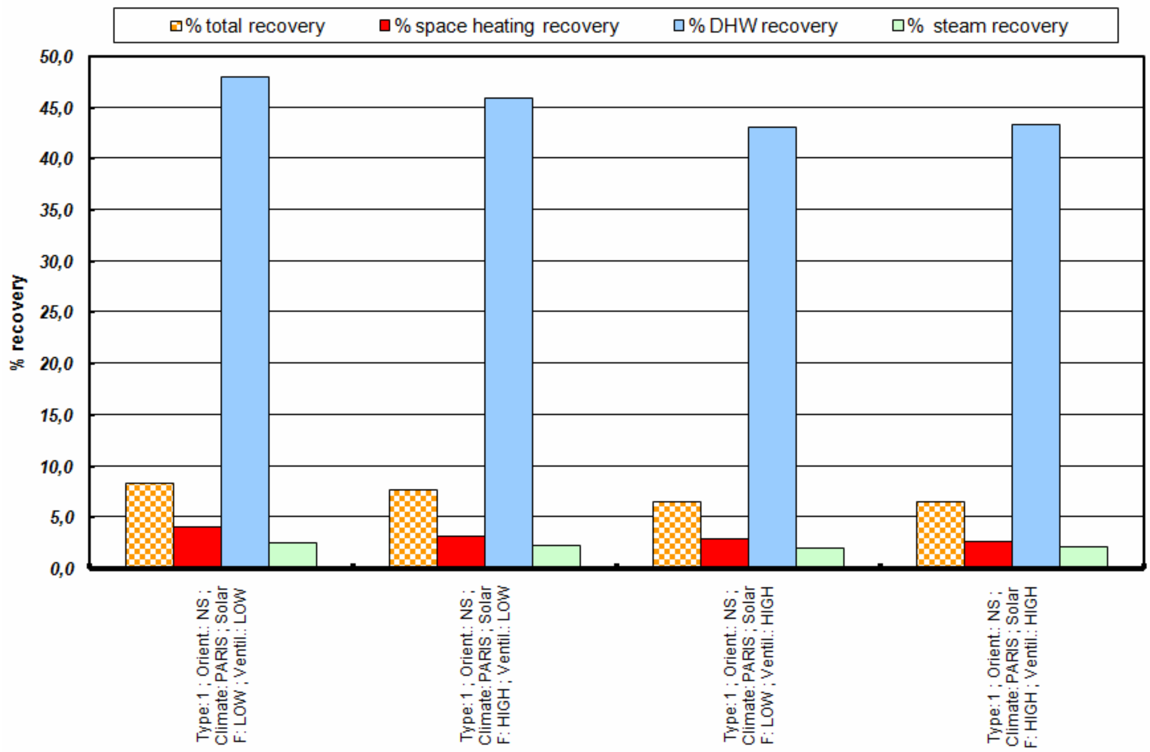


Figure 41: Recovery potential in% in large hospitals for <u>different ventilation rates and solar factors</u>

3 Conclusions

The analysis of building loads gives the following conclusions:

Office buildings offer a high potential of reversible use of the chillers since the maximum cooling power is close to the maximum heating power whatever the building type. Moreover, the potential is more interesting in temperate climates (Mid -North of Europe, France, North of Italy and North of Spain) than in hot climates (periphery of Mediterranean Sea). On the contrary, the recovery potential on chiller condenser for space heating is the most interesting in Southern Europe.
Furthermore, the building orientation does not appear as an important parameter on the reversibility and recovery potentials.
The data presented here before for office buildings give an overview of energy saving potentials on chiller for space heating.
The simulated office buildings are supposed to be representative of the European **air-conditioned** office building stock.

Health care institutions offer a high opportunity of energy recovery essentially for Domestic Hot Water. Large hospitals are more interesting than rest homes since the cooling demand and the domestic hot water demand are larger. The energy recovery for space heating is less interesting in all the cases. In large hospitals with humidity controlled operation room, the potential of recovery for steam humidification is not negligible (up to 1,5 kWh) but very low compared to the Domestic Hot Water one. Moreover, **the reversibility potential is high in fully air conditioned rest homes. In large hospitals, the reversibility potential is quite high in moderate climate and medium in hot climates** (Athens and Lisbon). On the contrary, in partly air conditioned rest homes, the best climates for reversibility are Lisbon and Athens.

The recovery and reversibility potentials are theoretical and do not take into account technical problems to be solved and any payback time. Moreover, they are based on the energy demand and so it does not take into account the distribution system.
The reversibility potential is high, however, in the building stock, it is not all the time easy to take advantage of this potential since :

- The terminal units in existing buildings often operate at temperature levels inadequate to temperatures provided by the heat pump (higher than 60°C) ; Any change on terminal units is a costly action;
- The chiller are not always reversible so that some changes on the chiller are necessary (the reversing valve and the control unit at least); Hopefully, during these last years, most of the chillers sold on the market can be used as heat pumps ;
- The simultaneous need of cooling and heating cannot easily satisfied; A strategy with an alternate use of the chiller in cooling and heating mode should be developed to comply with comfort requirements at best;

The recovery potential faces also to the temperature levels required in terminal units and the costly and hard changes on the chiller.

In the following, the reversibility and recovery potentials are recalculated more finely in order to take into account the influence of the HVAC systems.

IV ANALYSIS OF SYSTEM PERFORMANCES

1 Introduction

The first objective of this chapter is to select the most representative HVAC equipments installed in office and heath care buildings. The second aim is to propose solutions for heat recovery on the chiller condenser and reversibility of the chiller. Then, the equipments are described in term of performances.
The data provided here will be used for energy consumption simulation in order to assess the energy saving potential of heat recovery and reversibility.

2 Reference HVAC systems

2.1 *Air conditioning share on European market*

In the following table, the share of air conditioning systems in office and health care buildings is given for Europe-15.

Table 16: Share of air conditioning systems in percentage of air conditioning surface [EEC, 2003]

	CAC* + Water distribution	CAC + air distribution	CAC split	CAC roof top	CAC VRF	RAC**
Office building	45.5%	27.7%	2.5%	1%	2.3%	21%
Health care institutions	58.6%	35.6%	0%	0%	0%	5.8%

*Central Air conditioner
** Room Air conditioner

The central air conditioners using chillers are predominant in office buildings and health care institutions. The chiller market is dominated by air cooled chillers compared to water cooled chillers, they represent 86% of the market [EEC, 2003]. Since VRF and Reversible Multi-split systems are marginal on the market, they will be excluded of this subtask. They will be studied later in order to be compared to other reversible systems.

2.2 *Distribution units*

Only all air and all water systems are considered here. They are supposed to be the most commonly used.
In case of water distribution, **Fan Coil Units** are the main system. Radiant panels are excluded here. A ventilation system is required. Two systems are considered in the following :

- Single flux ventilation;
- Double flux ventilation with heat recovery;

In case of air conditioning, we can find several designs of **Air Handling Units**. For the sake of simplicity, the study will be limited to two cases [REI, 1996] :

- Multizone Variable Airflow Volume (Heat recovery – Pre heating coil - Cooling coil – local heating coil); The VAV system is without any return air. Indeed, in multizone buildings with

different internal gains, the hygienic air flow rate can sometimes not be respected in some zones[18] [REC, 2001] ;
- Single zone Constant Airflow Volume (Return air damper – Pre-heating coil - Cooling coil – heating coil + option: Humidifier)

In the case of the operation room of the large hospital, the Air Handling Unit is always a CAV system with a humidifier.

2.3 *Heat and cold production*

The cold production is achieved by an air-cooled chiller or a water-cooled chiller.
The heat production is achieved by a boiler.

2.4 *Synthesis of systems to be studied*

Some systems are rarely implemented in some types of buildings. The following Table shows the correspondence between HVAC systems and building types which will be simulated except for Refrigerant-based systems.

Table 17: HVAC system type matched to building type [FIL, 2006]

	FCU + SF	FCU +DF	VAV	CAV	Refrigerant-based systems		
					VRF	Reversible multi SPLIT	Air to Air Heat Pump
Office type 1a		✓	✓	✓			
Office type 1b		✓	✓				
Office type 1c		✓	✓				
Office type 2	✓	✓			✓		✓
Office type 3	✓	✓			✓	✓	✓
Hospital type 1		✓		✓			
Rest home type 2	✓	✓			✓	✓	✓

Since Water-cooled chillers are more common in large buildings, the study of heat recovery on water cooled chillers will be limited to the office buildings of type 1 and the hospital.

The schemes of the studied reference HVAC systems are presented below.

[18] For instance, if high heating demand is requested in a North zone, the return air flow rate would be at the highest. If the heating demand in a South zone in the same time is low, the supply air flow rate would be low and thus the minimum hygienic air flow rate can be unsatisfied since the return air percentage is high.

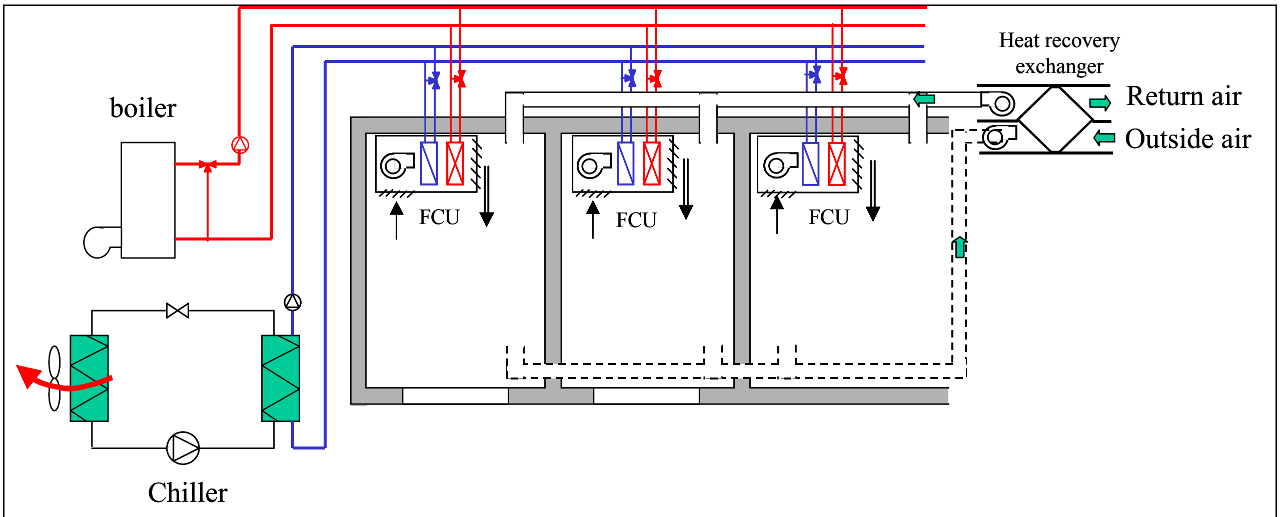


Figure 42 : Water distribution (FCU 4 pipes) + double flux ventilation with heat recovery (scheme according to [BOU, 1998])

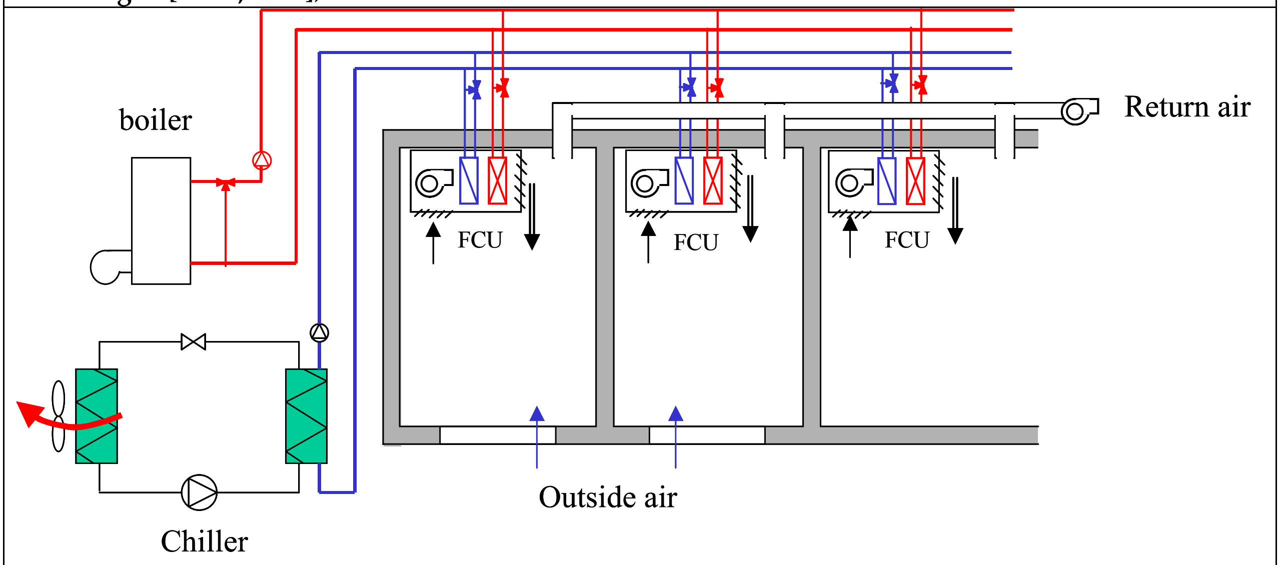


Figure 43 : Water distribution (FCU 4 pipes) + Single flux ventilation* (scheme according to [BOU, 1998])
*** In this case, the air in the conference room will be supplied by a double flux ventilation system without heat recovery**

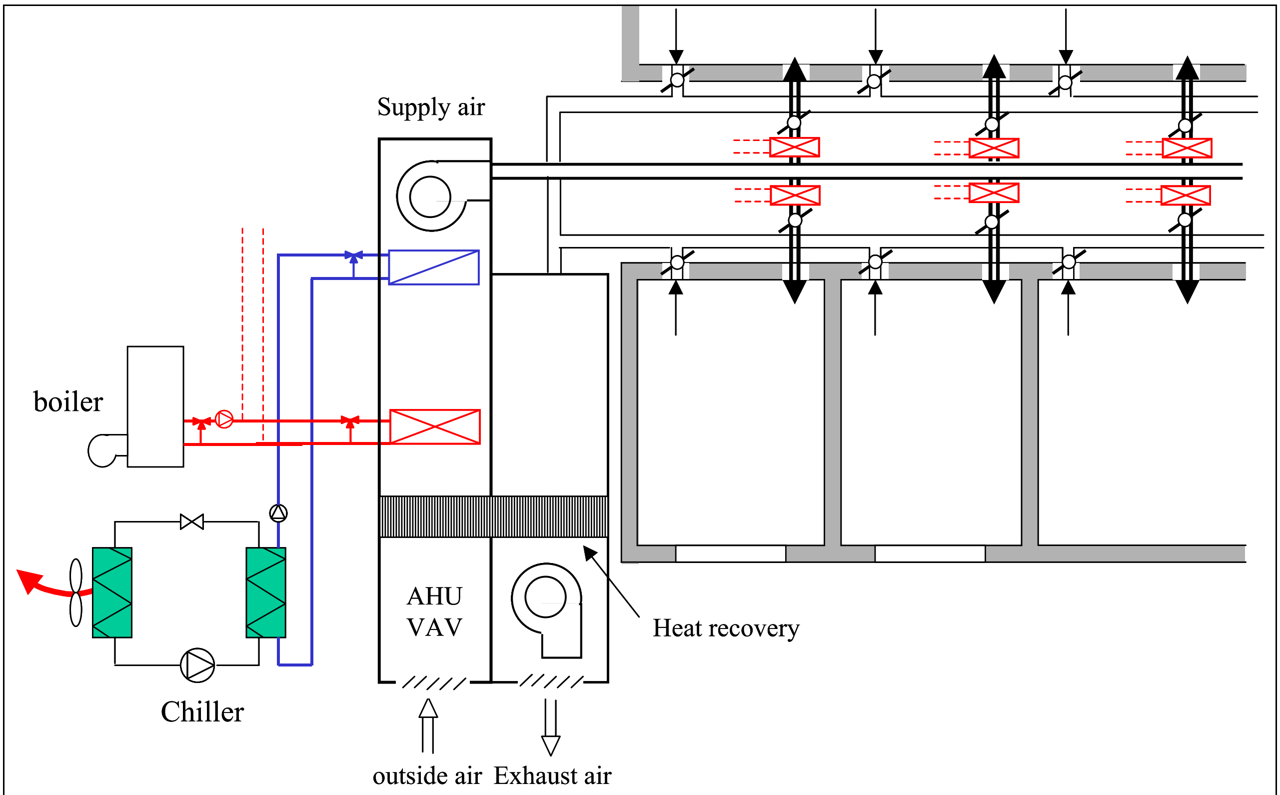


Figure 44 : Multi zone Variable Airflow Volume system - scheme according to [BOU, 1998]

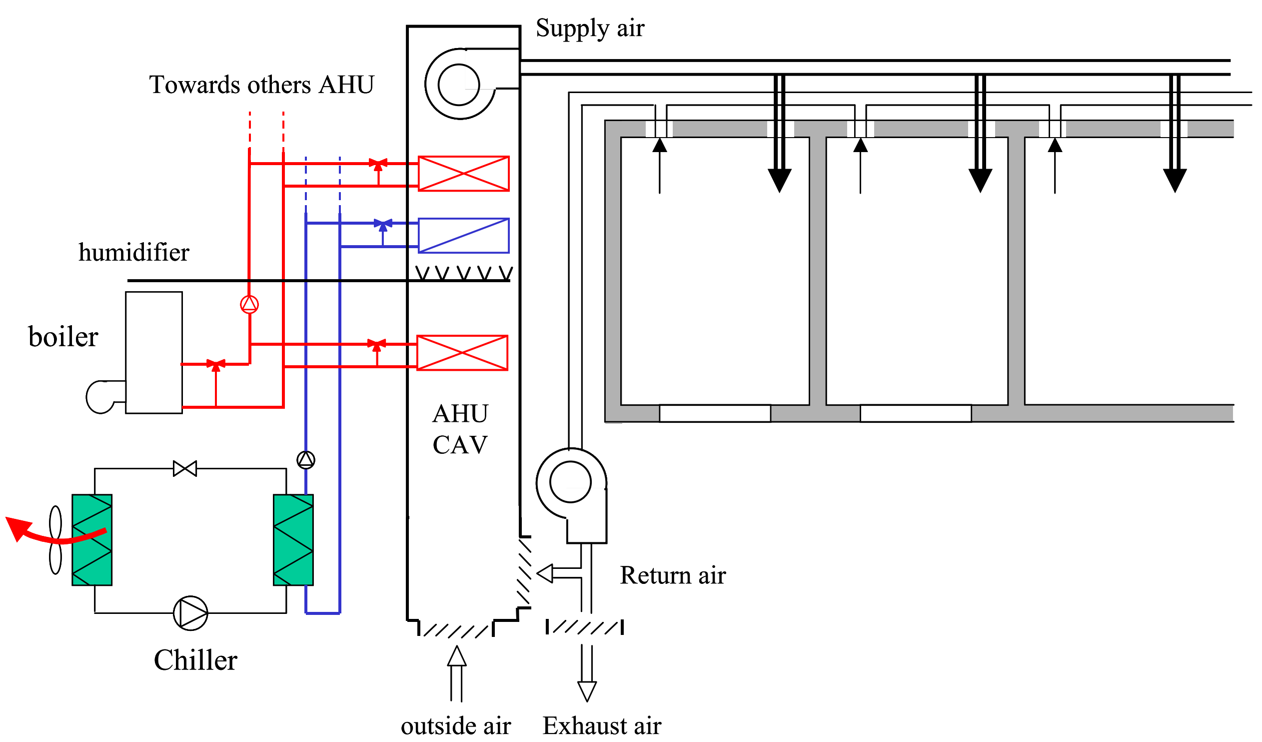


Figure 45 : Single zone Constant Airflow Volume system (humidifier is optional except for operation room) - scheme according to [BOU, 1998]

3 Reversibility and Heat Recovery systems

Three kinds of heat pump systems are distinguished :
- System with reversible heat pumping (no simultaneous cooling and heating demand);
- System with condenser heat recovery (simultaneous cooling and heating demand);
- System with reversible heat pumping and heat recovery.

The technologies are described in more details in the deliverable 1.4.

3.1 Reversible use of the chiller

Reversible heat pumps can be associated to fan coils units (FCU) or variable and constant air volume systems (VAV and CAV).
The reversible use of the chiller entails technical changes related to the changeover such as :
- Temperature level in distribution units;
- Management of the heating/cooling modes and the simultaneous heating and cooling demand;
- Reversible unit or reversible water circuit;
- Backup system.

3.1.1 Changeover : air cooled / water cooled chillers

The ***air cooled chillers*** are the most present technology on the market and a large offer of reversible units are proposed today.
Advantages:
- Availability of reversible units on the market
- Low costs

Drawbacks:
- Performance and capacity fall when the outside air is very cold in heating mode and when the outside air is very hot in cooling mode
- Simultaneous heating and cooling demands cannot be covered by the reversible Heat Pump

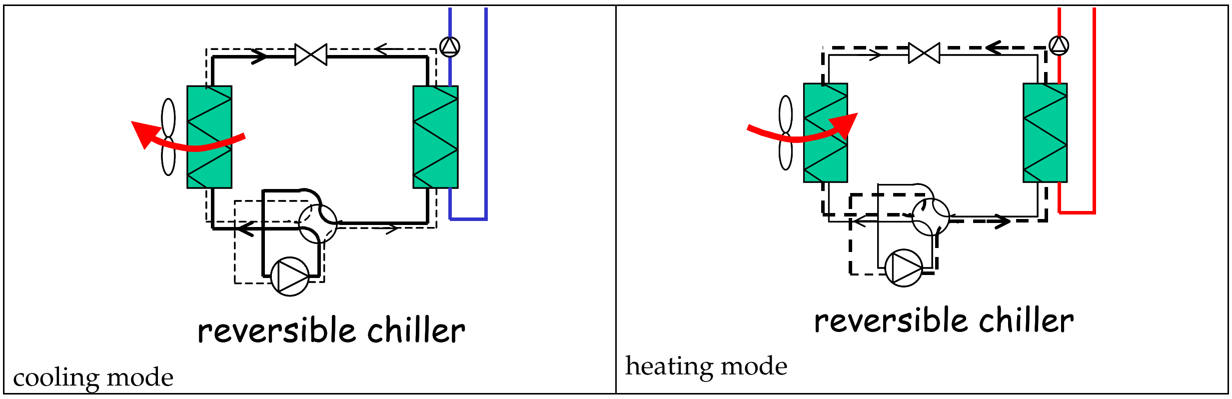


Figure 46: Air-cooled chiller with refrigerant change-over

The ***water-cooled chillers*** use essentially open cooling towers (for example, open cooling towers represented 92% of the heat rejection system market in Italy in 2000). This kind of heat rejection system is not appropriate for reversibility. The water cooled chillers appear to offer more opportunity for heat recovery.

If water or earth as heat source/sink are available, reversible heat pumping is an excellent solution. Since it can be considered as a marginal solution in commercial buildings up to now, these technologies will not be studied in this subtask.

3.1.2 Changeover : strategy

The reversible use of the chiller is then studied with the following assumptions:

- *Change over*:
  - In **high load buildings** (Figure 47), the cooling demand is almost all the year. However, as one can see on Figure 47 and Figure 48, the use of more ventilation and less restrictive set point temperatures can decrease strongly the cooling demand;
  - In l**ow load buildings**, the cooling demand in winter is roughly zero (Figure 49).

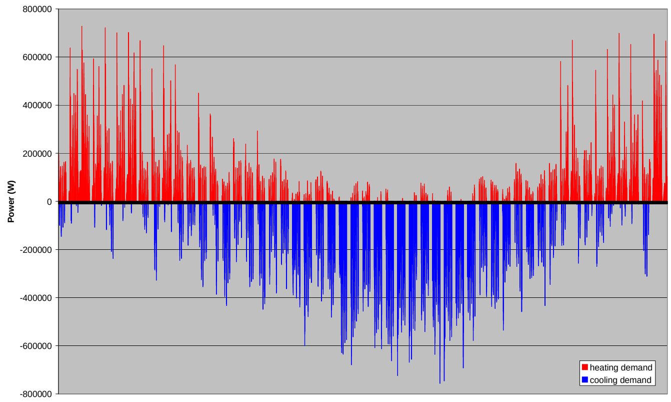


Figure 47: Hour by hour cooling and heating demand on one year (Type 1a, 21/24°C set points, low ventilation, high internal loads)

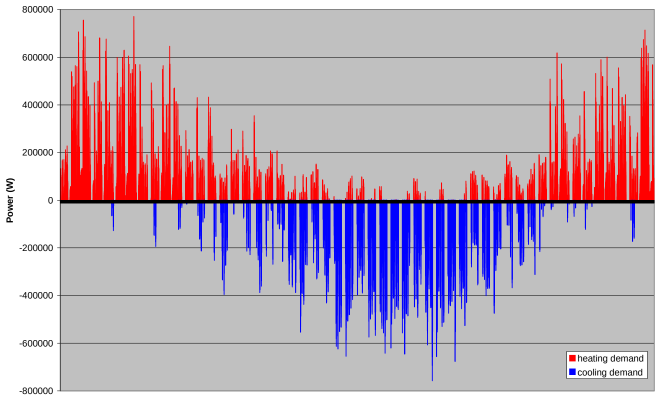


Figure 48: Hour by hour cooling and heating demand on one year (Type 1a, 20/25°C set points, High ventilation, high internal loads)

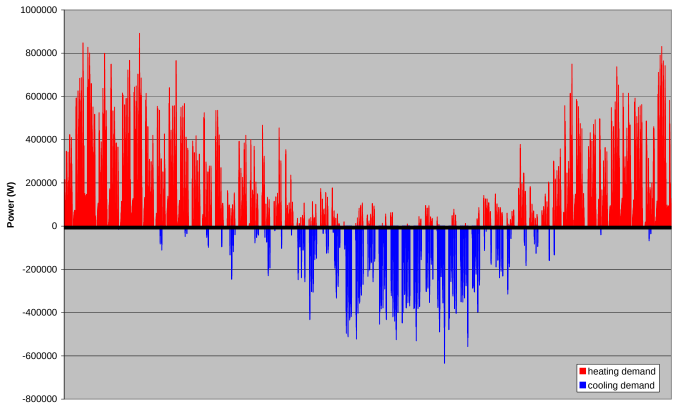


Figure 49: Hour by hour cooling and heating demand on one year (Type 1a, 20/25°C set points, high ventilation, low internal loads)

As daily change-overs do not seem realistic, a **simple change-over based on outside temperature will be studied** with the assumption that free cooling is available during the simultaneous cooling and heating demand days (for instance by the opening of the windows by the occupants).

3.1.3 Distribution units

A calculation on a typical FCU- 4 pipes (See in Annex 4 of the ANNEXES to this report [STA, 2008]) shows that the heat transfer is better in the cooling coil than in the heating coil. Even if water temperatures as high as 90°C are used in the heating coil, the same heat power can be transferred in the cooling coil with lower water temperatures, compatible with what can supply a Heat Pump. One assumes in the following that the cooling coil is used as a heating coil. This solution appears to be the simplest in terms of network retrofit.

3.1.4 Backup system

The boiler is used when the cooling and heating are required simultaneously and when only heating is required but the heat pumping cannot cover all the needs; In high load buildings, the chiller capacity is expected to cover nearly all the heating demand. In low load buildings, the boiler is expected to be used during the coldest days.

3.2 *Heat recovery*

The heat recovery on air-cooled chillers requires to change the refrigerant circuit (Figure 50). Concerning the water-cooled chillers, the change can be done on the water circuit (Figure 51). In office buildings, the simultaneity of heating and cooling demand is not very high; however the sequence of morning-time heating demand and afternoon-time cooling demand is more common. As a consequence, a heat storage could be considered.

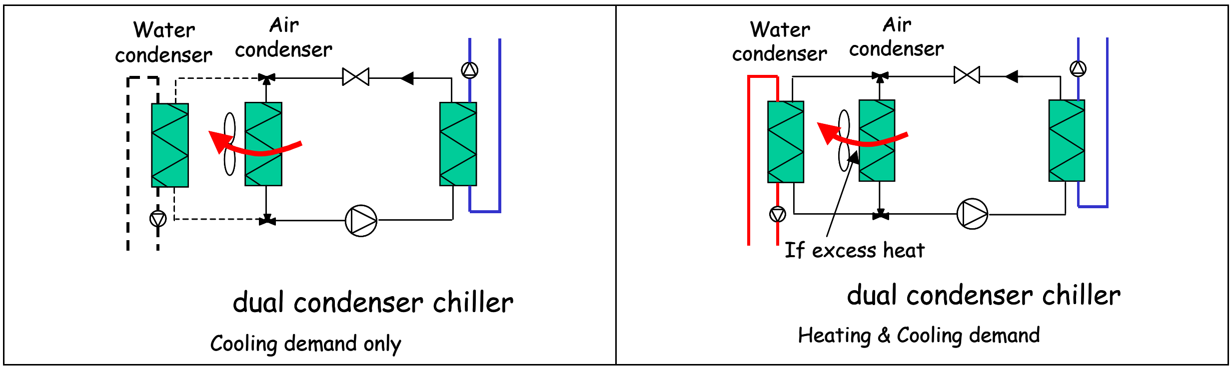


Figure 50: Heat recovery by using two condensers

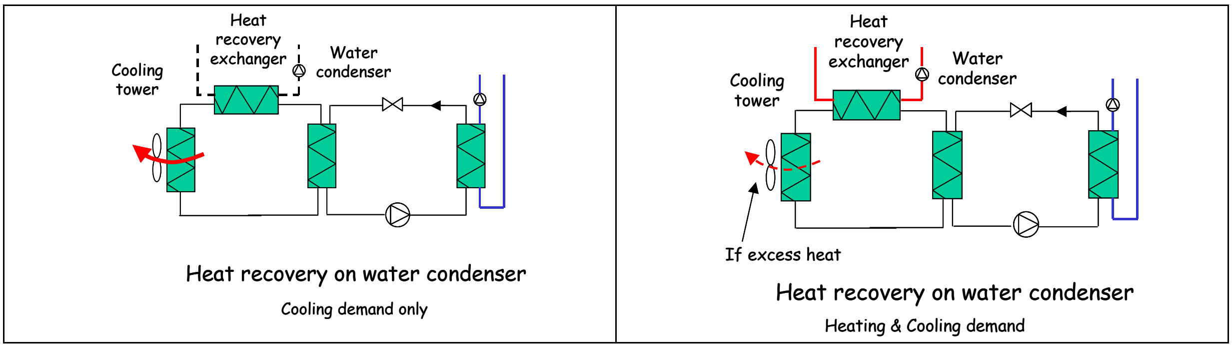


Figure 51: Heat recovery on a water condenser

3.3 *Systems with reversibility and heat recovery*

Outdoor air is a universal heat source/ sink for reversible heat pumps. A system with an air heat exchanger and two water heat exchangers can allow to operate in cooling or heating mode and to recover heat when cooling mode is selected as in Figure 52. This type of system is proposed for instance by CIAT.

