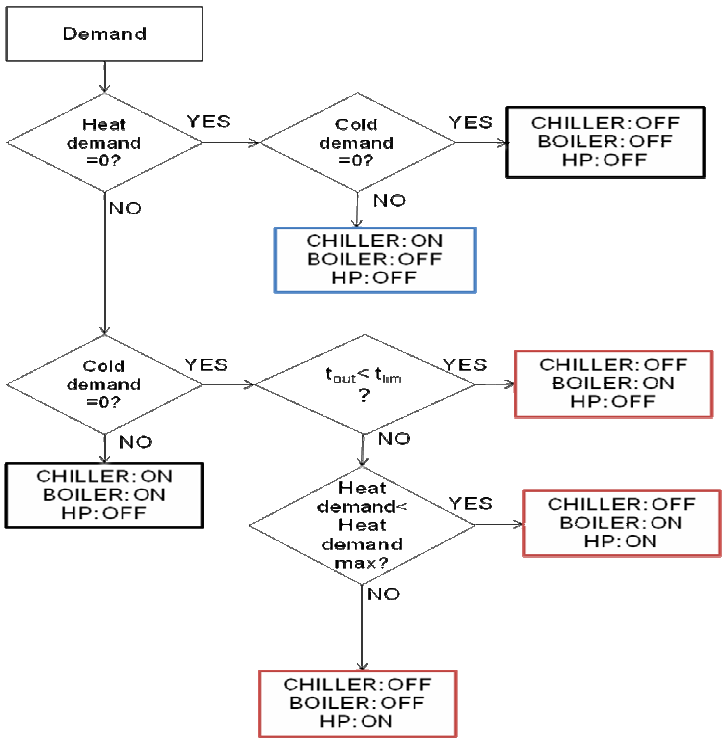
IEA-ECBCS ANNEX 48

Reversible Air-Conditioning

International Energy Agency
Energy Conservation in
Buildings and Community
Systems Programme

IEA 48 Simulation tools: Reference book



Authors :
Stéphane BERTAGNOLIO, Université de Liège, Belgium
Samuel GENDEBIEN, Université de Liège, Belgium
Benjamin SOCCAL, Université de Liège, Belgium
Pascal STABAT, Mines ParisTech, France

Internal Reviewers :
Arno DENTEL, IEG, Nürnberg, Germany

Date : 2011-05-24

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 presents the simulation tools specifically developed to assess the performance of reversibility or recovery-based heat pumping solutions.

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

Philippe ANDRE
Editor

Operating agents :
Jean LEBRUN, Laboratoire de thermodynamique – Campus du Sart-Tilman B49 (P33) B-4000 Liège (j.lebrun@ulg.ac.be)
Philippe ANDRE, Département des sciences de l'environnement – Avenue de Longwy, 185 B6700 Arlon (**p.andre@ulg.ac.be**)

Subtask leaders :
Dominique MARCHIO, Pascal STABAT, Ecole des Mines de Paris, France, 60 boulevard Saint Michel, 75272 Paris Cedex 06 (**dominique.marchio@ensmp.fr**, **pascal.stabat@ensmp.fr**)

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.

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.

Participants:

Université de Liège, Belgique

J.Lebrun
Co-operating Agent
S.Bertagnolio
Laboratoire de Thermodynamique
Campus du Sart-Tilman B49 (P33)
B-4000 Liège
mail : j.lebrun@ulg.ac.be
stephane.bertagnolio@ulg.ac.be
web : www.labothap.ulg.ac.be

P.André
Co-operating Agent
Département des sciences de l'environnement
Avenue de Longwy, 185
B-6700 Arlon
mail : p.andre@ulg.ac.be
web : http://www.dsge.ulg.ac.be/arlon/

CEA – INES, France

D. Corgier, V; Renzi
Co - Leader of Subtask 1
CEA – INES RDI
Laboratoire d'Intégration Solaire
Savoie Technolac - BP 332
50 Avenue du Lac Léman
F - 73377 Le Bourget du Lac
mail : david.corgier@cea.fr
virginie.renzi@cea.fr
web : www.cea.fr

Mines ParisTech, France

D.Marchio, P.Stabat
Co - Leaders of Subtask 1
Centre Énergétique et Procédés -
Mines ParisTech
60 Bd St Michel
F - 75272 Paris Cedex 06
mail : dominique.marchio@ensmp.fr
pascat.stabat@ensmp.fr
web : www.cep.ensmp.fr

Greth, France

B. Thonon
Leader of Subtask 5
Greth
Savoie Technolac - BP 302
50 Avenue du Lac Léman
F - 73377 Le Bourget du Lac
mail : bernard.thonon@greth.fr
web : www.greth.fr

Georg-Simon-Ohm-Hochschule Nürnberg, Germany

W. Stephan
Co – Leader of Subtask 2
Ieg Intstitut für Energie and Gebäude
Kesslerplatz 12
D – 90489 Nürnberg
mail : wolfram.stephan@t-online.de

HLK Stuttgart GmbH, Germany

J. Schmid
Co – Leader of Subtask 2
HLK Stuttgart GmbH
Pfaffenwaldring 6A
D – 70569 Stuttgart
mail : joerg.schmid@hlk-stuttgart.de
web : www.hlk-stuttgart.de

TEB GmbH, Germany

M. Madjidi, T. Dippel
Co – Leaders of Subtask 2
Transferzentrum Energieeffizientes Bauen GmbH
Kehlstr. 27/1
D – 71665 Vaihingen/Enz
mail : info@madjidi.de
dippel@teb-online.de
web : www.teb-online.de

Politecnico di Torino, Italy

M. Masoero
Leader of Subtask 4
Politecnico di Torino
mail : marco.masoero@polito.it
web : www.polito.it

INSA Rennes, France

P. Byrne
Laboratoire de Génie Civil et de Génie Mécanique
Equipe Matériaux et Thermo-Rhéologie
20, Avenue des Buttes de Coësmes
CS 70839
F - 35708 Rennes Cedex 7
mail: paul.byrne@insa-rennes.fr
web: www.insa-rennes.fr

Table of contents

I. Introduction

The aim of the project is to promote efficient solutions of heat pumping in commercial buildings in order to save primary energy and reduce CO2 emissions. The project targets are the retrofit of existing buildings and the design of new buildings.

The main goals of the first subtask of the Annex 48 project, *Analysis of building heating and cooling demands and of equipment performances*, are:

1. To evaluate the potential of heat recovery and reversible use of air conditioning systems, in order to save energy and to reduce GHG emissions;
2. To deliver simulation models for those systems;
3. To develop a simplified tool which could be used by decision makers in order to evaluate the energy and economic potential of these solutions for their buildings, while entering a limited number of easily accessible parameters.

The document presents the specifications of the tools developed in the framework of IEA-ECBCS Annex 48. Two types of tools are proposed:

- A heat recovery and reversibility potential assessment tool; This tool aims to help practitioners and decision makers in identifying the interest of using a reversible heat pump solution.
- An energy and economic assessment tool of the selected heat pump systems. This tool aims to allow quick and easy assessment of selected heat pump systems

The First Assessment Tool (FAST) is a webtool relied on the results of an extensive parametric study described in [1].

The energy and economic assessment tool is based on a set of HP systems simulation models implemented in an equation solver [2].

II. *FAST* ("First Assessment Tool")

1. General purpose

A web tool has been developed in the aim to give a fast evaluation of the potential of reconverting a chiller in a reversible unit in commercial buildings. This tool is addressed to stakeholders, HVAC consultants, architects and installers. This tool can be used in retrofit buildings but also in the case of new buildings.

The online tool allows the user to define in few clicks the main parameters concerning the building, the climate, the HVAC system installed and its current performance, the heat pump performance to install and the heating consumption of the building. Some default values for economic and environmental parameters are provided to the user but they can be changed at any time.

The results give an overview on the fuel consumption reduction, primary energy, CO2 and costs savings if any.
The results are based on the reading of a database of simulation results carried out in the subtask 1 of this Annex [1].
The tool cannot replace a deep analysis of the potential of primary energy, CO2 and costs savings. A design guide and a simulation tool have been developed in the framework of this annex to help the designers in going further in the assessment of heat pump solutions in commercial buildings.

This Web tool is available on the http://www.greth.fr/ecbcs-48/outil.php and a link with the web page http://www-ecbcs-48.org will be added soon.

2. Methodology

This tool relies on the results of an extensive parametric study described in the report 1 of the subtask 1 of this annex [1].

The methodology can be described step by step:

Step	Inputs	Intermediate or final outputs
1	Select among the library of building cases the one which is the closest of the building that the user want to study	• Potential fuel savings in % of total fuel consumption; • Ratios of seasonal performance on rating performance: o Air-to-water and Water-to-water HP, o Standard and low temperature boilers
2	Define the present heat production (fuel type, fuel consumption, the boiler type and boiler efficiency) and select a heat pump system (HP type, COP)	• Calculation of fuel savings based on fuel consumption • Calculation of electricity consumption of the heat pump in heating mode
3	Provide environmental and cost parameters	Calculation of primary energy, CO_2 and costs savings

2.1. Step 1: Selection of the building and its associated HVAC system

A parametric study on several commercial buildings has been carried out by using a dynamic building energy simulation tool called Consoclim [2]. This study covers typical office and health care buildings with classical HVAC systems in different climatic zones. The simulation assumptions are described in [1].

The parametric study includes:

- 5 offices buildings
    - TYPE 1: Large building with deep plan in open space; with 12 storeys, glazing ratio[1] of 50% and 15 000 m² floor area;
    - TYPE 2: Large building with shallow plan in a cellular layout; with 12 storeys, glazing ratio of 50% and 15 000 m² floor area;
    - TYPE 3: Medium building with deep plan in a cellular layout; with 4 storeys, glazing ratio of 27.5% and 5 000 m² floor area;
    - TYPE 4: Small building with deep plan in a cellular layout; with 2 storeys, glazing ratio of 34% and 1 000 m² floor area;
- 2 heath care buildings
    - LARGE HOSPITAL: Large building with deep plan in a cellular layout; with 5 storeys, glazing ratio of 27.5% and 30 300 m² floor area;
    - REST HOME: Medium building with deep plan in a cellular layout; with 4 storeys, glazing ratio of 30% and 3 900 m² floor area;
- 4 HVAC systems:
    - Fan Coil Unit and Mechanical Extraction Ventilation (only in small buildings: OFFICES TYPE 3 and 4, and REST HOME);
    - Fan Coil Unit and Mechanical Balanced Mixing Ventilation (all building cases);
    - Build-up Variable Air Volume including heat recovery, pre-heating coil, cooling coil and heating coil (only in large office buildings: TYPES 1 and 2);
    - Build-up Constant Air Volume including return air damper, pre-heating coil, cooling coil and heating coil (only in large buildings: Large Hospital and OFFICE TYPE 1);
- Two ventilation rates:
    - LOW: 25 m³/h/pers in offices and bedrooms and 30 m³/h/pers in conferences rooms
    - HIGH: 40 m³/h/pers in offices and bedrooms and 40 m³/h/pers in conferences rooms
- Two internal load levels in office buildings only:
    - LOW: 10W/m² of installed lighting power and 7.5W/m² of installed electric appliances power;
    - HIGH: 18W/m² of installed lighting power and 15W/m² of installed electric appliances power;

Then, five European climatic zones have been defined based on heating and cooling degree-days [1].

[1] % of vertical surface of the building

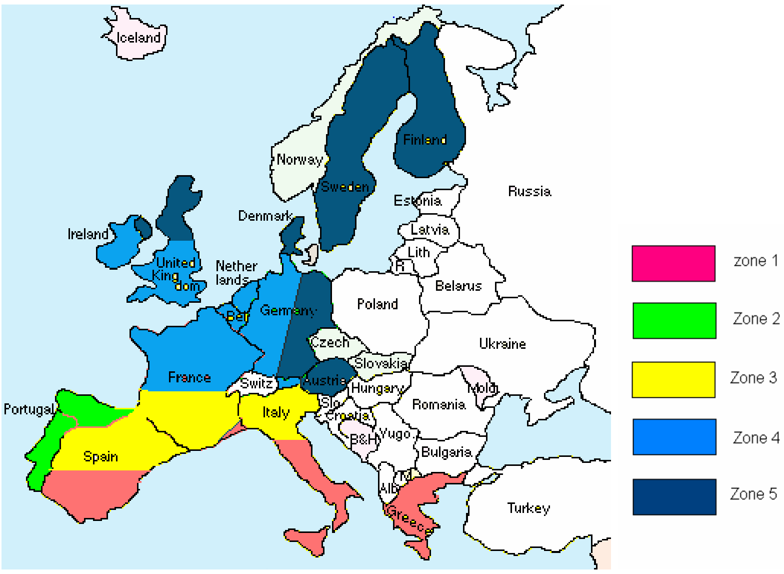


Figure 1: European climatic zones

Two series of simulations have been achieved, one with a conventional system (boiler for heating and chiller for cooling) and another one with the use of a reversible heat pump for cooling and heating sized on cooling demand (a back up boiler is used in case of peak heating loads or simultaneity of cooling and heating demands).
The heating capacity of the reversible heat pump at rating conditions has been assessed to 1.1 times those of the cooling capacity at rating conditions.
In case where the reversible heat pump sized on cooling were too oversized compared to the heating power required (the installed cooling power can be more important in some buildings in the south of Europe than the installed heating power), the heat pump is sized as follows:

$$\text{IF } (1.15 \times P^{max}_{heating} < 66\% \times 1.1 \times P^{max}_{cooling}) \quad \text{THEN} \quad P^{HP}_{heating} = 1.15 \times P^{max}_{heating} \quad \text{ELSE} \quad P^{HP}_{heating} = 1.1 \times P^{max}_{cooling}$$

This means if the reversible HP sized on cooling capacity has too high heating capacity, we would split it in two units, a reversible unit sized for heating and a chiller to bring the extra peak cooling power. A sizing with a security factor of 1.15 at minimum is assumed. Furthermore, the factor 1.1 corresponds to an average ratio between the heating capacity at rating and the cooling capacity at rating for air-to-water heat pump. For water-to-water HP, this average ratio is around 1.15.

The energy savings are then assessed by comparing the conventional system (boiler + chiller) to the use of a reversible heat pump on reference yearly weather data.

To use the tool, the user should first select the building characteristics among those proposed in the database, which are the closest to the user's building case. Then the user should enter the annual fuel consumption and the building floor surface. The percentage of fuel savings extracted from the database for the selected case is applied to the real annual consumption of the user's building.
However, there is behind the database, several assumptions:

- Reference weather data in five cities representatives of European climates;
- Default values for thermal insulation of buildings;
- Main orientation of buildings: north/south;
- Default values for thermal inertia;
- Default set points: 21°C in heating, 24°C in cooling;
- Default values solar protection value and scenario of use;
- Standard occupation, appliance use and lighting scenarios;

The further the building use from these assumptions the worse the assessment of the fuel savings will be. In particular, this FAST tool is not recommended to be used for:

- Buildings without cooling demands;

- Buildings with high simultaneous heating and cooling demands (including large data centres for instance) where the heat recovery on chiller condenser could be an opportunity;
- Buildings with specific occupation (for instance, week end or night occupation in office building types);

2.2. Step 2: Definition of the heating system

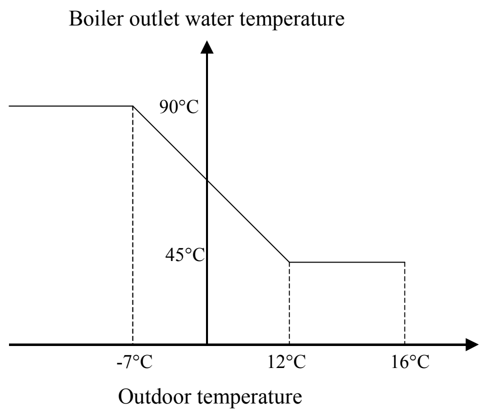
2.2.1. Boiler

The user should enter the performance characteristics of its current boiler. A selection between two types of boiler is proposed:

- A standard boiler operating at constant temperature implying a performance degradation at part load;
- A low temperature boiler with temperature regulation maintaining a good efficiency at part load up to 20% of load.

Then, the user should enter the full load efficiency of the boiler at 70°C based on Lower Heating Value.
The boiler options do not include the condensation boilers where the performance at part load increases slightly before decreasing at low part load (<20%).

The heating water distribution in the building is assumed to vary with outdoor temperature according to the following regulation law:



The table below shows the difference in terms of seasonal performance between the low temperature and the standard boiler for few office building cases.

Table 1: Ratio of seasonal boiler efficiency on rating boiler efficiency in the case of an office building

	Paris	Torino	Athens	Munich	Lisbon
Standard boiler	0.77	0.76	0.75	0.75	0.73
Low temperature boiler	0.95	0.94	0.96	0.95	0.96

2.2.2. Heat pump

Concerning the heat pump to install, two options are proposed:

- Air-to-water heat pump;
- Water-to-water heat pump;

The air-to-water heat pump is easy to install but its performance decreases when outdoor temperature decreases. The choice of a water-to-water heat pump depends on the possibility to install buried pipes in the ground or to take profit of a water source.

The COP of the heat pump should be given at rating conditions, that is:

- 7°C outdoor air temperature, 40/45°C water temperature for air-to-water HP;
- 10°C water source temperature, 40/45°C water temperature for water-to-water HP;

The air-to-water performance at full load decreases according a typical performance curve of a HP, quite linearly with the outdoor air temperature decrease (Figure 2). Same trend is found for a water-to-water HP (Figure 3).

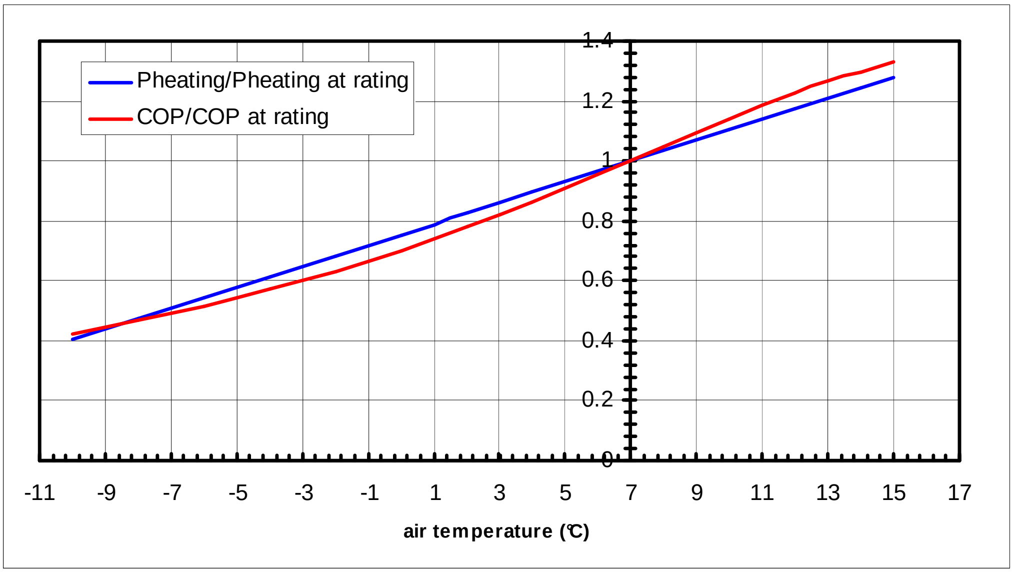


Figure 2: Variation of heating capacity and COP of the air-to-water HP with outdoor temperature

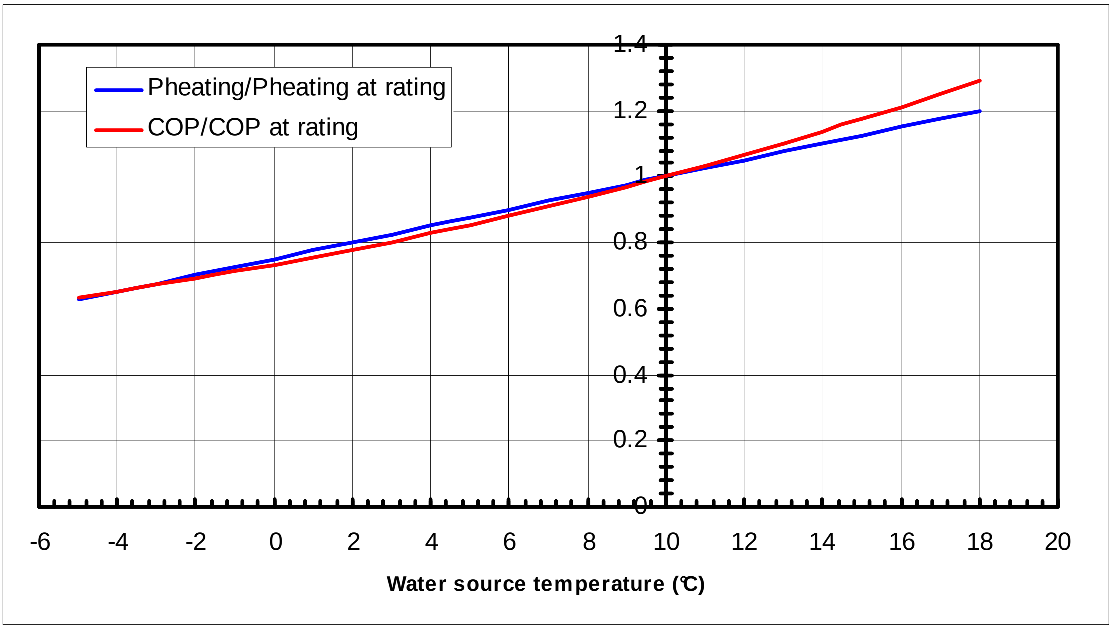


Figure 3: Variation of heating capacity and COP of the water-to-water HP with water temperature

The choice of the rating COP should be done by the user. In order to help him in selecting a value, a study of the heat pump performances has been carried out based on the EUROVENT DATABASE (www.eurovent-certification.com). The following figures show the breadth of COPs of HP by capacity range.