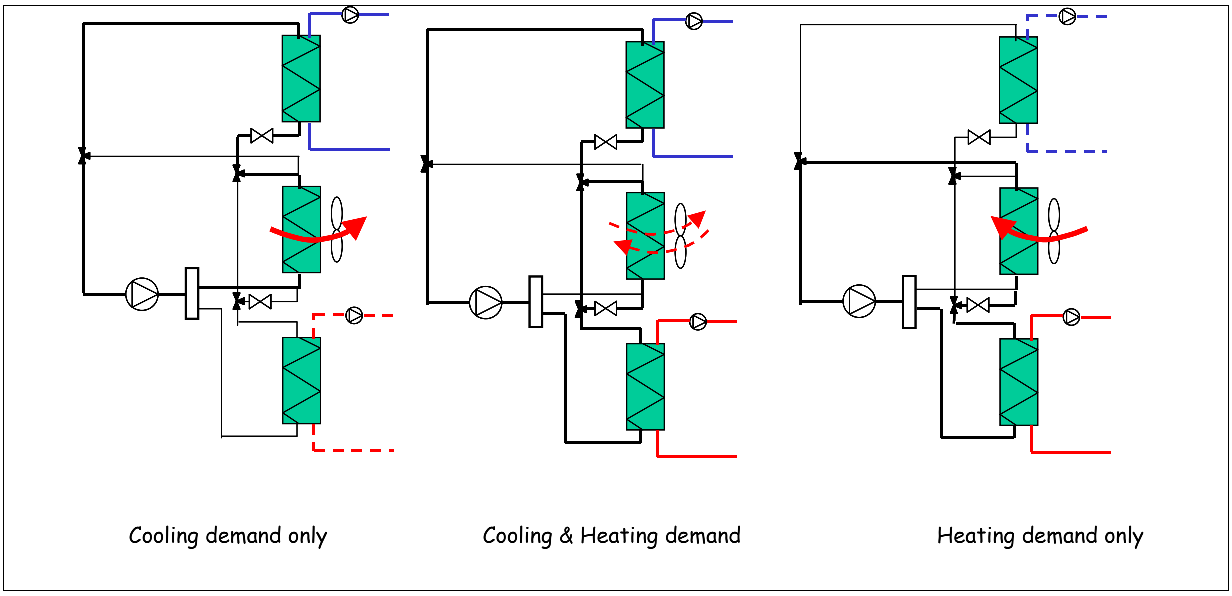


Figure 52: Water/Air/Water Heat Pump for reversibility use and heat recovery

Other heat sources/sinks can be considered such as the ground (Figure 53), water source (Figure 54), lakes, rivers…. The availability of this kind of heat source/sink is limited. A storage system can be here also installed to better take advantage of heat recovery.

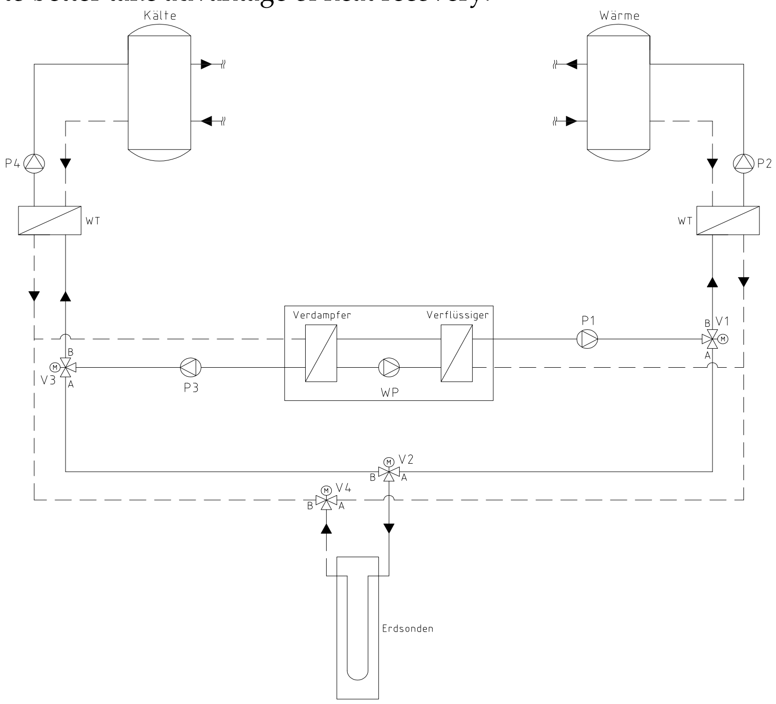


Figure 53: Heat pump with heat recovery and ground coupled for heat/cold surplus rejection

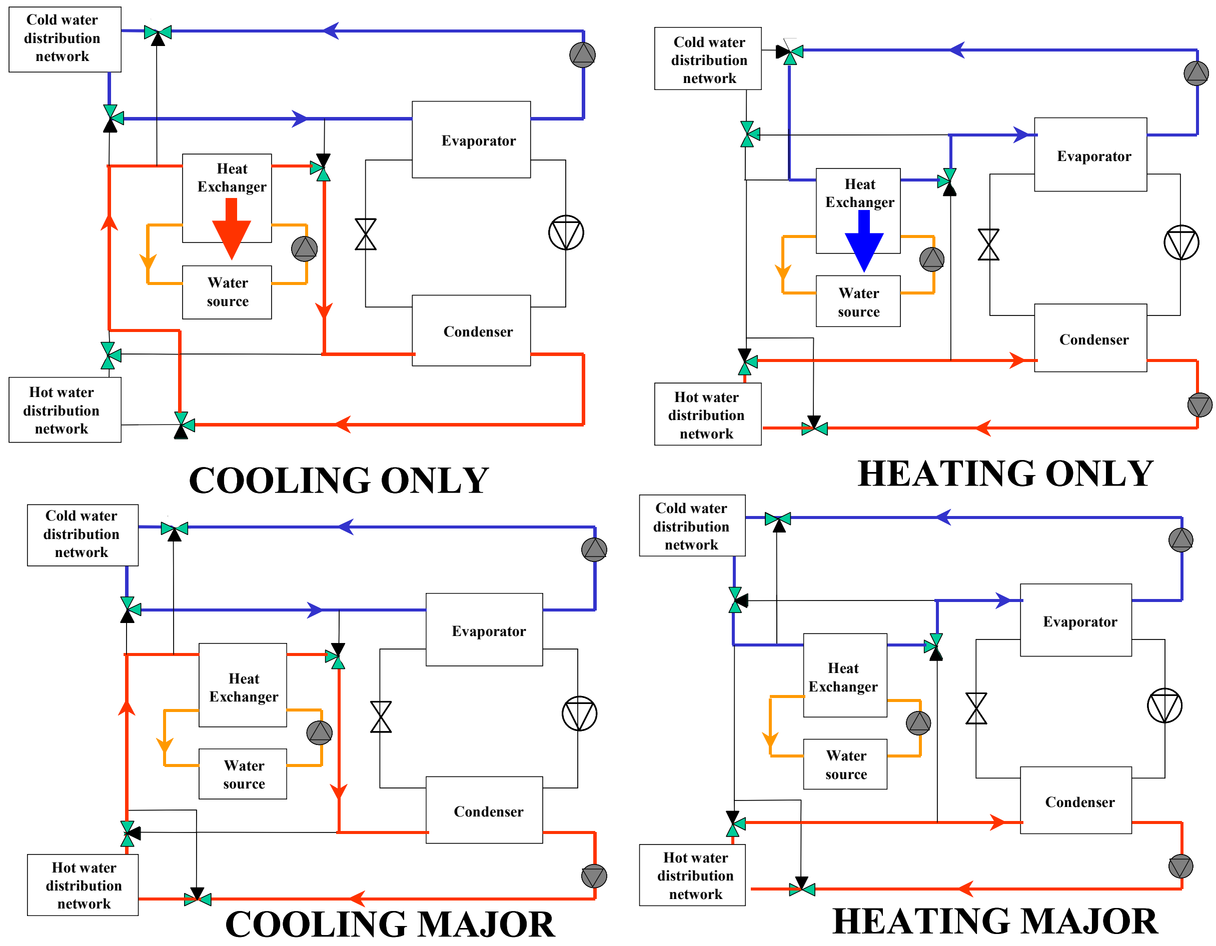


Figure 54: Heat pump with heat recovery and water source coupled for heat/cold surplus rejection

3.4 Conclusions

As a priority, one proposes to study the most common cases in **existing buildings** :

- Replacement of an air-cooled chiller by a reversible air-cooled chiller for space heating;
- Heat recovery on a water cooled chiller by adding a heat exchanger for space heating in offices and for Domestic Hot Water in Health care institutions.

Based on the analysis of heating and cooling demand results, the case studies will be limited for reversibility to :

- 4 office buildings and 2 health care institutions (type 1b office building is excluded since the heating and cooling demand results are closed to type 2);
- 2 parameters (ventilation rate, internal loads); Solar factor and building orientation are excluded since they don't have a strong impact; only one set point 21/24°C will be studied;
- 5 climates.

For recovery, the potential of recovery appears to be lower except in few cases for space heating. Only few extreme cases will be studied in order to confirm the results in heating and cooling demand:

- Office types 1a and hospital (water-cooled chiller are supposed to be used only in large buildings);
- Climates : Paris and Lisbon;
- Sensibility of ventilation rate, internal loads and set points only on one building type.
- Only the system with FCU and Double flux will be studied;

In the case of heat recovery for Domestic Hot Water, the potential is higher. Few cases will be studied :

- Hospital and rest home;
- Climates: Paris, Turin, Athens; (Results for Munich are closed to Paris and those for Lisbon are closed to Athens);
- Only the system with FCU and Double flux will be studied;

The summary of the simulation cases to be achieved is presented in the following table :

Table 18: Simulation cases

Reference cases			Reversibility & Heat Recovery		
description	Building types	Countries	description	Building types	Countries
Air-cooled chiller + boiler Fan Coil Unit + **double flux ventilation**	All offices except 1b All health care institutions	All Europe	Reversible Air-cooled chiller + boiler Fan Coil Unit + **double flux ventilation**	All offices except 1b All health care institutions	All Europe
Air-cooled chiller + boiler Fan Coil Unit + **single flux ventilation**	Office type 2&3 Rest home	All Europe	Reversible Air-cooled chiller + boiler Fan Coil Unit + **single flux ventilation**	Office type 3 Rest home	All Europe
Air-cooled chiller + boiler Air Handling Unit of **Variable Air Volume** type	Offices type 1	All Europe	Reversible Air-cooled chiller + boiler Air Handling Unit of **Variable Air Volume** type	Offices type 1 Large hospital	All Europe
Air-cooled chiller + boiler Air Handling Unit of **Constant Air Volume** type	Office type 1a Large hospital	All Europe	Reversible Air-cooled chiller + boiler Air Handling Unit of **Constant Air Volume** type	Office type 1a Large hospital	All Europe
Water-cooled chiller + boiler Fan Coil Unit + double flux ventilation	Offices type 1a Hospital Rest Home	Few climates	**Heat recovery on Water-cooled chiller With and without thermal storage** + boiler Fan Coil Unit + double flux ventilation	Offices type 1a and hospital (space heating) Hospital and Rest home (DHW)	Few climates

In a next step, other systems will be studied :

- VRF;
- Multi split systems;
- Dual source chiller for heat recovery;
- Heat pump coupled with ground as excess energy storage medium;
- Heat pump coupled with water source as excess energy storage medium;
- Heat pump coupled with air as excess energy storage medium;
- …

Heat pump coupled with ground or water source present a high efficiency but they are marginal today and probably more adapted to new buildings.
VRF systems are in development on the market in small office buildings (+28% in France in 2006).

In terms of heat/cold distribution, radiant panels (ceiling/floor) will be also studied.

4 Assumptions

4.1 Fan Coil Units

The Fan Coil Units are sized in offices and conference rooms based on a standard unit from a manufacturer's catalogue with the following characteristics:

Table 19: Cooling coil characteristics in FCU

Fan coil Unit (four pipes)	maximum speed	medium speed
Air Flow rate	310 m³/h	215 m³/h
Cold water flow rate	0.092 kg/s	0.066 kg/s
Hot water flow rate	0.036 kg/s	0.028 kg/s
Heating capacity*	3020 W	2380 W
Sensible cooling capacity**	1360 W	970 W
Total cooling capacity**	1920 W	1380 W
Fan power	48 W	36 W

*water temperature regime 90 – 70°C / air conditions 19°C and 38%
**water temperature regime 7 – 12°C / air conditions 27°C and 50%

The FCU are sized based on a heating water temperature regime of 90°C/70°C and a cooling water temperature regime of 7/12°C. The number of Fan coil units in each thermal zone is calculated based on maximum required between heating mode and cooling mode. For cooling mode, the number of FCU is determined as the maximum sensible cooling demand required divided by sensible cooling capacity (at medium speed). No over-sizing coefficient is taken into account. For heating mode, the number of FCU is calculated as the maximum heating demand divided by the heating capacity (at medium speed). The FCU in office rooms are essentially sized on cooling mode, whereas the most privileged mode for FCU sizing in conference rooms is the heating mode. No FCU are considered in toilets and circulations.
In heat pumping mode, the cooling coil is assumed to operate in heating mode. The water flow rate is equal to those in cooling mode. The water inlet temperature is to 45°C. The water side heat exchange coefficient is multiplied by 1.42 to account for the change of thermo-physical properties of water from 7/12°C to 45/40°C. The heating coil is used only:

- When the chiller operates in cooling mode but space heating is requested in some zones ;
- When the heating capacity of the heat pump is insufficient, a boiler delivers heat at higher temperature level and the heating coils are used for boosting.

The fan consumption ratio is taken to 1 W per 40 W of cooling power which corresponds to an intermediate model compared to fan coil units- four pipes of other 5 manufacturers (see figure below).

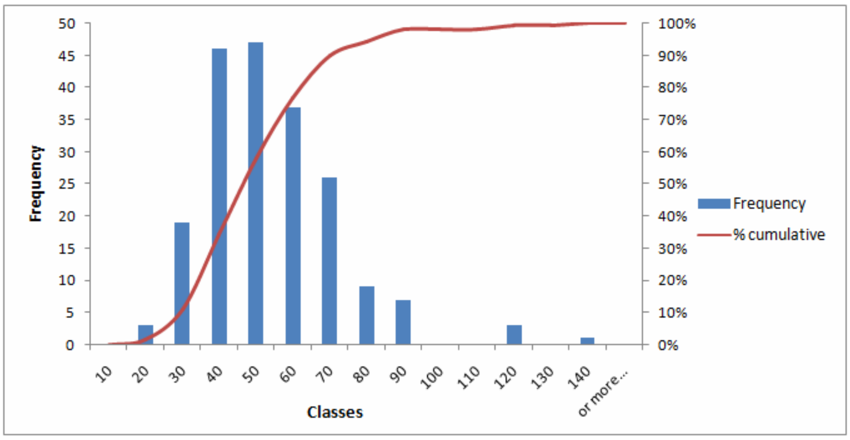


Figure 55: Histogram of cooling power to fan power ratios based on manufacturers' data

The water temperature regime in the FCU is taken to 7/12°C in cooling mode and 70/90°C in heating mode.
FCU are assumed to operate at medium fan speed (215 m^3/h). The FCU are stopped during inoccupancy.

4.2 *Air Handling Units*

The **fan consumption** is calculated based on pressure drop in AHU according to values from [EN 13779, 2007] and fan overall efficiency of 0.7 (see Annex 5 of the ANNEXES to this report [STA, 2008]).

Table 20: Assumptions on fan consumption in AHU

		ΔP AHU[19]	ΔP duct	SFP $W/(m^3/s)$	SFP $W/(m^3/h)$	Classification according to [EN 13779,2007]
CAV	Supply	930	600	2186	0.61	SFP 5
	Exhaust	250	600	1214	0.34	SFP 3
VAV	Supply	1080	700	2543	0.71	SFP 5
	Exhaust	400	700	1571	0.44	SFP 4
DF	Supply	700	600	1857	0.52	SFP 4
	Exhaust	400	600	1429	0.40	SFP 4
SF	Exhaust	150	600	1071	0.30	SFP 3
DF [20]	Supply	500	400	1286	0.36	SFP 4
	Exhaust	250	400	929	0.26	SFP 3

For VAV, the fan characteristic is described based on a controllable pitch blades and one speed motor fan as follows:

$$\frac{P}{P_n} = 0.875\left(\frac{\dot{V}}{\dot{V}_n}\right)^2 - 0.425\left(\frac{\dot{V}}{\dot{V}_n}\right) + 0.55$$

In the case of the double flux AHU, the **heat recovery system is an air to air plate heat exchanger**. The efficiency is determined based on EUROVENT data (5 manufacturers). The **median of heat recovery efficiency is about 52%** (see figure below). This value will be selected for double flux ventilation system and VAV system.

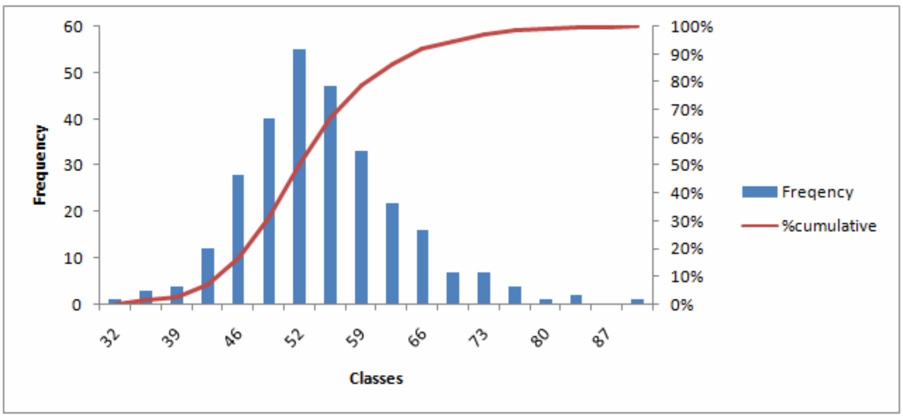


Figure 56: Distribution of plate heat exchanger efficiencies based on manufacturers' data

[19] See values in Annex
[20] DF without Heat recovery (only air filtration) for conference rooms in case of single flux ventilation building

The **cooling and the heating coils** are selected based on manufacturer's catalogue:

Table 21: Heating and Cooling coil characteristics in AHU

Cooling coil		Heating coil	
Water temperatures	7/12°C	Water temperatures	90/70°C
Air inlet conditions	27°C – 50%	Air inlet conditions	0°C
cooling capacity	5 W/(m³/h air)	Heating capacity	14 W/(m³/h air)

In heat pumping mode, the cooling coil is assumed to operate in heating mode as for FCU. The water flow rate is equal to those in cooling mode. The water inlet temperature is to 45°C. The water side heat exchange coefficient is multiplied by 1.42 to account for the change of thermo-physical properties of water from 7/12°C to 45/40°C. The heating coil is used only:

- When the chiller operates in cooling mode but space heating is requested in some zones ;
- When the heating capacity of the heat pump is insufficient, a boiler delivers heat at higher temperature level and the heating coil is used for boosting.

VAV system

The VAV unit operates without recycled air (100% outside air) since VAV systems with recycled air can not satisfy hygienic air flow rate in all zones when high differences of demand exist between zones. The VAV is equipped with an air-to-air heat recovery system.
In inoccupation, the VAV is stopped.

CAV system
The CAV operates in mono zone. One assumes one CAV unit per zone.
In heating mode, in occupancy, the outside air flow rate is minimal. In inoccupancy, all the supply air is recycled air;
In neutral mode, the outside air flow rate is still at minimum up to a middle indoor temperature between the heating and cooling set points temperatures. When the middle temperature is reached, the recycled air rate decreases gradually if the outside is below the middle temperature. If the outside temperature is higher than the middle temperature, the outside air flow rate is still minimal (Figure below).
In cooling mode, if the outside temperature is lower than the cooling set point temperature, no recycled air is used otherwise recycled air rate is maximum. In inoccupancy, all the supply air is recycled air.

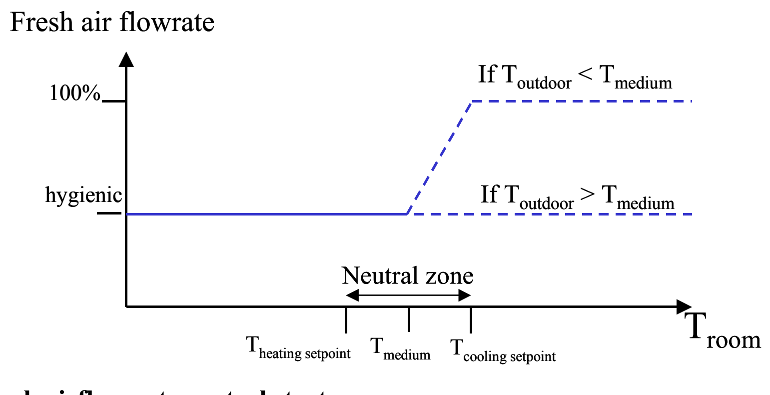


Figure 57: AHU fresh airflow rate control strategy

The ratio of heat losses which are not recovered by building is fixed to 100% for AHU.

Double Flux and Single Flux
Fans are stopped during inoccupancy.

An **air leakage of 7 %** is taken into account in CAV ,VAV and DF units from AHU inlet to supply outlets.
Losses due to control of VAV and CAV systems are difficult to taken into account and an arbitrary factor of **5%** will be applied on the final result of VAV and CAV consumption.
Heat losses in the AHU are neglected.

4.3 Water Pipe network

The pressure drop in the emitters and in the pipes is based essentially on values from [RT 2005]. The longest pipe is defined arbitrary with a limited to 500m. In case of the cooling and heating coils are in the AHU, it is assumed here that the heat and cold productions are in the vicinity of the air Handling unit (50 m). In the case of VAV systems, the heating coils are distributed locally which imply a long pipe network.

Table 22 : Values for pressure drop in pipe network

Building type		Office 1a	Office 1b	Office 1c	Office 2	Office 3	Health care 1	Health care 2
Pressure drop in FCU (kPa) [RT 2005]		20	20	20	20	20	20	20
Pressure drop in AHU (kPa) [RT 2005]		35	35	35	35	35	35	35
Longest pipe length in FCU & AHU (local emitters) (m)		400	400	400	300	100	500	300
Longest pipe length in AHU (central emitter) (m)		50	50	50			50	
Pressure drop (kPa) With 150 Pa/m [RT 2005]	FCU	**80**	**80**	**80**	**65**	**35**	**95**	**65**
	AHU (local emitter)	**95**	**95**	**95**				
	AHU (central emitter)	**42.5**	**42.5**	**42.5**			**42.5**	

Pump efficiency = 40%

The overall pipe network length for heat loss calculation is defined arbitrary in the following table:

Table 23: Overall Pipe network length for heat loss calculation

	Building type	Office 1a	Office 1b	Office 1c	Office 2	Office 3	Health care 1	Health care 2
FCU	Pipe length (m)	1800	1500	1500	600	100	2200	300
CAV	Pipe length (m)	150					50	
VAV (cooling coil)	Pipe length (m)	50	50	50				
VAV (local heating coils)	Pipe length (m)	1500	1300	1300				

The specific U value of the pipes is taken to 0.28 W/m/K (heat loss per length).
The ratio of heat losses which are not recovered by building is fixed to 20% for FCU.
The ratio of heat losses which are not recovered by building is fixed to 100% for AHU in case of central emitters and 20% in case of local emitters.

4.4 Cold Production

The air-cooled chiller has been sized considering an EER of 2.5, which is an average value for air cooled chillers (Figure 58) based on Eurovent statistics [EEC, 2003]. The cooling capacity of the units is calculated as the sum of the maximum cooling demands in each cooled zones with an over-sizing coefficient of 15% to account for dehumidification loads.

In heat pumping mode, the COP is about 10% higher than the EER (Figure 59) and the heating power is also about 10% (Figure 61) higher than the cooling power. Notice that the EER and the COP are provided at standard rating conditions of 7/12°C for water and 35°C at outdoor and 40/45°C for water and 7°C at outdoor, respectively. No part load performance improvement or reduction is taken into account in the following.
The following values have been chosen :
COP= 1.1 * EER = 2.75
Pheating = 1.1 * Pcooling

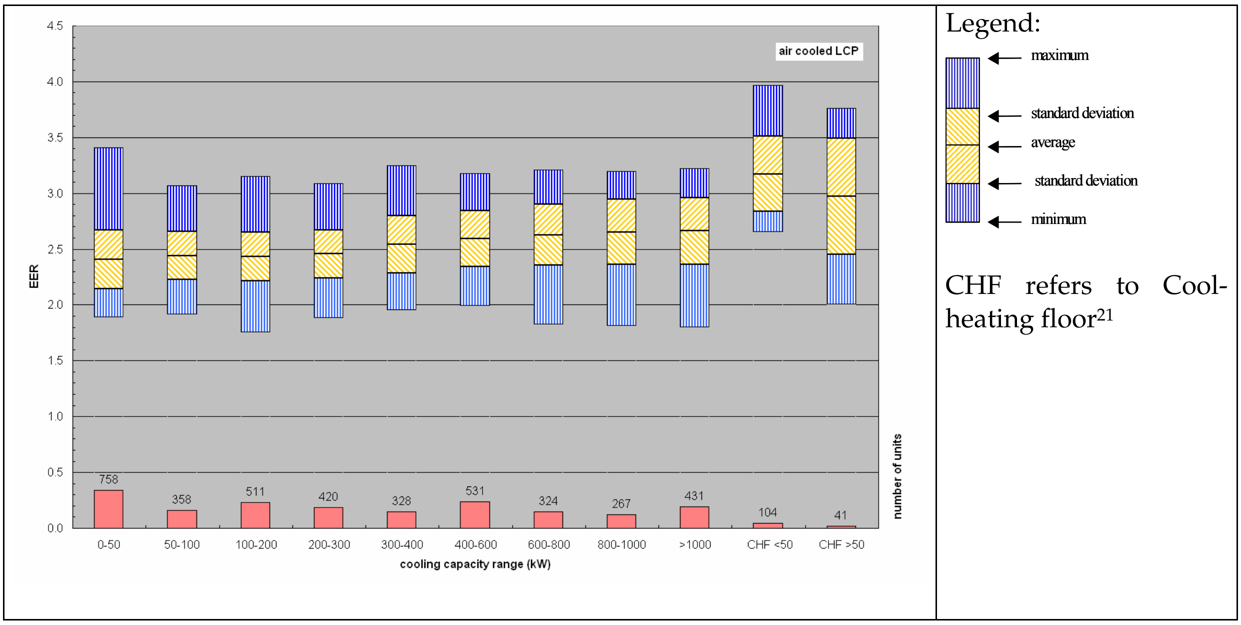


Figure 58: Average EER of air cooled LCP by range of cooling capacity based on Eurovent data 2005

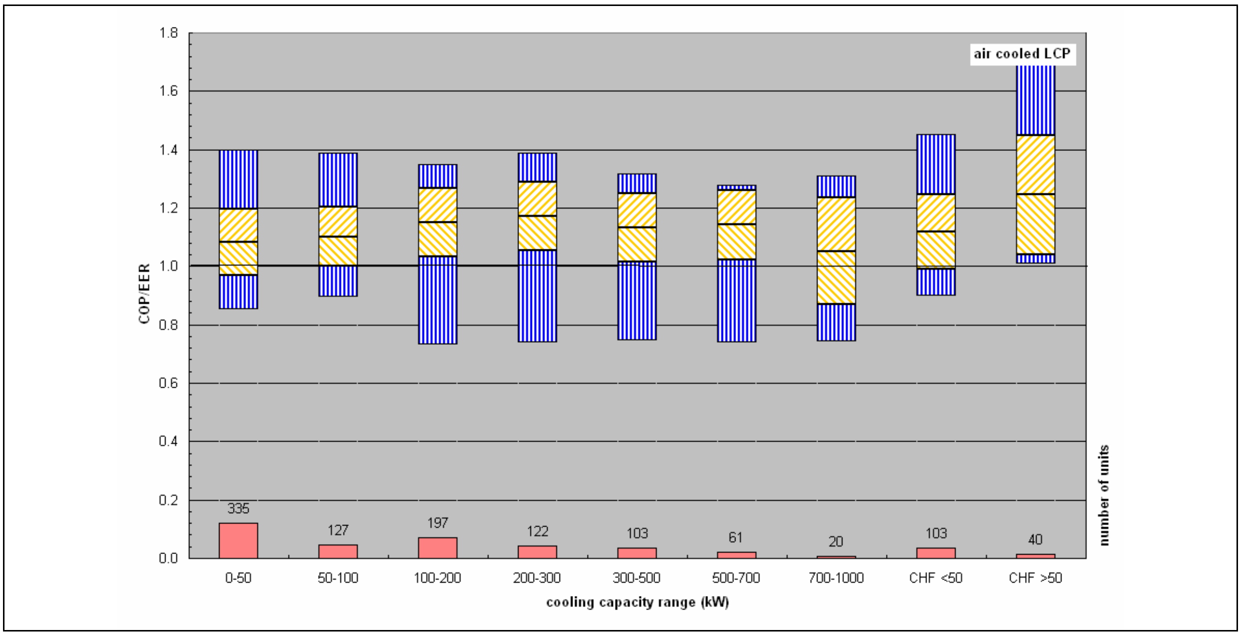


Figure 59: Ratio between COP and EER of reversible air cooled chillers in function of cooling capacity range

In heat pumping mode, the water temperature leaving the heat pump is adjusted in function of outside temperature as described in the following figure:

[21] The standard rating conditions in case of chillers on CHF are 23/18 °C for water and 35 °C for air which are more favourable conditions to obtain higher EER.

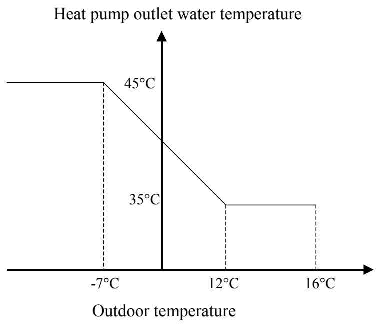


Figure 60 : Control of water distribution temperature in case of Heat pump

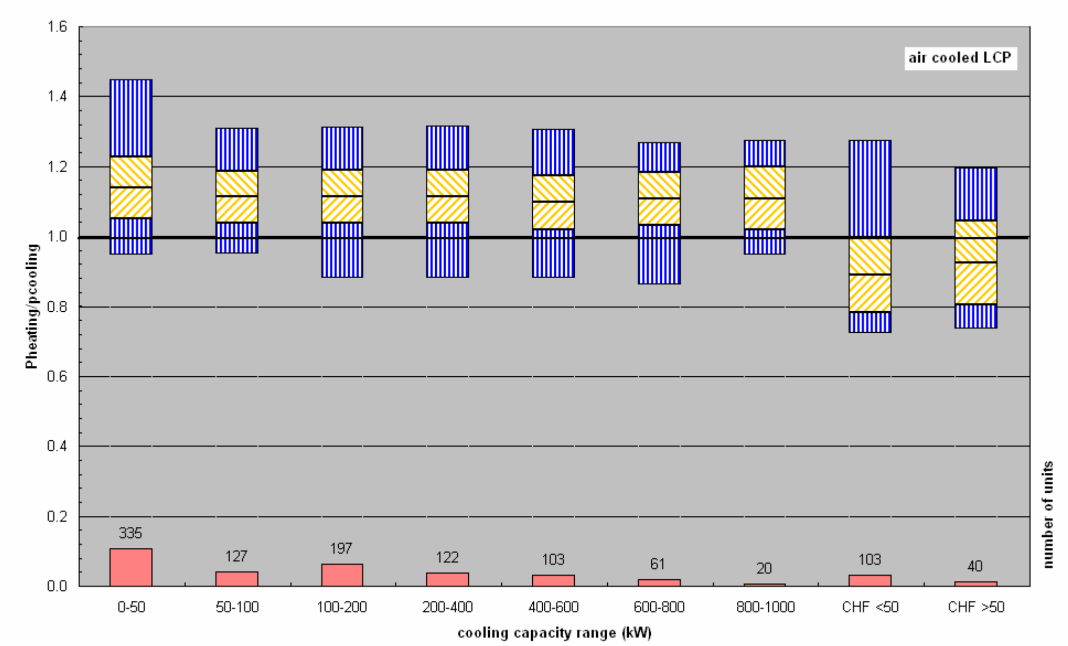


Figure 61: Ratio between heating and cooling capacities of reversible air cooled chillers in function of cooling capacity range

For heat recovery, the chosen chiller is a water-cooled one. An EER of 4.15 is selected (Figure 62).