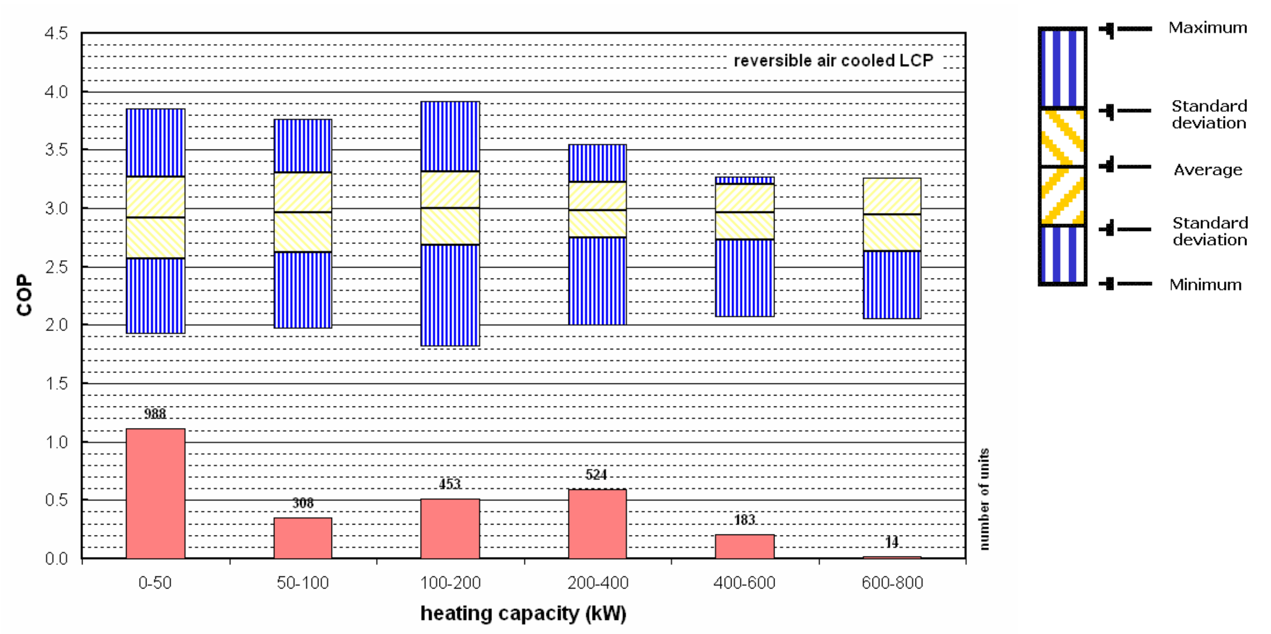


Figure 4: COP of air-to-water heat pumps according to the heating capacity range (EUROVENT database 2009)

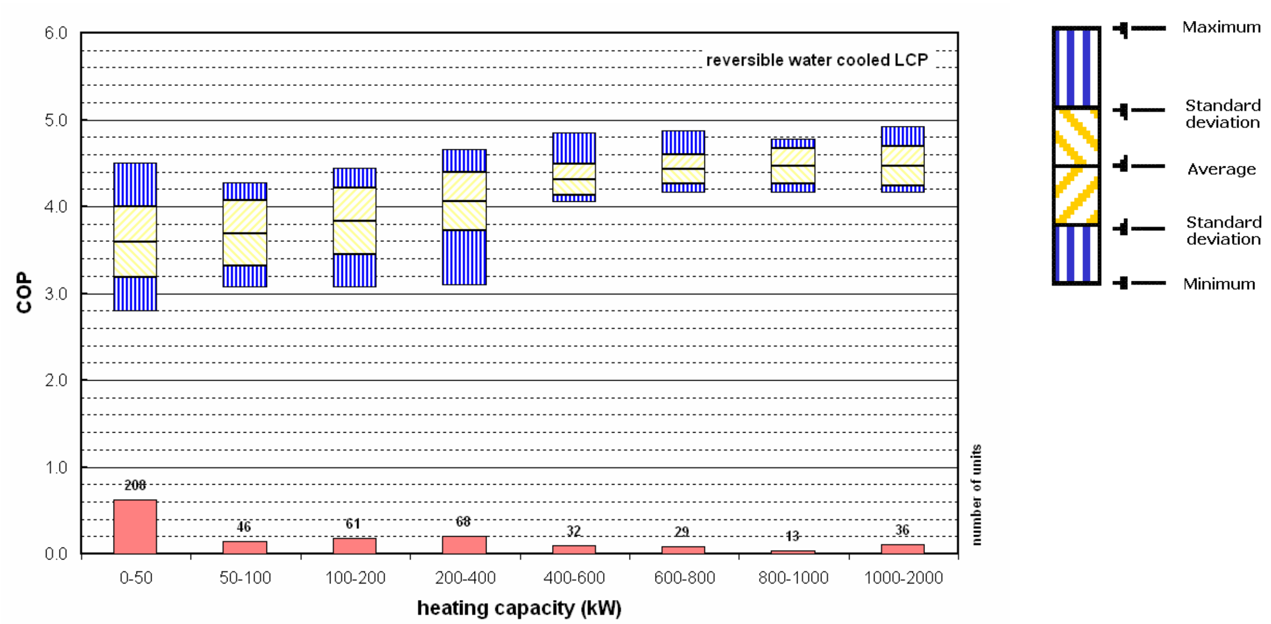
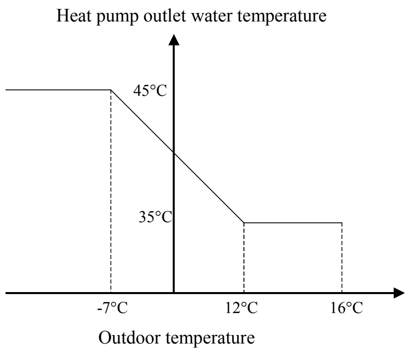


Figure 5: COP of water-to-water heat pumps according to the heating capacity range (EUROVENT database 2009)

The heat pump is assumed to provide heating water in the building with a water temperature regulation law as follows:



Thus, the tool is focused on medium temperature emitters such as fan coil units. The uses of low distribution emitters such as radiant ceiling/floor would increase the seasonal performance of the heat pump but it is not implemented in the tool.

In the case of water-to-water HP, it is assumed that the HP is coupled to ground by buried pipes. An infinite cylindrical source model is used in order to determine the temperature difference between the borehole radius and the undisturbed ground associated with a multiple load aggregation algorithm [3]. Here below is presented the variation of the water temperature exiting the borehole during a reference year. The undisturbed ground temperature is taken as the average air temperature for each climatic zone.
This borehole exit profile is used to determine the seasonal performance of the heat pump. Notice that the profile of the borehole exit water temperature depends on the sizing of the boreholes.

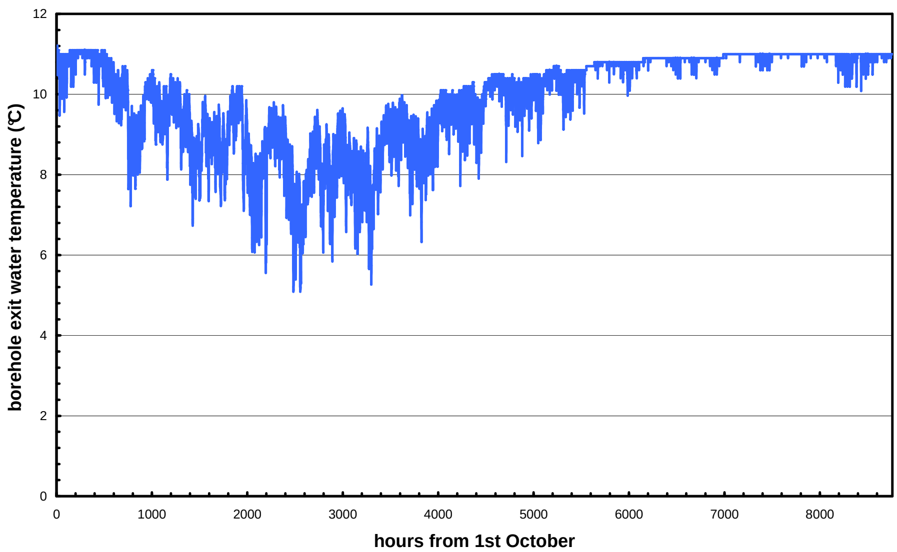


Figure 6: Profile of borehole exit water temperature along a reference year in the case of a hospital in Paris

At last, the performance of HP at part load is characterized. The performance curves at part load vary from a technology to another (variable speed compressor, bi compressor, On/Off). A default curve has been defined for the air-to-water HP considering a bi-compressor since the variable speed compressor is not so common for high capacity heat pumps. (Figure 7) An ON/OFF heat pump would have generally lower part load performance than a bi-compressor HP. Concerning the water/water HP, a part load factor versus part load ratio (Figure 8) has been defined according to [4].
It should be noticed that the sizing of the heat pump has a great importance since an oversizing can degrade strongly the seasonal performance.

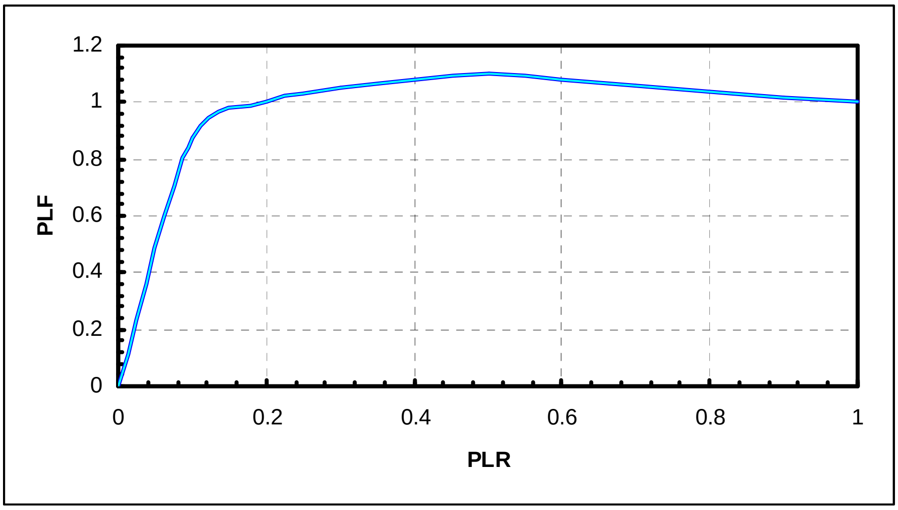


Figure 7: Part load factor versus part load ratio for the air-to-water HP (assumption of bi-compressor heat pump)

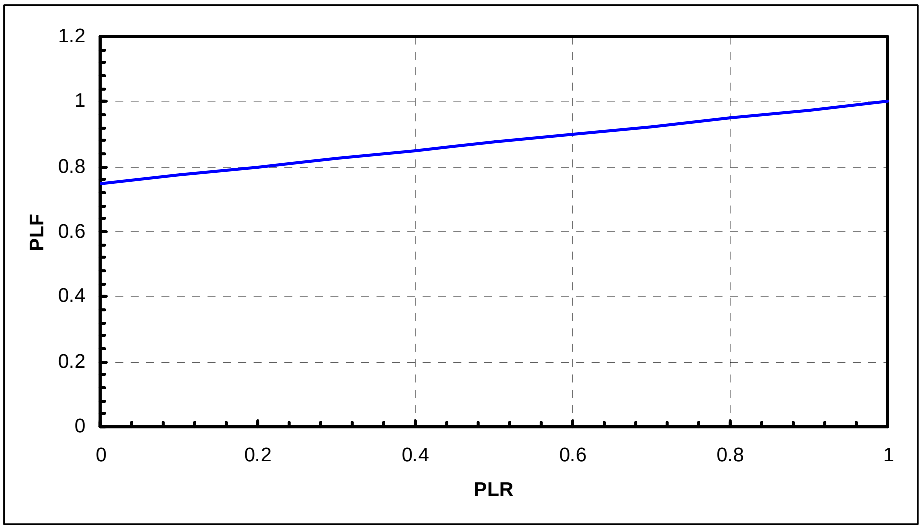


Figure 8: Part load factor versus part load ratio for the water-to-water HP

3. Results of the FAST tool

From the assumptions described here above on the building, HVAC system parameters, the tool estimates the primary energy savings, CO_2 emissions savings and the costs savings using primary energy factors, CO_2 emission factors and energy costs.
Some reference data are proposed to the user for all these factors (see following tables).

Table 2: Reference data for CO2 content of electricity and fuel

Average emission factors of electricity production by country [5]		
Austria	0.205	$kgCO_2/kWh$
Belgium	0.268	$kgCO_2/kWh$
Denmark	0.334	$kgCO_2/kWh$
Finland	0.253	$kgCO_2/kWh$

France	0.084	$kgCO_2/kWh$
Germany	0.517	$kgCO_2/kWh$
Greece	0.814	$kgCO_2/kWh$
Ireland	0.645	$kgCO_2/kWh$
Italy	0.510	$kgCO_2/kWh$
Luxemburg	0.304	$kgCO_2/kWh$
Portugal	0.502	$kgCO_2/kWh$
Spain	0.429	$kgCO_2/kWh$
Sweden	0.044	$kgCO_2/kWh$
The Netherlands	0.440	$kgCO_2/kWh$
United kingdom	0.455	$kgCO_2/kWh$
Emission factor of electricity for space heating in France [6]		
average content	0.180	$kgCO_2/kWh$
marginal content	0.500 to 0.600	$kgCO_2/kWh$
Emission factors of fossil fuels including extraction, treatment and transport [7]		
fuel oil and gasoil	0.301	$kgCO_2/kWh$
natural gas	0.231	$kgCO_2/kWh$
Emission factors of electricity and fuels according to [8] (these factors include the transformation and the transportation)		
fuel oil and gasoil	0.330	$kgCO_2/kWh$
natural gas	0.277	$kgCO_2/kWh$
electricity Mix UCPTE	0.617	$kgCO_2/kWh$

Table 3: Reference data for primary energy factor of electricity and fuel

Primary energy factors of fossil fuels and electricity according to [8]	
fuel oil energy	1.35
gas energy	1.36
electricity	3.31
Primary energy factors of fossil fuels and electricity in France[9]	
fuel energy	1.0
electricity	2.58
Primary energy factors of electricity according to ESD directive [10]	
electricity	2.5

Table 4: Example of average prices of energy in France [11]

Electricity (1000 kW during 3000h)	0.0756	€/kWh excl. taxes	(February 09)
Electricity (10000 kW during 6000h)	0.0583	€/kWh excl. taxes	(February 09)
Gas	0.039	€/kWh(HHV) excl. taxes	(February 09)
Domestic Fuel (2000 to 4999 litres)	39.01	€/hl excl. taxes	(May 09)
Domestic Fuel (> 27000 litres)	34.78	€/hl excl. taxes	(May 09)

First of all, the tool calculates the heating consumption of the building in kWh/m^2 in order to give an overview of the building efficiency and check if the inputs are coherent. A cursor gives just an indication of the building heating consumption. The thresholds are defined based on benchmarking values with some corrections by climatic zone and by sector. The correction factors applied are issued from the parametric simulation study (table 5).

Table 5: Position of the cursor of heating consumption

CURSOR (kWh/m²)	10	VERY LOW	30	LOW	80	MEDIUM	150	HIGH	200	VERY HIGH	300
Reference for office in climatic zone 4		Passiv Haus [12] (15 kWh/m²)				Average in France [13] (150 kWh/m²) Good practice office buildings [14] (79 –107 kWh/m²)		Typical office buildings [14] (151 –201 kWh/m²)			
Correction for climatic zone 1	-65%										
Correction for climatic zone 2	-70%										
Correction for climatic zone 3	-20%										
Correction for climatic zone 5	+10%										
Correction for health care	+85%										
Example for health care in zone 4	18.5		55.5		148		277.5		370		555
						Good practice hospitals [14] (215 – 322 kWh/m²)		Typical hospitals [14] (249 – 345 kWh/m²)			

Figure 9: Example of "results" web page

Then, the webtool assess the potential primary energy, costs and CO_2 savings based on the following formulas.

The HP electric consumption for heating is calculated as follows:

Electric_overconsumption (kWh) = Fuel_savings (kWh) × Seasonal_boiler_efficiency / Seasonal_COP

The seasonal boiler efficiency and the seasonal COP are calculated as their rating values corrected by a seasonal part load factor issued from simulation results.

The bill PE and CO_2 are then calculated as follows:

Bill_savings(€) = Fuel_savings (kWhLHV) × 1.111 (HHV/LHV) × Fuel_costs (€/kWhHHV)
- Electric_overconsumption(kWh) × Electricity_costs (€/kWh)

PE_savings(kWhPE) = Fuel_savings (kWhLHV) × Fuel_PE_factor
- Electric_overconsumption (kWh) × Electricity_PE_factor

CO2_savings(kg CO2) = Fuel_savings (kWhLHV) × Fuel_CO2_factor (kgCO2/kWhLHV)
- Electric_overconsumption (kWh) × Electricity_CO2_factor (kgCO2/kWhLHV)

If the values are negatives, this means that there is no saving but wasteful. The interest of heat pumps depends of the comparative values of the seasonal HP COP and the seasonal boiler performance. The comparative costs of electricity and fuel (gas or fuel oil) can also influence strongly the choice to install a HP or not.

4. Conclusion

This tool gives an indication of the primary energy, CO2 emissions and costs savings potentials which can be made by using a "reversible" air-to-water or water-to-water heat pump by typing 16 inputs only. The results are based on pre-selected values of the building and HVAC system characteristics, which differ more or less from the user's case.

This webtool aims to encourage stakeholder to go further on the assessment of heat pump solutions and invite them to use the design handbook and the more detailed tools developed in the framework of this annex.

References :

1. Stabat P. et Al. (2009). *Analysis of heating and cooling demands in the prupose of assessing the reversibility and heat recovery potentials*. Report 1 of subtask 1, IEA-ECBCS Annex 48

2. Bohler A., Casari R., Fleury E;, Marchio D., Millet J.R., Morisot O. (2000). *Consoclim – ensemble coordonné d'algorithmes pour le calcul des consommations d'énergie des bâtiments*. Cahiers du CSRB N° 3187. pp. 1-12.

3. Philippe M. (2009). *Dimensionnement et calcul des performances des échangeurs géothermiques verticaux et horizontaux pour le chauffage de bâtiments résidentiels.* Internal document.

4. Henderson H.I., Parker D. and Huang Y.J. (2000) *Improving DOE-2's RESYS routine: user defined functions to provide more accurate part load energy use and humidity predictions* http://gundog.lbl.gov/

5. IEA figures from 2004 (cited in "Bilan Carbone - guide des facteurs d'émissions version 5.0", ADEME & MIES - January 2007

6. RTE - ADEME, (2007). *Le contenu en CO2 du kWh électrique: Avantages comparés du contenu marginal et du contenu par usage sur la base de l'historique*, note détaillée

7. ADEME (2007). *Bilan Carbone - guide des facteurs d'émissions version 5.*", January 2007

8. PrEN15607. (2007) *Energy performance of buildings - overall energy use, CO2 emissions and definition of energy ratings*. CEN TC 89 and 228

9. RT (2005). *Réglementation thermique 2005*. CSTB

10. Directive 2006/32/CE du Parlement européen et du Conseil du 5 avril 2006 relative à l'efficacité énergétique dans les utilisations finales et aux services énergétiques

11. PEGASE Database - Ministère de l'Économie, des Finances et de l'Industrie - http://www.developpement-durable.gouv.fr/energie/statisti/f1e_stats.htm

12. Feist W. (2007). *Certification as „quality approved Passive House", Criteria for non-residential apssive house buildings.* Passiv Haus Institut

13. ADEME (2007). *Chiffres clés du bâtiment*, Edition 2007

14. Carbon Trust (2003). *Energy consumption guideline 19: "Energy use in offices"*, The Carbon Trust, http://www.thecarbontrust.co.uk/energy/pages/home.asp, April 2003

III. *AHPSAT* (“Advanced Heat Pump System Assessment Toolbox“)

5. General requirements and specifications

Once having applied the *FAST* tool to a given case, the practitioner is invited to perform a more detailed assessment of available heat pumping solutions described in [3] by running some simulations. This second part of the assessment methodology relies on the use of generated or recorded hourly heating and cooling demands (see §6 for details). Depending of the heat pump system considered, these demands consist in ”building demands” (provided by the secondary HVAC system to maintain comfort conditions in the zone) or ”system loads” (provided by the primary HVAC system by the secondary HVAC components).

The Thermodynamics Laboratory of University of Liège has developed and implemented calculation tools and heat pump system simulation models in an equation solver [2].
The tools are packaged in a suite of assessment tools usable to perform:

- Quantification of theoretical recovery and reversibility potentials;
- Pre-sizing of a selected heat pump system (including heat pump, heat source(s), heat sink(s) and auxiliaries);
- CO2, energy, primary energy and economical assessment;
- Comparison to a classical primary HVAC system (separated heat and cold productions).

The final objectives of such toolbox are:

- To assess the energy savings, CO_2 emission reductions and the economic interest.
- To help the decision-maker to select the best solution in terms of costs/environmental impact.
- To provide a first information on the environmental, economical and energetic performance of the considered system.

This suite of simulation tools uses pre-computed hourly heating and cooling demand profiles and corresponding weather data to assess the performances of the considered heat pump system (Figure 100).

The analysis consists in using the hourly heating and cooling demands to run a one-year simulation of a selected heat pump system. Only a small number of parameters are asked to the user to perform simulation. The main part of the parameters is automatically computed by the simulation tool based on the peak heating and cooling loads (automatic pre-sizing). Rule of thumb and default values are provided to the user and implemented in the tool to allow quick economic and energy assessments of the heat pump system (in terms of CO2, primary energy and money).

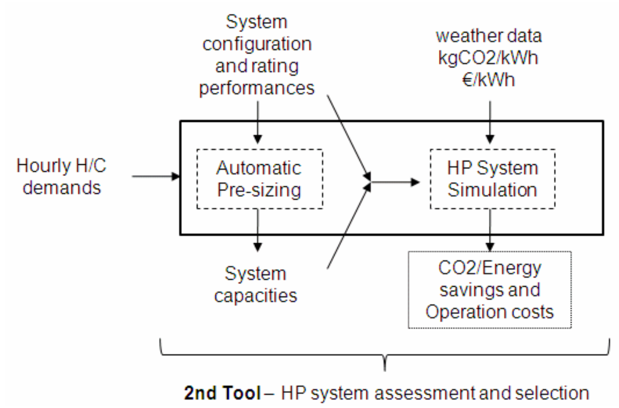


Figure 100: "AHPSAT" block diagram

Nine configurations are considered:

- System 0: classical separated heat and cool production by chiller(s) and boiler(s) (reference case)
- System 1: reversible air-to-water heat pump coupled to back up boiler;
- System 2: exhaust ventilation air heat pump coupled to back up boiler;
- System 3: dual condenser heat pump coupled to back up boiler;
- System 4A: water loop heat pump system using a heat rejection device (cooling tower or dry cooler) and a boiler;
- System 4B: water loop system using a vertical ground exchanger and a heat rejection device (cooling tower or dry cooler).
- System 4C: water loop system using a vertical ground exchanger and a boiler.
- System 5A: ground coupled reversible heat pump (no passive cooling)
- System 5B: ground coupled heat pump system with passive cooling heat exchanger

Heating/cooling building demands and system loads must be distinguished:

- In the case of primary HVAC systems (systems 0, 1, 2, 3, 5A and 5B) supplying heat and/or cold to a given secondary HVAC system (including potential dehumidifcation/humidification), heating and cooling system loads should be generated and used to perform the simulation of the HP system. In the present work, no attention is given to the secondary HVAC system which is supposed to be adapted to be operated with low temperature hot water.
- In the case of an "integrated" HP system (systems 4A, 4B and 4C) with no additional secondary HVAC system or not supplying any secondary HVAC system, net sensible building heating and cooling demands must be used.

In both cases, domestic hot water production is not taken into account and only space heating and cooling are considered.

Both inputs and parameters of the "AHPSAT" tool are chosen to be in good accordance with the information actually available at early design stages. Figure 11 shows a generic block diagram of the tools.

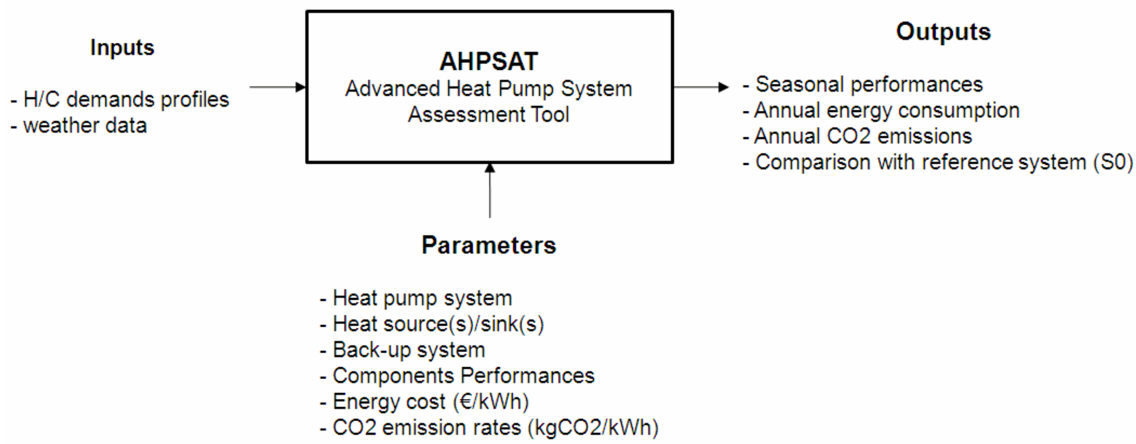


Figure 11: "AHPSAT" inputs/parameters/outputs

6. How to get hourly heating and cooling demands?

First, two situations should be distinguished:

- Retrofit project
- New building design project

In the case of a retrofit project, heating and cooling demands could be obtained from measurements (heat and cool counters) or from a calibrated building energy simulation model. For the design of a new building, heating and cooling demands should be generated by means of a detailed baseline building energy simulation model.

A number well-known programs are currently on the market: Energy+, TRNSYS, ESP-r,…

In the case of building energy simulation, two steps are necessary:

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) and calibration to reproduce recorded energy consumptions (in the case of an existing building);
2) Energy simulation of the Building-HVAC System on a typical year to compute hourly values of heating and cooling loads/demands.

The 8760 values of heating and cooling demands and the corresponding weather data are then used as inputs in "AHPSAT".

Modeling

3.1. Components modeling

3.1.1. Heat pumps

The amount of energy consumed by the heat pumps can represent up to 45% of the total consumption of the HVAC (Heating ventilation air-conditioning) system in the case of a water loop system [4]. In this context having an accurate model of the heat pump in order to determine the consumption is essential.

The heat pump simulation model is based on regression curves generated based on data collected from heat pump manufacturers. The aim of the model is to accurately simulate the operation and the electricity consumption of each type of heat pump especially when it operates at part load (PLR < 1)[2].

In this study, six types of heat pump are considered:

1. The air cooled chiller (cooling only)

2. The reversible air-to-water heat pump (heating and cooling)

3. The water-to-water or brine-to-water heat pump

4. The reversible water-to-water or brine-to-water heat pump

5. The dual condenser (air and water) heat pump

6. The reversible water-to-air heat pump (WLHP systems)

The DOE-2 method [5] is used to simulate the performance of large heat pump units (Systems 0, 1, 2, 3, 5A and 5B). Regression curves coefficients have been calibrated as prescribed by [6]. This method aims to develop a set of curves including the following influences:

- Effect of entering or leaving evaporator water temperature;

- Effect of entering or leaving condenser water temperature;

- Effect of part load operation.

A typical part load curve [7] is used for all the heat pumps and can be adjusted by the user.

For water-to-air heat pumps (Systems 4A, 4B and 4C), the model described in [8] is used. This model allows computing latent load on evaporator side in cooling mode.

The complete models are described in APPENDIX 2.

[2] PLR= Part Load Ratio. It quantifies the operation of the heat pump with respect to a full load operation.

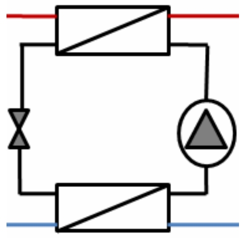


Figure 12: Schematic view of a heat pump

3.1.2. Water Pump

Pumping water represents a significant part of the energy consumption in every HVAC system. In the case of a water loop system it could represent more than half of the entire HVAC consumption [4]. For the sake of simplicity, only constant flow pumps will be considered in the present work.

In this work, no rigorous pump models were developed as the entire hydraulic network pressure drop is difficult to estimate. In fact, the pump model is based on ratios established between the water flow rate and the electricity consumption (specific pump power in W/kg/s). These ratios are different depending on the type of hydraulic network considered (hot water or cold water networks, water loop...). The model is described in APPENDIX 2.

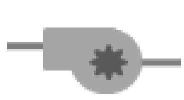


Figure 13: Schematic view of a pump

3.1.3. Boiler

According to [9], the supply water temperature should not be below a certain point defined close to 60°C for a usual boiler. This problem will be addressed with the assumption that a mixing hydraulic network is placed before entering the boiler and can perfectly control the valves in order to preheat the entering water.

The boiler performance simulation model it has been developed in two steps.

- Firstly, a reference boiler simulation model [10] has been implemented and used to simulate the performance of typical water boilers. This model is based on the heat exchanger method and uses an adiabatic burner, a water-gases heat exchanger and a water-ambient air heat exchanger. Each heat exchanger is simulated by means of the $\varepsilon - NTU$ method. Secondly, the reference model has been used to generate typical regression curves representing the actual efficiency of the boiler as function of the part load ratio.

Figure 14: Schematic view of a boiler

The complete model is briefly described in theAPPENDIX 2: Components modeling.

3.1.4. Cooling tower and Dry cooler

Two types of fluid coolers have been considered in the present work: indirect contact cooling tower and dry cooler.

The cooling tower and dry fluid cooler simulation models are based on [7], [10] and [11]. Both modes are based on $\varepsilon - NTU$ method and simulate the coolers as counter flow heat (and mass) exchangers. The simulation involves the three following steps:

1. Sizing of the cooling tower/dry cooler consisting in the identification of the parameters of the model (heat transfer coefficients and auxiliary fans and pumps nominal consumptions)
2. Simulation of the cooling tower/dry cooler in the actual operating conditions at maximal air and water flow rates (full load) and calculation of the full load cooling capacity.
3. Calculation of the part load ratio (ratio between actual and full load cooling rates) and of the corresponding auxiliary fans and pumps consumptions.

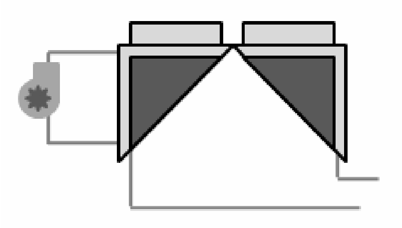


Figure 15: Schematic view of the indirect contact cooling tower

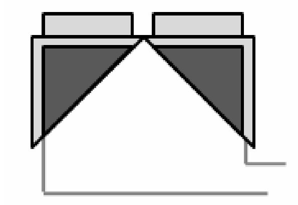


Figure 16: Schematic view of a dry cooler

Default and rule of thumb values of nominal water and air flow rates and corresponding auxiliary fans and pumps consumptions have been identified based on manufacturer data and are provided to the user. The complete model is briefly described in the APPENDIX 2: Components modeling.

3.1.5. Cooling coil

This device is only used in the exhaust heat pump system (System 2) in order to absorb as much energy as possible from the exhaust ventilation air which energy will be used as heat source for heat pumping. Like the cooling tower/dry cooler model, sizing and the simulation algorithms are used.

The sizing algorithms aims to determine the nominal heat transfer coefficient thanks to nominal data and using the model described in [7]. Manufacturer's data were used to determine correlations between the air and water heat transfer coefficient and the exhaust air flow rate.
The second part of the model simulates the operation of the cooling coil in order to determine the exhaust water temperature and the corresponding capacity. The model is described in APPENDIX 2: Components modeling.

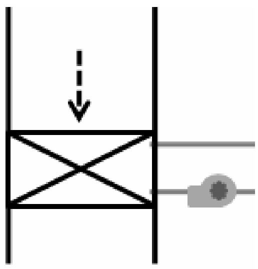


Figure 17: Schematic view of a cooling coil

The complete model is briefly described in the APPENDIX 2.

3.1.6. Ground heat exchanger

In this work only the vertical closed-loop ground heat exchangers will be considered as this system is the most common in non-residential applications.

The model used in this work is based on the work of Bernier [12] [13] [14] [15] [16]. This model involves the following steps:

1. Calculating the temperature of the inner surface of the borehole (global heat exchange) by means of the Cylindrical Heat Source method and an improved time superposition algorithm (Multiple Load Aggregation Algorithm)

2. Calculating the average fluid temperature by considering only static heat transfer in the borehole between the borehole's wall the fluid (local heat exchange)

3. Calculating the exhaust fluid temperature

Considering that only one-year simulation are performed in the present work, long-term influences (interferences between boreholes of a same borefield) are not taken into account. However, a simple calculation method is provided if multi-years simulations [16] must be performed.

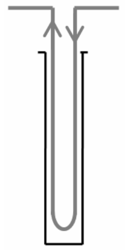


Figure 18: Schematic view of a ground exchanger

The complete model is briefly described in the APPENDIX 2.

3.1.7. Thermal storage

In the present work, the thermal storage model is used in order to model the effect of the thermal inertia of a water network (water loop heat pump system) or the behaviour of a water storage tank. The simulation model relies on a first order differentia equations and do not take into account the effect of stratification. Thermal losses are taken into account by specifying a global AU value for the whole tank.

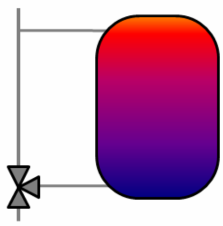


Figure 119: Schematic view of a thermal storage

The complete model is briefly described in the APPENDIX 2.

3.2. System modeling

3.2.1 Implementation

All the components simulation models have been developed and implemented in an Engineering Equation Solver [2]. Each configuration is implemented separately but all the files are structured in a similar way. The different parts of each file are described hereafter.

Introduction of procedures

According to EES terminology [2], a procedure consists in a subprogram which can be called from anywhere in the code. The specificity of a procedure is that all equations are developed in an explicit form, which means no iteration is required in order to solve them. Each component model has been developed as a set of components procedures. This choice has been motivated by two main reasons:

- Having explicit equations increases the robustness of the program as no iteration is performed;

- Interchanging components is easier as each procedure is only defined by its inputs/outputs.

Procedures have also been used to perform a “pre-sizing” of the considered system and to identify the parameters of the models. According to EES syntax, a procedure must be placed before any other equation.

Control algorithm

This is the decision tree of the system. It allows changing the operating mode of the system according to the conditions of use. In each part of the control algorithm different equations are implemented regarding the mode in which the system operates. The main outputs of this part are the consumption, the capacity of the different devices and various temperatures.

Inputs/parameters required

The inputs needed perform the simulation are:

- Heat and cold demands on a hourly basis (in W)

- Outdoor temperature (in °C) and humidity ratio (in kg/kg) on a hourly basis

The parameters of the models are:

- Temperature set points (°C) and variations in nominal conditions (in °K)

- Components performance (nominal EER, COP and efficiencies), capacities and sizes (specific pump power, storage tank volume...)

- Primary energy conversion factor

- Specific energy costs (€/kWh)

- CO2 emission rates (kgCO_2/kWh)

In the case of a lack of data, some of these parameters can be automatically estimated based on rule of thumb values (storage tank volume, specific pump power, heat pump capacity, cooling tower fan size...) or manufacturer data (EER, COP...).

Other parameters (such as nominal flow rates) are automatically computed by the model.

Time

The time step and boundaries can be modified by the user. A time step of one hour is generally used.

Integrations

This part of the program follows with the calculation of all the integrated values, such as electrical consumption or the total amount of fuel burned.

Comparison

The last part of the program deals with the comparison of the system considered with the reference model (except for the reference model itself, of course). General outcomes of the program such as the primary energy consumption, the CO_2 emission and the costs are calculated.

1.2.1. General hypothesis

Many assumptions have been made on components (already developed) and on systems which are essential to reduce the number of inputs needed to evaluate the environmental and economic performance of the considered system. Moreover, these assumptions also allow reducing the computation time.

General operation

The first hypothesis is related to the heat pump system which is assumed to be designed to supply the entire demand. All devices are sized to first supply the cooling demand or the heating demand. In case of a heat pump supplying the heating demand, a backup heater is used if required.

The second hypothesis is linked to the secondary HVAC system. In this study it will be considered that the secondary HVAC system is designed to work in the temperature range provided by the heat pump system (e.g. 35-45°C in heating mode and 7-12°C in cooling mode)
This allows focusing the program on the heating and cooling production system only.

Thirdly no defrosting losses of the equipments will be considered.

Fluid properties

In the selected systems four types of fluids are used and it is obviously not possible to link a variation law to each of the fluid properties. Many of these have been established as fixed under nominal conditions. The following table summarises the main characteristics for those fluids. All these parameters can be modified by the user if needed.

	Water	Glycol Water (25%)	Natural Gas	Air
c_p [J/kg-K]	4190	3960	1800	1006
ρ [kg/m³]	1000	1022		1.16
LHV [J/kg]			43 10^6	

Table 6: Hypothesis on fluid properties

Heat losses in hydraulic networks

Modelling the heat loss in all the hydraulic networks is a quite difficult task as such losses depend on many parameters such as the ambient temperature, the temperature of the fluid flowing in the pipes, length of the network and the properties of the insulation material. In the present work, heat losses and gains are taken into account in a very simple way. A ratio determines the lost/gained power based on the peak heating and cooling demands. In a first time, this ratio is arbitrarily fixed to 2.5%.

In water loop system, heat losses are usually not taken into account as the temperature range of the water loop stays close to the ambient temperature. However, heat losses and gains could be specified by the user if needed.

1.2.2. System 0: Classical separated heat and cool productions (reference case)

General description

This system will be used later as a reference for the other ones as it represents the simplest way to satisfy cooling and heating demands. The two heat and cold networks are completely separated; they are respectively supplied by the boiler and the air-to-water chiller.

The components which are modelled in this system are:

- Water cooled chiller
- Boiler
- Pump

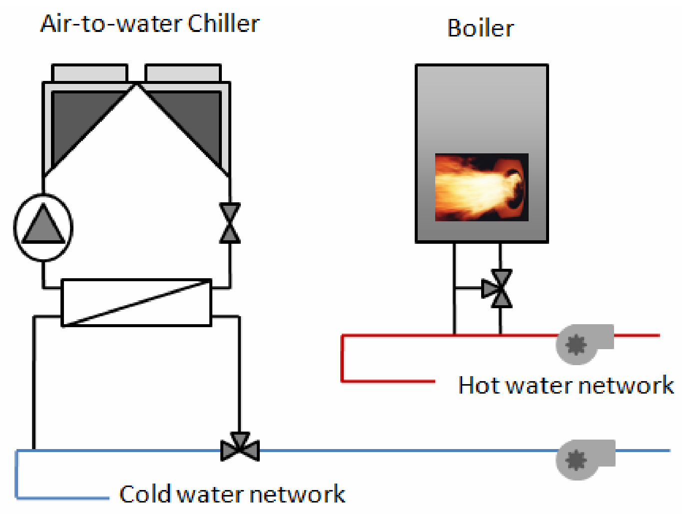


Figure 20: Model 0: Chiller and boiler independent

Heat source/sink

In this model only the outdoor air is considered as a heat sink as it is quite often used in heat pump systems. The main disadvantage of the outdoor air is its very variable temperature. In cooling mode high outdoor temperatures reduce the performance and the capacity of the heat pump.

Control algorithm

With this system, control algorithm is the simplest one as the two parts are independent; each network has its own control algorithm. The only test to be done is whether the demand is null or not.