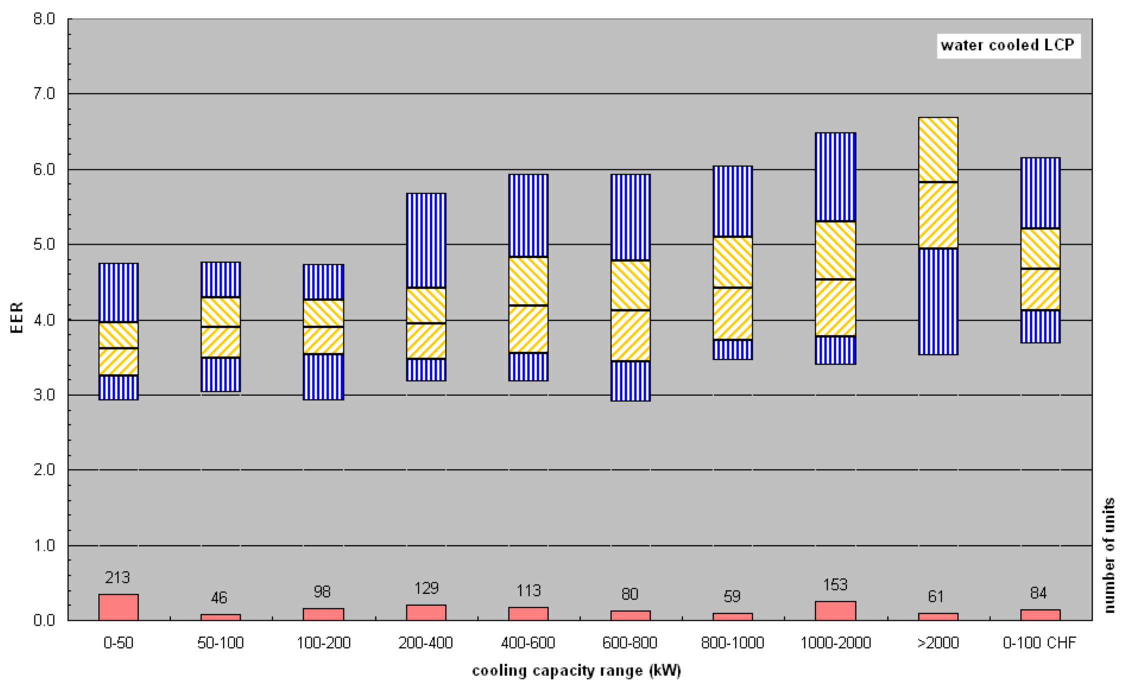


Figure 62: **Average EER of water cooled LCP by range of cooling capacity based on Eurovent data 2005**

4.5 Heat production

The boiler has been sized using the standard boiler values from the French Thermal Regulation 2000 (RT 2000).
The efficiency at 100% load is equal to η=89.2 (water temperature 70°C)
The efficiency at 30% load is equal to η=88.2 (water temperature 50°C)
The pilot-burner power is of 120 W and the recovery efficiency on the back-burner power is of 70%.
The losses at 0% load are equal to 1275 W if rating heat capacity is higher than 400 kW. Otherwise, the heat losses at 0% load, is defined as Q (kW) = Pn/100*(1.75-0.55*log (Pn)) with Pn the rating heat capacity in kW.

The maximum heating capacity is calculated as the sum of maximum heating demand of each zone. No oversizing coefficient is taken into account since the maximum heating demands in each zone are not simultaneous.

The water temperature leaving the boiler is adjusted in function of outside temperature as described in the following figure:

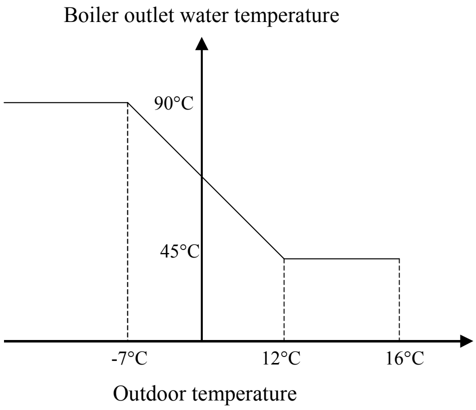


Figure 63 : Control of water distribution temperature in case of boiler

4.6 Other HVAC components

<u>Cooling Tower and Heat exchanger performances for heat recovery</u> (contribution from L. Sarrade – CEA-INES)
The cooling tower is of counter-flow type with centrifugal fan. It is sized according to the following data:
Inlet water temperature: 35 °C
Outlet water temperature: 30°C
Wet-bulb temperature according to the climate :

Paris : 20°C
Torino : 21°C
Athens : 21°C
Munich : 19°C
Lisbon : 21°C

The water flow rate is assessed based on the maximum heat to be rejected on condenser and a water temperature difference of 5°C at condenser.

<u>Humidifier</u>
The humidification is considered only is performed in the operation room of the hospital. It is achieved by electric steam humidification. The vapour temperature is fixed to 110°C.

V ANALYSIS OF BUILDING ENERGY CONSUMPTION

1 Introduction

The study of heating and cooling demands (chapter III) has allowed us to define the most interesting cases. Moreover, the most typical HVAC have been selected (Chapter IV). From this work, few representative simulation cases in energy consumption have been defined in order to assess the primary energy savings and CO_2 emissions avoided which can be obtained by opting for a reversible heat pump solution or energy recovery on condenser.

The scope of the building energy study is here limited to the cases in the following Table.

Table 24: Building energy consumption simulation cases

REVERSIBILITY POTENTIAL (five climates, two ventilation rate and two internal load levels)	Single Flux ventilation and Fan Coil Unit	Double Flux ventilation and Fan Coil Unit	Variable Air Volume	Constant Air Volume	HEAT RECOVERY POTENTIAL (two climates, two ventilation rate and two internal load levels for offices, five climates for health care)	Double Flux ventilation and Fan Coil Unit
Office building Type 1a		✓	✓	✓		✓
Office building Type 1c		✓	✓			
Office building Type 2	✓	✓				
Office building Type 3	✓	✓				
Health care type 1		✓		✓		✓
Health care type 2	✓	✓				✓

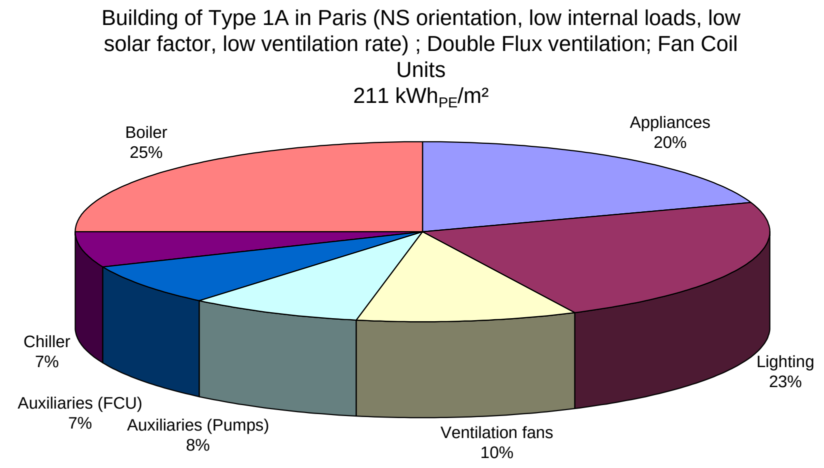
Notice that few simulations are carried out for the study of the heat recovery potential. Indeed, the analysis of building demand showed that the heat recovery potential is low in most of the cases, except for heat recovery for Domestic Hot Water.

2 Results of building energy consumption simulations

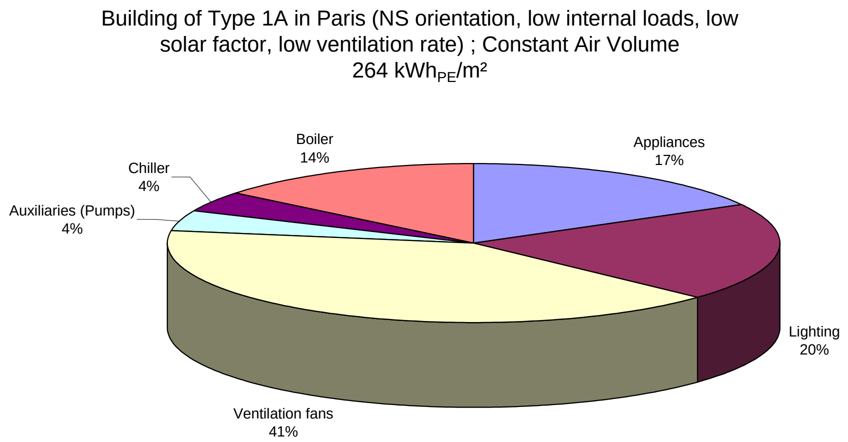
2.1 Office Buildings results

b) Energy consumption

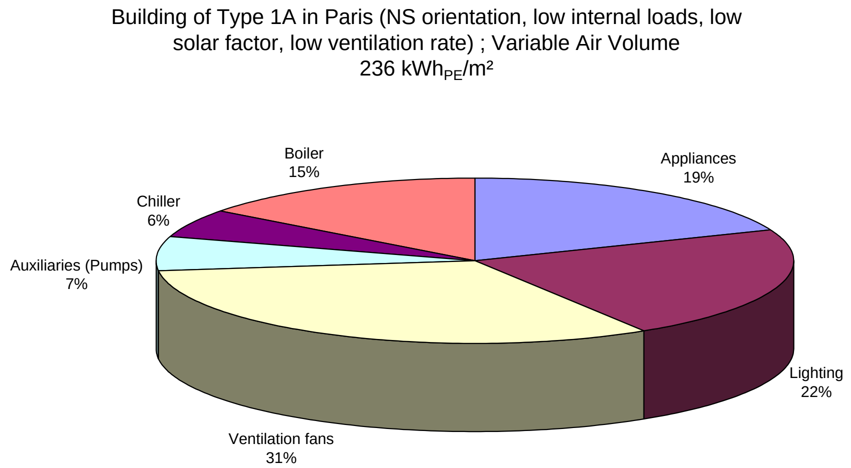
The energy consumption in office building has been calculated by using CONSOCLIM. Figure 64, Figure 65 and Figure 66 show the share of energy consumption in different types of buildings with different types of HVAC systems by end use. The percentages are based on the values in kWh/m² end-use. On the other hand, the total energy consumption is expressed in Primary Energy (a PE factor of 2.58 is used for electricity and 1.0 for fuel according to France standard).
Notice that the auxiliaries are split between the consumption of the hot and cold water circulation pumps and the consumption of Fan Coil Units if any, while ventilation fans are counted separately.
The electricity consumption due to office equipment and lighting represents between 25% and 44% of the total energy consumption in the shown cases. Considering all the simulated cases, the office equipment and lighting can represent up to 66.5% of total end-use consumption.
The chiller consumption represents in the shown cases less than 7% of the total energy consumption while the auxiliary (pumps and fans) stands for at least 13% of the total energy. In all the simulation cases, the chiller can reach 20% of the total consumption while the auxiliaries can represent up to 57% of the total consumption.



TYPE 1A + DF + FCU : 211 kWh_{PE}/m^2

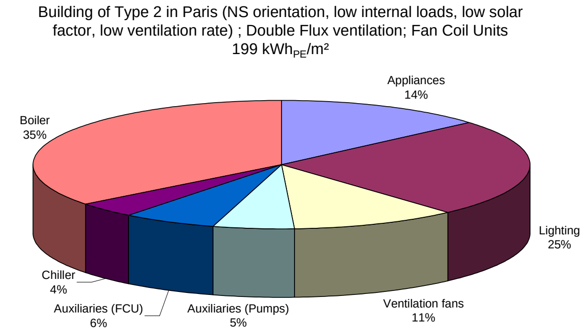


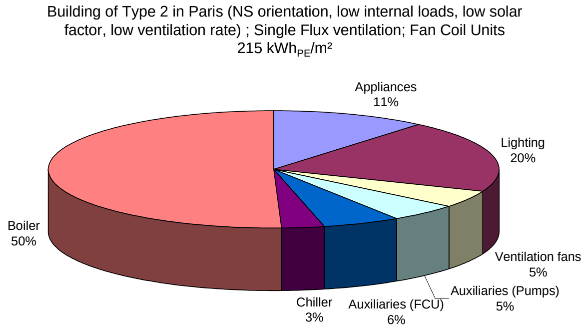
TYPE 1A + CAV : 264 kWh_{PE}/m^2



TYPE 1A + VAV : 236 kWh_{PE}/m^2

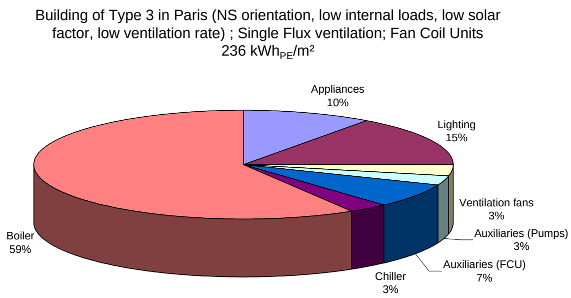
Figure 64: Share of yearly final energy consumption for different HVAC systems in Type 1A office building

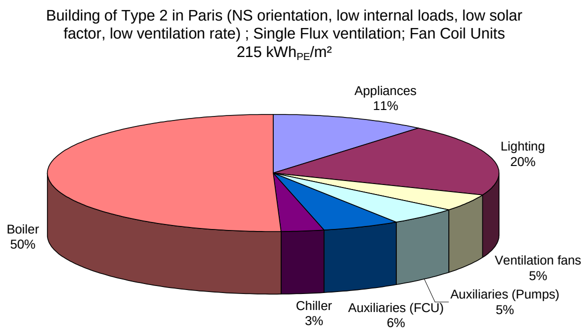




TYPE 2 + DF + FCU : 199 kWh_{PE}/m^2 TYPE 2 + SF + FCU : 215 kWh_{PE}/m^2

Figure 65: Share of yearly final energy consumption for different HVAC systems in Type 2 office building





TYPE 3 + SF + FCU : 236 kWh_{PE}/m^2 TYPE 2 + SF + FCU : 215 kWh_{PE}/m^2

Figure 66: Comparison of share of yearly final energy consumption between Type 3 and Type 2 office building with SF+FCU

c) Heat Pumping

The previous paragraph presented some results in the case of reference heat and cold production (chiller +boiler). The aim is now to compare the reference cases to a solution taking advantage of heat pumping. The simulations on heat pumping mode have been carried out according to the assumptions presented in chapter IV.

The results of type 1C office building show the climatic impact on annual energy consumption on Figure 67. The type 1C office building is taken as in example; similar trends are obtained in the other office building types. On this Figure, the consumption of the reference case and the heat pumping case are presented. The boiler consumption is strongly decreased when the chiller is used in reversible mode (heat pumping). However, the electric consumption of the chiller operating in reversible mode (called heat pump on the graph) compared to the chiller (only for cooling) is strongly increased in temperate climates (high heating demand) and slightly increased in hot European climates (low heating demand). Auxiliaries includes here circulation pumps and fan coil units.
Figure 68 shows that whatever the HVAC, the potential of reversibility of the chiller is high. On the example of type 1A office building, the boiler consumption is strongly reduced. However, it seems difficult to remove the auxiliary heating system (boiler or other system) since it is still useful during the coldest days (until now, most of the air-to-water heat pumps cannot operate below –10°C) and during the simultaneous heat and cooling load days.

Figure 69 shows a comparison between a single flux and a double flux ventilation in a type 3 office building. The heating consumption in the office building with a single flux ventilation is higher and so the fuel savings are more important by using the chiller in heat pump mode.
Figure 70 shows that whatever the internal loads and ventilation rates, the reversibility potential in Type1a building in Paris is high.

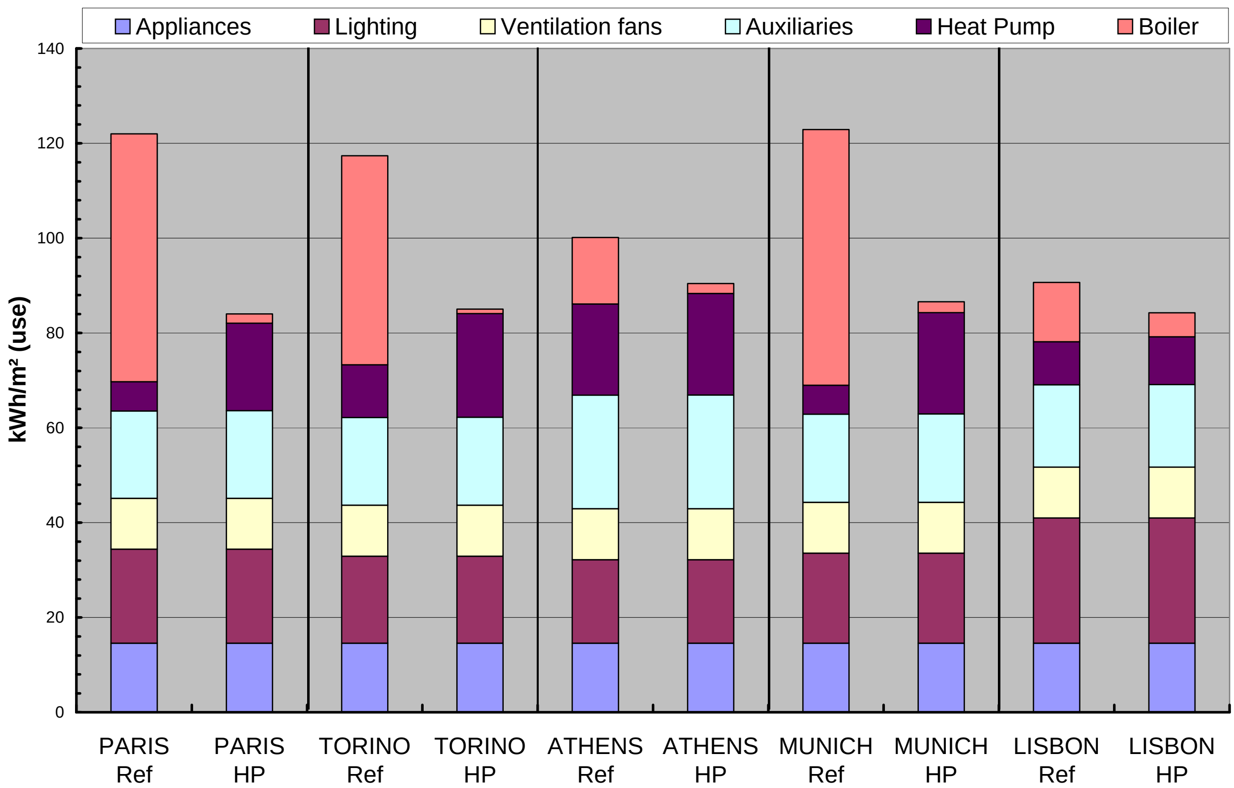


Figure 67: Impact of <u>climatic zone</u> on annual energy consumption reduction due to Heat Pump use (example: building of TYPE 1C, with a North/south orientation, low internal loads, low solar factor and low ventilation rate; equipped with a Double flux ventilation and Fan coil units)

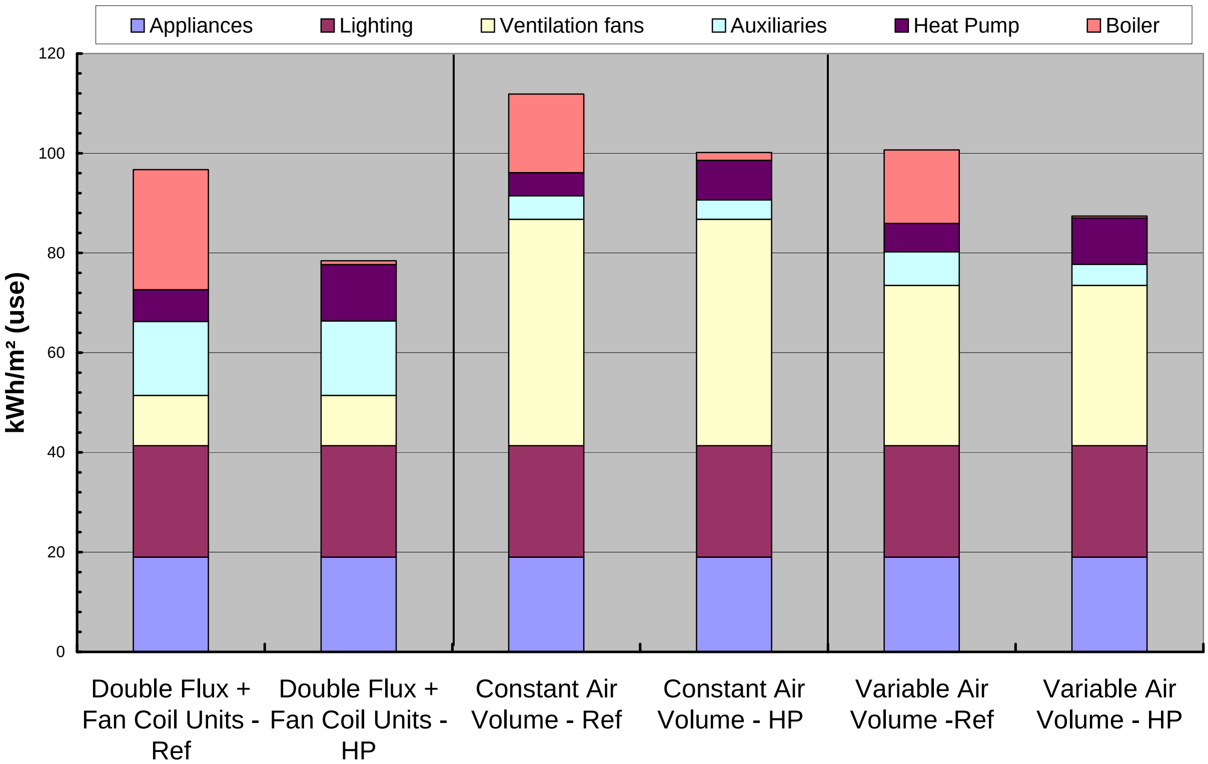


Figure 68: Impact of HVAC system on annual energy consumption reduction due to heat pump use (example: building of TYPE 1A in Paris, with a North/south orientation, low internal loads, low solar factor and low ventilation rate)

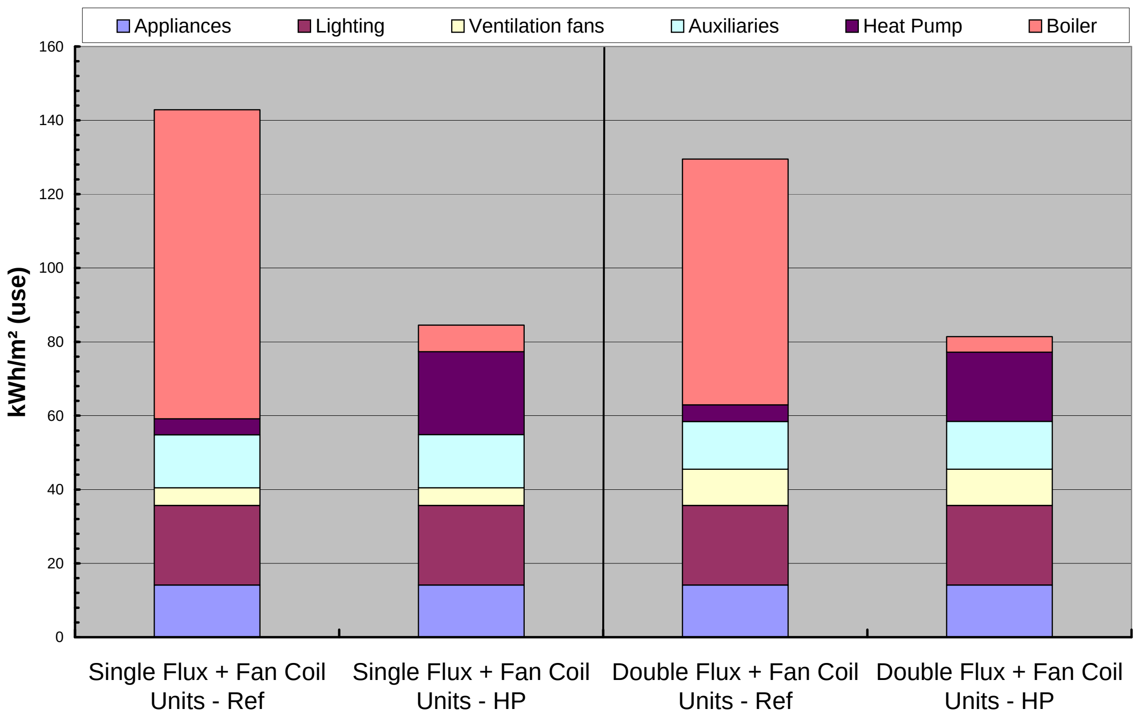


Figure 69: Impact of HVAC system on annual energy consumption reduction due to Heat Pump use (example: building of TYPE 3 in Paris, with a North/south orientation, low internal loads, low solar factor and low ventilation rate)

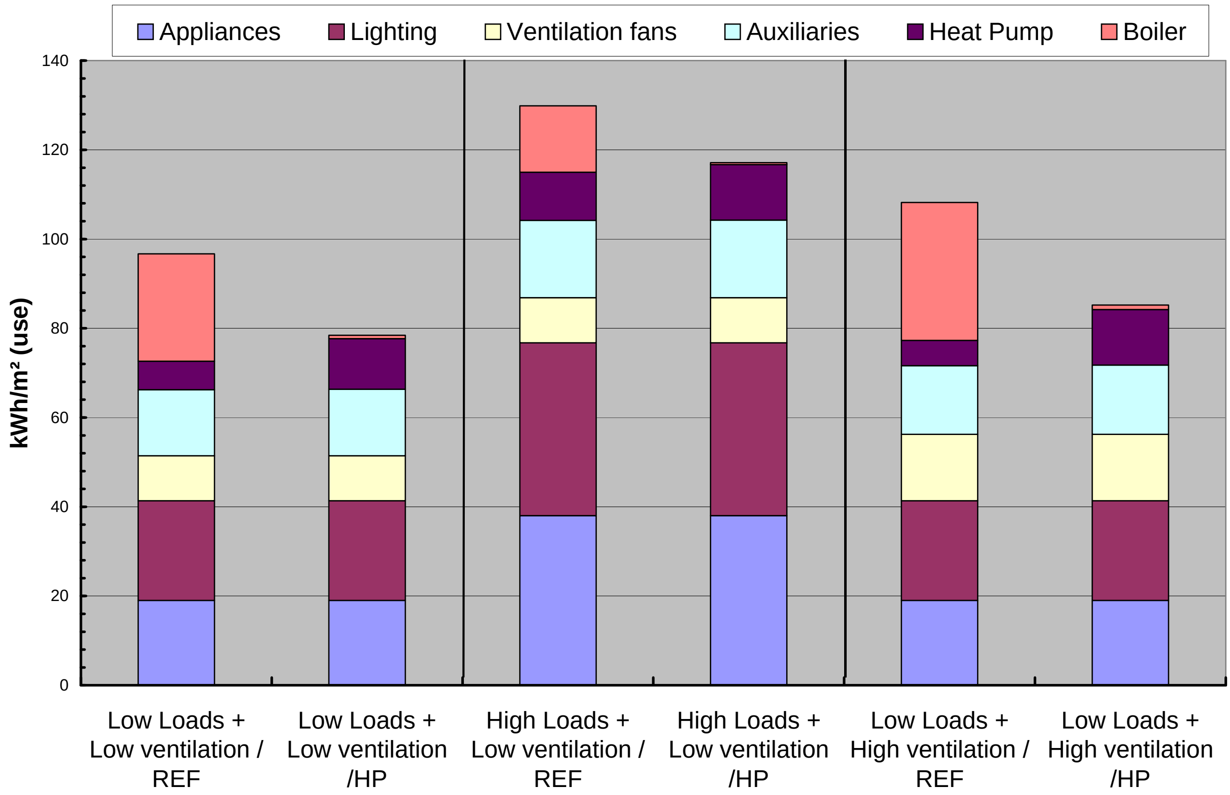


Figure 70: Impact of <u>internal loads and ventilation rates</u> on annual energy consumption (example: building of TYPE 1A, in Paris, with low solar factor; equipped with Double flux ventilation and Fan coil units)

d) Heat recovery on condenser (Based on the works of Laurent Sarrade – CEA-INES)

The heat recovery is considered in office buildings only for space heating. The study has been limited to one office building type, (Type 1a) with two levels of ventilation rate, internal loads, and set points and in two climates (Lisbon and Paris). The heat recovery is studied only in the case of water-cooled chiller as described on the figure below. An heat exchanger is implemented in the cooling water loop before the open circuit cooling tower. The heat exchanger is connected on its other side to the space heating loop before the boiler.

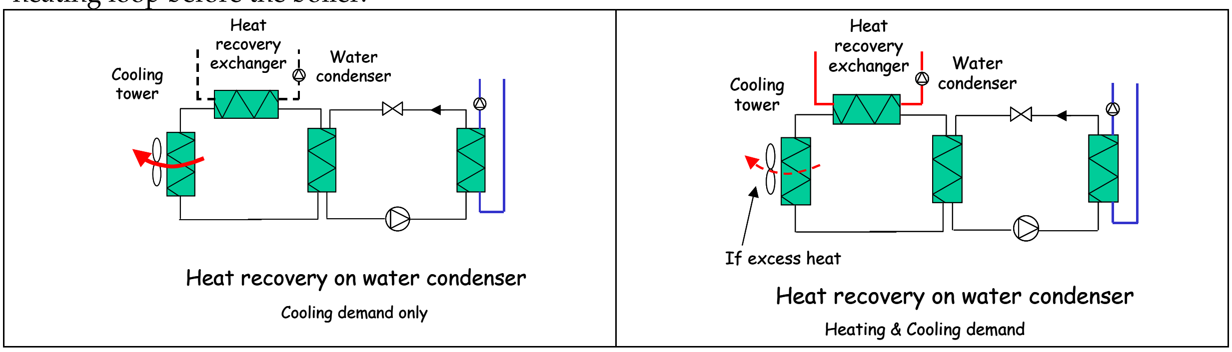
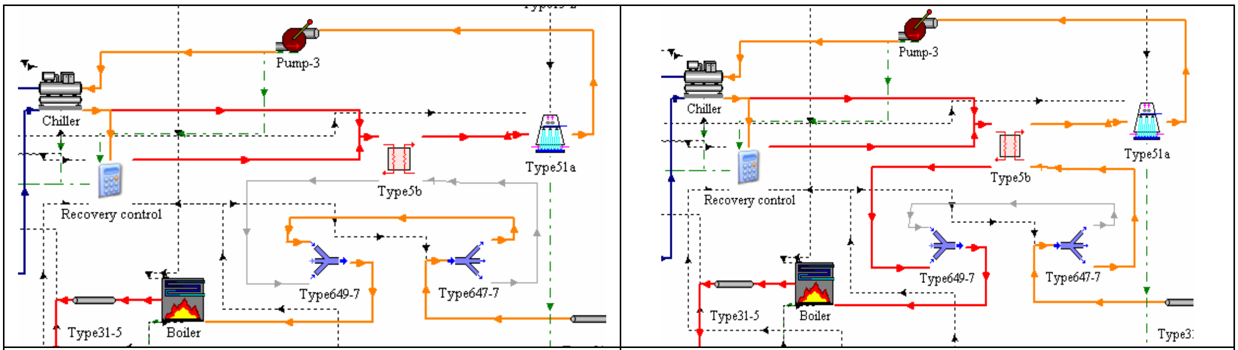


Figure 71 : Heat recovery on a water-cooled chiller

For heat recovery, the simulations have been carried out with TRNSYS 16. The water temperature regime in the Fan coil units is supposed to be equal to 40-35°C. During the simultaneous heating and cooling demand periods, the water temperature exiting from the condenser is assumed to be at 42°C in order to heat the space heating water to 40°C. Since the water temperature in the condenser is increased, the performance of the chiller is degraded. If there is no simultaneity of heating and cooling demand, the heat exchanger is by-passed, and the space heating water goes directly to the boiler. This mode of operation is modelled in TRNSYS as described in the following figure.



No simultaneous heating and cooling demands	Simultaneity of heating and cooling demands

Figure 72 : TRNSYS representation of operation modes of heat recovery

The main results are compiled in the Table 25. Primary energy savings have been calculated based on 2.58 electricity conversion factor and 1.0 for fuel. The potential of fuel savings range between 0% and 28% whereas the chiller electricity consumption increases between 0.1% and 3.2%. Finally, the space heating primary energy savings would not overpass 2% (by using a 2.58 primary energy factor of electricity and 1.0 for fuel). These results are lower than the one defined from heating and cooling demands. In fact, the energy calculations are based on a double flux ventilation with heat recovery whereas the analysis in demand was based on a single flux ventilation. Indeed, the number of simultaneity hours is divided by 3 to 20 when the single flux ventilation is replaced by double flux ventilation. Moreover, the double flux ventilation decreases the heat demand and thus the potential of heat savings.

Table 25: Results of heat recovery for the simulation cases in office buildings

						without heat recovery		with heat recovery		Primary energy savings (kWh/m²)	Primary energy savings (%)
n°	Building type	set points	Internal loads	Ventilation Rate	Climate	chiller consumption	boiler consumption (kWh/m²)	chiller consumption	boiler consumption		
1	TYPE 1A	20/25	High	High	LISBON	11.3	4.5	11.5	4.1	0.0	0.3
2	TYPE 1A	20/25	High	High	PARIS	6.2	24.9	6.3	24.7	-0.1	0.0
3	TYPE 1A	20/25	High	Low	LISBON	13.7	3.0	14.0	2.3	0.0	0.0
4	TYPE 1A	20/25	High	Low	PARIS	8.2	16.5	8.4	16.1	0.0	0.3
5	TYPE 1A	20/25	Low	High	LISBON	6.5	8.6	6.6	8.6	0.0	0.0
6	TYPE 1A	20/25	Low	High	PARIS	3.8	39.9	3.8	39.9	0.0	0.0
7	TYPE 1A	20/25	Low	Low	LISBON	8.0	5.2	8.0	5.1	0.0	0.0
8	TYPE 1A	20/25	Low	Low	PARIS	4.9	26.5	5.0	26.5	-0.1	0.0
9	TYPE 1A	21/24	High	High	LISBON	12.8	6.2	13.2	5.2	0.0	0.0
10	TYPE 1A	21/24	High	High	PARIS	7.3	30.2	7.5	29.4	0.2	0.5
11	TYPE 1A	21/24	High	Low	LISBON	15.0	4.3	15.5	3.1	0.0	0.0
12	TYPE 1A	21/24	High	Low	PARIS	9.4	20.4	9.7	19.2	0.4	2.0
13	TYPE 1A	21/24	Low	High	LISBON	7.7	12.0	7.8	11.6	0.1	0.8
14	TYPE 1A	21/24	Low	High	PARIS	4.6	46.8	4.7	46.7	-0.1	0.0
15	TYPE 1A	21/24	Low	Low	LISBON	8.9	7.6	9.1	7.1	0.0	0.3
16	TYPE 1A	21/24	Low	Low	PARIS	5.9	31.7	6.0	31.4	-0.1	0.0

The following figure shows that the fuel savings are low when the heating demand is the lowest (low internal loads, 20/25°C set points). The result confirms that the potential is higher in Lisbon than in Paris. The degradation of chiller performance in heat recovery mode penalizes the savings of primary energy. In terms of CO_2 emissions, the heat recovery could be interesting in countries with low CO_2 content of electricity.

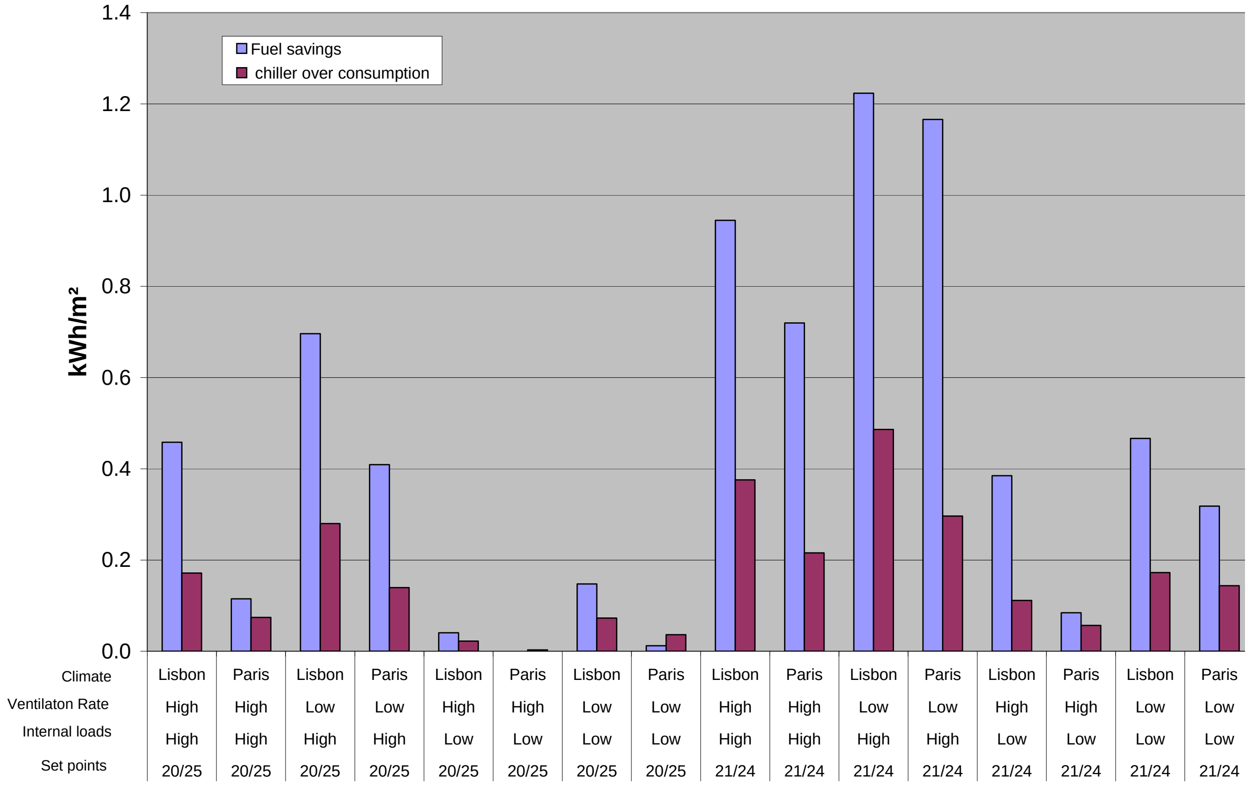


Figure 73: Impact of climate, ventilation rate, internal loads and set points on the fuel savings

As a conclusion, the heat recovery on chiller condenser for space heating in office buildings does not appear as a promising solution. However, in particular buildings with very different internal loads between zones (such as the presence of a data centre [SCH, 1999]), the heat recovery on chiller condenser deserves to be studied.

2.2 *Health care institutions*

a) Annual energy consumption

The energy consumption in health care institutions has been calculated by using CONSOCLIM. Figure 74 shows the share of energy consumption in different types of buildings with different types of HVAC systems by end use. The percentages are based on the values in kWh/m² end-use. On the other end, the total energy consumption is expressed in Primary Energy (a PE factor of 2.58 is used for electricity and 1.0 for fuel according to France standard).
The electricity consumption due to office equipment and lighting represents between 19% and 27% of the total end-use consumption in the shown cases. Considering all the simulated cases, the office equipment and lighting can represent up to 39% of consumption. Lighting in all cases represents still at least 12% of the energy consumption.
The chiller consumption represents between 0% and 14% of the total energy consumption while the auxiliaries (pumps and fans) stand for at least 11% of the consumed energy and at maximum 42%.
The steam humidification in large hospital represents from 10% to 19% of the total energy consumption.
The domestic hot water is produced by an independent boiler with a seasonal efficiency on gross calorific value taken to 0.57 [BAE, 2003]. It represents from 6% to 24% of the end-use consumption.

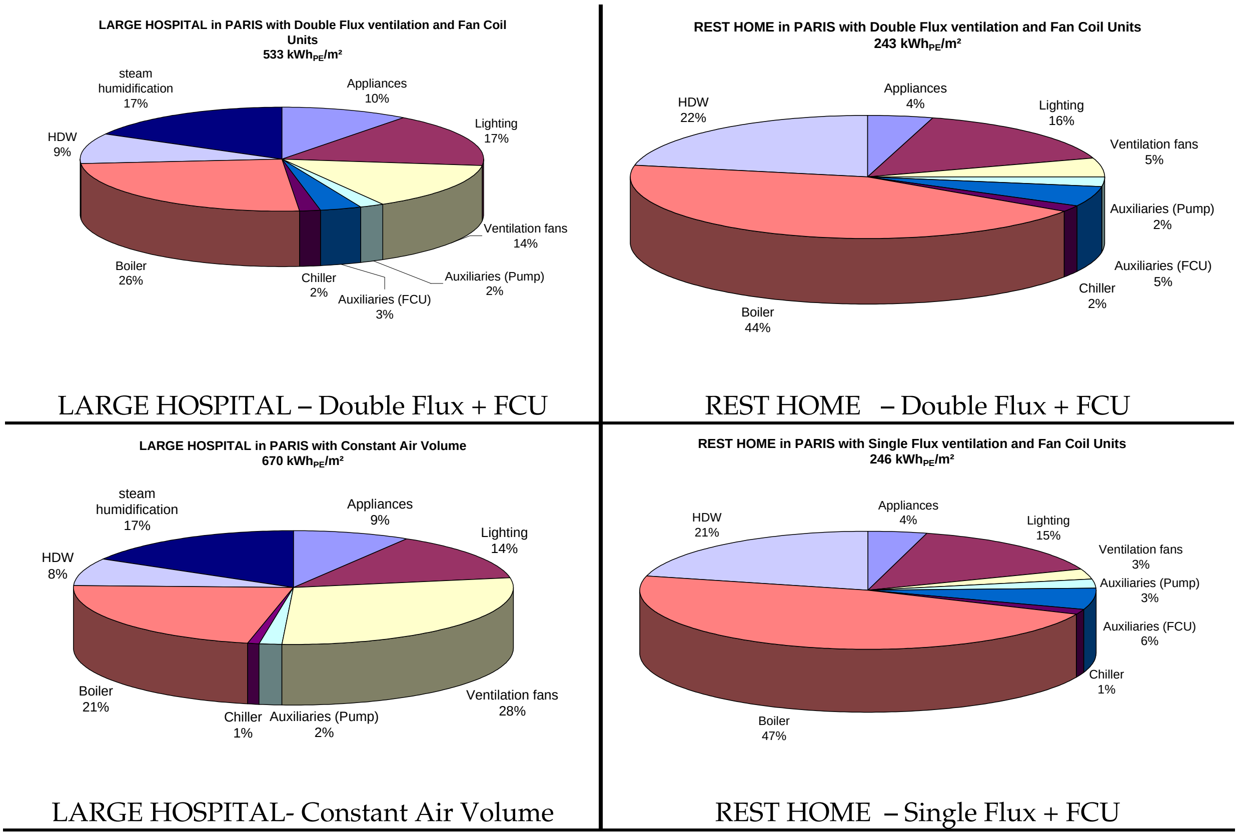


Figure 74: Share of yearly energy consumption in different health care institutions and HVAC systems

b) Heat pumping

The use of a reversible chiller for heating is then compared to the reference cases presented in the previous paragraph. The simulations on heat pumping mode have been carried out according to the assumptions presented in chapter IV. The heat pump mode is first used only for space heating. Figure 75 and Figure 76 show the energy consumption by climate in the reference case and in heat pump mode for the hospital and the rest home respectively. The boiler consumption is strongly decreased when the chiller is used in reversible mode (heat pumping). However, the electric consumption of the chiller operating in reversible mode (called heat pump on the graph) compared to the chiller (only for cooling) is strongly increased in temperate climates (high heating demand) and slightly increased in hot European climates (low heating demand).

Figure 77 shows that the HVAC system has a low impact on the potential of reversibility of the chiller. A backup boiler is still necessary whatever the health care institution type.

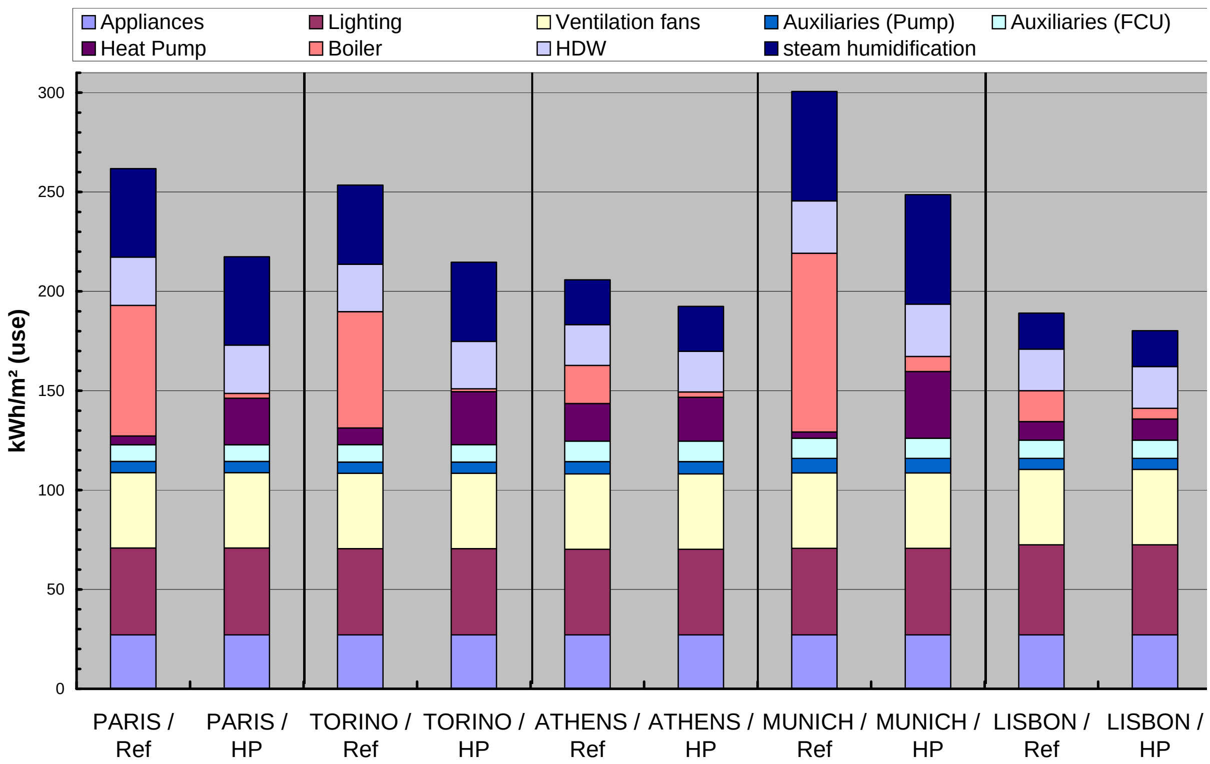


Figure 75: Impact of climatic zone on annual energy consumption reduction due to Heat Pump use (Large Hospital with double flux + FCU)

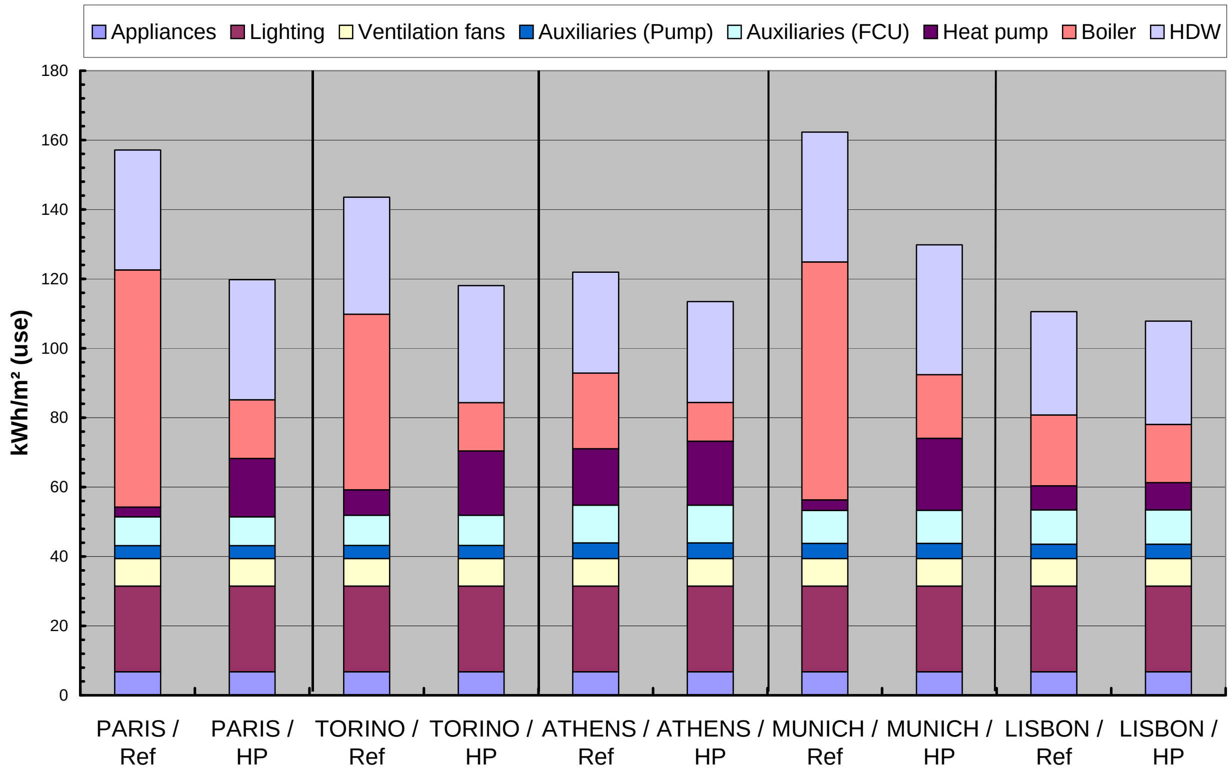


Figure 76: Impact of climatic zone on annual energy consumption reduction due to Heat Pump use (Rest Home with double flux + FCU)

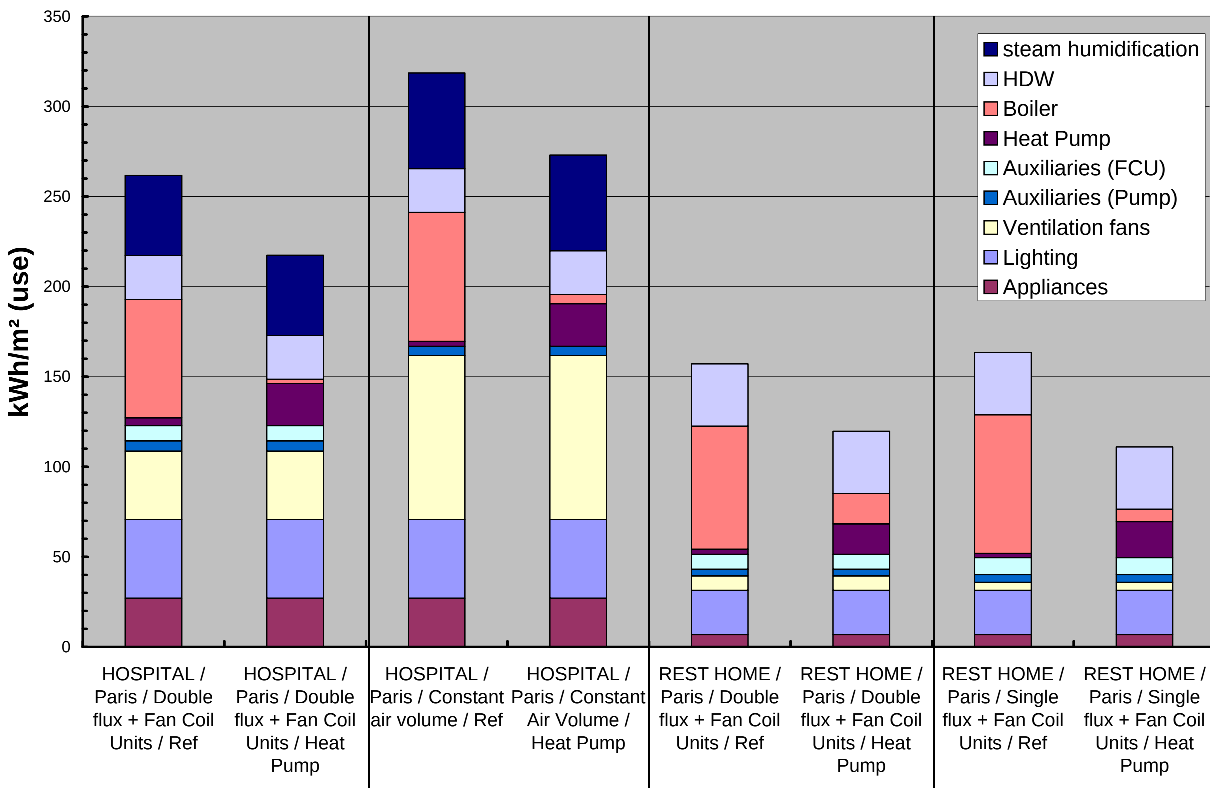


Figure 77: Impact of HVAC system on annual energy consumption reduction due to Heat Pump use (example: Hospital and rest home in Paris)

Figure 75 to Figure 77 showed results in case of heat pumping for space heating only. However, the heat pumping could also be used for domestic hot water. It has been assumed that when the chiller is in heat pump mode and DHW heating is requested, the chiller can preheat the DHW to 45°C and the boiler provides the set point temperature of 55°C.
Figure 78 shows that the use of a heat pump (sized on maximum cooling power) for Domestic Hot Water can save up to 14 kWh/m² of fuel in rest homes and 10 kWh/m² in large hospitals. The climatic zones corresponding to Paris, Torino and Munich are the most interesting whereas the potential in Lisbon and Athens is low. The annual primary energy savings can reach 4 kWh/m² in rest homes and 2 kWh/m² in large hospitals. The primary energy saving potential is lower in Athens and Lisbon climates and in large hospitals since the cooling demand is higher and the chiller is less used in heat pump mode. For example in a rest home in Paris, if there is no limit on the heating capacity of the chiller, only 5% of fuel savings in more could be achieved compared to a chiller sized on cooling demand. If the heat pump was used to heat water at 55°C, the fuel savings would be larger but the primary energy consumption would increase since the COP at 55°C decreases strongly.

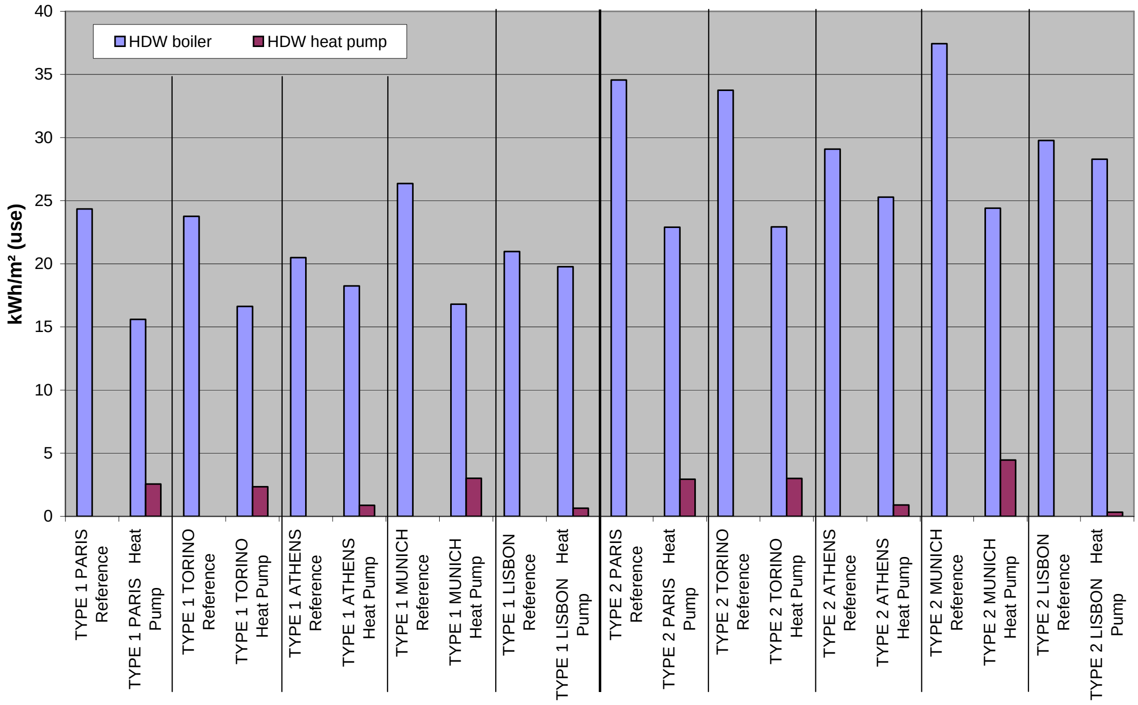


Figure 78: Energy consumption reduction for Domestic Hot Water production

c) Heat recovery on condenser (Based on works of Laurent Sarrade CEA-INES)

It has been shown in chapter III that the best potential of heat recovery in health care institutions is for Domestic Hot Water. Simulations have been carried for the hospital and the rest home in the five climates.
The heat recovery system is simulated by an heat exchanger as for heat recovery in office buildings but which preheats the domestic water when there is simultaneity between cooling demand and domestic hot water drawing. During the simultaneity period, the water temperature in the condenser is fixed to 53°C. This temperature is limited by the refrigerant characteristics, which was supposed in this study to be R407C. The boiler provides the supplement heat required to bring water up to 55°C.
The scheme simulated with TRNSYS is described on the Figure below.

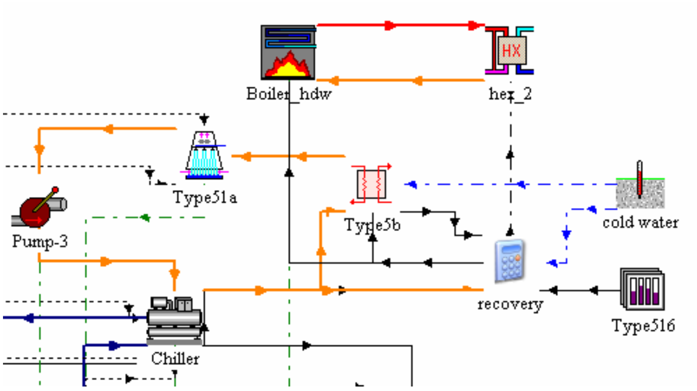


Figure 79: TRNSYS representation of domestic hot water pre heating by the chiller

The main results are compiled in the Table 26. The boiler consumption indicated in the Table corresponds only to the heating of Domestic Hot Water. Also, the percentages of primary energy savings are expressed with regard to heat consumption dedicated to Domestic Hot Water. Primary energy savings have been calculated based on 2.58 electricity conversion factor and 1.0 for fuel.

Table 26: Results of heat recovery for domestic hot water in health care institutions

			without heat recovery		with heat recovery		Primary energy savings (kWh/m²)	Primary energy savings (%)
n°	Building type	Climate	chiller consumption (kWh/m²)	boiler consumption for HDW (kWh/m²)	chiller consumption (kWh/m²)	boiler consumption for HDW (kWh/m²)		
1	Hospital	Paris	6.0	53.6	6.9	21.8	29.31	54.7
2	Hospital	Torino	8.6	52.4	9.9	18.2	30.96	59.1
3	Hospital	Athens	15.5	51.1	17.7	6.7	38.69	75.7
4	Hospital	Munich	4.9	54.8	5.6	22.5	30.27	55.2
5	Hospital	Lisbon	10.5	52.4	12.2	3.8	44.17	84.4
6	Rest Home	Paris	2.8	26.3	3.2	21.4	3.84	14.6
7	Rest Home	Torino	6.5	25.7	7.3	17.0	6.79	26.4
8	Rest Home	Athens	12.8	25.1	14.3	12.9	8.41	33.5
9	Rest Home	Munich	2.5	26.9	2.9	21.2	4.75	17.7
10	Rest Home	Lisbon	6.8	25.7	7.7	15.1	8.24	32.1

The Figure below shows that important fuel savings are possible in health care institutions. Since the cooling demand is higher in hospitals, the heat recovery potential is more important. For the same reason, the hot climates (Lisbon and Athens) presents the best potential of energy savings for hot water production.

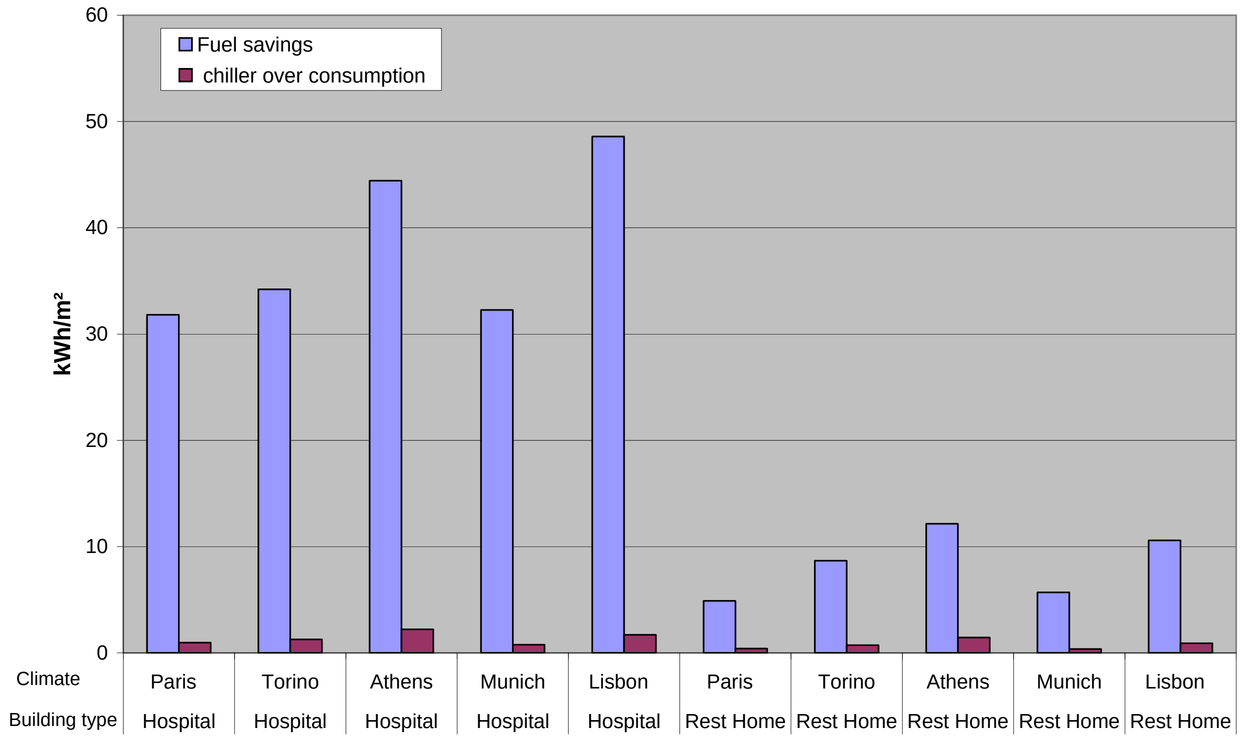


Figure 80: Impact of health care type and climate on the heat recovery potential for domestic hot water

3 Main conclusions on energy consumption reduction potentials

3.1 Main results on reversibility potentials

3.1.1 Office buildings

If we compare the reversibility potentials in the different office buildings cases in terms of HVAC systems, there are no big differences (Figure 81). By looking Figures 82, 83 and 84, one can see that the reversibility potential is a bit better in office buildings with VAV systems. Since the simultaneous heating and cooling demands are more limiting for reversibility than the peak heating power demands, the air volume systems are more adapted. Indeed, since the hours of simultaneous heating and cooling demands often occur when outside air is below setpoint inlet temperatures, the rooms which request cooling can be cooled by high ventilation rates of outside air whereas the heat pump can operate for the heating of the other rooms.