When a cooling or heating demand is detected by the system, the chiller or the boiler is switched on. The solving algorithm is summarised in Figure:

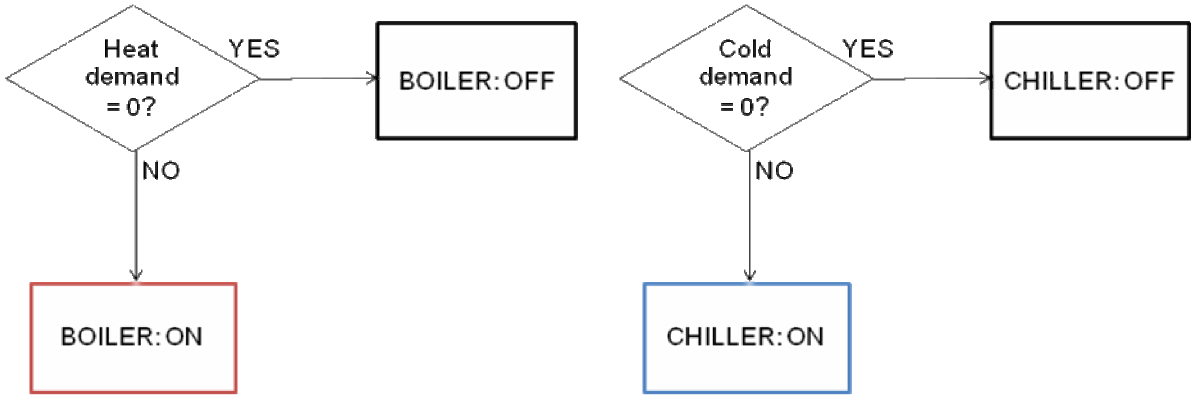


Figure21: Model 0 – Solving procedure

1.2.3. System 1: Air-to-water reversible heat pump with backup boiler

General description

The air-cooled chillers are the dominant technology on the European air-conditioning market, representing 85 % of chillers sold [64].
The system can be reversed by means of a refrigerant change-over[3] (4 ways valve) which inverses the refrigerant flow passage into the two exchangers:

- **In cooling mode** the air exchanger works as a condenser, rejecting heat, while the water-exchanger works as an evaporator, transferring cooling power to the distribution system.

- **In heating mode** the air exchanger works as an evaporator, absorbing heat from outside air, while the water exchanger works as a condenser, transferring heating power to the distribution system.

In most cases, reversible air-cooled units are however installed in combination with backup boilers, for the following reasons:

1. To supply heating power when heating and cooling loads are simultaneous: when the chiller works in cooling mode, it cannot provide any heating power. Because its condenser is air cooled, the condenser heat cannot be recovered.

2. To complete heating power for low outdoor air temperatures, in case the air-to-water unit is not sized for low outdoor temperatures (the heating power at –5°C can be 30% lower than heating power at nominal conditions [55].

3. To supply the entire heating power when, at very low outdoor temperatures, the boiler has better performances[4] than the air-to-water unit, or when the outdoor temperature is under the unit working range.

Such system can reach good efficiency if the heating and cooling demands are alternate, as the system is designed to be reversible.

[3] Another possibility is the "system change-over" changing the hydraulic network but will not be considered here.
[4] in terms of primary energy consumption and/or CO_2 emissions

This type of heat pump modelled the following components:

- Air-to-water reversible heat pump
- Boiler
- Pump

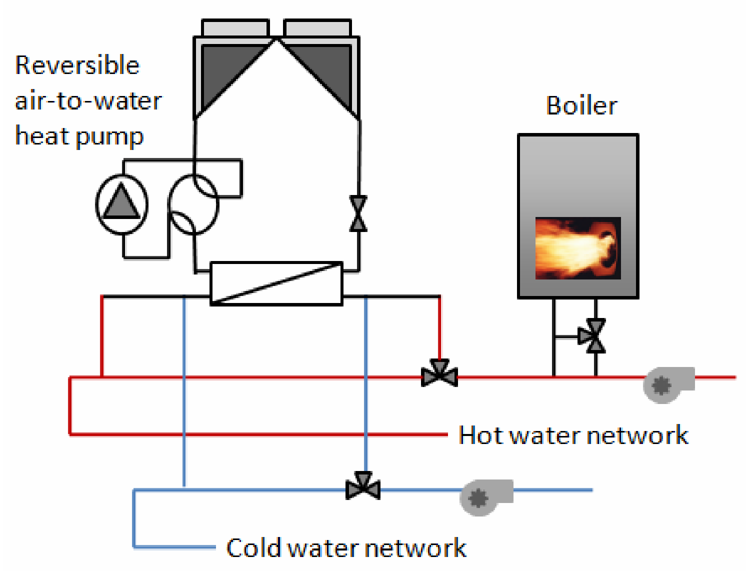


Figure22: Model 1: Reversible heat pump

Heat source/sink

During winter and in heating mode, low outdoor temperatures limit the capacity of the heat pump and decrease its performance which could be due to the presence of frost. Like in the model 0, in cooling mode, high outdoor temperatures reduce the performance and the capacity of the heat pump. Moreover, during winter, defrosting strategies have to be used and decrease the performances and the heating capacity of the heat pump. The defrosting process of the heat pump will not be considered in our modelling of the system.

Control algorithm

The control algorithm explained below rigorously follows the sequential steps of the control procedure. The system is cooling driven which means that the priority is given to cold generation if there is a cooling demand. The control algorithm starts with the declaration of the variable as it is the necessary step to avoid any compilation problems. Variables defined in this part are mainly the outputs of the control procedure.

DECLARATION OF VARIABLES

The main temperatures which are used in the following control algorithm are the return temperatures of the network defined as:

$$t_{w,hw,return} = t_{w,hw,set} - \frac{\dot{Q}_{heat}}{c_{p,w} * \dot{M}_{hw}}$$

$$t_{w,cw,return} = t_{w,cw,set} + \frac{\dot{Q}_{cold}}{c_{p,w} * \dot{M}_{cw}}$$

Applying the hypothesis previously explained that the set point is achieved (even with a backup boiler if necessary) and that the secondary system (ventilation) is able to supply the entire demand. The control algorithm follows with the evaluation of the heat demand.

IF $\dot{Q}_{heat} = 0$ THEN

IF $\dot{Q}_{cold} = 0$ THEN
NOTHING
ELSE
HEAT LOSSES CALCULATION Cold network
CALL CHILLER PROCEDURE
PUMP CONSUMPTION CALCULATION
ENDIF

When the heat and cold demand are equal to zero, no consumption is calculated and heat losses are not taken into account. Obviously, electricity consumption of the pumps is also equal to zero.

In case of a cooling demand and no heating demand, the heat pump in chiller mode is activated. The entire cooling demand to supply is equal to the sum of the cooling demand and the heat losses in the cold network. At this step of the program the chiller mode procedure is called, which determines the electricity consumption and the EER of the component using the DOE-2 method. The pump electricity consumption is taken into account.

ELSE

IF $\dot{Q}_{cold} = 0$ THEN
HEAT LOSSES CALCULATION Hot network
IF $t_{out} < t_{\lim}$ THEN
CALL BOILER PROCEDURE
PUMP CONSUMPTION CALCULATION

In case of a heating demand and no cooling demand, the heat losses in the hot network are calculated before doing anything else. The heat losses in the hot network are summed to the heating demand to obtain the entire heating demand. Further, a simple measurement of the outdoor air temperature defines whether the heat pump will be activated or not. The control algorithm does not allow the heat pump to operate in case of too low outside temperature to avoid operating at a poor COP. In the context of this work, the limit temperature of -5°C is considered. If the outdoor air is below such limit the heat pump is switched off and the boiler supplies the entire heat demand. Thereafter, the calculation of the pump electricity consumption is performed through the correlations.

ELSE
MAXIMUM CAPACITY CALCULATION
IF $\dot{Q}_{heat} < \dot{Q}_{heat,max}$ THEN
$t_{w,ex,HP} = t_{w,ex,set}$
CALL HEAT PUMP PROCEDURE
ELSE

$$\dot{Q}_{HP} = \dot{Q}_{heat,max}$$

$$t_{w,ex,HP} = t_{w,su,HP} + \frac{\dot{Q}_{HP}}{c_{p,w} * \dot{M}_{w,hw}}$$

CALL HEAT PUMP PROCEDURE
CALL BOILER PROCEDURE
ENDIF
PUMP CONSUMPTION CALCULATION
ENDIF

Considering that the ambient air temperature is high enough, the maximum capacity of the heat pump is defined in relation with the operating conditions. It is calculated through a procedure with uses the DOE-2 method with a PLR=1 (full load). When such parameter is not exceeded the system determines the electricity consumption of the heat pump assuming that the exhaust water temperature is the set one. In the opposite case, the system needs the help of the boiler to reach the set temperature. The exhaust heat pump water temperature is determined assuming that the device operates at its maximum capacity. As the system functions in heat pump mode only the heat pump set of correlations is used to determine the consumption of the system.

```
        ELSE
        HEAT LOSSES CALCULATION Hot network
        HEAT LOSSES CALCULATION Cold network
        CALL CHILLER PROCEDURE
        CALL BOILER PROCEDURE
        PUMP CONSUMPTION CALCULATION
        ENDIF
ENDIF
```

When heating and cooling demands are simultaneous both networks are considered as separated as it was the case in the Model 0: Chiller and boiler independent. The heat losses in the cold and the hot network are calculated in order to determine the entire cooling and heating demands.

The control algorithm can be summarised in the Figure23:

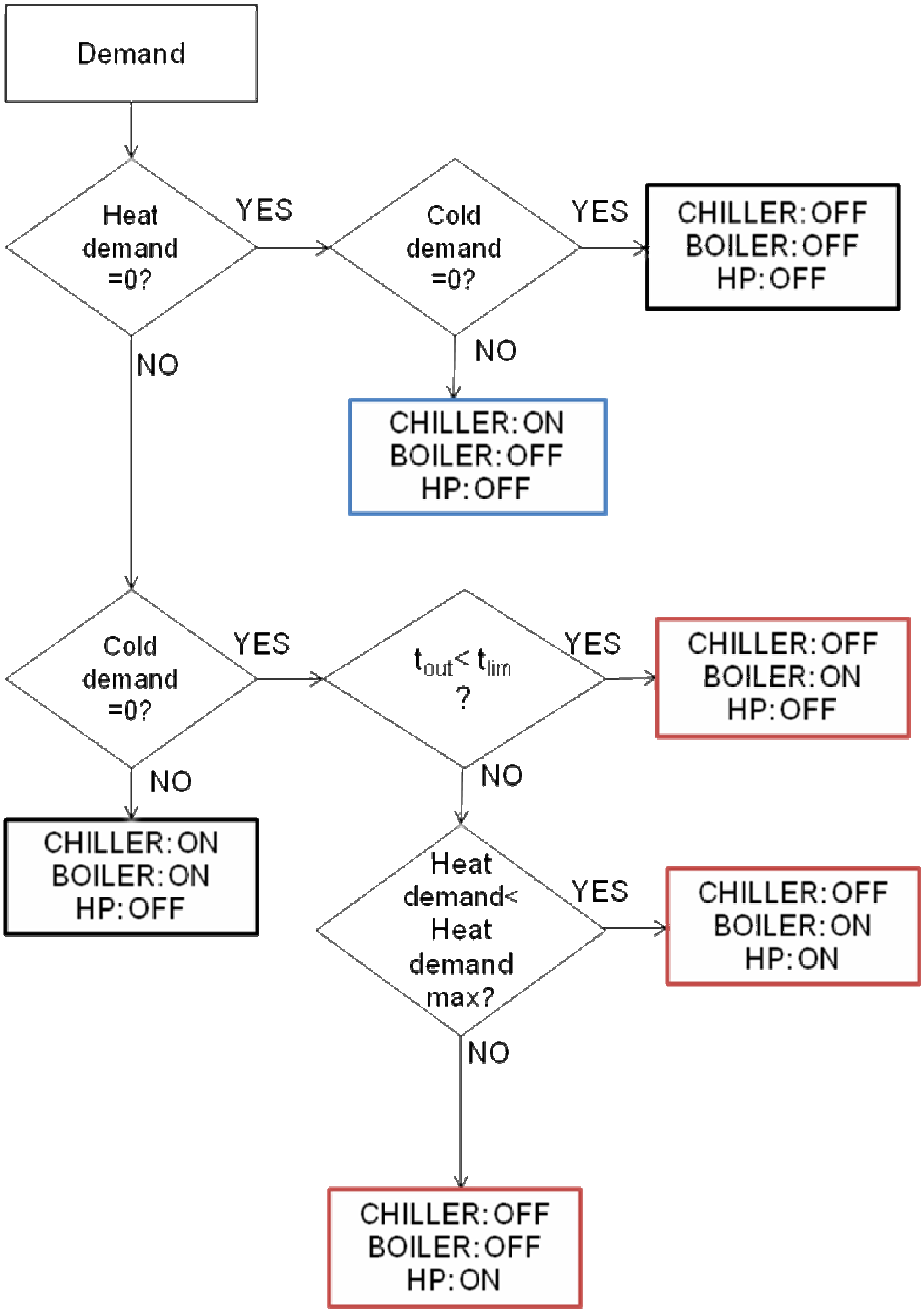


Figure 23: Model 1 – Solving procedure

1.2.4. System 2: Exhaust ventilation air heat pump with backup boiler

General description

In this system, extracted ventilation air is used as heat source for heat pumping. Given that the non residential building demand could be quite large, the exhaust air flow could be insufficient to supply heat. In that case an additional heat generator (boiler) is required. The heat pump is not reversible and each water network is independent from the other. This type of heat pump can achieve a good performance level as the temperature of the heat source is quite constant all along the year. For this system and although it could be possible, the condenser heat is not recovered. The recovery possibilities will be studied in other configurations.

This type of heat pump uses the following components:

- Water-to-water heat pump
- Boiler
- Pump
- Cooling tower/Dry fluid cooler
- Air-water cooling coil

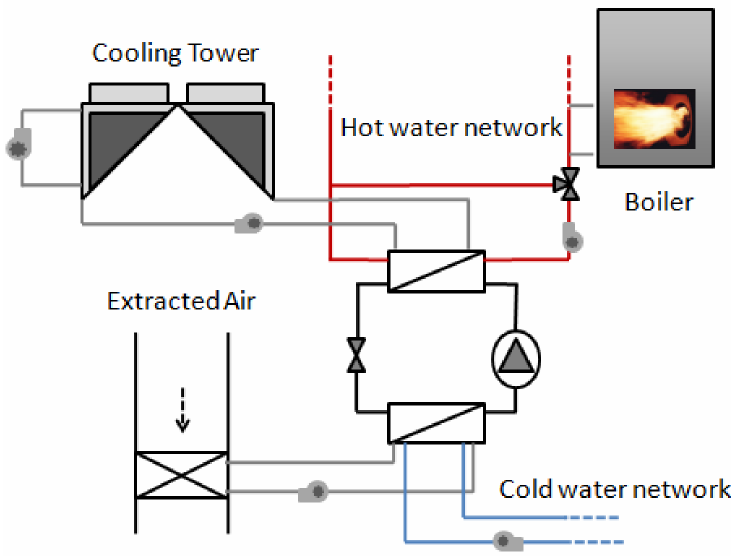


Figure24: Model 2: Exhaust air heat pump

Heat source/sink

Exhaust air represents a very interesting heat source because of availability, good coincidence with needs and its very constant temperature. Condensing the water contained in extracted air allows recovering part of the latent energy and increases the heating capacity. However, exhaust ventilation rates are restrictive in terms of capacity so that an additional heat source may be required.

Control algorithm

Like in the previous model, it is cooling driven. The control algorithm starts with the declaration of variable followed by the evaluation of the demand.

DECLARATION OF VARIABLES

$$t_{w,hw,return} = t_{w,hw,set} - \frac{\dot{Q}_{heat}}{c_{p,w} \cdot \dot{M}_{hw}}$$

$$t_{w,cw,return} = t_{w,cw,set} + \frac{\dot{Q}_{cold}}{c_{p,w} \cdot \dot{M}_{cw}}$$

IF $\dot{Q}_{heat} = 0$ THEN

IF $\dot{Q}_{cold} = 0$ THEN
HEAT LOSSES CALCULATION Hot network
HEAT LOSSES CALCULATION Cold network
ELSE
HEAT LOSSES CALCULATION Hot network
CALL CHILLER PROCEDURE

$$\dot{Q}_{cd} = \frac{EER + 1}{EER} * \dot{Q}_{ev}$$

$$t_{w,ex,set,CT} = t_{w,ex,cd} - \frac{\dot{Q}_{cd}}{c_{p,w} * \dot{M}_{w,cd}}$$

CALL COOLING TOWER PROCEDURES
PUMP CONSUMPTION CALCULATION
ENDIF

Like in other models each device is switched off when there is no demand and the heat losses are calculated. When only a cooling demand is present, the chiller procedure allows calculating the electricity consumption of the device (and the EER) knowing the power of the evaporator. The power of the condenser is determined through the EER which is used to calculate the theoretical exhaust water temperature of the cooling tower. Finally the cooling tower procedure is called to determine the electricity consumption of the component. In fact two procedures are successively called for the cooling tower: the first one is the sizing procedure of the cooling tower which determines heat transfer coefficients according to the nominal point; the second one simulates the cooling tower and determines the electricity consumption.

ELSE
IF $\dot{Q}_{cold} = 0$ THEN
HEAT LOSSES CALCULATION Cold network
CALL COOLING COIL PROCEDURE
CALL HEAT PUMP PROCEDURE

$$\dot{Q}_{cd} = \frac{COP}{COP - 1} * \dot{Q}_{ev}$$

If only a heating demand is present, the cooling coil water loop is switched on. The procedure called allows knowing the capacity potentially supplied by the exhaust air which determines the capacity of the evaporator. The heat pump procedure determines the COP which is used to calculate the corresponding condenser capacity.

IF $\dot{Q}_{cd} < \dot{Q}_{heat}$ THEN

$$t_{w,ex,HP} = t_{w_{su,HP}} + \frac{\dot{Q}_{cd}}{(c_{p,w} * \dot{M}_{w,cd})}$$

CALL BOILER PROCEDURE
PUMP CONSUMPTION CALCULATION
ELSE
$\dot{Q}_{cd} = \dot{Q}_{heat}$
$t_{w,ex,HP} = t_{w,hw,set}$
CALL HEAT PUMP PROCEDURE
PUMP CONSUMPTION CALCULATION
ENDIF

At this stage of the control, we have to differentiate when the heat pump is able to supply the entire demand or when it is not. In the first case, the heating demand is larger than what the condenser can provide. In this context, the exhaust condenser water temperature is below the set temperature and the PLR is equal to 1; the boiler will provide the additional heat needed to reach the temperature set. In the second case the set temperature is reached and the PLR of the heat pump is below or equal to 1.

ELSE
CALL CHILLER PROCEDURE

$$\dot{Q}_{cd} = \frac{EER + 1}{EER} \cdot \dot{Q}_{ev}$$

$$t_{w,ex,set,CT} = t_{w,ex,cd} - \frac{\dot{Q}_{cd}}{c_{p,w} \cdot \dot{M}_{w,cd}}$$

CALL COOLING TOWER PROCEDURES
CALL BOILER PROCEDURE
PUMP CONSUMPTION CALCULATION
ENDIF
ENDIF

In the case of simultaneous demands there are 2 possibilities:

- It can be assumed that both networks are separated and are operating independently;
- Or part of the heat rejected is considered to be collected in the water exchanger and the exceeding heat is rejected at the cooling tower.

At this stage of the research we will focus on the first option as the second one will be considered in the next model (Model 3: Dual condenser heat pump). In particular, in case of simultaneous demands, the chiller and the cooling tower procedures are called for the cooling demand while the boiler procedure is called for the heating demand operating completely separately.
The control algorithm can be summarised in Figure25:

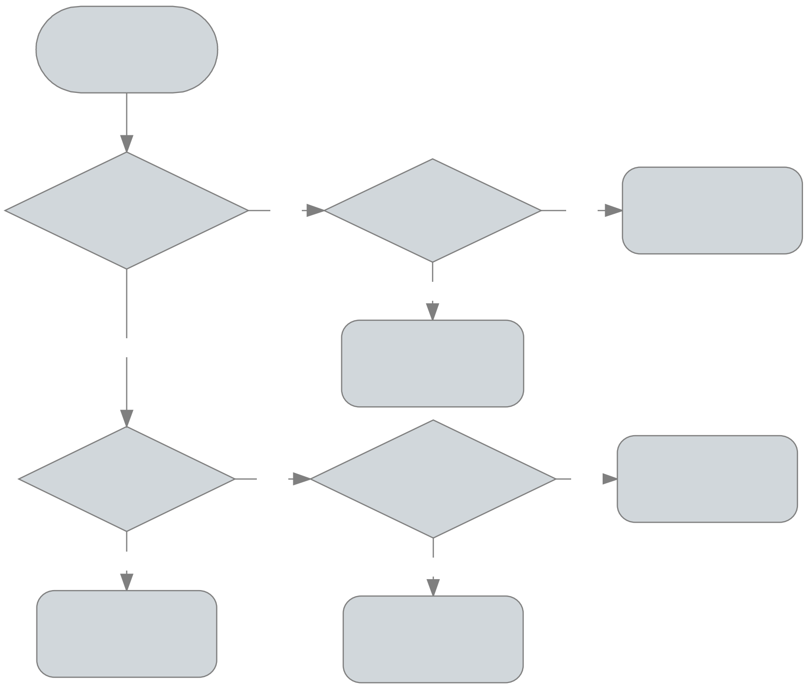


Figure 25: Control algorithm of Model 2

1.2.5. System 3: Dual condenser heat pump with backup boiler

General description

The dual-condenser chiller is made of a chiller equipped with two condensers, one air-cooled and the other one water-cooled. The main advantage of this type of heat pump is that it allows the heat recovery, so heating and cooling demands can be simultaneous. There are a lot of different possibilities to integrate both condensers in the system. In this configuration both condensers are installed in parallel. Double-bundle condenser system has been selected because historically it has been the most often used system for chiller heat recovery [17]. Refrigerant gas from the compressor flows into condenser shells, allowing heat rejection to one or both condensers. Usually the each condenser is sized for full heat output of the chiller. According to [17] the main advantages of this type of double-bundle condenser in relation to others are:

- Standard construction can be used;
- Efficiency is maintained during cooling only operation;

It is assumed that a back up boiler can operate to:

- cover the whole heating demand when there is no cooling demand;
- cover the peak heating demand.