On the Figures 82, 83 and 84, one can see a good correlation by climatic zone between the reference fuel consumption in the buildings and the fuel savings which can be achieved by using a reversible heat pump.
In buildings with single flux ventilation and fan coil units, the reversibility potential (Figure 82) is medium in Lisbon climatic zones (~55% of fuel consumption can be saved), quite good in Athens climatic zones (~80% of fuel consumption can be saved) and good in Torino, Paris and Munich climatic zones (~90% of fuel consumption can be saved).
In buildings with double flux ventilation and fan coil units, the reversibility potential (Figure 83) is medium in Lisbon climatic zones (~55% of fuel consumption can be saved), quite good in Athens climatic zones (~80% of fuel consumption can be saved) and good in Torino, Paris and Munich climatic zones (~95% of fuel consumption can be saved).
In buildings with Variable Air Volume Units, the reversibility potential (Figure 84) is good in Lisbon climatic zones (~65% of fuel consumption can be saved), very good in Athens and Munich climatic zones (~90% of fuel consumption can be saved) and excellent in Torino, Paris climatic zones (almost

100% of fuel consumption can be saved). Buildings with Constant Air Volume units offer also a good potential (~70% of fuel consumption can be saved in Lisbon and Athens and ~90% in other climates).

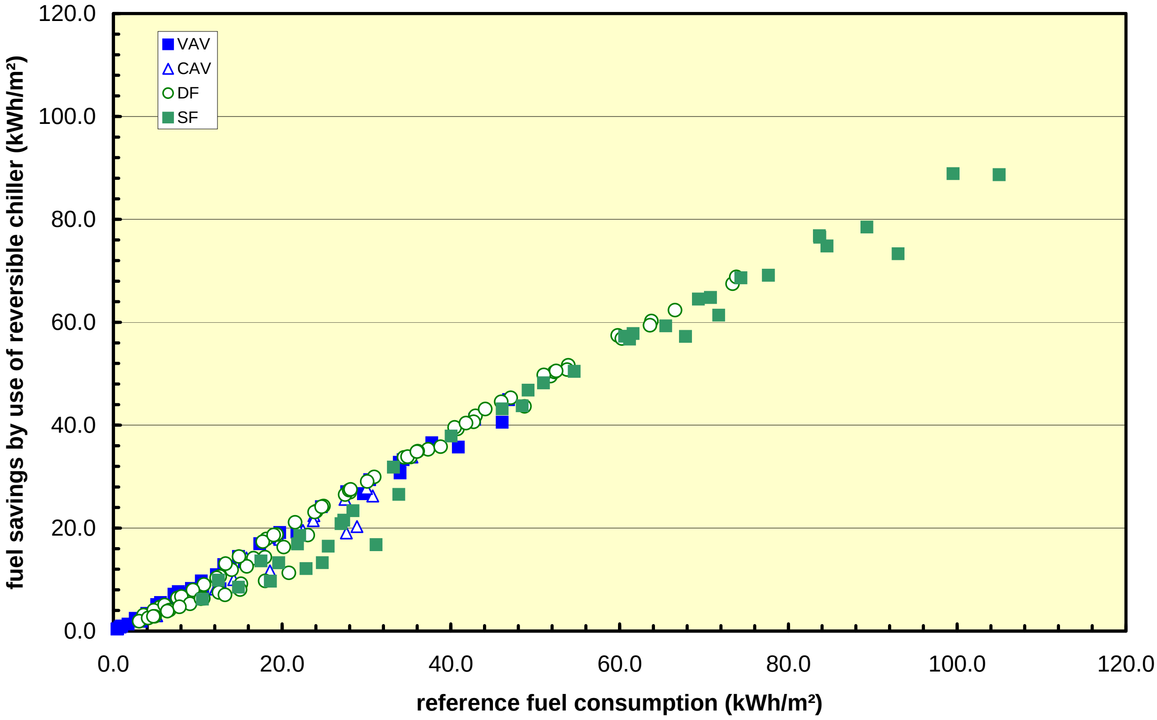


Figure 81: Comparison between different HVAC systems in terms of reversibility potential for all the simulation cases

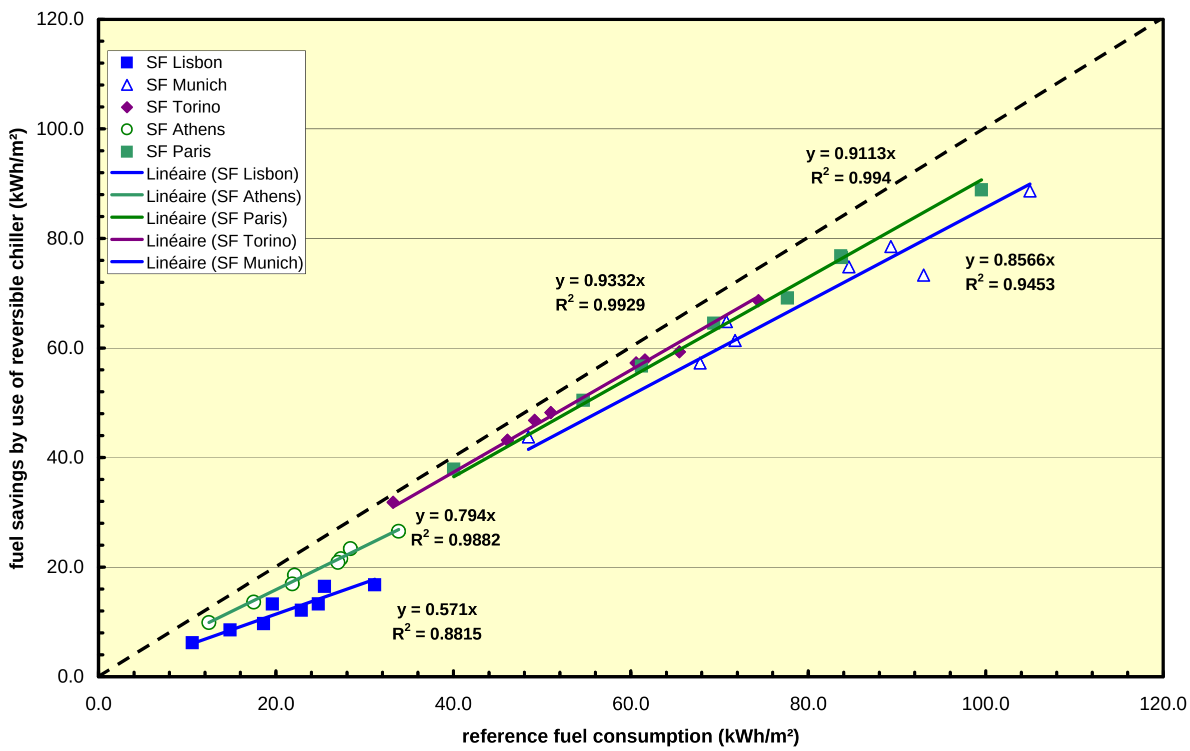


Figure 82: Fuel savings in office buildings with single flux ventilation and Fan Coil Units

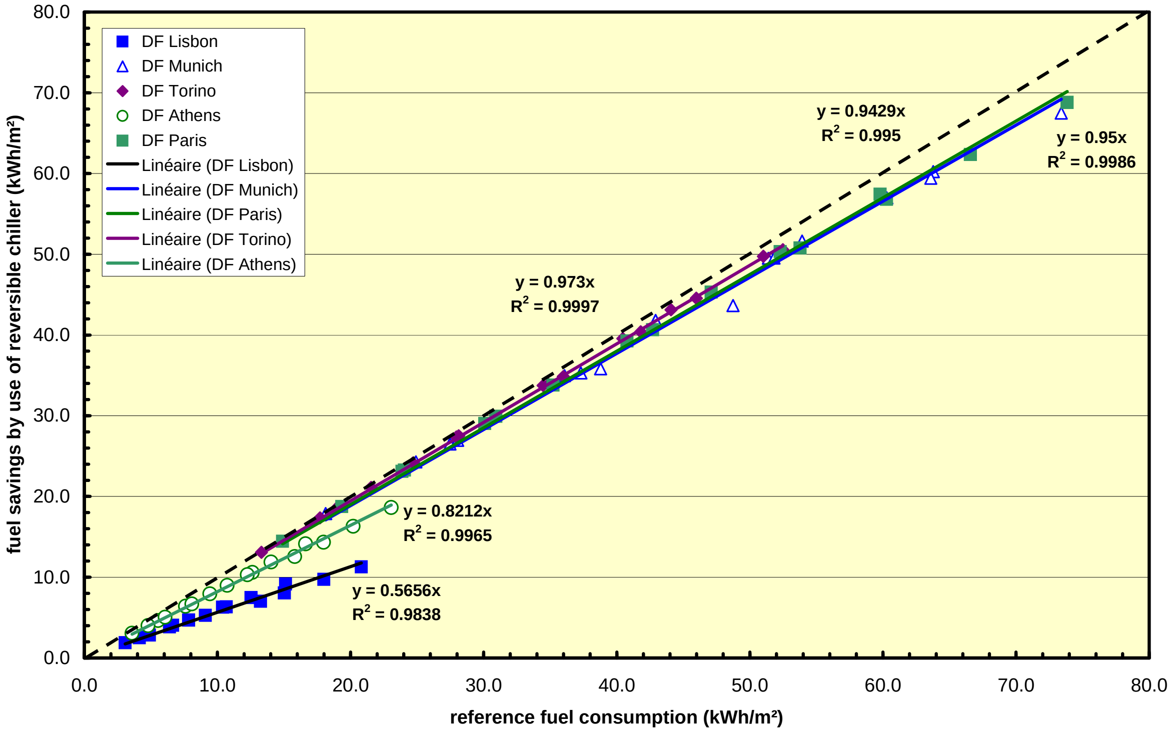


Figure 83: Fuel savings in office buildings with double flux ventilation and Fan Coil Units

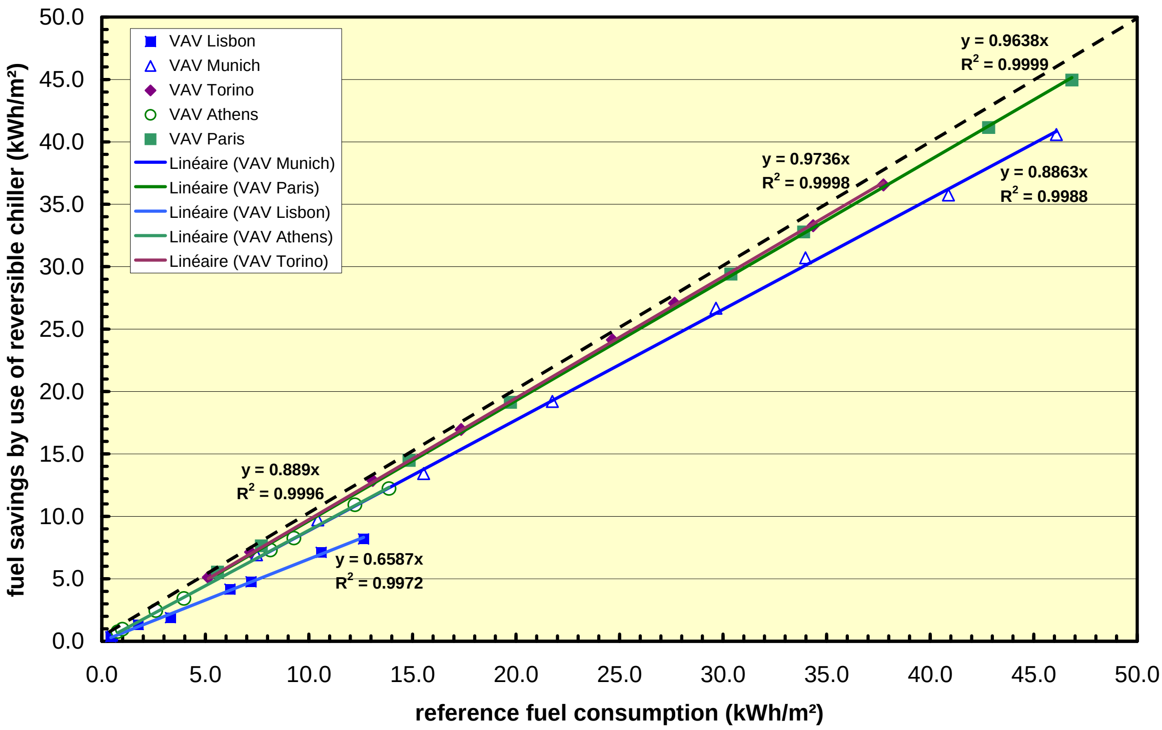


Figure 84 : Fuel savings in office buildings with Variable Air Volume system

In the Table 27, the fuel saving potentials are summarized according to HVAC systems and climate. These results are valid only for the studied building cases. They have been set for 21/24°C set point temperatures for heating and cooling respectively. The potentials can be expected even better for more distant set point temperatures between heating and cooling.

Table 27: Summary of reversibility potential according to HVAC system and climate

Climatic zone / HVAC system	PARIS	TORINO	ATHENS	MUNICH	LISBON
SF + FCU	☺☺☺	☺☺☺	☺☺	☺☺	😐
DF + FCU	☺☺☺	☺☺☺☺	☺☺	☺☺☺	😐
VAV	☺☺☺☺	☺☺☺☺	☺☺☺	☺☺☺	☺☺
CAV	☺☺☺	☺☺☺	☺☺	☺☺☺	☺☺

3.1.2 Health care institutions

The potential of fuel savings by using a reversible chiller is high in Large hospitals (Figure 85) whatever the HVAC system used (CAV or DF+FCU). Indeed, 85% to 95% of fuel consumption can be avoided in all climates except in Lisbon climatic zone where the boiler consumption can be reduced only of about 60%. The fully air-conditioned rest homes with single flux ventilation in climatic zones of Paris, Munich and Torino offer a very high potential of energy savings (>90% of fuel consumption can be avoided). The potential is a bit lower in the case of double flux ventilation, around 75%. In Lisbon climate, whatever the HVAC system, the potential is low in Rest Homes. In Athens climate, the potential is medium in Rest home, with about 50% of fuel savings for double flux ventilation and 70% for single flux ventilation.
In partially air-conditioned rest homes, the potential would be largely lower if the reversible heat pump is sized on the peak cooling demand. On the contrary, the potential would be similar to fully air-conditioned rest homes if the heat pump is sized on peak heating demand.

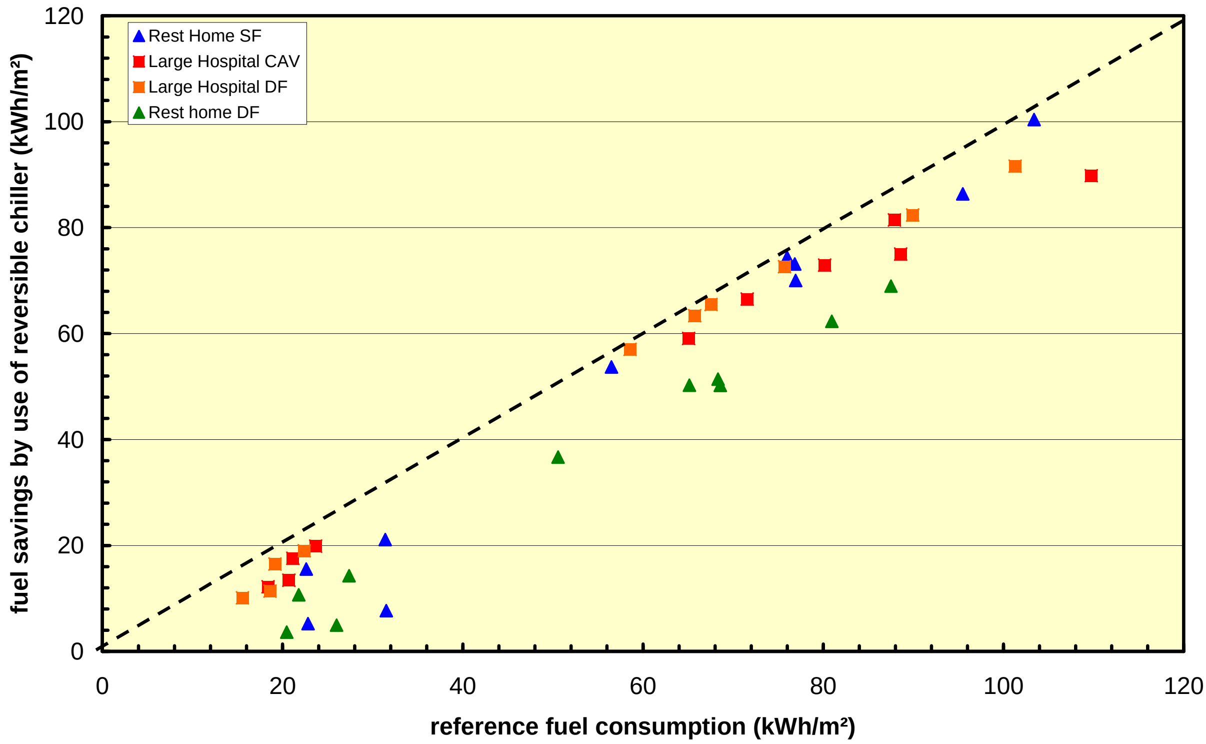


Figure 85 : Fuel savings in health care institutions by using reversible chiller

In theTable 28, the fuel saving potentials are summarized according to HVAC systems, climate and health care building type. These results are valid only for the studied building cases.

Table 28: Summary of reversibility potential according to HVAC system, climate and health care building type

Building type		Large Hospital					Rest Home				
Climatic zone		PARIS	TORINO	ATHENS	MUNICH	LISBON	PARIS	TORINO	ATHENS	MUNICH	LISBON
HVAC system	SF + FCU						☺☺☺	☺☺☺☺	☺☺	☺☺☺☺	😐
	DF + FCU	☺☺☺☺	☺☺☺☺	☺☺☺	☺☺☺	☺☺	☺☺	☺☺	😐	☺☺	😐
	CAV	☺☺☺	☺☺☺	☺☺☺	☺☺☺	☺☺					

3.2 *Main results on heat recovery potentials*

If we consider the studied office buildings, the heat recovery potential is very low. However, in data centre, the cooling demand is high and faintly dependent of weather conditions. Thus, the potential of heat recovery in winter would be quite high. This case has not been studied further in this document.

The heat recovery on chiller condenser for Domestic Hot Water preparation can save a lot of fuel, in particular in health care institutions where DHW consumptions are high. The large hospitals present the better saving potentials compared to Rest Homes. Lisbon and Athens climatic zones are the most appropriate for fuel savings since air-conditioning system operates more often during the year. However, even in a fully air-conditioned rest home located in Paris, we found, in the simulated cases, 5 kWh/m^2 of fuel savings by recovering energy from chiller condenser.

4 Simulated buildings versus European building stock

Information about the energy consumption in office buildings in Europe is poor. [EPA, 2006] reports some figures on energy consumption in office buildings in Europe. However, the sources of these data are often not clear.
In order to compare simulated buildings to European building stock, several sources of data are used (Figure 86 and Figure 87). The first source is a survey on 69 office buildings in Europe (Germany, France, Italy, UK, Austria, Spain, Czek Republic) with a majority located in Germany, Austria and Czek Republic (71%) [GRE 2005]. The information is based on energy bills and gross floor area. The buildings are split in five types: Naturally ventilated cellular, naturally ventilated open-plan, air-conditioned standard, air-conditioned prestige, air-conditioned skyscraper (>50m). The four ones are defined according to [ECG, 2003], the last one has been added.
The other sources correspond to standard values in UK [ECG, 2003] and in France [ADE, 2005]. The French value is an average based on a national survey. The UK benchmark data are derived from surveys on a large number of occupied buildings.
A last important source is a database of 132 energy audits on office buildings in Europe [SEA, 2005]. However, it has not been reported in Figure 86 and Figure 87 since the studied buildings are almost exclusively heated by district heating and cooled by district cooling. So the thermal consumption do not correspond to fuel consumption for heating but represents only the heating demand of the building.

In terms of electricity consumption, the simulated buildings are in the range of the building stock data (Figure 86). Notice that the simulated data do not include escalators, electric domestic hot water... Moreover, no low values of electric consumption are found in our simulated buildings. One can expect that the buildings with very low internal loads do not required any air-conditioning system and so they are not in the scope of our study. If one compares by end use, similar figures are obtained for lighting, office equipment and ventilation (Table 29).

Table 29 : Comparison of installed power and electric consumption by end use between simulation cases and literature data

		Green effect	SEA	UK	Simulation
Lighting power (W/m²)	Min	3	7 (average for landscape offices)	12	10
	Max	45	12.9 (average for individual offices)	20	18
	Average	13			
Lighting (kWh/m²/year)	Min	5	6.6	14	17.6
	Max	99	57.2	60	48.8
	Average	21	22.7		28.9
Equipment (W/m²)	Min	3		10	7.5
	Max	45		18	15
	Average	13			
Equipment (kWh/m²/year)	Min	1.5	3.6	12	13.5
	Max	114	367	32	38
	Average	23.5	23.4		23.6
Ventilation (kWh/m²/year)	Min	0.2	0.8	22	4.8
	Max	75.7	56.9	44	99.6
	Average	21.9	17.8		21.7

Humidification is rare in office building in UK according to [ECG, 2003]. Moreover, only two buildings in [GRE, 2005] present consumptions due to humidification.

In terms of heat consumption, our results are quite low compared to the data issued from European studies. One should first notice that the European surveys presented here are focused on north European countries where the heating demand is the highest (70% of the studied building of Green effect project are in the north east of Europe and most of the other sources comes from UK and France). However, the average annual fuel consumption of simulated buildings limited to Paris and Munich is 50 kWh/m² which is still low compared to north Europe data [GRE, 2005], [ECG, 2003], [ADE, 2005] (around 100-150 kWh/m²). On the other hand, [FRA, 2007] gives much closer values with 68 kWh/m² for standard office buildings in Germany.
One should notice that the simulated buildings refer to quite new buildings which are expected to be more frequently equipped of chillers and to be better insulated than old buildings. Table 30 shows that the U-values of the simulated buildings are better than Green effect buildings.
However, the simulated buildings still appear to have an average heating consumption a bit low compared to the average heating consumption in air-conditioning office building stock. This difference can be explained by several reasons:

- The user behaviour in some buildings can increase strongly the heating consumption such as high heating set points, the opening of windows and doors etc.;
- The use of higher ventilation rates than usual data in some buildings (Table 30).

Table 30: Comparison of U–values and ventilation rate between simulation cases and green effect data

		Green effect	Simulation
U_{wall}	Min	0.4	0.6
	Max	1.6	0.8
	Average	0.9	
$U_{glazing}$	Min	0.7	2
	Max	3	3
	Average	1.8	-
Ventilation (m³/h/m²)	Min	0.3	2.7
	Max	17.8	4.4
	Average	5.9	-[22]

[22] 14 m³/h/m² in [ECG, 2003]

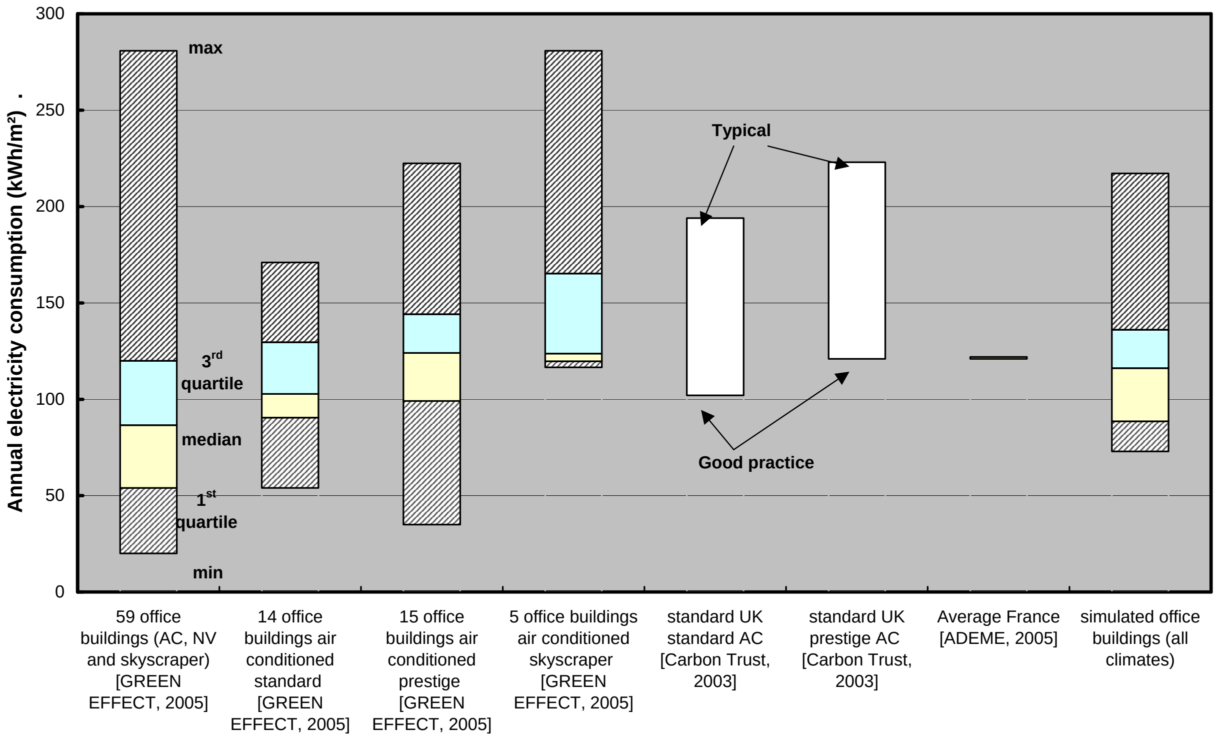


Figure 86: Comparison of annual electricity consumption between different sources and simulated buildings (values are expressed in kWh/m² of treated floor area except for Green Effect data which are in kWh/m² Gross floor area)

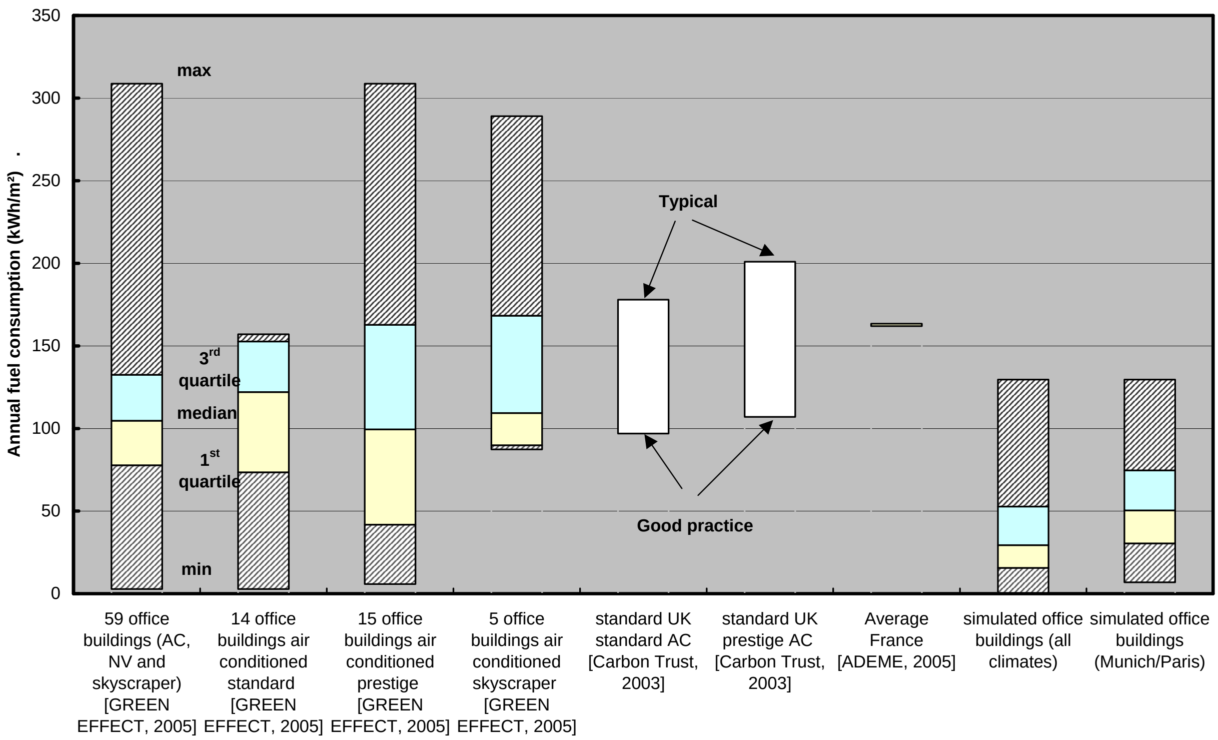


Figure 87: Comparison of annual thermal consumption between different sources and simulated buildings (values are expressed in kWh/m² of treated floor area except for Green Effect data which are in kWh/m² Gross floor area)

VI PERFORMANCE INDEX FOR CHILLER REVERSIBILITY AND HEAT RECOVERY POTENTIAL

1 Introduction

In this part, the objective is to elaborate a tool allowing a quick estimation of the heat recovery and the reversibility potentials, starting with limited information.

A first attempt at developing this tool, starting from the specifications which might be defined, was developed in [AND, 2007]. The tool was implemented in a software program which was able to use data originating from several sources and was applied to a first case study (an laboratory+office building) in Belgium.

Subsequent validation work carried out on a range of buildings will identify the limitations of the method [CAC, 2007]. Consequently, it was proposed to change the approach and to start from detailed simulation results using a model previously calibrated on global measurements performed on the building. This report will present this updated approach and the results obtained by applying it on a case study (a typical office building in the Walloon Region).

2 Summary of the first version of the identification tool

2.1 Specifications

Before starting the development of such a tool, the main specifications were formulated.

The objectives of this tool were:

- To quantify the heat recovery and/or the chiller reversibility potential in a given (office) building
- To identify the feasible technologies in that building
- To assess the energy (and economical) performance of a selected technology in a given building

This evaluation should be possible given limited information on the building, for instance using very rough design data or the level of information that might be extracted from the "as-built" files of the building project.

At a first glance, the possibilities of heat recovery or reversibility are dependent on the following characteristics:

- the heating and cooling demands profiles
- the technology eventually already in place (in case of retrofit)
- the choices already done in case of a new project (presence of air-conditioning, type of heating system,…)
- the levels of temperature observed or planned for the heating system as well as for the cooling system

Consequently, the decision variables of the identification process should be connected to the quantitative description of those characteristics.

The origin of the data which are feeding this tool is depending upon the status of the building project (new building or existing building).

For new buildings, data can be provided by the specifications of the project or by sizing calculations (if they are available) or by additional simulations (with programs like EES and/or TRNSYS). The level of detail of the data may be variable (monthly, daily or hourly, the latter in the case of the use of dynamic simulation).