This type of heat pump system requires the following elements:

- Water-to-water dual condenser heat pump model
- Boiler model
- Pump model

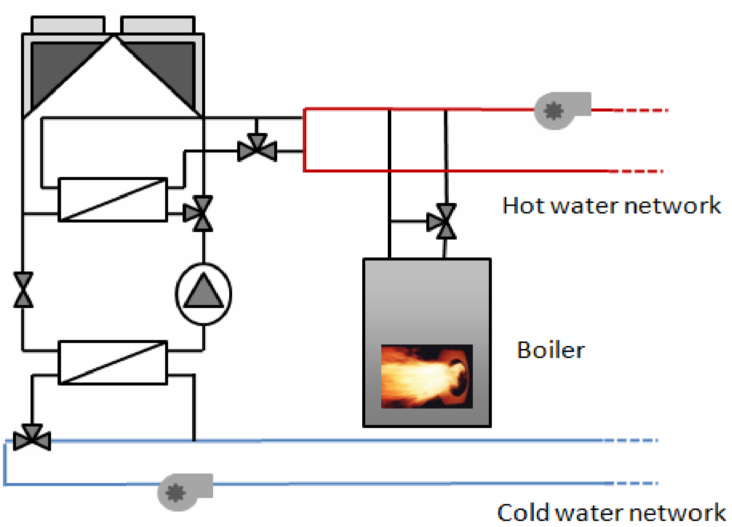


Figure26: Model 3: Dual condenser heat pump

Heat source/sink

As it is the case in some previous systems, the air is the heat sink when the heat pump operates in chiller mode. On the contrary, the heat source is either the boiler or the heat pump condenser. The availability of the second option clearly depends on the cold demand as there is a relation between the evaporator power and the condenser power.

Control algorithm

DECLARATION OF VARIABLES

$$t_{w,hw,return} = t_{w,hw,set} - \frac{\dot{Q}_{heat}}{c_{p,w} * \dot{M}_{hw}}$$

$$t_{w,cw,return} = t_{w,cw,set} + \frac{\dot{Q}_{cold}}{c_{p,w} * \dot{M}_{cw}}$$

IF $\dot{Q}_{heat} = 0$ THEN

IF $\dot{Q}_{cold} = 0$ THEN
NOTHING
ELSE
HEAT LOSSES CALCULATION Cold network
CALL CHILLER PROCEDURE
PUMP CONSUMPTION CALCULATION
ENDIF

In case of no cooling and no heating demand, chiller and pump electricity consumptions are equal to zero.
In case of a cooling demand, the heat pump only functions as a chiller and the air condenser is the only one to operate. The heat losses in the cold network are calculated in order to determine the entire cooling demand. The calculation of the pump electricity consumption is performed through the correlations. The consumption of the fan of the air condenser is already taken into account in the chiller procedure.

ELSE

IF $\dot{Q}_{cold} = 0$ THEN
HEAT LOSSES CALCULATION Hot network
CALL BOILER PROCEDURE
PUMP CONSUMPTION CALCULATION

If there is a heat demand, only the boiler functions as it is not possible to recover heat from the chiller. Once again, the heat losses in the hot network are calculated in order to determine the entire heating demand. So far the control algorithm is exactly the same as for the reference case.

ELSE
HEAT LOSSES CALCULATION Cold network
HEAT LOSSES CALCULATION Hot network
CALL CHILLER WITH RECOVERY PROCEDURE

$$\dot{Q}_{cd} = \frac{COP}{COP-1} \cdot \dot{Q}_{ev}$$

IF $\dot{Q}_{heat} > \dot{Q}_{cd}$ THEN

$$t_{w,ex,HP} = t_{w,su,HP} + \frac{\dot{Q}_{cd}}{c_{p,w} \cdot \dot{M}_{w,cd}}$$

CALL BOILER PROCEDURE
PUMP CONSUMPTION CALCULATION

If simultaneous demands are observed, the capacity which can be recovered is calculated. If the heat demand is greater than what the condenser can provide, only the water condenser works. As the set temperature cannot be achieved the boiler is switched on.

ELSE

$$t_{w,ex,HP} = t_{w,hw,set}$$

CALL CHILLER WITH RECOVERY PROCEDURE+FAN CONSUMPTION CALCULATION
PUMP CONSUMPTION CALCULATION
ENDIF

ENDIF

ENDIF

If the recovery is greater than the heat demand, the boiler is no longer needed and the excess heat is rejected by the air condenser. The consumption of the fan of the air condenser is added to the general consumption of the heat pump.

The control algorithm can be summarised in the Figure27:

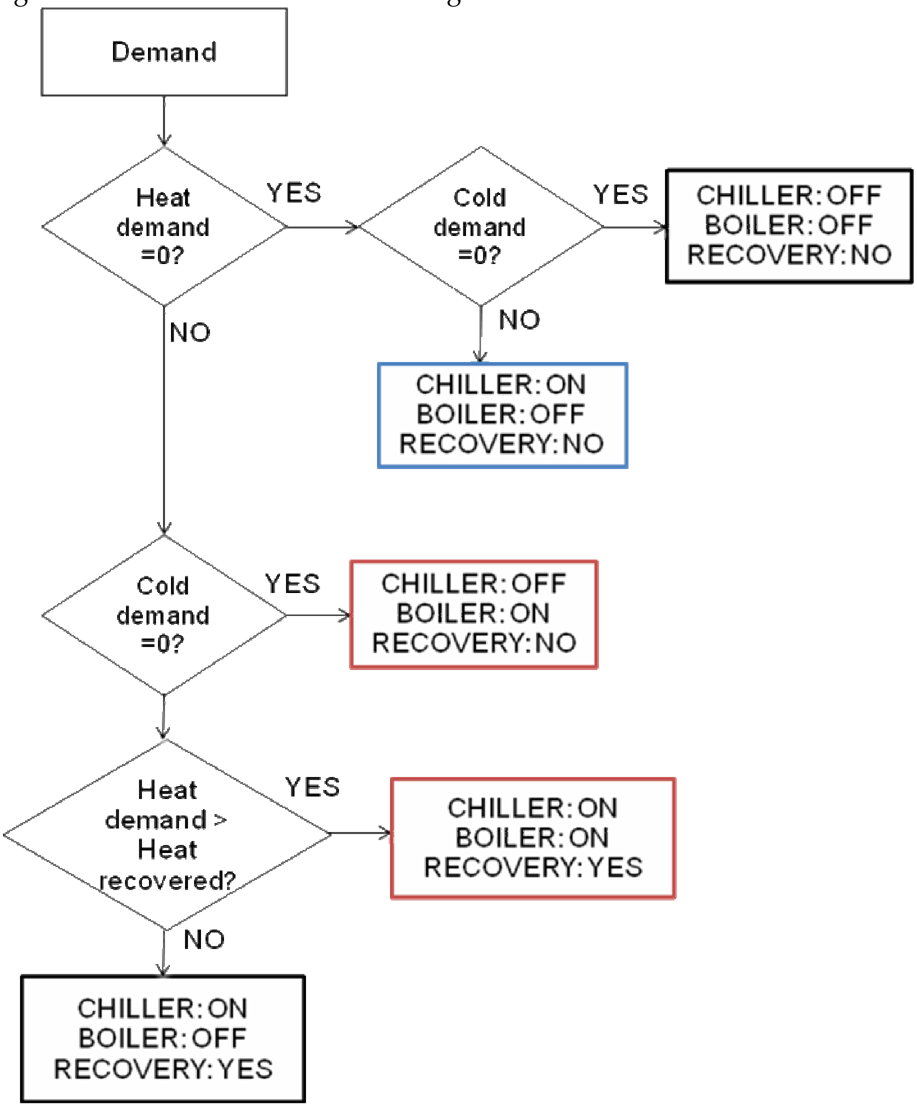


Figure27: Model 3 – Solving procedure

1.2.6. System 4: Water loop heat pump system

General description

A water loop heat pump system is made of several reversible air-to-water heat pump units, each serving a zone, connected to a closed water loop circulating throughout the building. Each heat pump unit uses the water loop as heat source/sink rejecting or absorbing heat to/from it. The main advantage of this system is that balanced simultaneous heating and cooling demands keep the temperature of the loop in a certain range of temperature without the intervention of an auxiliary heat source or heat rejector. If the temperature exceeds the upper limit a heat rejector must be switched on (generally a cooling tower or a dry fluid cooler). In the opposite case a heat source is used (generally a gas condensing boiler or an electrical boiler). Loop temperature is generally included between 16 and 32°C [18].

Hybrid configurations combining ground heat exchangers with a standard heat source (boiler) or a heat rejector (cooling tower or dry fluid cooler) are also considered [19]. It allows optimising the sizing of the ground heat exchangers as they can be used as a heat source or sink. It is important to keep in mind that in the case of the utilisation of a ground exchanger all the water loop has to be loaded with glycol (to avoid freezing). Moreover, even if it may appear more efficient to have an independent ground loop, isolation heat exchanger is rarely implemented. In some cases the main loop is separated from the heat rejector and heat source by isolation heat exchangers. In this study only a simplified water loop system without any secondary circuit will be considered. Some authors provide rule of thumb values to determine the fluid flow rate in the loop and the associated pumping power [18][20][21][22][23][24].

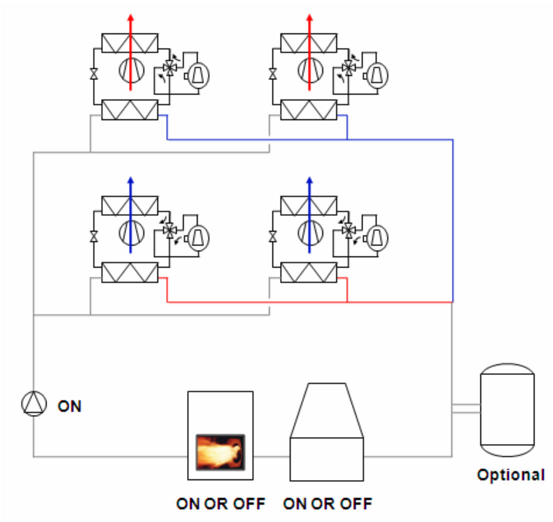
Adding a thermal storage allows a better utilisation of the water loop: the water loop temperature remains longer in the appropriate range without the use of the cooling tower or of the boiler during longer periods. This allows making some load shifting when heating and cooling demands are not simultaneous. Rule of thumb values exist to optimize the size of an additional thermal storage tank [18][20].

This type of heat pump system may include the following components:

- Water-to-air heat pump
- Boiler
- Cooling tower or dry cooler
- Water loop pump
- Ground exchanger
- Thermal storage

Three configurations are considered in the present work (Figure28): boiler + heat rejector, ground heat exchanger + heat rejector and ground heat exchanger + boiler. In Figure28, the peripheral zone uses heat pumps in heating mode while the core zones uses heat pumps in cooling mode. At the other side of the loop a combination of a heat generator and a heat rejector can be used to keep the water loop temperature in the right range.

System 4A



System 4B & 4C

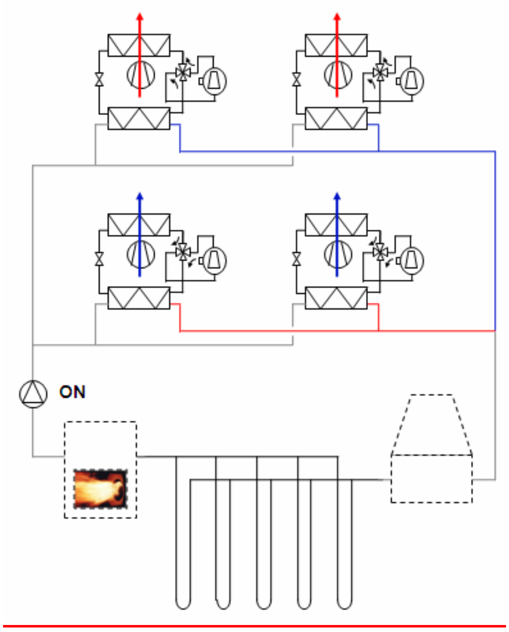


Figure28: Water loop heat pump systems configurations

Heat source/sink

The soil composition has a great influence on its thermal properties and on the overall performance of the ground heat exchanger. The thermal diffusivity of the soil is the dominant factor for the heat transfer in the ground. The availability of the ground in terms of temperature is excellent as the undisturbed ground temperature two meters below the surface is often between 5 and 10° C all year long.

Control algorithm

The control algorithm is depending on the type of water loop system considered. In general, the water loop system model solving procedure is defined as follow:

1. Given a supply heat pump temperature the exhaust heat pumps temperature is calculated through the evaluation water-to-air heat pump performances.

2. The water flow leaving the heat pump enters the thermal storage tank and the storage temperature is computed.

3. The control algorithm of the plant determines whether the heat source or the heat rejector should be activated or not. Fuel and electricity consumptions are computed

4. The plant exhaust temperature, corresponding to the heat pump supply temperature, is computed and the procedure is repeated.

A way to avoid iterations and to decrease computation time was to use the supply heat pump temperature of the previous time step to calculate the actual heat pump exhaust temperature and plant performance. No iteration is required so that the running time is drastically shortened and the convergence is ensured.

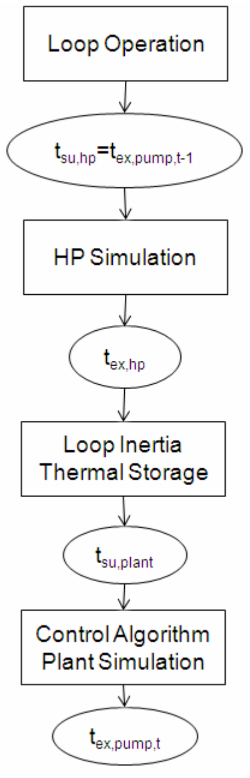


Figure 129: WLHP system model – solving procedure

Model 4a: Heat rejector and boiler

This type of water loop utilises a heat rejector and a heat source (here a gas boiler) with the aim of keeping the temperature of the loop in a certain range. The heat rejector can be either a cooling tower or a dry cooler.

$t_{f,su,hp} = t_{f,ex,pump,t-1}$ *//supply heat pump temp. = exhaust pump temp. at previous time step*

HEAT PUMP SIMULATION – COOLING

HEATPUMP SIMULATION – HEATING

$\dot{Q}_{loop} = \dot{Q}_{cd,hp,cooling} - \dot{Q}_{ev,hp,heating}$ *//Loop heat balance (>0 = rejected to the loop; <0 = absorbed from the loop)*

$$t_{f,ex,hp} = t_{f,su,hp} + \frac{\dot{Q}_{loop}}{\dot{M}_{f,loop} * c_{pf}}$$ *//exhaust heat pump temperature*

In this first part of the program, heat pumps operation is simulated in order to compute their performances (compressor and fan electricity consumptions) ant the amount of heat rejected to/absorbed from the loop. Heat pump consumption is null if there is no demand or if the loop is not in operation (circulation pumps are OFF).

LOOP THERMAL INERTIA AND THERMAL STORAGE CALCULATION (1st order model)

$t_{f,su,plant} = t_{f,ex,loop}$ *//supply plant temperature*

The thermal inertia of the loop and the thermal storage tank (if any) are simulated by means of the first order capacitance method. The plant supply temperature is equal to the exhaust thermal storage temperature.

```
IF f_loop > 0 THEN                                //Loop is in operation (pumps are working)

    IF t_f,ex,hp = t_f,su,hp THEN
    PERFECTLY BALANCED OPERATION OR NO DEMAND
    HEAT REJECTOR IS OFF
    HEAT SOURCE IS OFF
    ENDIF
```

If exhaust and supply heat pump temperatures are (almost) equal, the loop is perfectly balanced or there is no heating or cooling demand. In both case, the heat rejector and the heat source are not used and the loop temperature is floating.

```
    IF t_f,ex,hp < t_f,su,hp THEN                 //Heating dominated operation
    HEAT REJECTOR IS OFF
        IF t_f,su,plant > t_f,loop,min THEN
        HEAT SOURCE IS OFF                        // Free floating loop temperature
        ELSE
        HEAT SOURCE IS ON
        HEAT SOURCE SIMULATION                    //Fuel and auxiliaries consumptions
        ENDIF
    ENDIF
```

In case of "heating dominated operation", the heat source is used only if the loop temperature is below the minimal loop temperature set point. The value of this lower temperature limit can be determined as:

- keeping a constant limit at 16°C, for example;
- or varying the limit along the year. For instance the limit could be 28°C in summer time and 5°C in winter time.

The heat source is controlled in order to maintain this minimal temperature.

```
    IF t_f,ex,hp > t_f,su,hp THEN                 //Cooling dominated operation
        IF t_f,su,plant < t_f,loop,max THEN
        HEAT REJECTOR IS OFF                      // free floating loop temperature
        ELSE
            IF t_f,su,plant - t_su,CS > Δt_cs,ON THEN  //tsu,CS is outdoor drybulb temperature if DFC
                                                  or wetbulb temperature if CT
            HEAT REJECTOR IS ON
            HEAT REJECTOR SIMULATION              //Auxiliaries (fan and pump) consumption
            ELSE
            HEAT REJECTOR IS OFF
            ENDIF
        ENDIF
    HEAT SOURCE IS OFF
    ENDIF
```

In case of "cooling dominated operation", the heat source is not used and the heat rejector is used only if two conditions are satisfied. If the heat rejector supply temperature is above the upper loop temperature limit ($t_{f,loop,max}$), one must evaluate if switching on the heat rejector leads to a global benefit or not. Δ_T is defined as the difference between the water loop temperature at the exhaust of the thermal storage and the heat sink temperature (outdoor air). Outdoor air drybulb temperature is considered if the heat rejector is a Dry Fluid Cooler (DFC or DC). Wetbulb temperature is used in case of a Cooling Tower (CT). If such difference is greater than a given control value (often defined as 10°C) switching on the heat rejector is beneficial. On the opposite using the heat rejector would not be efficient and one can assume that the temperature of the loop can increase until the previous condition is achieved. When operated, the heat rejector is controlled in order to maintain the exhaust temperature to the maximal loop temperature set point.

Even if it is not implemented in the program and in case of heat wave, the heat rejector could operate at full load during the night in order to decrease the water loop temperature to a low level. During the day, the loop temperature would increase continuously. This control strategy would allow electrical load shifting from working hours to non-working hours.

PUMP IS ON
PUMP SIMULATION *//Pump consumption*

Finally, the pump consumption is computed and the plant exhaust temperature is computed as an output of the control algorithm.

ELSE *//Loop is not operated (pumps are off)*
PUMP IS OFF
HEAT REJECTOR IS OFF
HEAT SOURCE IS OFF

ENDIF

If the loop is not operated (for example, in the case of a scheduled control of the circulation pumps), no electrical consumption is considered.