For existing buildings, data can be provided by measurements of energy consumptions. In that case, the available level of detail is not very often better than monthly values and is also likely to be very global (total gas consumption or total electricity bill). For electricity, it is not straightforward to extract the specific consumption of HVAC systems. Another possible source of information is provided by the as-built files and by the description of the installed HVAC equipment (nominal power).

Two variables are useful to consider:

- the energy consumption on a given period (typically yearly ($Q_{y,heating}$ and $Q_{y,cooling}$) or monthly values: ($Q_{m,heating}$ (i) and $Q_{m,cooling}$(i)) or daily values: ($Q_{d,heating}$ (i,j) and $Q_{d,cooling}$(i,j))
- the peak power, which can reasonably be substituted by the installed power and consequently available from the sizing calculations: $P_{max,heating}$ and $P_{max,cooling}$. The ratio between both quantities gives a first impression on the balance between heating and cooling demands.

2.2 *Development of the identification tool*

2.2.1 Basic principle

As mentioned above, the identification tool should be able to use very limited information. Consequently, the idea of the tool is to start from very global data on the heating and cooling demands of the building and to progressively refine the identification of the potential when more detailed data become available.

The principle of the method is given by Figure 88.

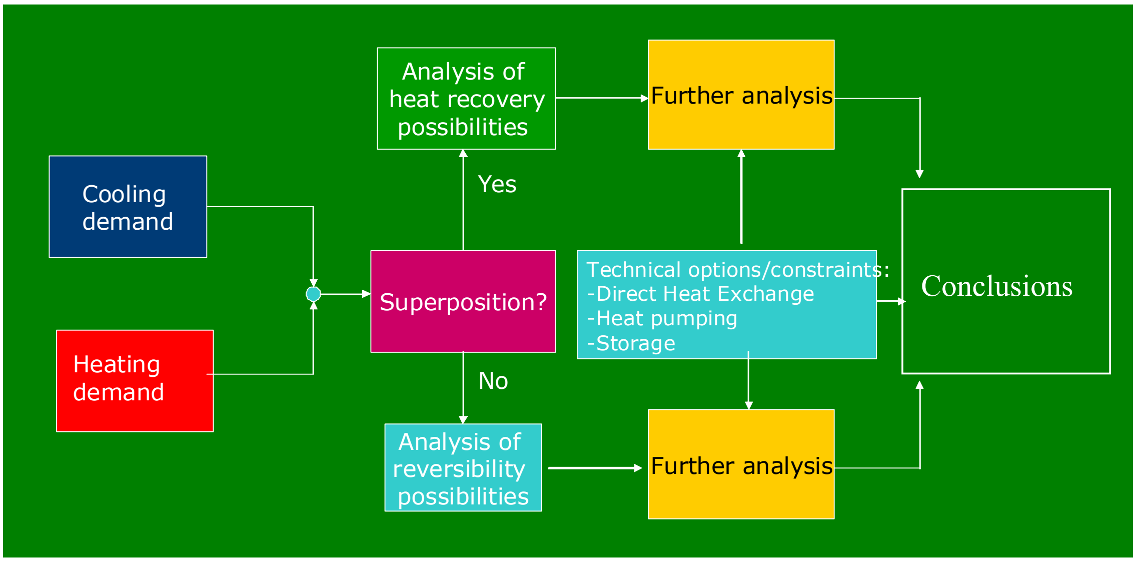


Figure 88: Principle of the identification strategy

The basis of the methodology was the comparison of the heating and cooling demands profiles.

A very first step consists in comparing the installed or nominal heating and cooling powers. The potential of energy savings is increasing when this ratio is closer to 1.

The next step requires yearly heating and cooling demands.

The first index is offered by the ratio of the yearly heating demand to the yearly cooling demand, which may be standardized in a [0,1] scale by defining it as:

$$Indice_{annual} = \frac{\min\left(\left|\sum_{year} Q_{heat}\right|, \left|\sum_{year} Q_{cool}\right|\right)}{\max\left(\left|\sum_{year} Q_{heat}\right|, \left|\sum_{year} Q_{cool}\right|\right)}$$

Two cases are interesting in function of the objectives of this work:

A. The case where the demands are clearly separated (a heating demand in winter and a cooling demand in summer) and are of the same order of magnitude: in this case the reversibility of the heat pump might be an option
B. The case where the demands are sufficiently superposed and again of the same order of magnitude: in that situation, the recovery of condensor heat (in air-conditioned buildings) or of the evaporator cool (in buildings heated by a heat pump) is an option

In both cases the energy savings potential is increasing when the order of magnitude of the heating and cooling power is closer.

Both cases can appear simultaneously in a building. For example, a chiller can produce cool in summer with heating recovery at the condenser and this chiller can be used as a heat pump in winter.

At this point, it is necessary to distinguish the two cases :

2.2.2 Reversibility of the heat pump

Reversibility of the chiller may be assessed in a first attempt from the calculation of the balance between the winter heating demand and the summer cooling demand. This balance can be estimated from the following index:

$$Indice_{season} = \frac{\min(\sum_{winter} Q_{heat}, \sum_{summer} Q_{cool})}{\max(\sum_{winter} Q_{heat}, \sum_{summer} Q_{cool})}$$

Ideally, this index should be around one (for an optimal reversible use) but a reasonable range may be defined between 0.5 and 1.

In many buildings, the cooling demand doesn't disappear, even in winter (some rooms like computer rooms may require cooling all the year long); on the other hand, some heating demand might be observed in summer (for the sanitary hot water preparation for instance). Consequently, the evaluation of the reversibility potential should be based on modified demands, ie the one obtained after removal of the "parasitic" demands.

An estimation of the parasitic demands can be obtained by resp. the summer heating demand and resp. by the winter cooling demand, assuming that these demands stay of the same order of magnitude in winter, resp. in summer.

A modified reversibility assessment index may consequently be calculated by:

$$Indice'_{season} = \frac{\min((\sum_{winter} Q_{heat} - (\sum_{summer} Q_{heat} \cdot \frac{nberwintermonths}{nbersummermonths})), (\sum_{summer} Q_{cool} - (\sum_{winter} Q_{cool} \cdot \frac{nbersummermonths}{nberwintermonths})))}{\max((\sum_{winter} Q_{heat} - (\sum_{summer} Q_{heat} \cdot \frac{nberwintermonths}{nbersummermonths})), (\sum_{summer} Q_{cool} - (\sum_{winter} Q_{cool} \cdot \frac{nbersummermonths}{nberwintermonths})))}$$

2.2.3 Recovery of condensing heat

If reversibility does not appear to be an option, heat/cool recovery has to be evaluated.

The second step for this case consists in identifying the periods of the year for which the potential is present. Therefore, it is necessary to have more details on the time evolutions of both demands. A monthly analysis is required in order to evaluate this potential. So, first the same calculation as above is repeated for each month, which allows locating the months of the year where the potential is interesting:

$I_m(i) = \min(|Q_{m,heating}(i)|, |Q_{m,cooling}(i)|) / \max(|Q_{m,heating}(i)|, |Q_{m,cooling}(i)|)$

Again, this index should not be too far from 1 to show a potential for heat recovery for a given month.

To recover the condensing heat, the heating demand and the cooling demand have to be "sufficiently" superposed.

If the cooling and the heating demand are exactly superposed, heating could be directly recovered at the condensing part of the chiller.
If there are some hours (maximum 1 day) lag between the two demands, a water or ice storage could be used.
If there is more than 1 day between the demands, a ground storage could be used but this solution is very complex to install in a existing building.

The next step could consists in identifying the superposition profiles, which requires to move down to a daily analysis : identification of days for which both demands are superposed define periods where heat recovery is theoretically possible.

In the next paragraph, we show that the index calculated by the monthly demands (that are most often available for an existing building) overestimates the results by about 15%. So the identification could manage with monthly information.

Finally, when the demands are "daily" superposed, the last level of analysis leads to an observation of the hourly demands profile: if this indicates a real superposition of the demands, heat recovery ("direct recovery") confirms to be an option; if the demands are "hourly" separated, a storage system could be installed to use the availability of heat later in the day ("delayed recovery")

$I_d(i,j) = \min(|Q_{d,heating}(i,j)|, |Q_{d,cooling}(i,j)|) / \max(|Q_{d,heating}(i,j|, |Q_{d,cooling}(i,j)|)$

The difference between the daily and hourly results could give an idea of the storage but , as the storage is often "realistic", the recovery potential could be well estimated by the monthly demands.

2.3 *Validation of the method*

Validation of the method involves different issues which were already presented in [AND, 2007]. In this first part of the work, only global validation was assessed and it was showed that, concerning heat recovery, the fraction predicted by the value of the index as developed here above, was in good correlation (Figure 78) with the value of the recoverable heat as calculated by:

$$P_{heat\ potential} \cdot = \frac{4}{3} \cdot P_{cool}$$

Associated to the following limits:
If $P_{heat\ potential} > P_{heat}$, $P_{heat\ effective} = P_{heat}$
If $P_{heat\ potential} < P_{heat}$, $P_{heat\ effective} = P_{heat\ potential}$

The calculation, on a monthly basis of the heat recovery index and the potential heat recovery fraction leads to the following figure, which shows a relatively good correlation appears between both indexes (Figure 89). The different regression curves correspond to different variants of a given building with different setpoints, occupancy schedules, internal gains,…

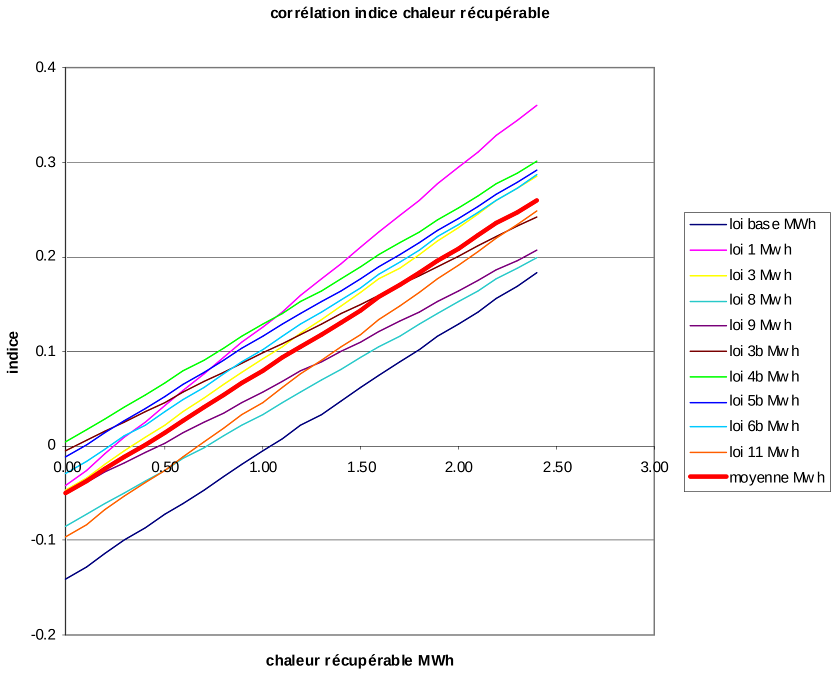


Figure 89: Correlation between heat recovery index and potential heat recovery fraction

A second validation step consisted in comparing the values predicted by the indexes to estimations of the savings generated by hourly calculation of heating and cooling demands. These calculations allow indeed a good prediction of the reversibility and recovery potentials [CAC, 2008] as defined hereunder.

2.3.1 Reversibility potential calculated from hourly simulations

The reversibility potential (REVpot) is defined, in absolute value, as the part of the total heating demand that can be covered by the chiller in heat pump mode, in MWh. The reversibility potential depends on the heating power capacity which can be reached by converting a chiller into a heat pump and on the non simultaneous demand of cooling and heating.
In order to standardize this value, it can be divided by the total heating demand, obtaining the relative reversibility potential (Irev,pot).

These two quantities are calculated hour by hour (Figure 16, chapter III), and they represent the value of the reversibility potential such as :

- The "chiller" operates in priority in cooling mode; so reversibility is possible only when no cooling is required ;
- The maximum heating power available is assessed to be 0.8 × maximum cooling power of the chiller[23], the supplementary demand is assumed to be covered by the boiler.

No consideration on the emitter temperature levels required is taken into account at this stage.

[23] This value is based on data from manufacturer's catalogues. Some differences can be noticed between chillers. For the selected reversible chiller, one can notice that at nominal conditions the cooling power is about 15% lower than the heating power. However, the nominal heating conditions are at 7°C outside air temperature which do not correspond to the worst conditions whereas the nominal cooling conditions are imposed to 35°C that is close to the worst conditions. If we considered the heating power at –5°C, the heating power is about 20% lower than cooling power at nominal conditions.

Typical office buildings have been simulated in five European climatic zones (meteorological files from Athens, Lisbon, Paris, Munich and Turin). At the end, 800 simulations have been achieved taking into consideration different levels of internal loads, solar protection, building typologies, building orientation, set points and ventilation rate (described in Chapter II).

2.3.2 Correlation with previously defined indexes

The simulation results have been examined in order to find some types of trends or correlations between the various indexes defined.

Figure 90 plots the growing curve of the relative reversibility potential of all the simulation cases. It can be noticed that reversibility potential seems not to have wide variation among the simulations, falling in most of the cases (700 on 800) in the range 80-100%. Only in very few cases it is under the 60%. About 100 simulations are under the 80 %. These results show that the reversibility potential is important in office buildings.

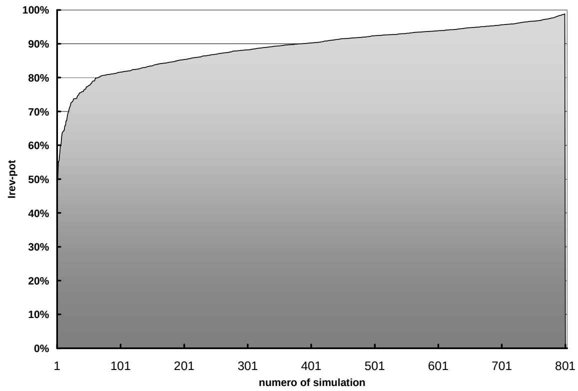


Figure 90: Growing curve of the reversibility potentials of the simulation cases

Consequently, plotting the absolute reversibility potential versus the heating demand (Figure 91), it results a very good correlation.

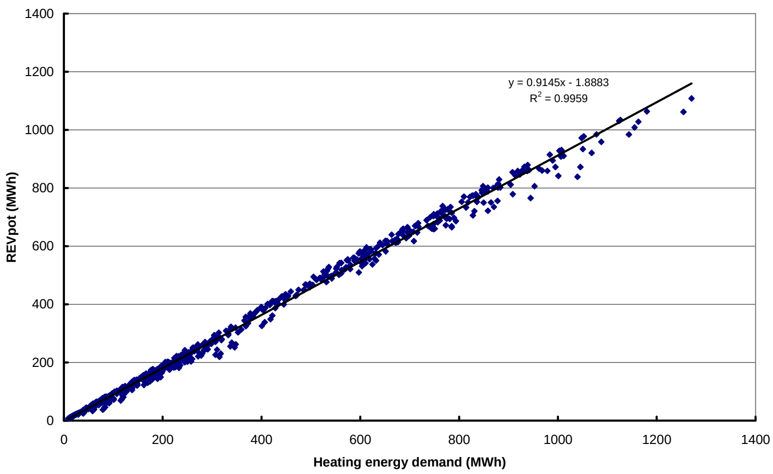


Figure 91: Absolute reversibility potential versus yearly heating energy demand

Afterwards, the reversibility potential, both absolute and relative, have been plotted versus annual index, seasonal index figures 92 to 95.

Observing these graphs, apparently no correlation or trend seems to exist.

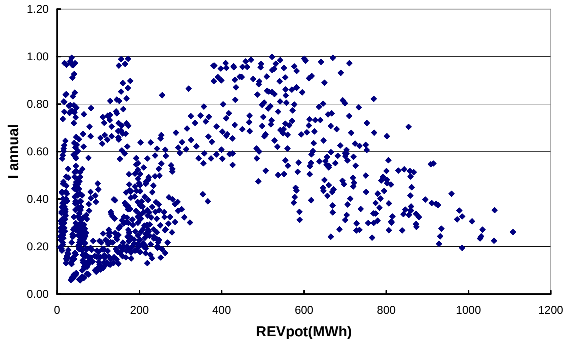


Figure 92

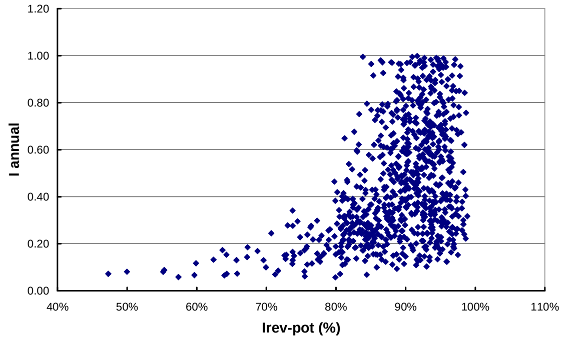


Figure 93

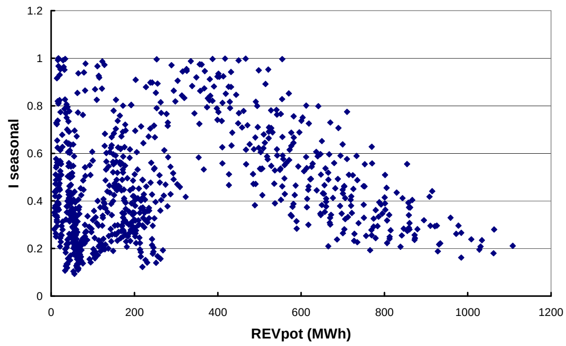


Figure 94

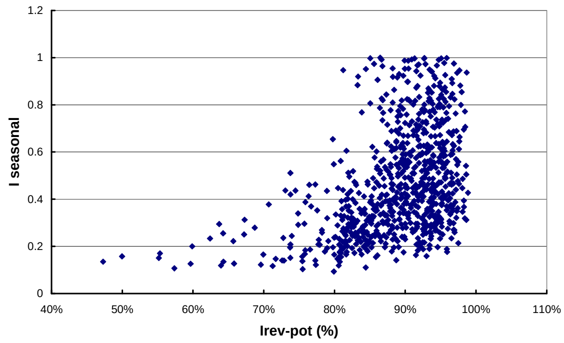


Figure 95

2.3.3 Proposition of other Indexes

Here, we define a coefficient calculated as the ratio between the cooling capacity [kW] and the heating capacity [kW] (one can assume that more the heating capacity is high compared to the cooling capacity, less the reversible chiller[24] would be able to cover the heating demand).

$$I_{power}=\min\left(\frac{P_{cooling}}{P_{heating}},1\right)$$

This index is supposed to be a measure of the "capacity" of the chiller to face the winter heating demand.

Finally, a winter cooling index is defined as the ratio between winter cooling demand and annual cooling demand:

$$I_{wcool}=\frac{Q_{summer,cooling}}{Q_{annual,cooling}}=1-\frac{Q_{winter,cooling}}{Q_{annual,cooling}}$$

This ratio is supposed to give a measure of the heating demand which could not be faced by the reversible chiller (more the cooling demand is high in winter, less the chiller can operate in heating mode during winter).

A similar ratio for summer heating could be determined but the results do not plead for its use.
The correlation between the reversibility potential and the product of the two indexes is presented in the Figures 96 to 100.

BUILDING TYPE 1A

[24] The term « reversible chiller » is used to specify that the chiller is sized to face the maximum cooling power required and the maximum heating power is then a consequence of the sizing in cooling mode.

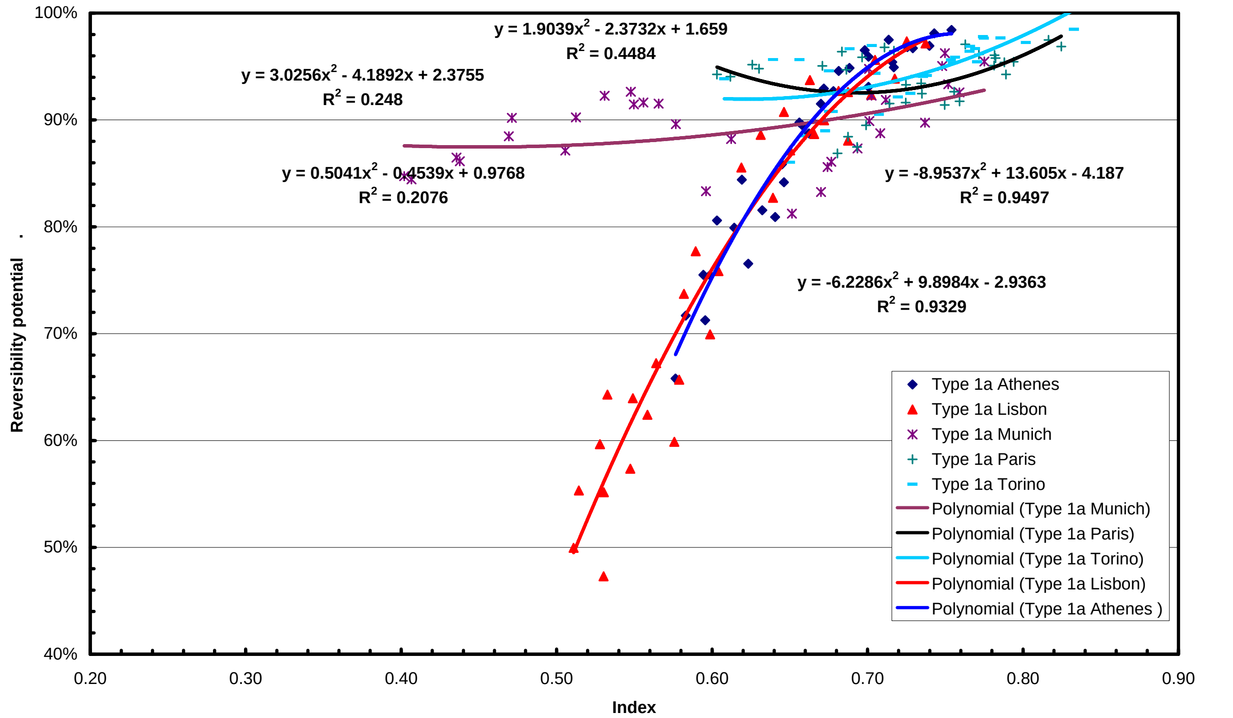


Figure 96: correlation between the reversibility potential ($I_{rev, pot}$) and the Index (I_{power} * I_{wcool}) for the different climates in building of TYPE 1A

BUILDING OF TYPE 1B

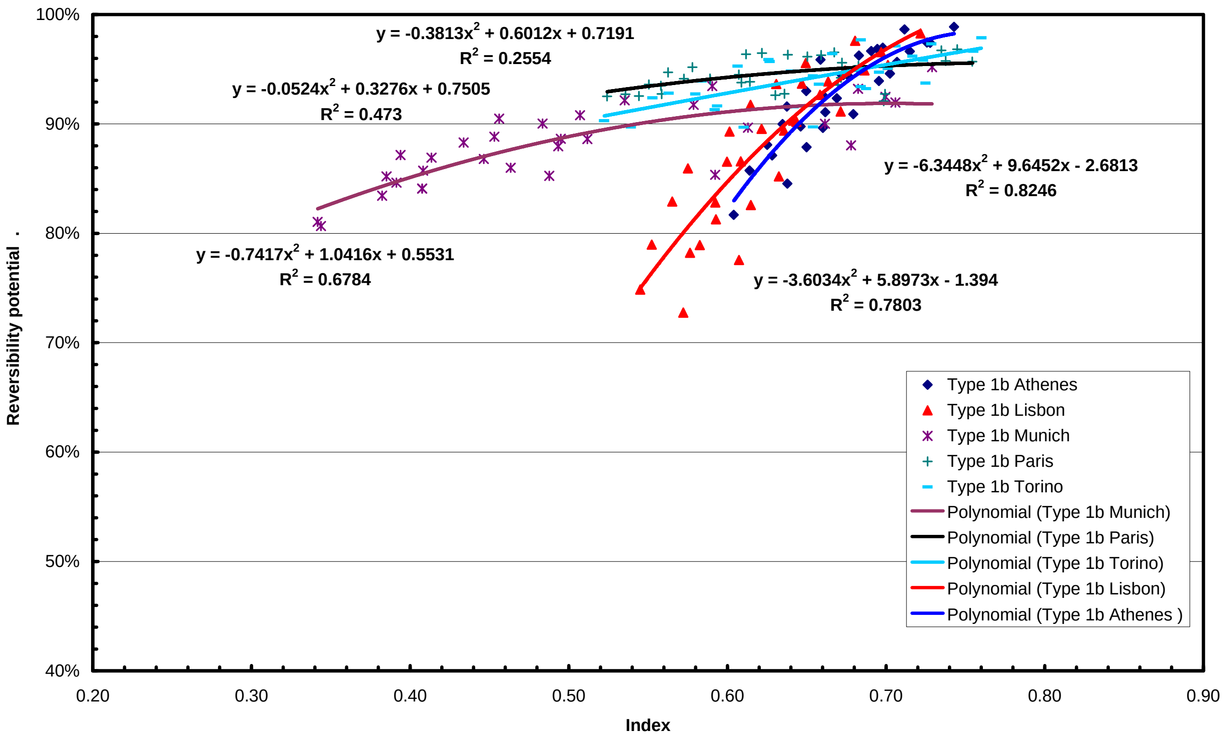


Figure 97: correlation between the reversibility potential ($I_{rev, pot}$) and the Index (I_{power} * I_{wcool}) for the different climates in building of TYPE 1B

BUILDING OF TYPE 1C

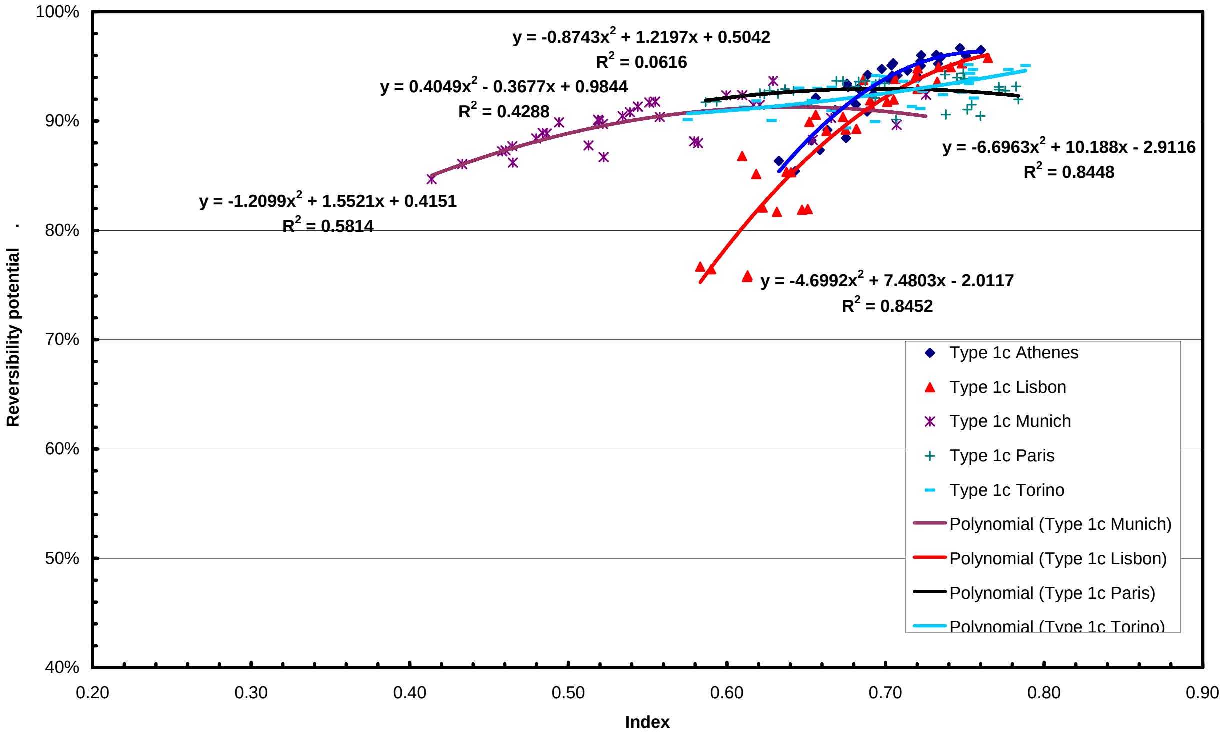


Figure 98: correlation between the reversibility potential ($I_{rev,\ pot}$) and the Index (I_{power} * I_{wcool}) for the different climates in building of TYPE 1C

BUILDING OF TYPE 2

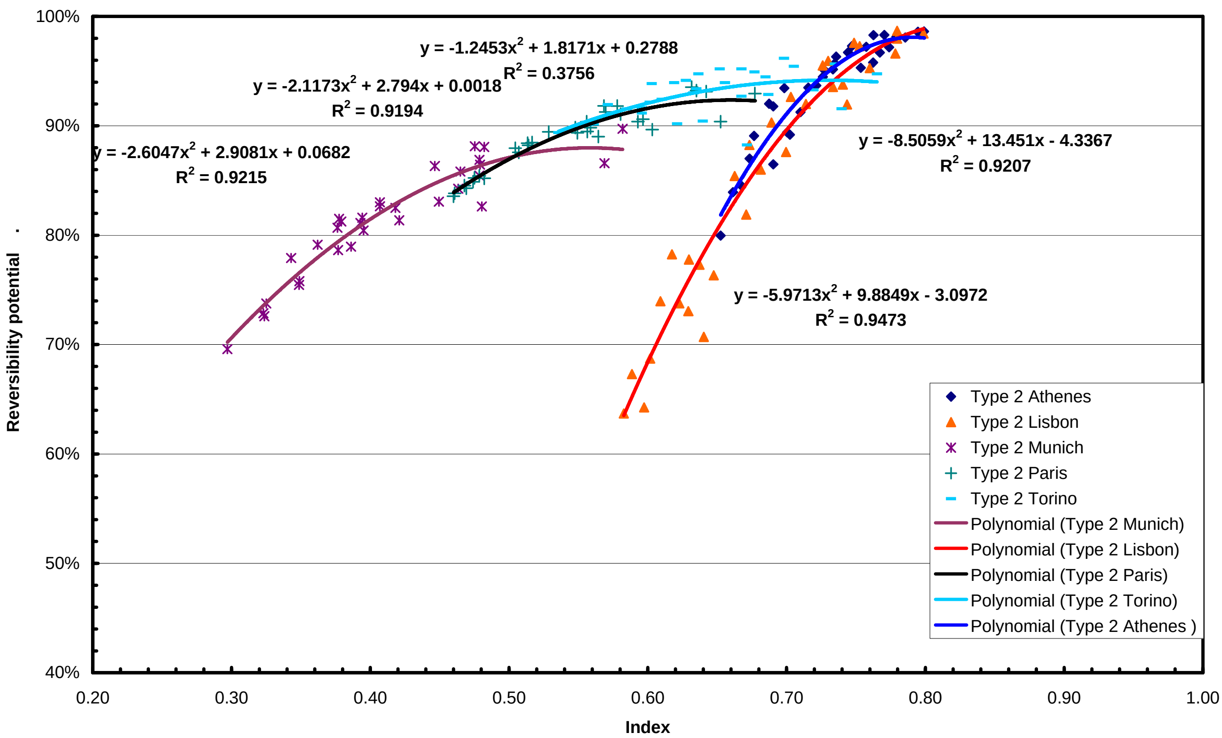


Figure 99: correlation between the reversibility potential ($I_{rev,\ pot}$) and the Index (I_{power} * I_{wcool}) for the different climates in building of TYPE 2

BUILDING OF TYPE 3

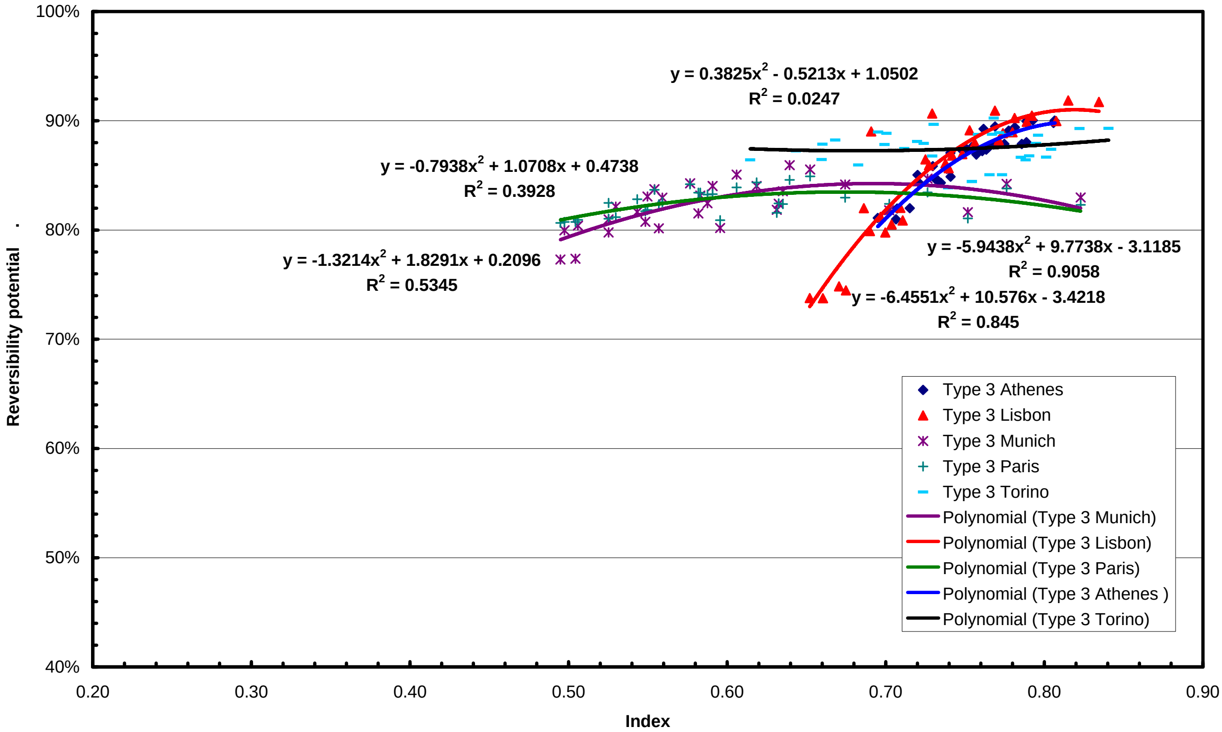


Figure 100: correlation between the reversibility potential ($I_{rev,\ pot}$) and the Index (I_{power} * I_{wcool}) for the different climates in building of TYPE 2

The building type and the climate have a noticeable impact on the reversibility potential. On the other hand, the internal loads and the cooling/heating set point temperatures have a lower impact on the reversibility potential. The building orientation, ventilation rate and solar protection have still lower impact on the reversibility potential.