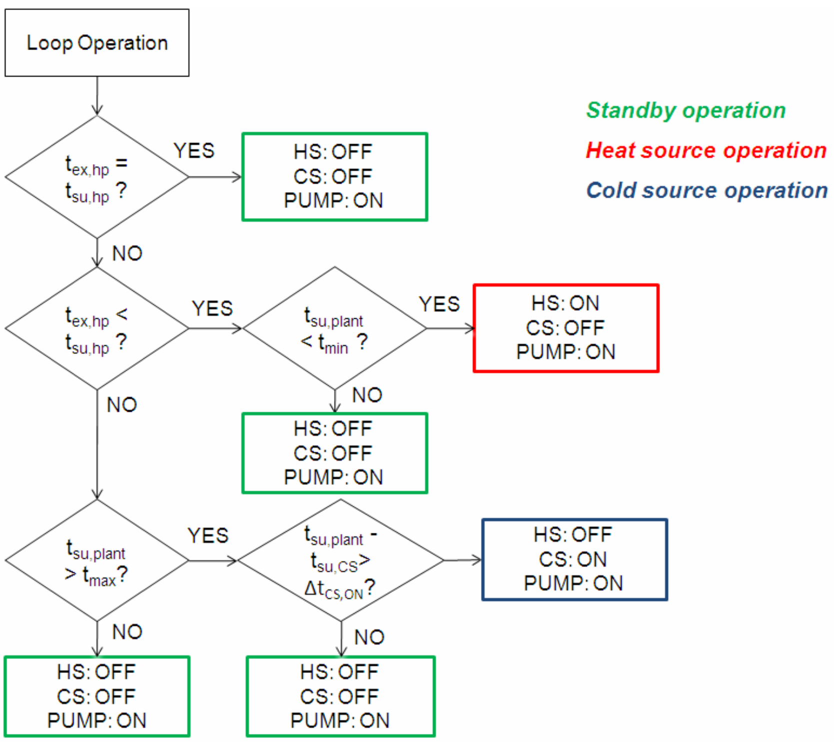


Figure 30: CT/DC (Cold Source – CS) and boiler (Heat Source – HS) water loop control algorithm

Model 4b: Heat rejector and GHX

This type of water loop considers that the ground heat exchanger (GHX) is used as a heat source device but also in order to reject heat when heat pumps operate in cooling mode. Like in the previous model the control algorithm starts with the calculation of the loop thermal balance. Because of the use of a GHX, the whole loop is circulated with glycol water.

$t_{f,su,hp} = t_{f,ex,pump,t-1}$ *//supply heat pump temp. = exhaust pump temp. at previous time step*
HEAT PUMP SIMULATION – COOLING
HEATPUMP SIMULATION – HEATING
$\dot{Q}_{loop} = \dot{Q}_{cd,hp,cooling} - \dot{Q}_{ev,hp,heating}$ *//Loop heat balance (>0 = rejected to the loop; <0 = absorbed from the loop)*

$t_{f,ex,hp} = t_{f,su,hp} + \frac{\dot{Q}_{loop}}{\dot{M}_{f,loop} * c_{pf}}$ *//exhaust heat pump temperature*

In this first part of the program, heat pumps operation is simulated in order to compute their performances (compressor and fan electricity consumptions) ant the amount of heat rejected to/absorbed from the loop. Heat pump consumption is null if there is no demand or if the loop is not in operation (circulation pumps are OFF).

LOOP THERMAL INERTIA CALCULATION (1st order model)
$t_{f,su,plant} = t_{f,ex,loop}$ *//supply plant temperature*

The thermal inertia of the loop is simulated by means of the first order capacitance method. The plant supply temperature is equal to the loop exhaust temperature.

IF $f_{loop} > 0$ THEN *//Loop is in operation (pumps are working)*

IF $t_{f,ex,hp} = t_{f,su,hp}$ THEN
PERFECTLY BALANCED OPERATION OR NO DEMAND
HEAT REJECTOR IS OFF
GHX IS OFF
ENDIF

If exhaust and supply heat pump temperatures are (almost) equal, the loop is perfectly balanced or there is no heating or cooling demand. In both case, the heat rejector and the GHX are not used and the loop temperature is floating.

IF $t_{f,ex,hp} < t_{f,su,hp}$ THEN *//Heating dominated operation*
HEAT REJECTOR IS OFF
IF $t_{f,su,plant} > t_{f,loop,min}$ THEN
GHX IS OFF *// Free floating loop temperature*
ELSE
GHX IS ON
GHX SIMULATION *//GHX pump consumption and exhaust temperature*
ENDIF
ENDIF

In case of "heating dominated operation", the GHX is used only if the loop temperature is below the minimal loop temperature set point. The value of this lower temperature limit is usually between 0 and 3°C (depending of the size of the borefield). If glycol water is circulated through the GHX, the GHX simulation model is used to compute the exhaust temperature and no flow bypasses the GHX. If the GHX is not used, the GHX is completely bypassed. This "On-Off" operation of the GHX results in a variable GHX exhaust temperature, depending on the ground temperature.

IF $t_{f,ex,hp} > t_{f,su,hp}$ THEN	*//Cooling dominated operation*
IF $t_{f,su,plant} < t_{f,loop,max}$ THEN	
HEAT REJECTOR IS OFF	*// free floating loop temperature*
IF $t_{f,ex,CS} > t_{f,loop,GHX}$ THEN	*// $t_{f,loop,GHX}$ is the temperature set point for using the GHX in cooling mode*
GHX IS ON	
GHX SIMULATION	*//GHX pump consumption and exhaust temperature*
ELSE	
GHX IS OFF	*// Free floating loop temperature*
ENDIF	
ELSE	
IF $t_{f,su,plant} - t_{su,CS} > \Delta t_{cs,ON}$ THEN	*//tsu,CS is outdoor drybulb temperature if DFC or wetbulb temperature if CT*
HEAT REJECTOR IS ON	
HEAT REJECTOR SIMULATION	*//Auxiliaries (fan and pump) consumption and exhaust temperature*
IF $t_{f,ex,CS} > t_{f,loop,GHX}$ THEN	*// $t_{f,loop,GHX}$ is the temperature set point for using the GHX in cooling mode*
GHX IS ON	
GHX SIMULATION	*//GHX pump consumption and exhaust temperature*
ELSE	
GHX IS OFF	*// Free floating loop temperature*
ENDIF	
ELSE	
HEAT REJECTOR IS OFF	
IF $t_{f,ex,CS} > t_{f,loop,GHX}$ THEN	*// $t_{f,loop,GHX}$ is the temperature set point for using the GHX in cooling mode*
GHX IS ON	
GHX SIMULATION	*//GHX pump consumption and exhaust temperature*
ELSE	
GHX IS OFF	*// Free floating loop temperature*
ENDIF	
ENDIF	
ENDIF	
HEAT SOURCE IS OFF	
ENDIF	

In case of "cooling dominated operation", the heat rejector is used only if two conditions are satisfied. If the heat rejector supply temperature is above the upper loop temperature limit ($t_{f,loop,max}$), one must evaluate if switching on the heat rejector leads to a global benefit or not. Δ_T is defined as the difference between the water loop temperature at the exhaust of the thermal storage and the heat sink temperature (outdoor air). Outdoor air drybulb temperature is considered if the heat rejector is a Dry Fluid Cooler (DFC or DC). Wetbulb temperature is used in case of a Cooling Tower (CT). If such difference is greater than a given control value (often defined as 10°C) switching on the heat rejector is beneficial. On the opposite using the heat rejector would not be efficient and one can assume that the temperature of the loop can increase until the previous condition is achieved. When operated, the heat rejector is controlled in order to maintain the exhaust temperature to the maximal loop temperature set point.

In addition to the heat rejector control, if the loop temperature is higher than the GHX cooling set point, the GHX is used for "passive cooling" of the loop. This temperature set point is usually 7 to 10°C below the heat rejector setpoint (30-35°C).

Once again, some optimized control strategies involving sequencing the operation of the GHX and the heat rejector to maximize "passive cooling" and reduce heat rejector energy consumption can be envisaged but are not implemented in the present program.

PUMP IS ON
PUMP SIMULATION *//Pump consumption*

Finally, the pump consumption is computed and the plant exhaust temperature is computed as an output of the control algorithm.

ELSE *//Loop is not operated (pumps are off)*
PUMP IS OFF
HEAT REJECTOR IS OFF
HEAT SOURCE IS OFF

ENDIF

If the loop is not operated (for example, in the case of a scheduled control of the circulation pumps), no electrical consumption is considered.

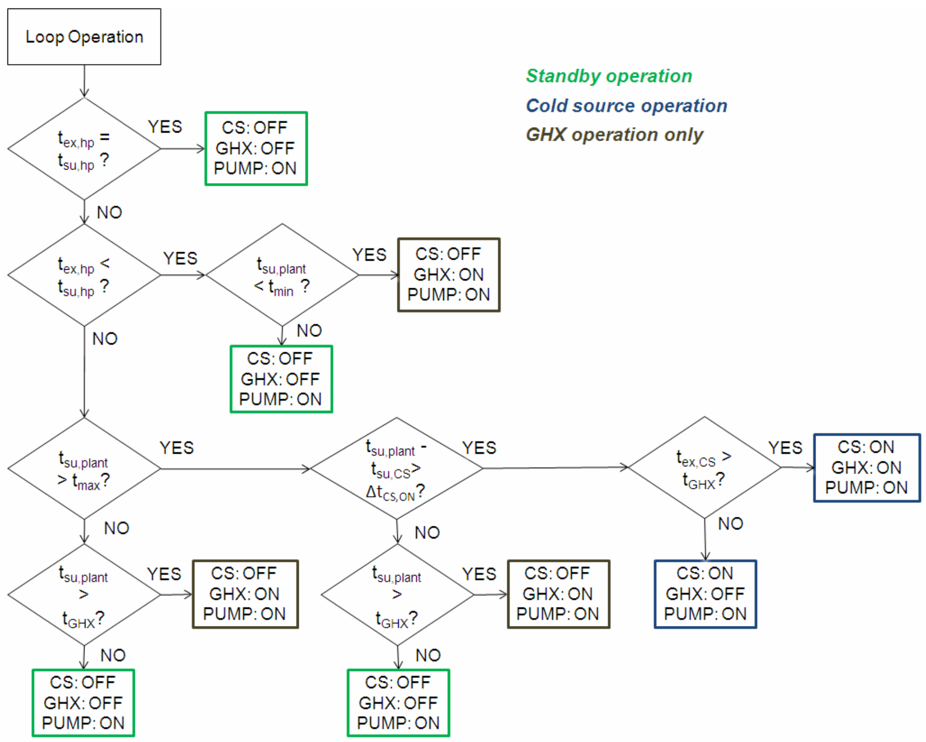


Figure 31 CT/DC and GHX water loop control algorithm

Model 4c: GHX and boiler

This type of water loop considers that the ground heat exchanger (GHX) is used as a heat rejector (heat sink) device but also as a heat source when heat pumps operate in heating mode. Like in the previous model the control algorithm starts with the calculation of the loop thermal balance. Because of the use of a GHX, the whole loop is circulated with glycol water. A boiler is used as a secondary heat source in case of a too limited capacity of the GHX.

$t_{f,su,hp} = t_{f,ex,pump,t-1}$ *//supply heat pump temp. = exhaust pump temp. at previous time step*
HEAT PUMP SIMULATION – COOLING
HEATPUMP SIMULATION – HEATING
$\dot{Q}_{loop} = \dot{Q}_{cd,hp,cooling} - \dot{Q}_{ev,hp,heating}$ *//Loop heat balance (>0 = rejected to the loop; <0 = absorbed from the loop)*

$t_{f,ex,hp} = t_{f,su,hp} + \frac{\dot{Q}_{loop}}{\dot{M}_{f,loop} * c_{pf}}$ *//exhaust heat pump temperature*

In this first part of the program, heat pumps operation is simulated in order to compute their performances (compressor and fan electricity consumptions) ant the amount of heat rejected to/absorbed from the loop. Heat pump consumption is null if there is no demand or if the loop is not in operation (circulation pumps are OFF).

LOOP THERMAL INERTIA CALCULATION (1st order model)
$t_{f,su,plant} = t_{f,ex,loop}$ *//supply plant temperature*

The thermal inertia of the loop is simulated by means of the first order capacitance method. The plant supply temperature is equal to the loop exhaust temperature.

IF $f_{loop} > 0$ THEN *//Loop is in operation (pumps are working)*

IF $t_{f,ex,hp} = t_{f,su,hp}$ THEN
PERFECTLY BALANCED OPERATION OR NO DEMAND
GHX IS OFF
HEAT SOURCE IS OFF
ENDIF

If exhaust and supply heat pump temperatures are (almost) equal, the loop is perfectly balanced or there is no heating or cooling demand. In both case, the the GHX and the heat source are not used and the loop temperature is floating.

IF $t_{f,ex,hp} < t_{f,su,hp}$ THEN *//Heating dominated operation*

IF $t_{f,su,plant} > t_{f,loop,min}$ THEN
GHX IS OFF *// Free floating loop temperature*
ELSE
GHX IS ON
GHX SIMULATION *//GHX pump consumption and exhaust temperature*

IF $t_{f,ex,GHX} > t_{f,loop,HS}$ THEN *// $t_{f,loop,HS}$ is the temperature set point for using the secondary HS in heating mode*
HEAT SOURCE IS OFF
ELSE
HEAT SOURCE IS ON

HEAT SOURCE SIMULATION //*Fuel and auxiliaries consumptions*
ENDIF
ENDIF
ENDIF

In case of "heating dominated operation", the GHX is used only if the loop temperature is below the low loop temperature set point. The value of this lower temperature limit is usually around 5°C (depending of the size of the borefield). If glycol water is circulated through the GHX, the GHX simulation model is used to compute the exhaust temperature and no flow bypasses the GHX. If the GHX is not used, the GHX is completely bypassed. This "On-Off" operation of the GHX results in a variable GHX exhaust temperature, depending on the ground temperature. If the GHX exhaust temperature is below the boiler operation set point (a few degrees below the GHX temperature set point), the boiler is used to increase this temperature. The use of the boiler is well adapted in case of highly heating dominated demand as it allows the loop temperature not to fall below a damageable point no matter how the ground heat exchanger is sized.

IF $t_{f,ex,hp} > t_{f,su,hp}$ THEN //*Cooling dominated operation*
IF $t_{f,su,plant} < t_{f,loop,max}$ THEN
GHX IS OFF // *free floating loop temperature*
ELSE
GHX IS ON
GHX SIMULATION
ENDIF
HEAT SOURCE IS OFF
ENDIF

As the temperature is still in the acceptable range the system uses its own thermal inertia; the ground heat exchanger is bypassed. The maximum temperature of the loop before the GHX is activated is here defined around 20°C.

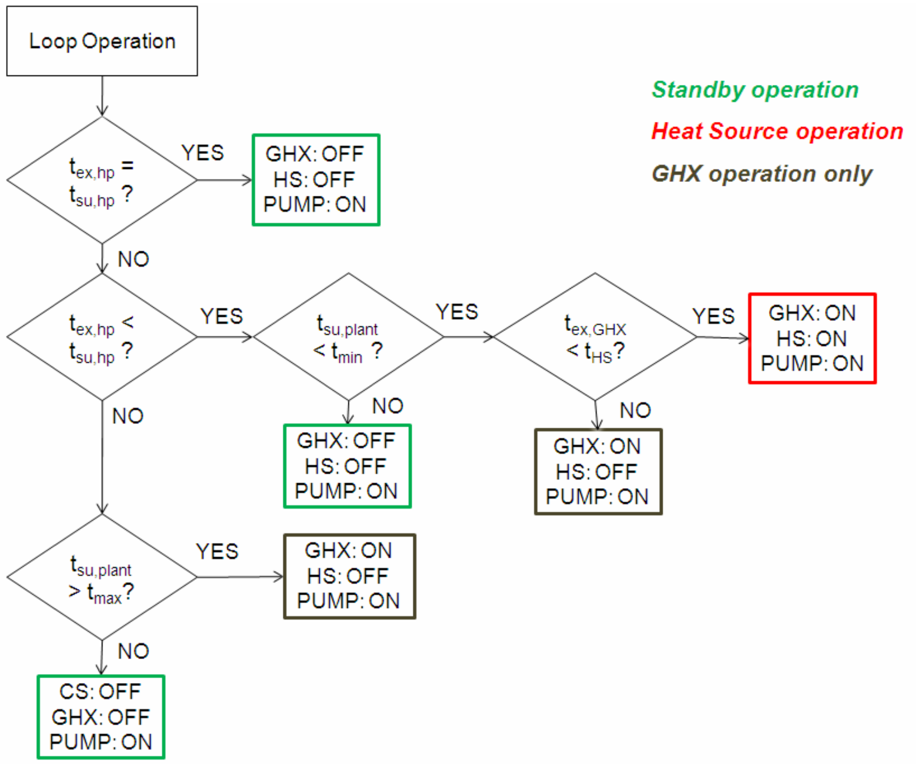


Figure 313: GHX and boiler water loop control algorithm

1.2.7. System 5: Ground coupled heat pump system

Model 5a: reversible heat pump coupled with a ground heat exchanger

General description

The system can be reversed by means of a refrigerant change over (4 ways valve) which inverses the refrigerant flow passage into the two exchangers:

- **In cooling mode,** heat of the condenser is rejected to the ground thanks to the ground heat exchanger, while the evaporator transfers cooling power to the distribution system.

- **In heating mode,** the evaporator absorbs heat from the ground heat exchanger, while the condenser transfers heating power to the distribution system.

The water-to-water reversible heat pump is installed in combination with back up boilers, for the following reasons:

1. To supply heating power when heating and cooling loads are simultaneous: when the chiller works in cooling mode, it cannot provide any heating power.
2. To complete heating power when the heat pump cannot provide the entire heating demand.
3. To supply the entire heating power when, at very low temperatures, the boiler has better performance than the reversible heat pump, or when the temperatures are under the unit working range.

Such systems can reach good efficiency if the heating and cooling demands are alternate, as the system is designed to be reversible.

This type of heat pump modeled the following components:

- Water-to-water reversible heat pump
- Boiler
- Ground heat exchanger
- Pumps

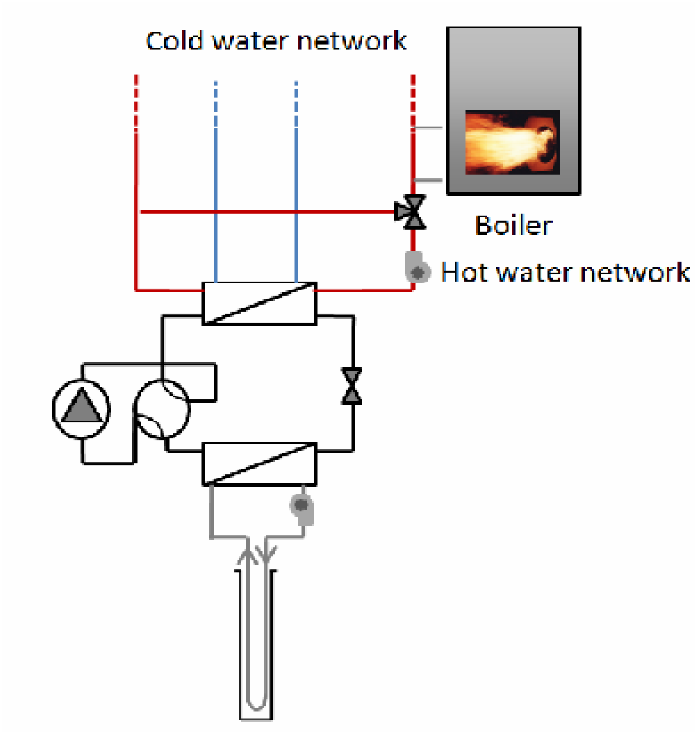


Figure33: Model 5a: reversible heat pump coupled with a ground heat exchanger

Heat source/sink

During winter and in heating mode, the ground provides heat to the evaporator. In cooling mode, the heat pump works as chiller and provides the cooling power to the cold water network.

Control algorithm

The control procedure is explained below. Once again, the priority is given to the cold generation if there is a cooling demand. As usual, the control algorithm starts with the declaration of the variables.

DECLARATION OF VARIABLES

The return temperatures of the networks are obviously essentials for the control algorithm.

$$t_{w,hw,return} = t_{w,hw,set} - \frac{\dot{Q}_{heat}}{c_{p,w} * \dot{M}_{hw}}$$

$$t_{w,cw,return} = t_{w,cw,set} + \frac{\dot{Q}_{cold}}{c_{p,w} * \dot{M}_{cw}}$$

Once again, it is assumed that the set points are achieved at any times.

IF $\dot{Q}_{heat} = 0$ THEN
 IF $\dot{Q}_{cold} = 0$ THEN
 NOTHING
 ELSE
 HEAT LOSSES CALCULATION Cold network
 CALL CHILLER PROCEDURE
 CALL GROUND HEAT EXCHANGER

PUMP CONSUMPTION CALCULATION
ENDIF

When there is simultaneously no heat and no cool demand, consumptions (of the pumps and of the heat pump) are equal to zero and heat losses are not taken into account.

If the cool demand is not equal to zero, the heat pump works as a chiller and provides the entire cooling power (equal to the sum of the cooling demand and the heat losses in the cold network). The chiller procedure determines the electric consumption of the chiller. The pump electricity consumption is also taken into account. The ground heat exchanger is used to reject heat to the ground.

ELSE
IF $\dot{Q}_{cold} = 0$ THEN
HEAT LOSSES CALCULATION Hot network
IF $t_{su;ev} < t_{\lim}$ THEN
CALL BOILER PROCEDURE
PUMP CONSUMPTION CALCULATION

When there is a heating demand and no cooling demand, the first step is to take the heat losses in the hot network into account. The supply evaporator temperature determines whether the heat pump will be activated or not in order to avoid a poor COP. If the supply evaporator temperature is below -5°C, the boiler provides the entire heating demand to the hot water network. As usual, the electric consumption due to the pumps is also calculated.