One can notice that the correlations for Lisbon and Athens are very closed and could be aggregated in one correlation. An aggregated correlation for Paris and Munich and in a lesser extent for Torino could also be considered.

Correlation between the index defined here and the reversibility potential is quite poor. However its calculation can provide a fast indication of the interest of using the chiller in heating mode in typical office buildings. The higher is the index, the higher is the reversibility potential in typical office buildings. Another important conclusion is that the reversibility potential is less sensitive to internal loads, solar loads, orientation, building type… in temperate climates such as Paris and Torino than in more "extreme" climates studied here (Munich, Athens and Lisbon).

2.5. Recovery potential

The recovery potential depends on the simultaneous heating and cooling demand and on the heat power available on chiller condenser (only consideration on energy recovery for space heating is taken into account here in office buildings, but heat recovery could also be possible for Domestic Hot water).
The recovery potential is calculated hour by hour (Figure 23 of chapter III) as the percentage of heating demand which could be provided by a chiller condenser under the following conditions :

- the "chiller" is in operation in order to provide the cooling demand;
- The maximum heating power available on the condenser is calculated based on energy conservation principle such as (EER+1)/EER × cooling power provided by the chiller at the step time, the supplementary heating demand is assumed to be covered by the boiler.

No consideration on the emitter temperature levels required is done here. The % of recovery is calculated as the ratio between the total heat recovery for space heating and the total space heating demand.

For all the cases, the recovery potential ranges between 0% and 45%.

3 Proposal of a new approach for assessment of savings

Given the limitations of an estimation based upon global values of heating and cooling demands, it is proposed here to use another approach, namely to generate by dynamic simulations hourly values of heating and cooling demands and to calculate the reversibility and recovery potentials as developed above from the results of the calculation. The connection with the measurements is done through tuning of the simulation model in order to reproduce not too bad the global values. Hourly simulations are carried out using models developed using EES (SimZone and Aggregate). The tools are described in [BER, 2008].

The tool consists in a package of directly executable simulation files together with an explanation note [BER, 2008] including the definition of an evaluation index about reversibility and recovery potentials.

Identification of the savings is carried out in 3 steps:

1) Tuning of the parameters of the model using information effectively available (main dimensions of the building, types of facades, windows, orientations, occupancy schedule and rate, internal gains, effective control of the air renewal rate, electricity and fuel consumptions, corresponding weather data)
2) Simulation of the system on a typical year with integration of the heating and cooling demands and corresponding consumptions
3) Aggregation of the demands for the analysed zones. Computation of the reversibility and recovery potentials, of the global index and system selection in view of an energy refurbishment.

The three steps are carried out using the two following tools :

- Tool n°1: Building simulation and calculation of demands
- Tool n°2: Aggregation of the demands and calculation of the recovery and reversibility potentials

A short description of the tools is given below. A more detailed presentation is given in [BER, 2008]:

TOOL n°1 ("SimZone") : mono-zone or multi-zone building simulation

The behaviour of the building can be simulated by three different methods:

- the building is considered as an equivalent single zone to which an HVAC system is connected. This HVAC system includes an Air Handling Unit and terminal units for both heating and cooling
- the building is analysed floor by floor. It is possible to distinguish between intermediate floors and floors below the roof
- the building is divided in sub-zones which can represent either a single room, a part (wing) of the building or a set of peripheral spaces (in contact with an external frontage) or a set of central spaces.

A dedicated simulation tool is available for each type of simulation. The selection of the tool has to be done by the user who has to judge of the necessity to simulate a small part of the building or a larger area. The main selection factors are the orientation and the levels of internal gains.

TOOL n° 2 (Aggregate) : aggregation of the demands and calculation of the recovery and reversibility potentials

Once the zone simulations have been completed, the demands from the different zones (8760 hourly values of heating and cooling demands) are introduced in the aggregation tool. The number of zones of a given type is specified (number of south orientated zones, number of central zones,…) and the loads are aggregated in order to determine the global demand profile.

The heating and cooling demands are compared and used in order to compute the recovery and reversibility potential as defined here above (see 2.5 and 2.3.1. respectively).

In both cases, the residual heating demand is supposed to be provided by a classical equipment (boiler).

4 Comparison of reversibility potentials based on heating and cooling demands and based on energy consumption

The reversibility potential has been first calculated based on heating and cooling demands (concerns the building only). The aim is now to compare these potentials to those obtained from building energy consumption simulations (includes the different HVAC systems):

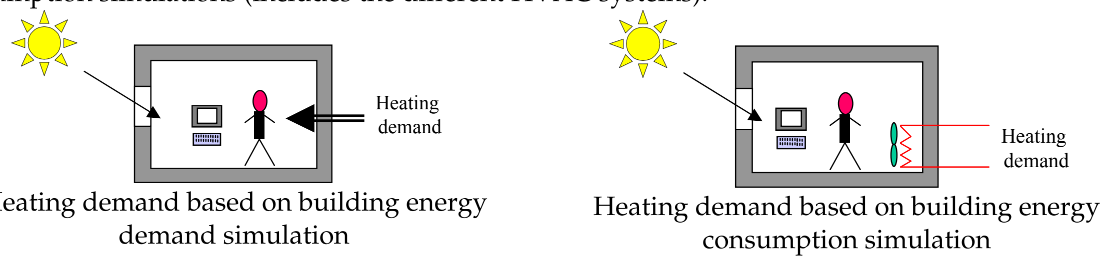


Figure 101: Scheme of heating demand with or without HVAC secondary system

First, one should notice that the reversibility potentials based on building energy demands were calculated for single flux ventilation, only.
Figure 102 shows a comparison on the heating demand between energy demand simulations and energy consumption simulations including various HVAC systems. It should be also noted that the heating demand calculated from building energy consumption simulation is calculated here without considering the hot water distribution losses. The heating demand is lower when the HVAC system is included since the heating provided by the fans of AHU or FCU reduces the heating demand to the boiler. With double flux ventilation, the heating demand is decreased. However, in case of Air handling units, in particular for Constant Air volume systems, there is a lower correlation between building heating demand based on system energy and based on building energy.

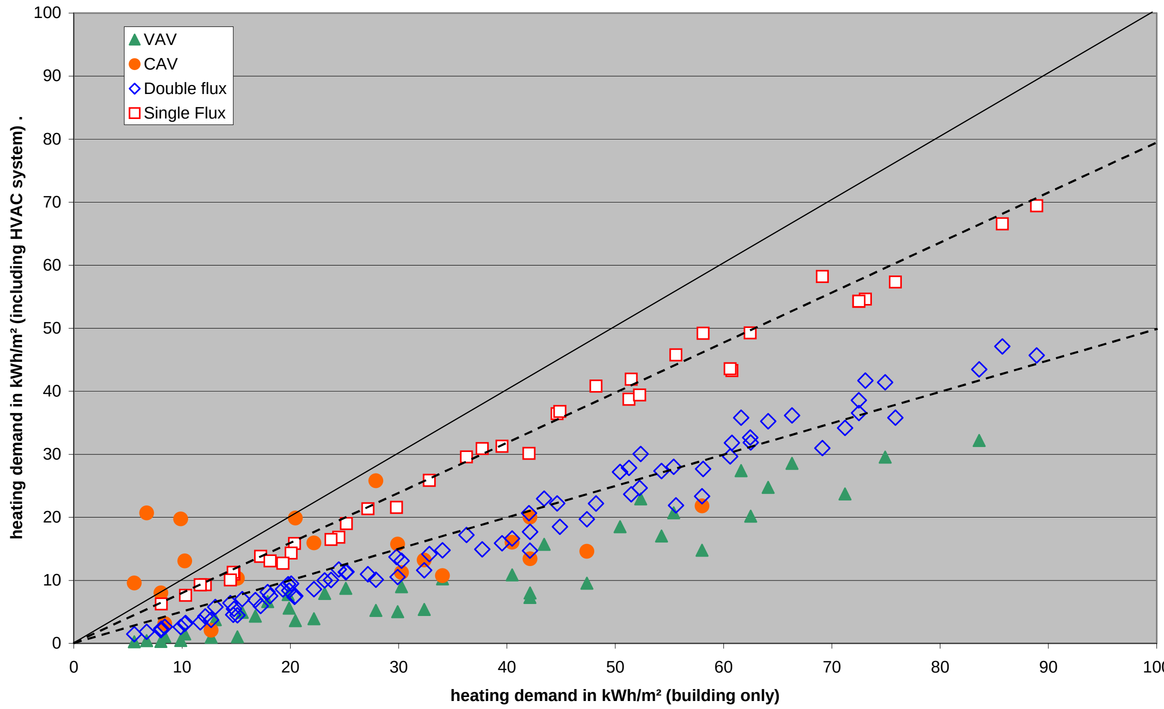


Figure 102: Comparison of heating demand based on building energy demand and HVAC system energy demand

The comparison of reversibility potential between simulations in energy demand and in energy consumption are presented only in the case of building energy simulations with single flux ventilation and Fan Coil Units (Figure 103). The results show a quite good correlation between results in energy demand and in energy consumption in climatic zone such as Paris, Munich and Torino with about 10% of difference. On the contrary, the reversibility potential based on energy demand are far from those based on energy consumption including the HVAC system in climatic zones such as Lisbon and Athens. This can be explained by the fact that the total heating demand in these climatic zones is low and small differences in the reversibility potential include large differences in %.

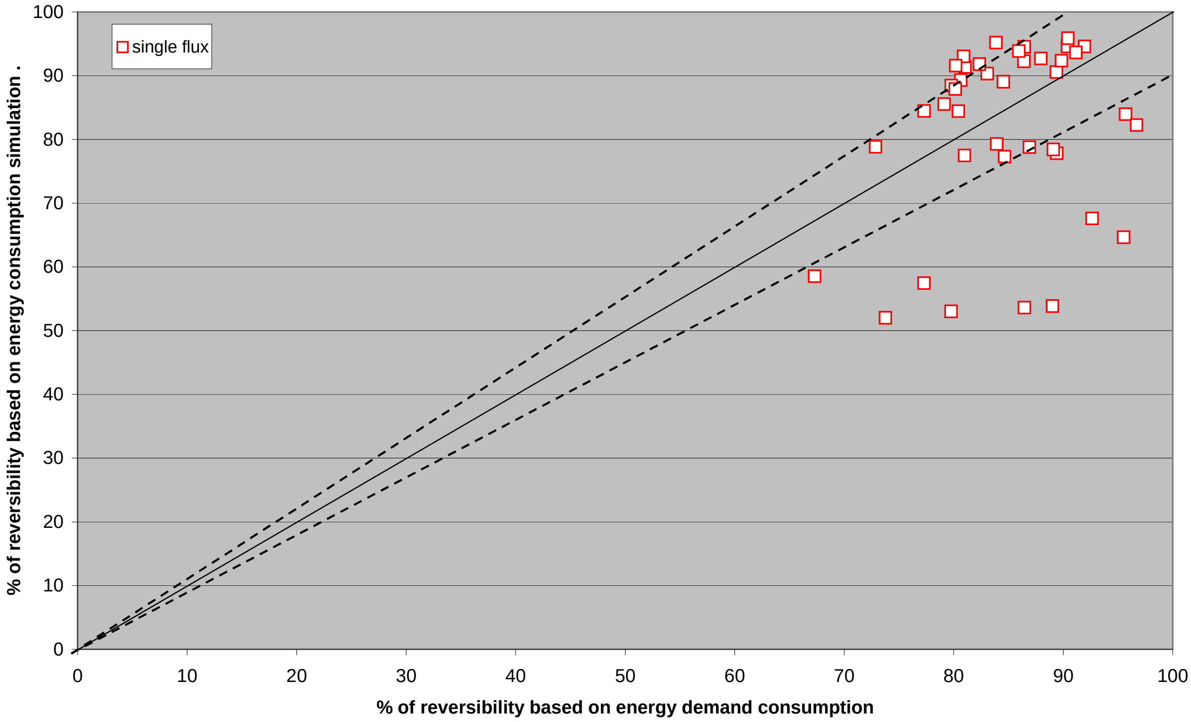


Figure 103: Comparison of percentage of reversibility calculated from building energy demand simulations and from building energy consumption simulations including HVAC systems

The HVAC system and its control strategy in particular has an influence on the reversibility potential. For instance, if one assumes a control strategy where the heat pump is stopped during night and so that the backup boiler maintained the set back temperatures, one gets some noticeable differences compared to an operation of the heat pump even during the night (results presented here). The fact to stop the heat pump during night reduces obviously the potential of reversibility.
Thus, the reversibility potentials calculated from heating and cooling demands are a quite good index but cannot presupposed the control strategy to be implemented on the reversible heat pump.

5 Application of the method to a typical office building in Belgium (Case Study n°6)

The building described in details in [AND, 2008] is used for the application of this method.

The first phase of the identification consists in choosing how the simulation has to be performed.

The main block of the building is analysed. This part of the building is made of 8 identical floors. Each floor can be divided into 5 zones (Figure 104) :

- one central zone
- 4 zones situated along the 4 facades of the building

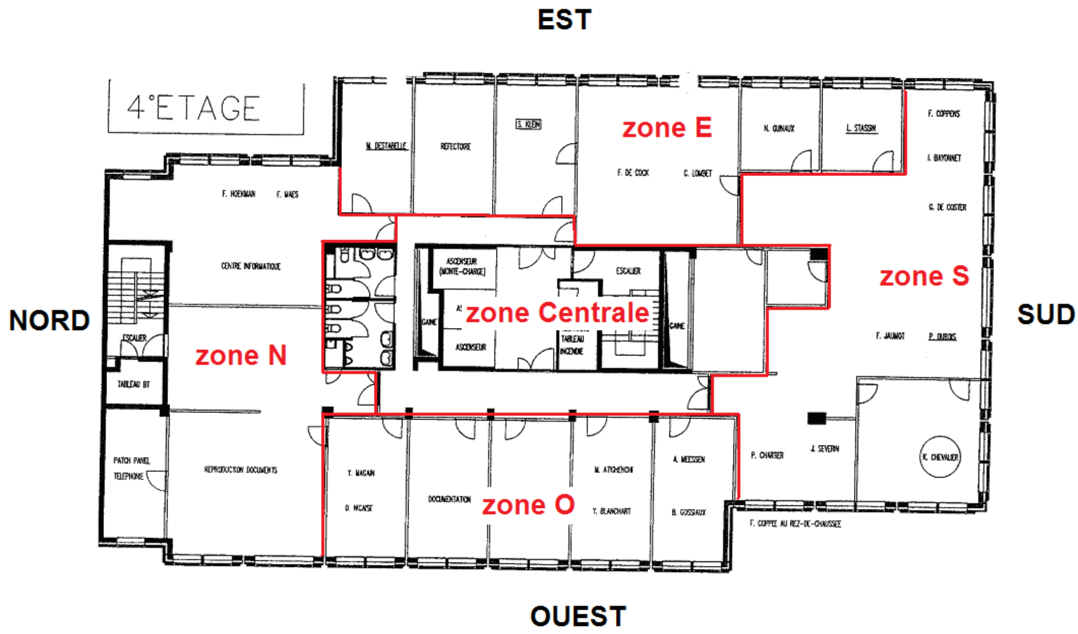


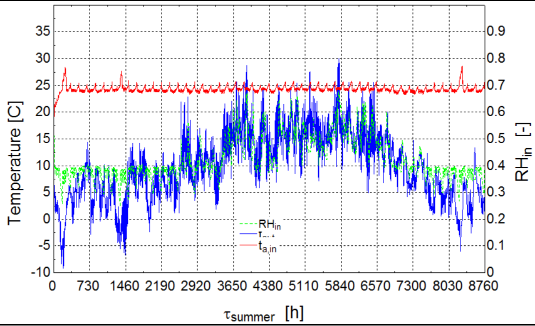
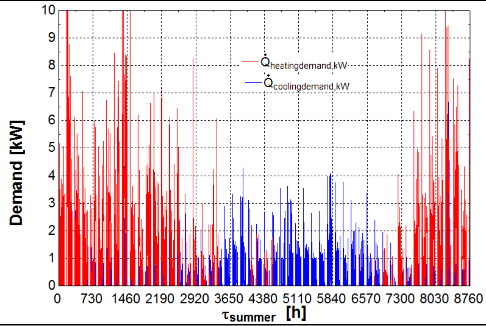
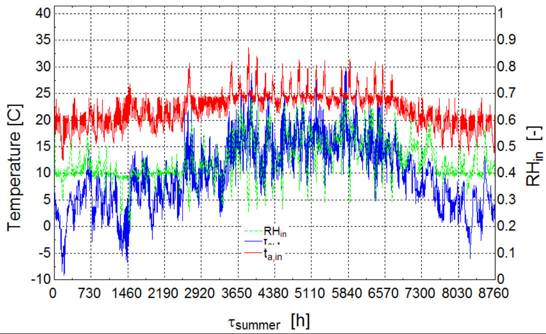
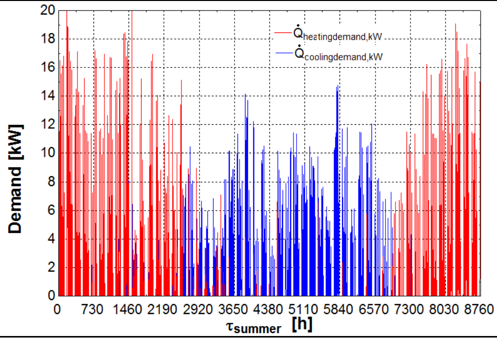
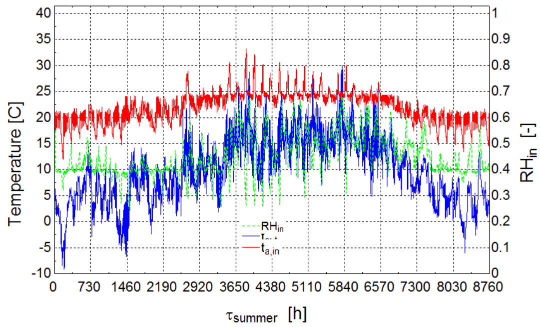
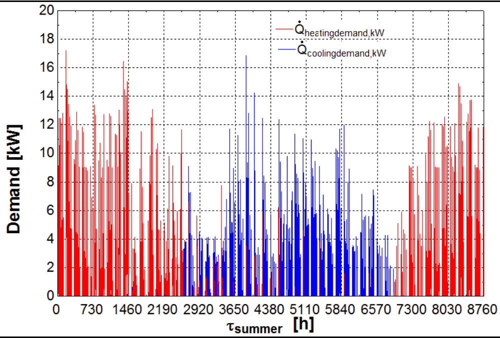
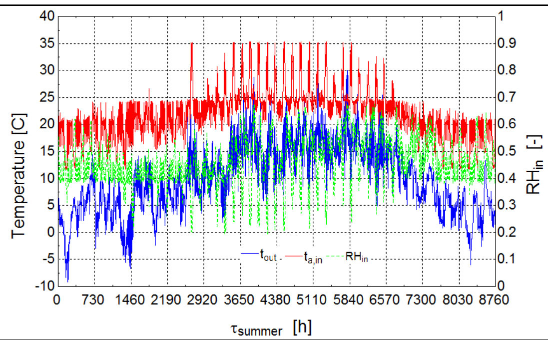
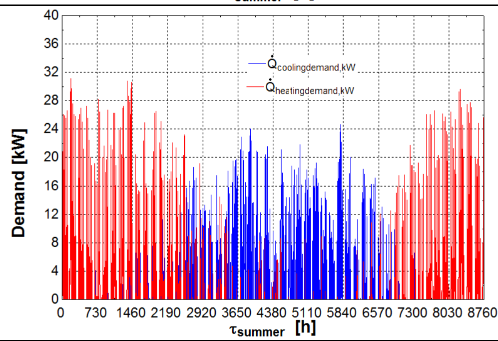
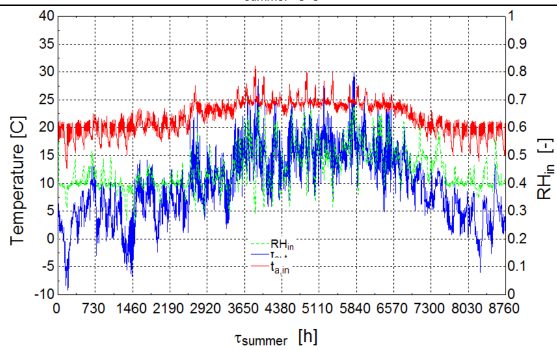
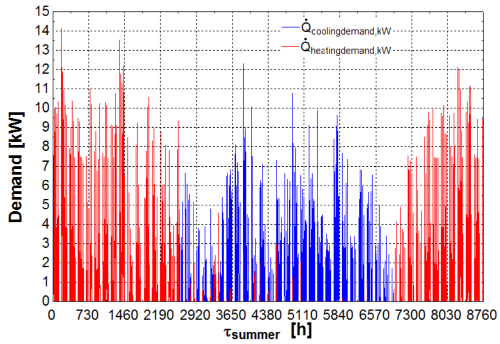
Figure 104: Zoning of the simulated flow

For this simulation, the building is supposed to be controlled by an ideal system. The control of humidity is made by the Air Handling Unit (adiabatic humidification and dehumidification by cooling) while the temperature control is supposed to be made by terminal units. There is no control of the supply temperature (no post-heating coil).

The characteristics of each zone are obtained thanks to the blue prints (dimensions, glazed area, internal walls,…) and the information provided by the building manager or during an audit (occupancy, internal loads, set points,…).

Table 31 shows the five heating and cooling demands loads profiles generated by the model.

Table 31: Simulation results

Zone	Temperature/RH profile	Heating and cooling demands
Central		
South		
East		
North		
West		

The demands are afterwards aggregated using tool n°2. Figure 105 shows the global demands of the simulated floor.

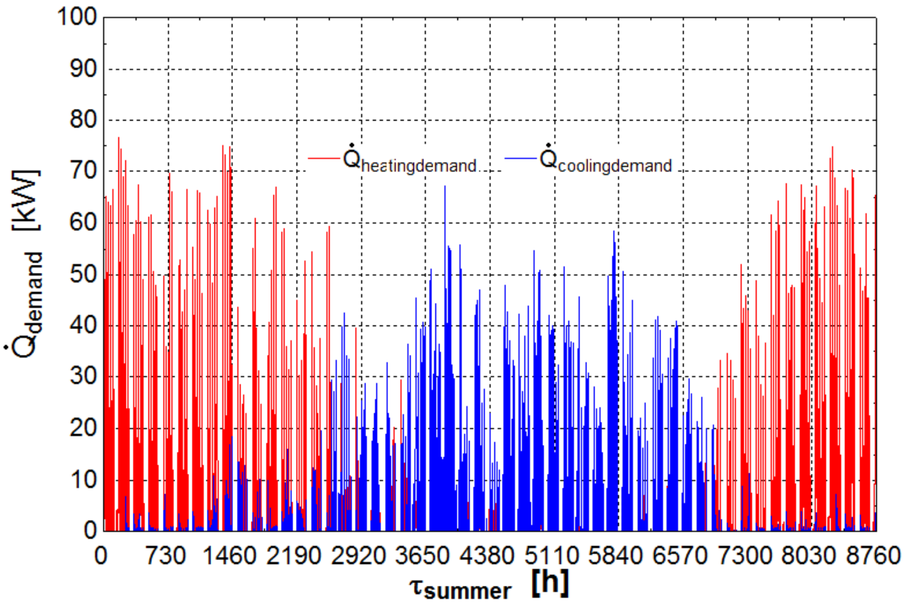


Figure 105: Global heating and cooling demands of the simulated floor

The corresponding reversibility and recovery potentials are as follows :

- Reversibility potential : 52 %
- Recovery potential : 13 %

As expected, the recovery potential stays relatively small under our average latitudes. Reversibility seems to be a more interesting opportunity as it seems able to cover more than 50% of the heating demand of the simulated floor.

6 Conclusions

This chapter has presented various identification methodologies in order to estimate the energy saving potentials which can be obtained from the application of heat pumping strategies in office buildings. The results show that it is no easy to use simple indexes to assess accurately the heat recovery and reversibility potentials.
In existing buildings, electricity and fuel bills are easy to get but they are generally at best monthly. Moreover, it is difficult to extract the chiller electricity consumption from the electricity bill. So, the simple indexes based on monthly heating and cooling demands appear difficult to use. Another idea could be to use the installed heating and cooling capacities. However, these data do not help so much in defining an efficient index.
At the end, the proposed methodology is to have recourse to simulation tools in order to assess the energy saving potentials. This solution imposes to make an audit of the building to look for the input data requested by the building energy dynamic simulation tool (building surface, building size, building orientation, window surface, occupancy scenarios…). The tool would be better if it includes the HVAC system since it will take into account the temperature levels in terminal units which can limit the application of heat pumping strategies.

VII PRIMARY ENERGY SAVINGS AND CO_2 EMISSION REDUCTION

In this part, the primary energy savings and CO_2 emission reductions in Europe-15 by using reversible chillers and heat recovery on chiller condenser are assessed.
The estimations of primary energy savings and CO_2 emission reduction are calculated by considering that the European-15 building stock equipped with air-cooled chillers could be retrofitted by reversible heat pumps and the stock equipped water-cooled chilled could be retrofitted by heat recovery on condenser. The main assumptions are that:

- The current building stock equipped with chillers is in parallel heated by boilers;
- Air-cooled chillers are only taken into account for reversibility potential assessment since their replacement by reverse chillers appears to be easier than for other air-conditioning systems (Room air conditioners, water-cooled chillers);
- Water-cooled chillers are only taken into account for heat recovery on condenser;
- The assessment does not take into account the growth of the building stock but only the retrofit of the existing buildings;
- The reversible chillers are assumed to be sized on the cooling demand;
- The backup boiler efficiency is the same as the average efficiency of the boiler in the reference case.

1 Assumptions

1.1 Building stock

The health care institution and office building floor areas air-conditioned by air-cooled chillers are calculated for each country of Europe-15 on the basis of the number of air-conditioning units sold and typical ratios of installed power, as described in [EECCAC, 2003]. The figures for air-cooled chillers are reported in Table 32 and Table 33. The figures are based on a forecast stock by 2005.

Table 32: Total office building area air-conditioned by air-cooled chillers in Europe-15 countries

Country	Mm^2	Share	Country	Mm^2	Share
Belgium	5.4	1.5%	Luxembourg	0.2	0.1%
Denmark	2.0	0.6%	The Netherlands	11.2	3.1%
Germany	42.3	11.5%	Austria	6.2	1.7%
Greece	22.5	6.1%	Portugal	7.3	2.0%
Spain	66.5	18.1%	Finland	4.4	1.2%
France	59.2	16.2%	Sweden	8.0	2.2%
Irland	2.2	0.6%	United Kingdom	39.1	10.7%
Italy	89.8	24.5%	TOTAL	366.4	100.0%

Table 33: Total health care institution area air-conditioned by air-cooled chillers in Europe-15 countries

Country	Mm^2	Share	Country	Mm^2	Share
Belgium	1.7	1.6%	Luxembourg	0.1	0.1%
Denmark	0.6	0.6%	The Netherlands	4.0	3.7%
Germany	13.1	12.1%	Austria	2.3	2.1%
Greece	5.3	4.9%	Portugal	1.3	1.2%
Spain	17.9	16.6%	Finland	1.7	1.6%
France	18.2	16.9%	Sweden	3.0	2.8%
Irland	0.7	0.6%	United Kingdom	11.8	10.9%
Italy	26.1	24.2%	TOTAL	107.9	100.0%

The following table provides figures for water cooled chillers in health care institutions. No heat recovery is considered in office buildings since it has been shown in the previous chapter that the potential is low.

Table 34: Total health care institution area air-conditioned by water-cooled chillers in Europe-15 countries

Country	Mm^2	Share	Country	Mm^2	Share
Belgium	0.2	1.6%	Luxembourg	0.0	0.1%
Denmark	0.1	0.6%	The Netherlands	0.6	3.7%
Germany	1.8	12.1%	Austria	0.3	2.1%
Greece	0.7	4.9%	Portugal	0.2	1.2%
Spain	2.5	16.6%	Finland	0.2	1.6%
France	2.5	16.9%	Sweden	0.4	2.8%
Irland	0.1	0.6%	United Kingdom	1.6	10.9%
Italy	3.6	24.2%	TOTAL	15.0	100.0%

1.2 Building consumption

The energy savings have been assessed based on simulation results by difference between reference cases (chiller + boiler) and Heat pumping case (reversible heat pump + backup boiler). The simulation results have been split as described in the following tables. It is assumed that Fan Coil Units represent 62% of the installed distribution systems and central and variable air volume systems represent 38%. Moreover, based on office building typology (See in Annex 1 of the ANNEXES to this report [STA, 2008]), it is supposed that 14% of office buildings are of type 1a, 33% of type 1c, 8% of type 2 and 25% of type 3. Since the analysis of the heating and cooling demands has shown that the type 1b is similar to type 2, the 20% of type 1b office buildings have been considered as type 2. When there is no information on ventilation rates, internal loads…, the equal probability of the simulation cases has been considered. Same assumptions have been made for health care institutions.

Table 35 : Share in % of simulated office buildings for assessing the average energy savings on the building stock

	Climatic zone					Internal loads		Ventilation rate		Air distribution 38%			Water distribution 62%					
	Paris	Torino	Athenes	Munich	Lisboa	low	high	low	high	type 1A VAV	Type 1C VAV	Type 1A CAV	Type 2 SF + FCU	Type 3 SF + FCU	Type 1A DF + FCU	Type 1C DF + FCU	Type 2 DF + FCU	Type 3 DF + FCU
Belgium	100							50	50									
Denmark				100				50	50									
Germany	50			50				50	50									
Greece			100					50	50									
Spain		40	30		30			50	50									
France	50	50						100										
Irland	100							50	50									
Italy		60	40			50	50		100	16	68	16	6.5	20.5	3	11	38.5	20.5
Luxembourg	100							50	50									
The Netherlands	100								100									
Austria	20			80				50	50									
Portugal					100			50	50									
Finland				100				50	50									
Sweden				100				50	50									
United Kingdom	80			20				50	50									

Table 36 : share in % of simulated health care institutions for assessing the average energy savings on the building stock

	Climatic zone					Internal loads		Ventilation rate		Air distribution	Water distribution		
	Paris	Torino	Athenes	Munich	Lisboa	low	high	low	high	type 1 CAV	Type 2 SF + FCU	Type 1 DF + FCU	Type 2 DF + FCU
Belgium	100							50	50				
Denmark				100				50	50				
Germany	50			50				50	50				
Greece			100					50	50				
Spain		40	30		30			50	50				
France	50	50						100					
Irland	100							50	50				
Italy		60	40			50	50		100	100	40	20	40
Luxembourg	100							50	50				
The Netherlands	100								100				
Austria	20			80				50	50				
Portugal					100			50	50				
Finland				100				50	50				
Sweden				100				50	50				
United Kingdom	80			20				50	50				

1.3 *Primary energy consumption*

Final energy consumption is transformed into primary energy consumption by means of a factor of 1.35 for fuel energy and 3.31 for electric energy according to [PrEN 15603, 2007]. These factors are assumed to be the same for all the European countries.

1.4 *CO_2 emissions*

For calculation of CO_2 emissions, electricity and fuel emission factors are reported in Table 37. The fuel emission factors are calculated on the basis of the relative proportion of gas and oil boiler in each country, given in [KEM, 2007], with a natural gas emission factor of 0,231 kg_{CO2}/kWh and a fuel oil one of 0,301 kg_{CO2}/kWh [ADE 2007]. The CO_2 emission factors of electricity stems from data published par IEA based on emissions due to electricity production for the reference year 2004 [ADE 2007]. More information is given in Annex 6 of the ANNEXES to this report [STA, 2008].

Table 37: CO_2 emission factor in Europe-15 countries

Country	Boiler emission factor	Electricity emission factor	Country	Boiler emission factor	Electricity emission factor
	Kg_{CO2} / kWh	Kg_{CO2} / kWh		Kg_{CO2} / kWh	Kg_{CO2} / kWh
France	0,279	0,084	**Ireland**	0,300	0,645
Germany	0,280	0,517	**Italy**	0,287	0,510
Austria	0,283	0,205	**Luxemburg**	0,266	0,304
Belgium	0,278	0,268	**Netherlands**	0,240	0,440
Denmark	0,279	0,334	**Portugal**	0,295	0,502
Spain	0,295	0,429	**UK**	0,273	0,455
Finland	0,266	0,253	**Sweden**	0,288	0,044
Greece	0,301	0,814			

2 Results

The results of the study are given in terms of potentials. The potentials correspond to primary energy (PE) savings and CO_2 emission reductions theoretically achievable through the adoption of reversible use of chiller for space heating in air-conditioned office building and in health care institution stocks or the heat recovery on water-cooled chillers for DHW production in health care institutions. The potentials are assessed by using the energy consumption calculations discussed in chapter V.

2.1 *Primary energy savings*

The average primary energy saving potential per unit of floor area (Figure 106 and Figure 107) depends mainly on the seasonal COP of the chiller in reversible mode and on the heating demand, both depending on the climatic zone.
Subsequently, primary energy saving potential in kWh/m² is larger in northern countries than in southern countries, as showed in Figure 106 and Figure 107. In the same way, the PE savings are greater in health care institutions than in office buildings.
Figure 108 shows the widespread range of PE savings by HVAC systems, by climatic zone and by office building types. The results show that the best potentials in kWh_{PE}/m^2 are for oceanic climate (Paris) and in the smallest buildings with the highest heating consumption (type 2 and 3). As a consequence single flux and fan coil unit system appears to be more interesting, but this is due to the fact that one assumed it was installed in "small" buildings only.
Figure 109 shows the widespread of PE savings by HVAC systems and by health care institution type. The potentials are better in large hospitals than in rest homes. More detail results are presented in Annex 7 and 8 of the ANNEXES to this report [STA, 2008].

Large reductions of Primary energy due to space heating of air-conditioned office buildings can be reached in Europe, in average of 37%. In air-conditioned health care institutions, an average of 22% of PE savings can be attained.

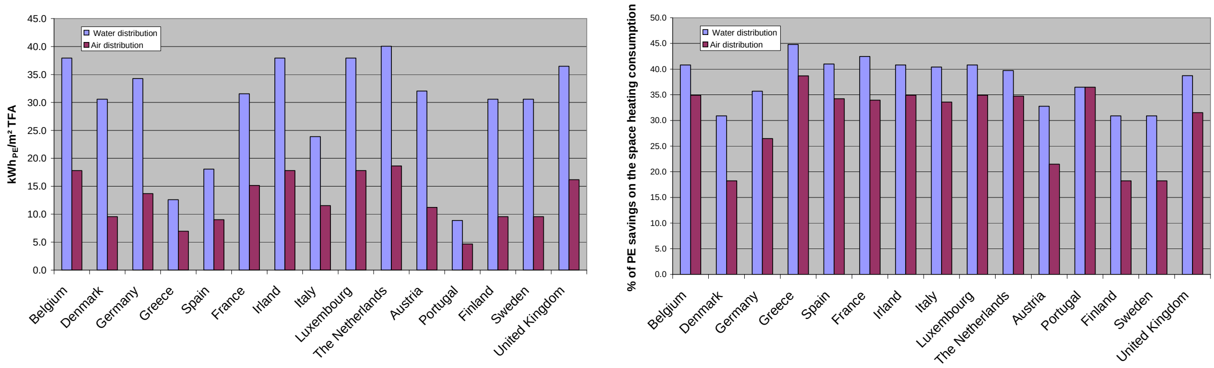


Figure 106: Potentials of annual primary energy savings by country and by distribution system in offices buildings (in kWh/m² TFA and in % on the space heating consumption of the reference case)

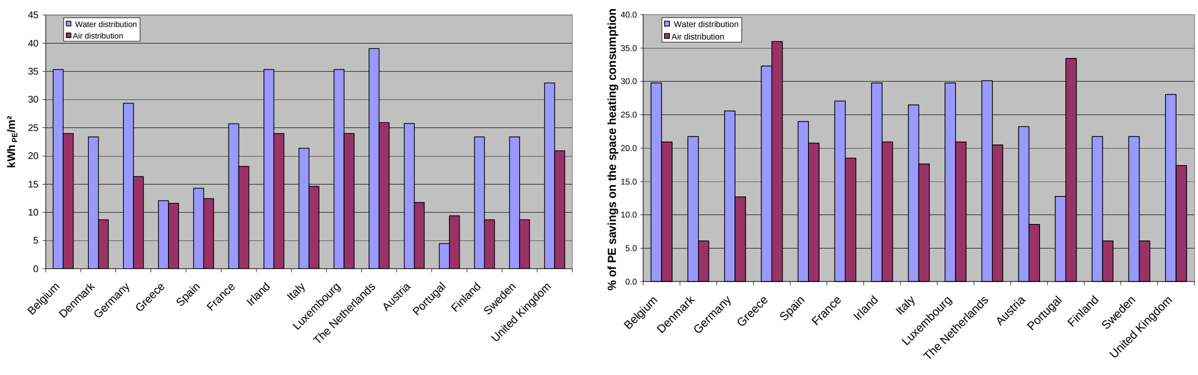


Figure 107: Potentials of annual primary energy savings by country and by distribution system in health care institutions (in kWh/m² TFA and in % on the space heating consumption of the reference case)

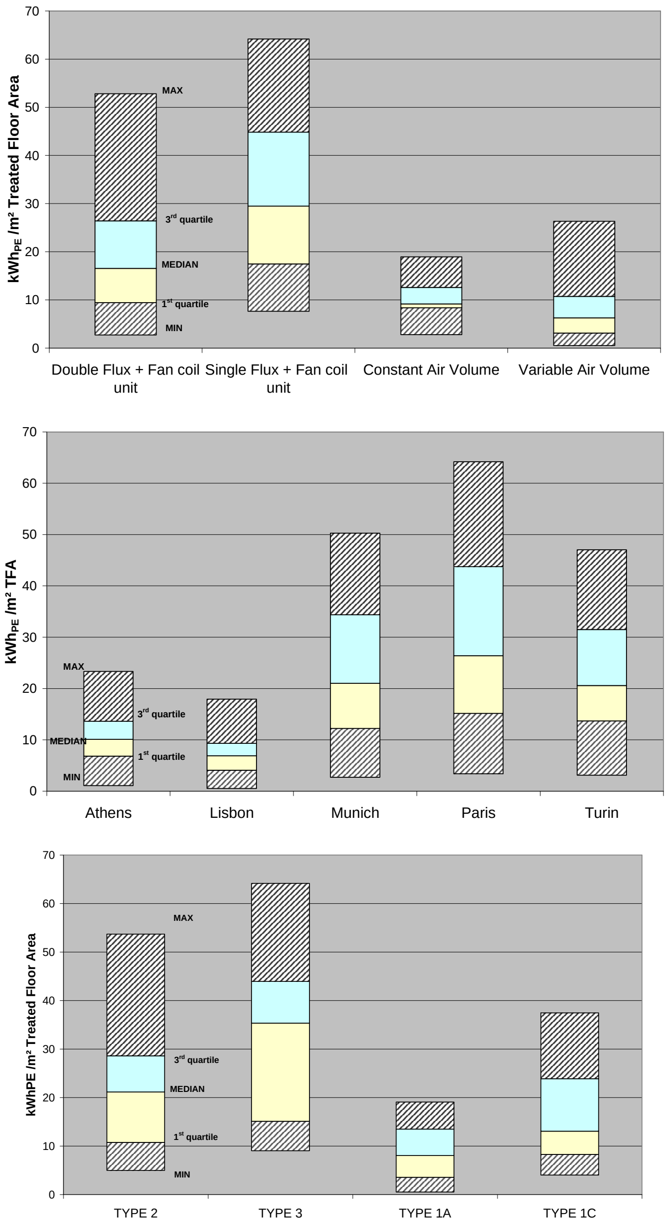


Figure 108: Range of annual primary energy saving potential per unit of floor area by HVAC system, by climatic zone and by office building type

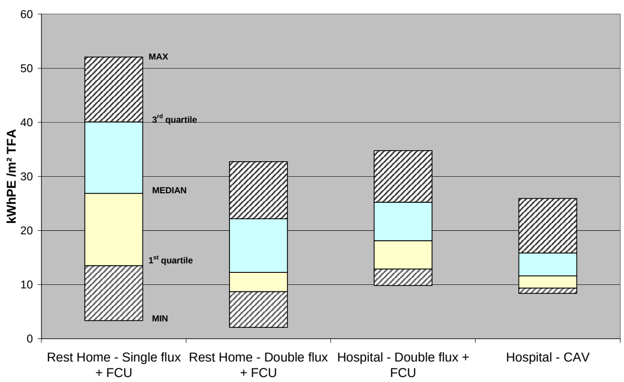


Figure 109: Range of annual primary energy saving potential per unit of floor area by HVAC system and by health care institution type

2.2 CO_2 *emission reduction*

The annual CO_2 emission reduction potential is affected by electricity emission factors of each country. Sweden is the country with the highest CO_2 reduction potential in kg CO_2/m^2, while Greece and Portugal present the lowest ones. In terms of percentage of CO_2 reduction, the potential is the best in France in office building and also in health care institutions (Figure 110 and Figure 111).

Large reductions of CO_2 due to space heating of air-conditioned office buildings can be reached in Europe, in average of 49%. In air-conditioned health care institutions, an average of 51% of CO_2 reduction can be attained.

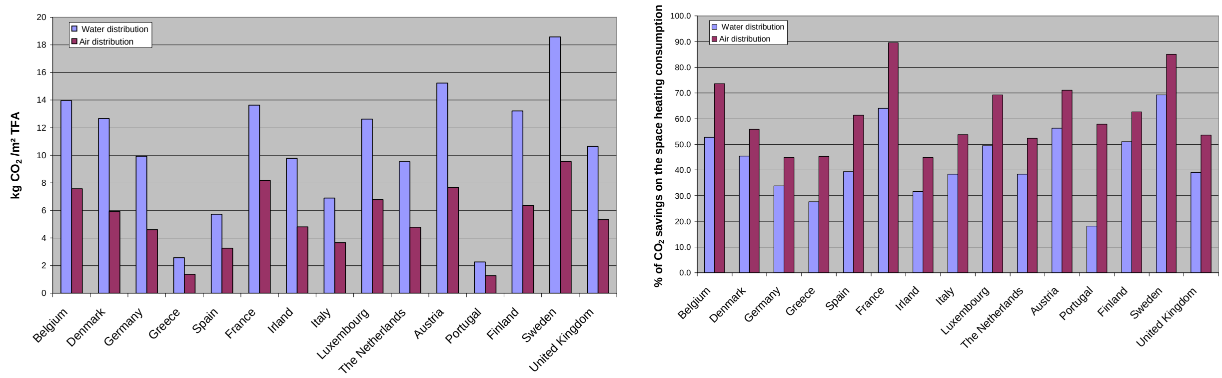


Figure 110 : Potentials of annual CO_2 reduction by country and by distribution system in offices buildings (in kg CO_2/m^2 Treated floor Area and in % on the space heating consumption of the reference case)

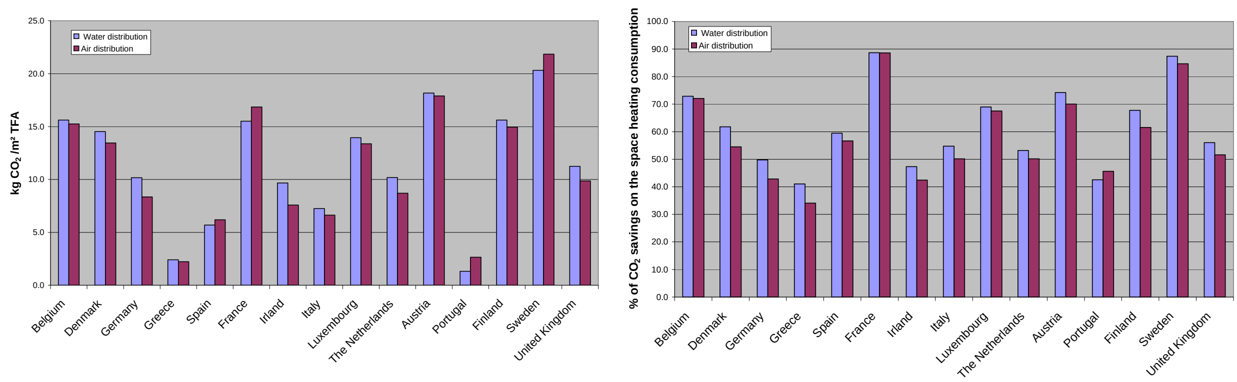


Figure 111: Potentials of annual CO_2 reduction per unit of floor area by country and by distribution system in health care institutions buildings (in kg CO_2/m^2 Treated Floor Area and in % on the space heating consumption of the reference case)

2.3 *Reversibility and heat recovery potentials in Europe*

Based on air-conditioned building stock data of 2005 [EEC, 2003] and simulation results, the **reversibility potential** can be estimated to be 7.8 $TWh_{\text{PE savings}}$ in air-conditioned office buildings (Figure 112) and 2.2 $TWh_{\text{PE savings}}$ in air-conditioned health care institutions (Figure 113) in Europe-15. In terms of CO_2 emissions, the annual saving potentials are of 2.7 millions of tons in air-conditioned office buildings and 1.06 millions of tons in air-conditioned health care institutions if all air-cooled chiller stock of 2005 in these sectors were replaced by reversible heat pumps.

The CO_2 emission reduction and PE saving results are however conservative since the simulated space heating consumption of the air conditioned building stock is underestimated.

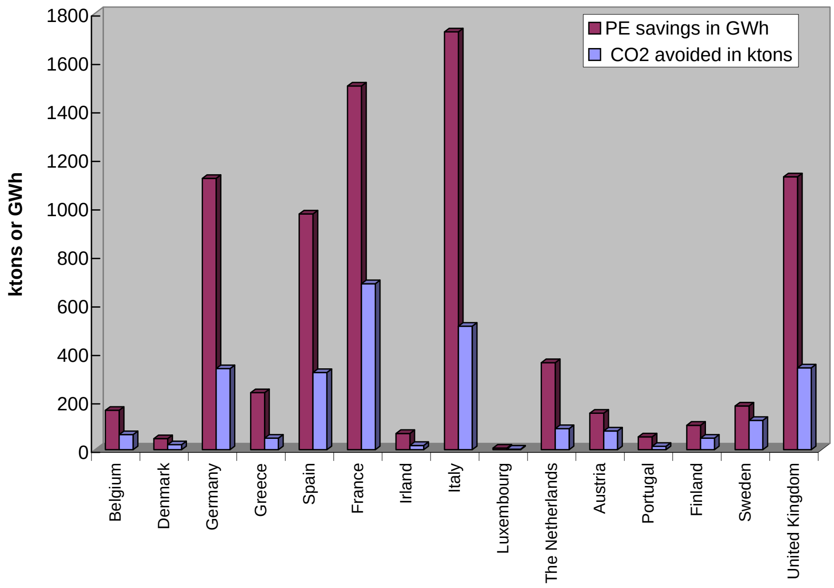


Figure 112: Annual primary energy savings and CO_2 emission reductions in air-conditioned office buildings in Europe-15 based on simulation results

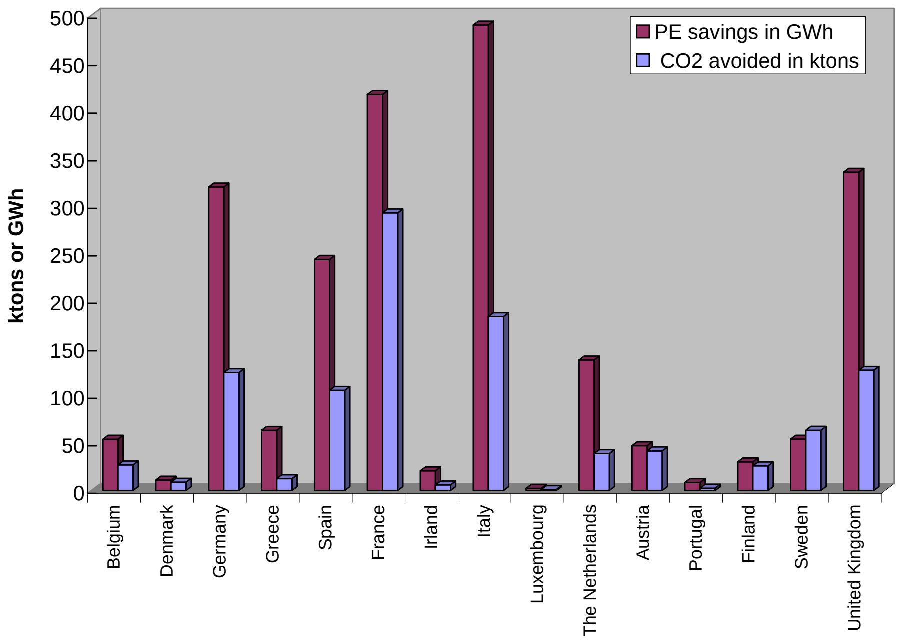


Figure 113: Annual primary energy savings and CO_2 emission reduction in air-conditioned health care institutions in Europe-15 based on simulation results

In terms of **heat recovery on chiller condenser for DHW** in health care institutions in Europe - 15, the potential is assessed to 430 GWh PE saved and 94 kilotons of CO_2 avoided (Figure 114).

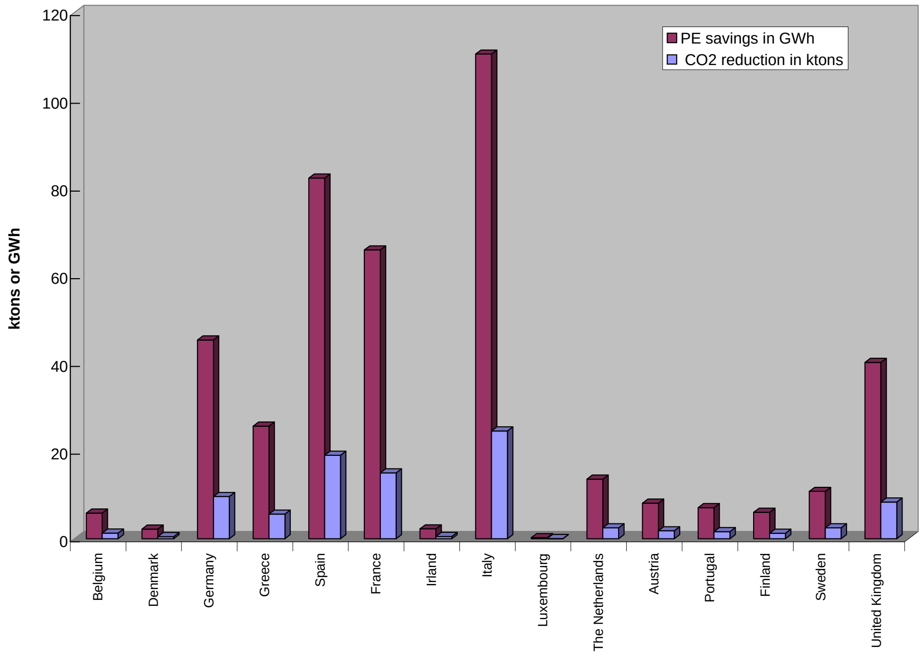


Figure 114: Annual primary energy savings and CO_2 emission reduction by heat recovery on chiller condenser for DHW in air-conditioned health care institutions in Europe-15 based on simulation results

The reversible use of the chiller for DHW production has been also studied for health care institutions. The savings of primary energy have been estimated to 370 GWh and the CO_2 emission reduction has been set to 160 kilotons (Figure 115). The PE savings are lower but the CO_2 reduction is higher since the reversibility potential is better in some countries of North Europe with low CO_2 content of electricity (mainly France).

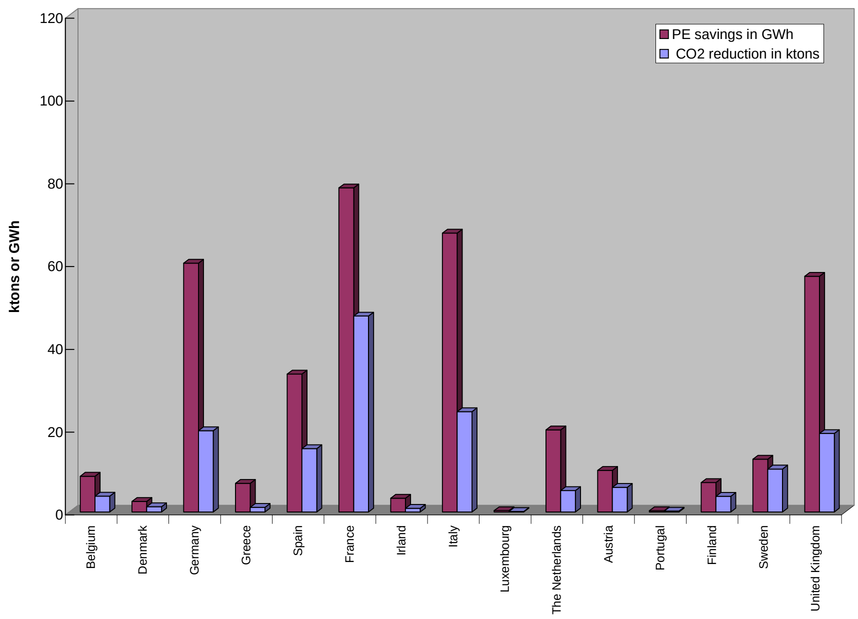


Figure 115: Annual primary energy savings and CO_2 emission reduction by reversible use of the chiller for DHW in air-conditioned health care institutions in Europe-15 based on simulation results

2.4 *Parametric study*

All previous analysis has been carried out supposing a COP of 2.75 at rating conditions. It has been shown in chapter IV that, even if the selected COP is the standard on the market, chillers with a nominal COP at rating conditions up to 4.5 are already available in the market.

As the COP is supposed to have a very large influence on reversibility potential, a sensitivity analysis has been achieved. Figure 116 and Figure 117 show the results of the analysis on European potentials.

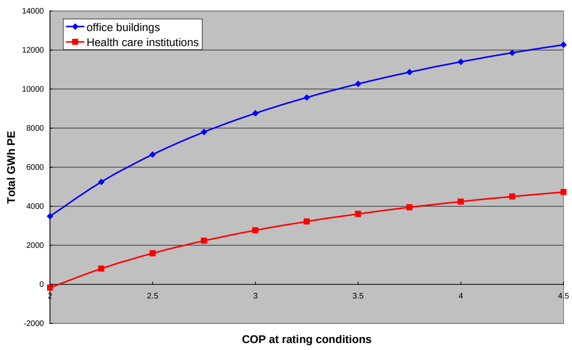


Figure 116: Variation of global primary energy potential in Europe-15 in dependence on COP of the chiller

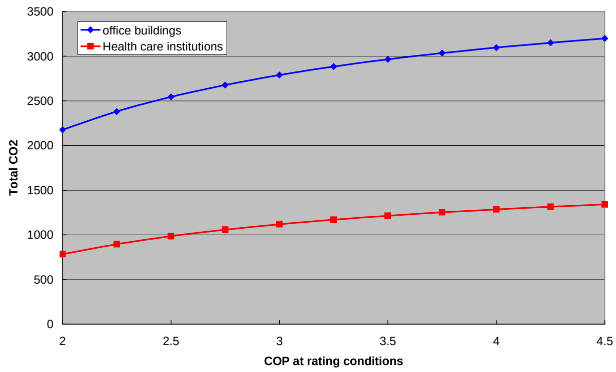


Figure 117: Variation of global CO_2 reduction potential in Europe-15 in dependence on COP of the chiller

As it could be expected, reversibility potential raise with better chiller performance. If all the chillers installed had a COP of 4, global European reversibility potentials would increase to about 11.4 TWh $_{\text{PE savings}}$ and 4.2 TWh $_{\text{PE savings}}$ in office buildings and in health care institutions respectively, and 3.1 and 1.3 millions of tons of CO_2 emissions reduction in office buildings and in health care institutions respectively.

As also boiler seasonal efficiency can be expected to increase, with the diffusion of condensing boilers and most efficient control systems, a last sensitivity analysis has been carried out in order to calculate the minimum average COP of chillers required to save primary energy or CO_2.

Figure 118 and Figure 119 show that, even at high boiler efficiencies, the minimum COP needed for primary energy savings lies in the order of medium standard chillers on the market, while minimum COP for a CO_2 emission reduction is very low. Notice that the COP values are assessed for water temperature at 45°C. If hot water distribution is set to 50°C or more, the required minimum COP at rating conditions would be higher. The seasonal COP has been assessed for each climate as an average of simulations (Table 38). Notice that the seasonal COP depends also on the heating demand of the building. The best seasonal COPs are obtained in Rest Homes. The reasons are that the heating capacity of the reversible chiller is low compared to the heating peak loads in rest homes and so the reversible chiller operates in heating mode more often in quite "warm" temperatures.

Table 38: Average seasonal COP of heat pump for different climates with a COP at rating conditions (40/45°C for water temperatures and 7°C for outside air temperature) of 2.75

Building type	Office buildings	hospitals	Rest homes
Climatic zone	SCOP		
Athens	3.0	2.6	3.3
Lisbon	3.1	2.9	3.5
Turin	2.6	1.8	2.1
Paris	2.7	2.2	2.8
Munich	2.1	2.1	2.5

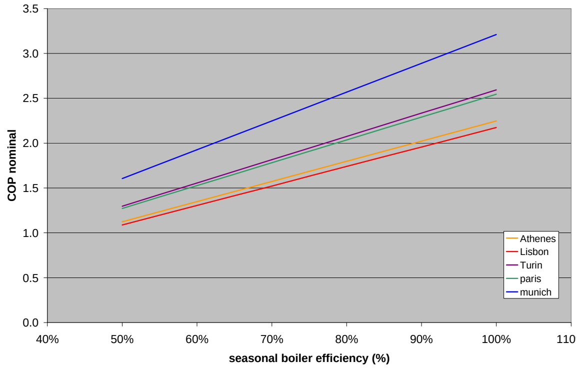


Figure 118 : Minimum COP required for primary energy savings versus seasonal boiler efficiency by climate (in office buildings)

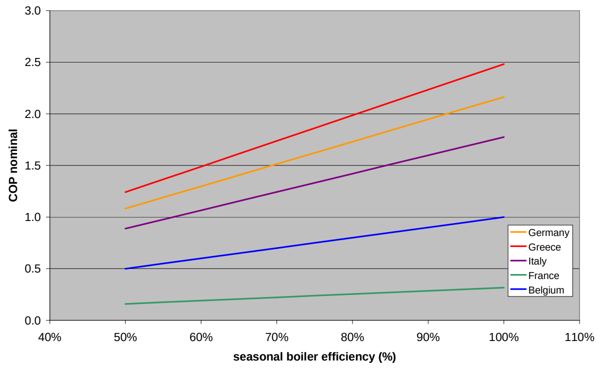


Figure 119: Minimum COP required for CO_2 emission reduction versus seasonal boiler efficiency for few countries (in office buildings)

CONCLUSIONS

In most of the studied cases, a reversible chiller can not cover all the heating demand mainly due to simultaneity of cooling and heating demand. A back up boiler is in almost every case still necessary. However, by an efficient use of free cooling, the boiler could be suppressed in most of the cases since the backup boiler is used essentially when simultaneous heating and cooling demands occur, in other terms when the outside temperature is not very high. In cold climates, a boiler is still required since most of the air-cooled heat pumps cannot operate at low outdoor temperatures (generally below –10°C). Furthermore, the performance of air-cooled heat pumps decreases strongly when the outdoor temperature decreases such as the use of the boiler can become preferable again.

The potential of primary energy savings and CO_2 emission reduction in European office buildings by using reversible chillers with COP of 4 would be higher than 8.8 TWh and 2.5 millions of tons, respectively. In health care institutions, PE savings of 4.2 TWh and CO_2 avoided of 1.3 millions of tons could be reached.

The heat recovery potential appears to be interesting only for DHW production in health care institutions. The primary energy savings on DHW production have been assessed between 55% and 85% in large hospitals and between 15% and 33% in rest homes. The heat recovery potential for space heating appears to be low in the simulated office and health care institution cases but could be interesting when a data centre exists [SCH, 1999].

The potential of primary energy savings and CO_2 emission reduction in European health care buildings by heat recovery on chiller condenser for DHW would be higher than 0.4 TWh and 0.1 million of tons, respectively.

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