ELSE
MAXIMUM CAPACITY CALCULATION
IF $\dot{Q}_{heat} < \dot{Q}_{HP,max}$ THEN
$t_{w,ex,HP} = t_{w,ex,set}$
CALL HEAT PUMP PROCEDURE
CALL GROUND HEAT EXCHANGER
ELSE
$$\dot{Q}_{HP} = \dot{Q}_{HP,max}$$
$$t_{w,ex,HP} = t_{w,su,HP} + \frac{\dot{Q}_{HP}}{c_{p,w} * \dot{M}_{w,hw}}$$
CALL HEAT PUMP PROCEDURE
CALL BOILER PROCEDURE
CALL GROUND HEAT EXCHANGER
ENDIF
PUMP CONSUMPTION CALCULATION
ENDIF

If the supply evaporator temperature is high enough, the maximum capacity of the heat pump is defined in relation with the operating conditions. If the maximum capacity is equal or larger than the heating demand, the heat pump is able to provide the entire heating demand. In the opposite case, the system needs the help of the boiler to reach the set temperature. The pump and the heat pump electricity consumptions are obviously determined.

ELSE
HEAT LOSSES CALCULATION Hot network
HEAT LOSSES CALCULATION Cold network
CALL CHILLER PROCEDURE

CALL BOILER PROCEDURE
PUMP CONSUMPTION CALCULATION
CALL GROUND HEAT EXCHANGER
ENDIF
ENDIF

When there is a simultaneous heating and cooling demand, the heat losses in the hot and cold networks are taken into account. The heat pump works as a chiller and provides the entire cooling demand while the boiler provides the entire heating demand. Both networks are considered separately and we are in the same configurations as in the Model 0: Chiller and boiler independent case.

The control algorithm can be summarized in the following figure:

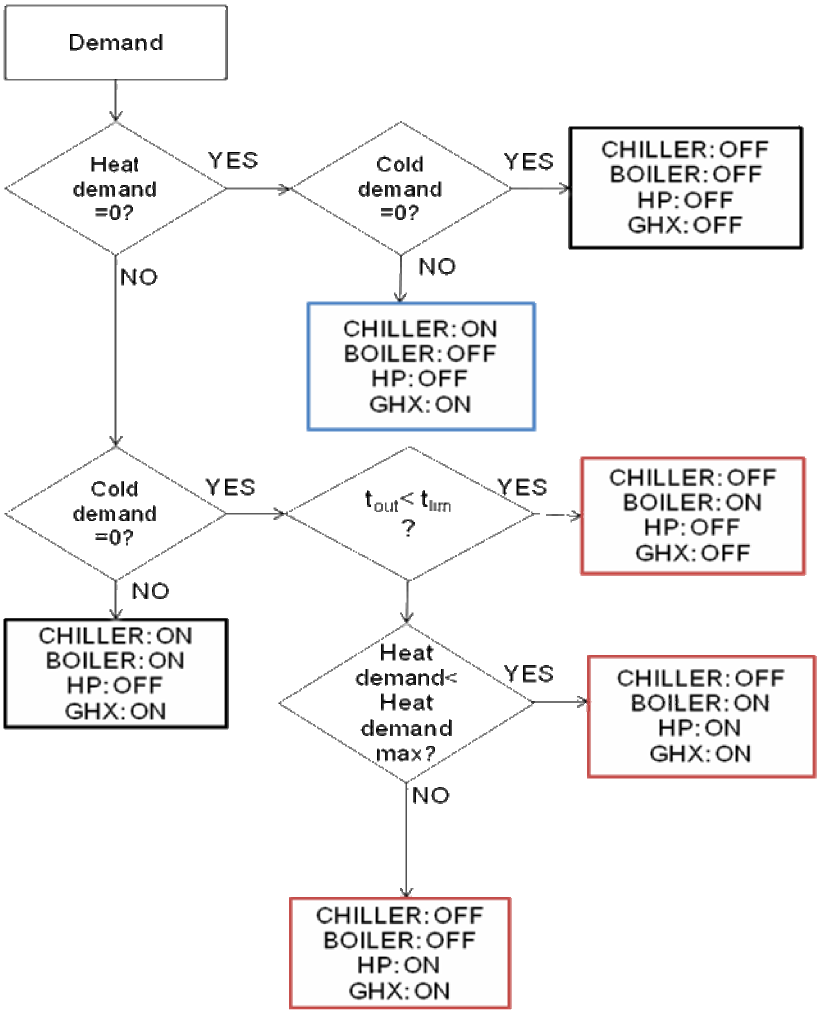


Figure 34: Model 5a – Solving procedure

Model 5b: ground coupled heat pump for both heating and cooling

General description

This system also uses a ground heat exchanger to reject or absorb heat from the ground but contrary to the system 5a, the heat pump is not reversible.

- **In cooling mode**, there are two possibilities:

- the ground is able to provide the cooling demand (passive cooling).

- the heat pump works as a chiller and the condenser heat is rejected to the ground thanks to a heat exchanger in contact with the ground heat exchanger network.

- **In heating mode,** the heat pump (and eventually the back up boiler) transfers heating power to the hot network. In this case, the evaporator absorbs heat from the ground thanks to the ground heat exchanger.

The boiler works when the heat pump is not able to provide the entire heating demand.

This type of system requires the following components:

- Water-to-water heat pump
- Back up boiler
- Ground heat exchanger
- Heat exchanger
- Pumps

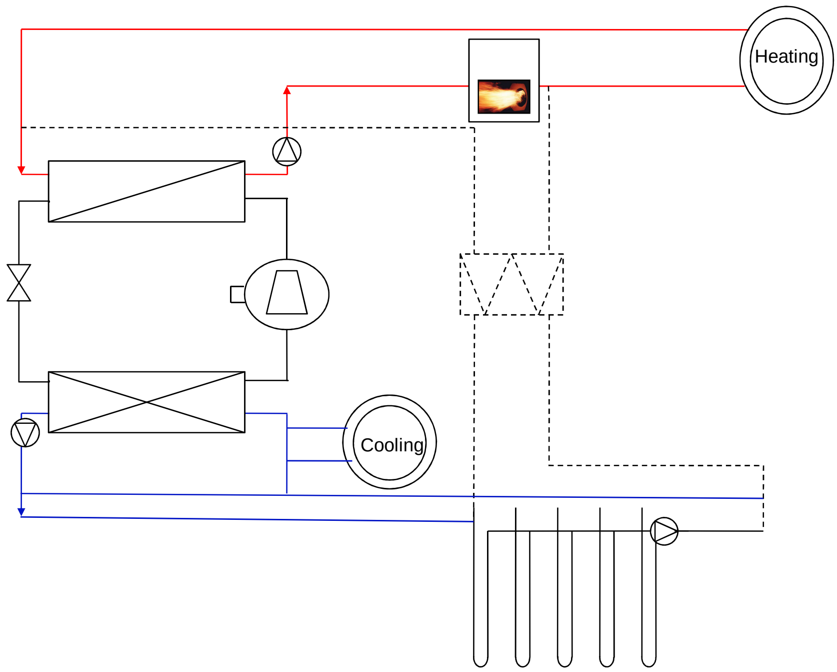


Figure35: Model 5b: ground coupled heat pump for both heating and cooling

Heat source/sink

During winter and in heating mode, the ground provides heat to the evaporator.
In cooling mode, the heat pump works as chiller and provides the cooling power to the cold water networks. The ground absorbs heat from the condenser.

Control algorithm

As usual, the control algorithm starts with the declaration of the variable.

DECLARATION OF THE VARIABLES

The return temperatures of the networks are obviously used in the control algorithm and are defined by:

$$t_{w,hw,return} = t_{w,hw,set} - \frac{\dot{Q}_{heat}}{c_{p,w} * \dot{M}_{hw}}$$

$$t_{w,cw,return} = t_{w,cw,set} + \frac{\dot{Q}_{cold}}{c_{p,w} * \dot{M}_{cw}}$$

IF $\dot{Q}_{heat} = 0$ THEN

IF $\dot{Q}_{cold} = 0$ THEN
NOTHING

When the heating and the cooling demand are equal to zero, no consumption is calculated.

ELSE
CALL GROUND HEAT EXCHANGER
HEAT LOSSES CALCULATION Cold network
IF $t_{w;ex;loop} < 7$ THEN
PUMP CONSUMPTION CALCULATION
ELSE
CALL CHILLER PROCEDURE
CALL GROUND HEAT EXCHANGER
PUMP CONSUMPTION CALCULATION
END
ENDIF

If the cooling demand is not equal to zero, the first step is to determine if the passive chilling is feasible (thanks to the ground heat exchanger procedure). In this case (if the exhaust heat exchanger water temperature is lower than 7°C), only pumps consumption is calculated. In the opposite case, the evaporator provides the cooling power to the cold network. The condenser heat is rejected to the ground thanks to the heat exchanger in contact with the ground heat exchanger network. Pumps and chiller consumption is obviously calculated.

ELSE

IF $\dot{Q}_{cold} = 0$ THEN
HEAT LOSSES CALCULATION Hot network
MAXIMUM CAPACITY CALCULATION
IF $\dot{Q}_{heat} < \dot{Q}_{heat,max}$ THEN
$t_{w,ex,HP} = t_{w,ex,set}$
CALL HEAT PUMP PROCEDURE
CALL GROUND HEAT EXCHANGER
ELSE
$\dot{Q}_{HP} = \dot{Q}_{heat,max}$

$$t_{w,ex,HP} = t_{w,su,HP} + \frac{\dot{Q}_{HP}}{c_{p,w} * \dot{M}_{w,hw}}$$

CALL HEAT PUMP PROCEDURE

CALL BOILER PROCEDURE
CALL GROUND HEAT EXCHANGER
ENDIF
PUMP CONSUMPTION CALCULATION

When there is only heating demand, the maximum capacity of the heat pump is defined in relation with the operating conditions. If the maximum capacity is equal or larger than the entire heating demand (which is the sum of the heating demand and the heat losses in the hot network), the heat pump is able to provide the entire heating demand. In the opposite case, the system needs the help of the boiler to reach the set temperature. The supply boiler temperature is the exhaust heat pump temperature assuming that the device at its maximum capacity. The ground heat exchanger provides the heat to the evaporator. For both cases, the pump and the heat pump electricity consumption is obviously determined.

ELSE
HEAT LOSSES CALCULATION Hot network
HEAT LOSSES CALCULATION Cold network
CALL GROUND HEAT EXCHANGER
IF $t_{w;ex;loop} < 7$ THEN
MAXIMUM CAPACITY CALCULATION
IF $\dot{Q}_{heat} < \dot{Q}_{heat,max}$ THEN
$t_{w,ex,HP} = t_{w,ex,set}$
CALL HEAT PUMP PROCEDURE
CALL GROUND HEAT EXCHANGER
ELSE
$\dot{Q}_{HP} = \dot{Q}_{heat,max}$
$$t_{w,ex,HP} = t_{w,su,HP} + \frac{\dot{Q}_{HP}}{c_{p,w} * \dot{M}_{w,hw}}$$
CALL HEAT PUMP PROCEDURE
CALL BOILER PROCEDURE
CALL GROUND HEAT EXCHANGER
ENDIF
PUMP CONSUMPTION CALCULATION

When there is a simultaneous heating and cooling demand, the first step is to determine if the passive cooling is feasible. In this case, the heating demand is satisfied thanks to the heat pump (and eventually the boiler).

ELSE
CALL CHILLER PROCEDURE
IF $\dot{Q}_{heat} < \dot{Q}_{cd}$ THEN
$t_{w,ex,HP} = t_{w,ex,set}$
CALL HEAT PUMP PROCEDURE
CALL GROUND HEAT EXCHANGER
ELSE
$$t_{w,ex,HP} = t_{w,su,HP} + \frac{\dot{Q}_{cd}}{c_{p,w} * \dot{M}_{w,hw}}$$
CALL HEAT PUMP PROCEDURE
CALL BOILER PROCEDURE
CALL GROUND HEAT EXCHANGER
ENDIF

```
                              PUMP CONSUMPTION CALCULATION
          ENDIF
          ENDIF
ENDIF
```

If passive cooling is not feasible, the heat pump provides the cooling demand to the cold network thanks to the evaporator. In this case, recovery is possible. If the heat provided by the condenser is not large enough to supply the entire heating demand, the system needs the help of the boiler to reach the set temperature. For these cases, the pump and the heat pump electricity consumption is obviously determined.

The control algorithm can be summarized in the following figure:

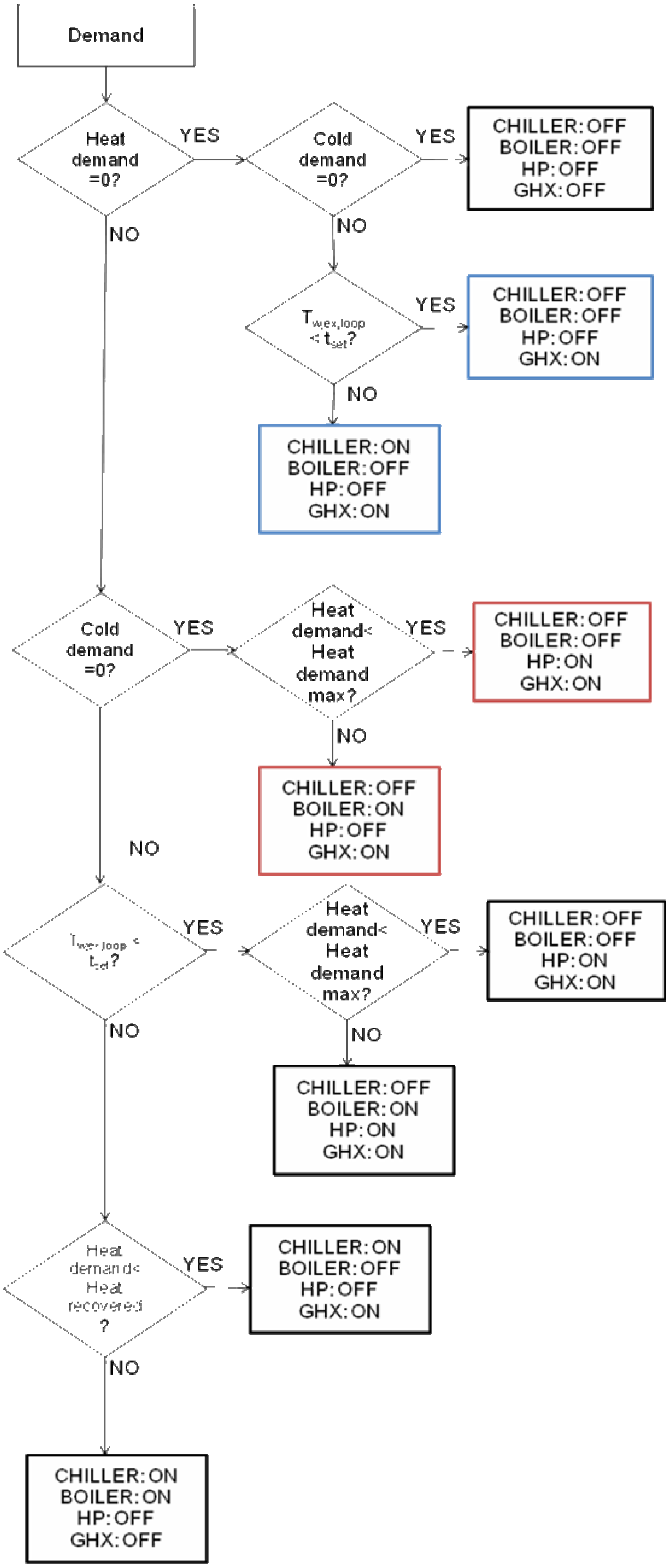


Figure 36: Model 5b – Solving procedure

IV. Parametric study

1. Introduction

This chapter aims to assess the saving potentials in terms of primary energy and CO2 emissions of different heat pump systems in office buildings by means of a parametric study. Four types of air-conditioned office buildings, defined based on a study of the French building stock, are considered for the simulations: from small suburban office buildings (1000 m^2) to large office buildings (15000 m^2). Five climatic zones, defined to be representative of the Europe-15 are used for the simulations.

2. Methodology

Parametric study cases

The first step of this study was to generate the heating and cooling demand profiles for the considered cases. This was done by means of the building energy simulation tool Consoclim [7]. Twenty cases have been considered at this stage of the study, based on 5 different climatic zones and 4 types of buildings. Then, 5 different heat pump systems have been considered and compared. Finally, one hundred simulation runs have been done in this parametric study (Figure 37: Parametric study cases).

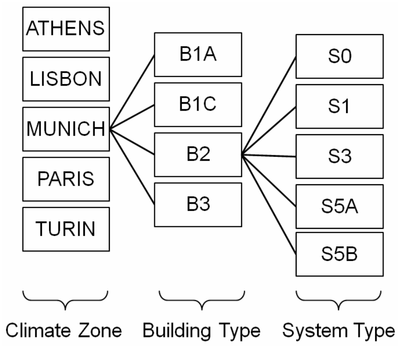


Figure 37: Parametric study cases

Climatic zones

Europe-15 is divided into five distinct climatic zones (Figure 148). The locations of a same climatic zone are characterized by similar heating and cooling degree days. According to [1], each climatic zone is represented by one location whose typical weather data are available: Athens, Lisbon, Turin, Paris and Munich.

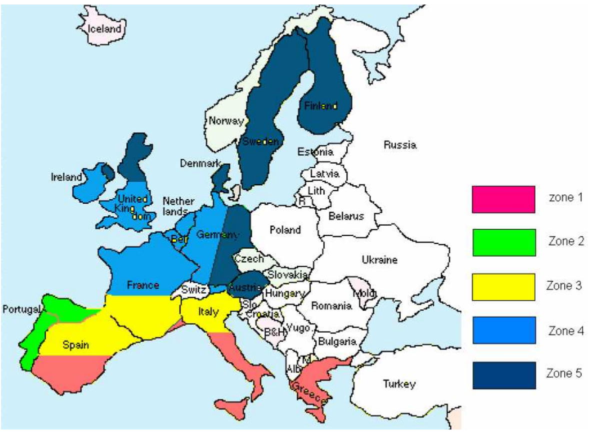


Figure 148: Climatic zones

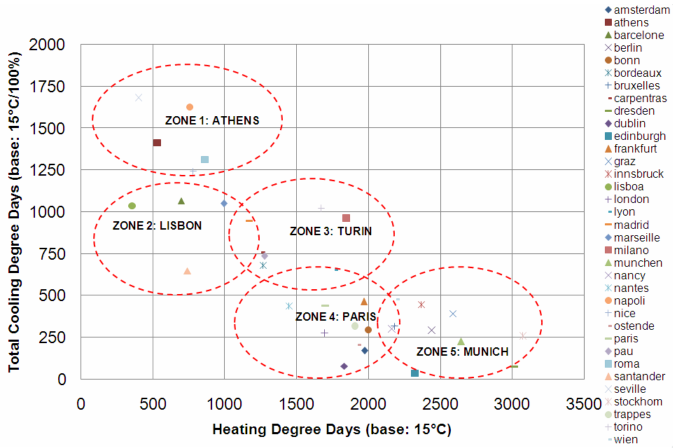


Figure 159: Heating degree days

Building Types

Four types of buildings are considered in the present study:

- Type 1A: large glazed office buildings with peripheral zones (offices) and core zones (meeting rooms); floor: 15000 m² ; window-to-wall ratio : 35%
- Type 1C: large glazed office buildings with peripheral zones only (offices and meeting rooms); floor: 15000 m² ; window-to-wall ratio : 45%
- Type 2: medium office buildings; floor : 5000 m² ; window-to-wall ratio : 22%
- Type 3: small office buildings; floor : 1000 m² ; window-to-wall ratio : 40%

All the buildings have similar orientations, envelope components, internal loads, occupancy profiles, internal set points and infiltration and ventilation rates. All these parameters have been fixed to average values. [1] give more details about the considered cases.

Secondary HVAC system

Only air-water systems (single duct CAV system and fan coil units) are considered in the present study. Fan coil units installed in the zones provide local heating and cooling. The CAV air handling unit provides fresh air and includes a heat recovery system. Humidity control is not taken into account.

In the next steps of this study, all the heating secondary HVAC components are supposed to be operated with low temperature hot water at a (constant) temperature of 45°C. The (constant) chilled water temperature is fixed at 7°C.

Heat pump system configurations

Five heat and cold production systems are considered in the present parametric study:

- System 0: classical separated heat and cold productions (independent boiler and chiller)
- System 1: reversible air-to-water heat pump system sized for cooling (with backup boiler for heating). Priority is given to cooling.
- System 3: dual-condenser heat pump system sized for cooling (with backup boiler for heating). Priority is given to cooling.
- System 5A: reversible ground coupled heat pump (sized for cooling) with backup boiler.
- System 5B: ground coupled heat pump (sized for cooling) with direct ground cooling and backup boiler.

Inputs/Parameters/Outputs

The main inputs of the simulation models developed are the pre-computed hourly heating and cooling demands and the corresponding weather data.

System	EER	COP
0	2.5	-
1	2.5	2.7
3	2.5	3.5
5A	3.8	3.6
5B	3.8	3.6

Table 7: Heat pumps rating performance

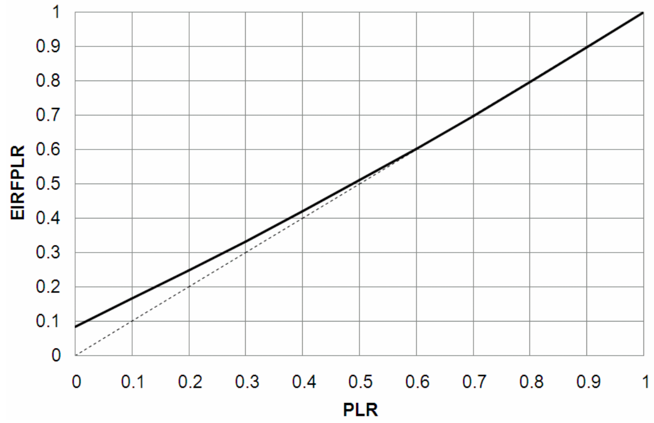


Figure 40: Heat pumps part load performance (Electricity Input Ratio as function of the Part Load Ratio)

The parameters of the models include performance and capacities of the considered HVAC components. The capacities of the equipments vary from case to case and are determined based on the same rules of thumb (depending of the heating and cooling peaks). The rating and full load performance of the heat pumps depend of the configuration and are based on actual manufacturer data (for air-to-water, water-to-water and dual-condenser heat pumps) chosen as representative of the actual heat pump market. On the contrary, the part load performances follow the same generic trend (Figure40) for the different systems. The rating conditions for each system are defined according to [63]. A nominal boiler efficiency of 90% is used for all the simulations.

3. Results

Final and primary energy consumption

Computed electricity and gas consumptions have been compared in terms of final and primary energy consumptions. Because up to data values of conversion factors were not available, the average EU values given in [62] have been used to compare the considered heat pump systems (1, 3, 5A and 5B) to the reference system (0). A conversion factor of 3.31 is used for electricity and a factor of 1.35 is used for natural gas.

The relative variations (in % of the reference consumption, positive values correspond to an increase of the consumption and negative values correspond to a decrease of the consumption) are given in Table 8. Since the differences between the considered building types are limited, average values are presented. It appears that the reversible heat pump systems (1,5A and 5B) provide larger natural gas savings than heat pump systems able to handle heat recovery only (system 3). This is mainly due to the fact that the selected cases (buildings and climate zones) have rarely simultaneous heating and cooling demands. This trend is confirmed when looking at primary energy savings. It should be noticed that, the very small natural gas savings provided by the system 3 do not balance the increases of the electricity consumption related to the use of the heat pump. This finally results in an increase of the primary energy consumption.

%	HP system	Athens	Lisbon	Munich	Paris	Turin
Natural Gas	1	-95	-92	-70	-97	-92
	3	-4	-6	-2	-1	-2
	5A	-95	-92	-98	-99	-97
	5B	-99	-99	-100	-100	-100
Electricity	1	18	23	108	138	75
	3	8	8	7	6	7
	5A	-3	7	107	107	50
	5B	-3	8	106	106	50
Primary Energy	1	0	-1	2	1	3
	3	6	5	2	2	3
	5A	-18	-14	-15	-13	-13
	5B	-19	-15	-16	-14	-15

Table 8 : Relative energy consumption variation compared to system S0 (average for the four building types considered)

Natural gas savings resulting of the use of the system 1 (reversible air-to-water heat pump) are represented as a function of the percentage of overlap between heating and cooling demands (compared on an hourly basis) in Figure 41: Natural gas savings of the system 1. As mentioned above, the potential of this heat pump system is quite high in the considered cases and decreases slowly as the percentage of overlap between the two demand profiles increases. The effect of the sizing can also be highlighted: on the contrary to the other cases, the high difference between the heating and cooling peaks in Munich leads to a non-sufficient capacity of the heat pump (sized for cooling) in heating mode. The balanced peak demands of Paris and the bigger cooling peaks in Turin, Lisbon and Athens lead to a sufficient capacity of the heat pump in heating mode.

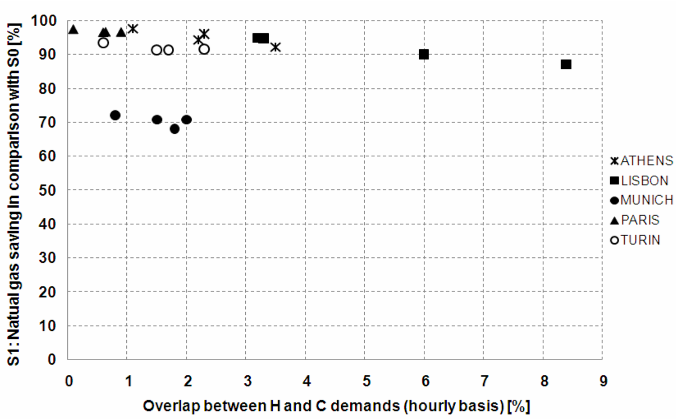


Figure 41: Natural gas savings of the system 1

Seasonal performances of the considered systems can also be studied by means of the simulation models used. The values of the average annual heating COP of the reversible air-to-water heat pump are given in Table 9. As expected, COPs are higher in Athens and Lisbon but stay under 2.3 due to low part load operation (air-to-water heat pump sized for cooling). In colder regions (Paris, Munich and Turin), the average annual COPs are lower and vary between 1.97 and 2.1. Indeed, in these regions, even if heat pumps are operating a higher load rate (balanced peak heating and cooling loads), the negative effect of the lower outdoor temperatures leads to lower seasonal performances.

Location	EER_{avg}	COP_{avg}
ATHENS	2.79	2.19
LISBON	2.79	2.29
MUNICH	2.85	1.97
PARIS	2.81	2.10
TURIN	2.80	2.02

Table 9: Average values of EER and COP (for the system 1)

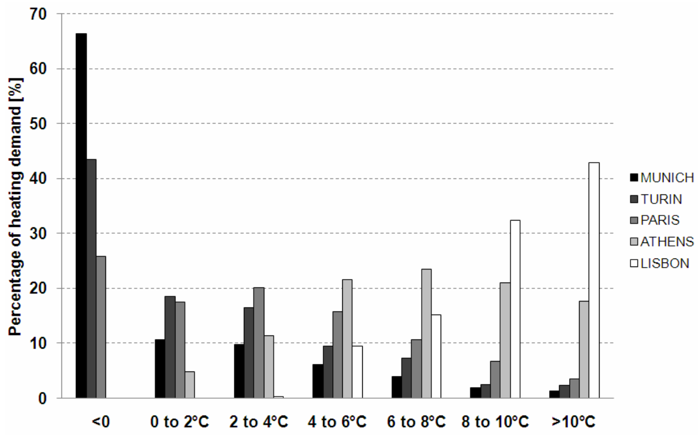


Figure 42: Percentage of heating demand [%] per temperature ranges

Seasonal performances of the considered systems can also be studied by means of the simulation models used. The values of the average annual heating COP of the reversible air-to-water heat pump are given in Table 9: Average values of EER and COP (for the system 1). As expected, COPs are higher in Athens and Lisbon but stay under 2.3 due to low part load operation (air-to-water heat pump sized for cooling). In colder regions (Paris, Munich and Turin), the average annual COPs are lower and vary between 1.97 and 2.1. Indeed, in these regions, even if heat pumps are operating a higher load rate (balanced peak heating and cooling loads), the negative effect of the lower outdoor temperatures leads to lower seasonal performances.

As shown in Figure, the reversible air-to-water heat pump (system 1) results in very small primary energy savings in mild climates (Athens and Lisbon). In colder climates (Munich, Paris and Turin), this system leads to an increase of the primary energy consumption since the reduction of the natural gas consumption is more than compensated by the increase of the electricity consumption. Even if the recovery heat pump (system 3) is less adapted than the other systems in the considered cases (smaller recovery potential), it leads to a reduction of the total primary consumption mainly because of the better EER in cooling mode. Ground coupled heat pump systems (systems 5A and 5B) lead to reductions of total primary energy consumption in all cases (between 13 and 19%) because of their higher COPs in heating mode (water-to-water heat pump with high and quite stable evaporation temperature) and EERs in cooling mode (water cooled chiller with low condensation temperature) compared to other systems.

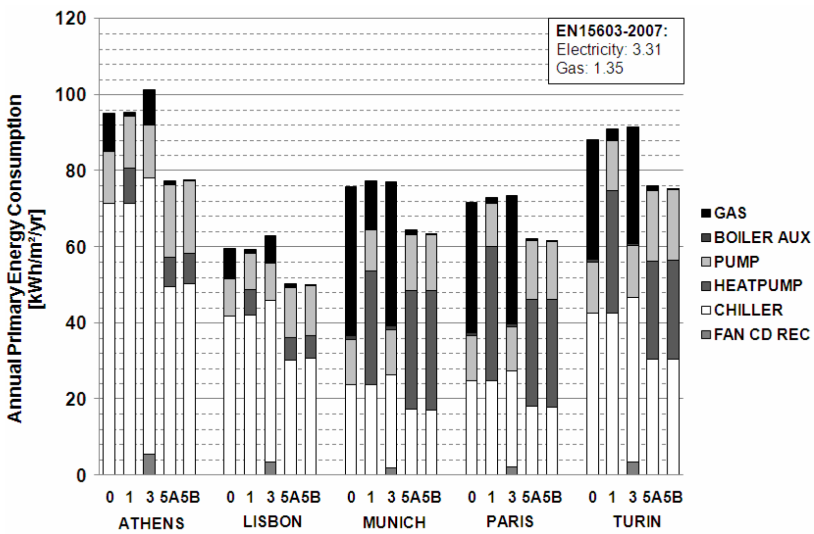


Figure 43: Annual primary energy consumption per square meter of floor area (Type 1A building)

It should be noticed that the small differences in the results provided by systems 5A and 5B are due to the variations of the EER/COP of the heat pump (different for reversible and a non-reversible machines) since the potential gain related to the use of direct ground cooling stays marginal and has not been highlighted in the present study because of the considered simulation hypotheses.

Of course, the previous observations are highly dependent of the arbitrary conversion factors used to convert gas and electricity consumptions in primary energy consumption and of the performance of the considered units. For example, the reversible air-to-water heat pump provides more than 97% of the heating demand of the building located in Paris with an average annual heating COP of 2.10 (compared to a boiler efficiency of 90%) but leads to an increase of 1.7% the primary consumption within the meaning of the European standards (conversion factor of 3.31 for electricity and 1.35 for natural gas). In the meaning of the French standard (2.58 for electricity and 1 for natural gas), the same results lead to an increase of 4.1% of the primary energy consumption.

CO2 emissions

The seasonal performance of the HP systems can also be compared in terms of CO2 emissions. National average annual values of CO2 emissions per consumed kWh of electricity [45] for Greece, Portugal, Germany, France and Italy have been used for the comparison. The same average CO2 emission rate for natural gas has been used for the five considered countries.

In Athens (highest average CO2 emission rate in Europe-15: 0.814 kg CO2/kWh), the reversible air-to-water heat pump (systems) lead to a slight increase of the annual CO2 emissions (+5%). Indeed the ratio of the "electrical" CO2 emission rate (0.814 kg/kWh) to the "gas" CO2 emission rate (0.231 kg/kWh) do not allow to decrease the global CO2 emissions with this system. Some reductions of CO2 emissions (between 14%) are possible but are mainly due to the better cooling EER of the systems 5A and 5B (allowing consistent reduction of the electricity consumption related to cold production). Situations in Munich and Turin are similar and only ground coupled systems can lead to sensible reductions of the annual CO2 emissions (FigureTable 10: Percentage of CO2 emissions variations compared to system S0 (average for the four building types **considered)** and Table 10: Percentage of CO2 emissions variations compared to system S0 (average for the four building types considered).

%	HP system	Athens	Lisbon	Munich	Paris	Turin
CO2	1	5	-3	-2	-74	-1
	3	7	5	2	0	3
	5A	-14	-16	-19	-79	-17
	5B	-14	-17	-21	-80	-19

Table 10: Percentage of CO2 emissions variations compared to system S0 (average for the four building types considered)

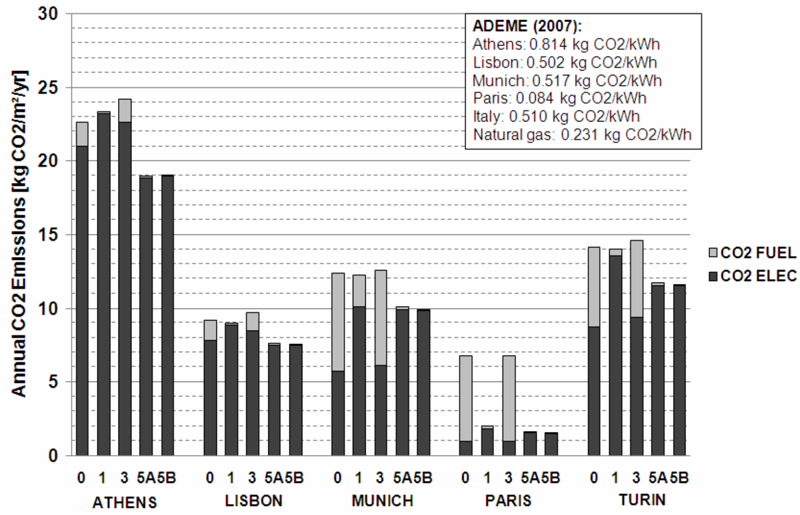


Figure 44: Annual CO_2 emissions per square meter of floor area (Type 1A building)

The very low CO_2 emission rate of France (due to the large part of the electricity produced by nuclear plants) leads to very important reduction of the total CO_2 emissions of the considered buildings (between, and 74 and 80%). Only the recovery heat pump system (system 3) does not lead to a sensible reduction of the CO_2 emissions. It also appears that the very simple reversible air-to-water heat pump system with a quite low seasonal COP (2.1) can provide environmental performance as good as high performance ground coupled heat pump systems.

DISCUSSION

The major influences (effects of the temperature of heat sources and sinks, part load behavior…) and components (heat pump, heat source and heat sink, auxiliaries…) are taken into account in the present models.

Four buildings located in five climate zones and equipped with 5 different heat pump systems have been considered. The analysis of the simulation results led to the following observations:

- Better performances could be obtained by using more efficient heat pumps (higher rating COPs) and low temperature emitters (radiant heating system).
- Only primary HVAC system consumptions have been considered in the present study. This methodology is efficient to assess the potential of such systems but does not allow optimizing the system operation (including secondary HVAC system).
- Sizing and part load performances have a large influence on the seasonal performance of a heat pump system.
- In general, heat pump systems are interesting in terms of primary energy only if the global efficiency of the considered electrical network is sufficiently high. In some European countries, improvements should be brought to electrical power plants before considering heat pump systems as an opportunity for space heating.
- In countries where the CO_2 emission rate per produced kWh is quite low (because of the intensive use of renewable energy sources or nuclear plants to produce electricity, as in France), even very simple heat pump systems (such as reversible air-to-water heat pump) can lead to interesting environmental results.
- If the CO_2 emission rate per produced kWh is high (e.g. in Greece), only high performance heat pump systems (e.g. ground coupled heat pump systems) should be considered for space heating. Low COP systems could even lead to supplemental CO_2 emissions.
- In the case of building retrofit and design studies, sophisticated heat pump systems should only be considered if a global energy efficiency approach aimed to reduce building's global energy consumption (including lighting, appliances, mechanical ventilation…) is followed.

References

1. Stabat, P., Andre, P., Bertagnolio, P. et al. 2010. Analysis of building heating and cooling demands in the purpose of assessing the reversibility and heat recovery potentials. IEA-ECBCS Annex48 project final report. www.ecbcs-48.org
2. Klein., S.A. 2009. EES: Engineering Equation Solver, User manual. F-chart software. Madison: University of Wisconsin, Madison, USA.
3. Bertagnolio, S., Stabat, P., Caciolo, M., Corgier, D. 2010. Review of heat recovery and heat pumping solutions. IEA-ECBCS Annex48 project final report. www.ecbcs-48.org
4. Chapon, C., Delandre, M. 1991. Air Conditioning by heat pumps on a water loop. ASHRAE Transactions 97(1): 283-286.
5. DOE (Department of Energy). 1980. DOE 2 Reference Manual Part 1, Version 2.1. Berkeley, Calif.: Lawrence Berkeley National Laboratories
6. Hydeman, M., and K. Gillespie. 2002. Tools and techniques to calibrate electric chiller component models. ASHRAE Transactions 108(1): 733-741.
7. Consoclim. 2002. Cahier des Algorithmes – version 2.3. In-french.
8. Shenoy, A. Simulation, Modeling and Analysis of Water to Air Heat Pump. M.S. Thesis, Oklahoma State University, December 2004.
9. Cooper, W. (1994). Operative experience with water loop heat pump systems. ASHRAE Transactions: NO-94-26-1.
10. Lebrun, J., Bourdouxhe, J.-P., Grodent, M. 1999. HVAC1KIT: A toolkit for primary HVAC system energy calculation. Atlanta, GA, American Society of Heating, Refrigerating and Air-Conditioning Engineers.
11. Stabat, P. And Marchio, D. 2004. Simplified model for indirect-contact evaporative cooling-tower behavior. Applied Energy 78: 433-451.
12. Bernier, M. 2000. A Review of the Cylindrical Heat Source Method for the Design and Analysis of Vertical Ground-Coupled Heat Pump Systems. 4th International Conference on Heat Pumps in Cold Climates, p. 1-14.
13. Bernier, M. A. 2001. Ground-coupled heat pump system simulation. ASHRAE Transactions 107(1): 605-616.
14. Bernier, M.A., Labib, R., Pinel, P., Paillot, R. 2004. A Multiple Load Aggregation Algorithm for Annual Hourly Simulations of GCHP Systems. HVAC&R Research 10(4): 471-487.
15. Bernier, M.A. 2006. Closed-Loop Ground-Coupled Heat Pump Systems. ASHRAE Journal 48(9): 12-18.
16. Bernier, M., Chahla, A., Pinel, P. 2008. Long-term ground temperature changes in geo-exchange systems. ASHRAE Transactions 114(2).
17. Dorgan, C., Linder, R., & Dorgan, C. (1999). *Chiller heat recovery: Application guide.* Atlanta: American society of heating, refrigerating and air-conditioning engineers.
18. Pietsch, J. 1990. Water-loop heat pump systems assessment. ASHRAE Trans. 96, pp. 1029-1038.
19. Hughes, P.J. 1990. Survey of water-source heat pump system configuration in current practice. ASHRAE Trans. 96, pp. 1021-1028.
20. Kavanaugh, S.P. 1991. Design of water-to-air heat pumps with high cooling efficiency for ground-coupled applications. ASHRAE Trans. 97, pp. 889-894.
21. Howell, R.H., Zaidi, J.H. 1990. Analysis of heat recovery in water-loop heat pump systems. ASHRAE Trans. 96, pp. 1039-1047.
22. Henderson, H.I., Khattar, M.K., Carlson, S.W., Walburger, A.C. 2000. The implications of the measured performance of variable flow pumping systems in geothermal and WLHP applications. ASHRAE Trans. 106, pp. 533-542.
23. Cane, D., Clemes, B., Morrison, A. 1996. Operating Experiences with commercial ground-source heat pumps - Part 1. ASHRAE Trans. 102, pp.911 - 916.
24. Cane, D., Morrison, A. 1998. Operating Experiences with commercial ground-source heat pumps - Part 2. ASHRAE Trans. 104(2) : pp.677-686.
25. Stabat, P., & Marchio, D. (2004). *Simplified model for indirect-contact evaporative cooling-tower behavior.* Applied Energy 78: 433-451.

26. Marchio, D., & Morisot, O. (1999). *Modelisation simplifiée d'un aéroréfrigerant dans l'optique d'un calcul des consommations d'énergie d'une installation de climatisation dans un bâtiment tertiaire.* ENEA/CVA-n°98.162R, CSTB.
27. Lebrun, J., X. Ding, J.-P. Eppe, and M.Wasac. 1990. Cooling Coil Models to be used in Transient and/or Wet Regimes. Theoritical Analysis and Experimental Validation. Proceedings of SSB 1990. Liège:405-411
28. Morisot, O., Marchio, D. (2002). Simplified Model for the Operation of Chilled Water Cooling Coils Under Nonnominal Conditions. HVAC&R Research 8(2): 135-158.
29. Brandemuehl, M.J., S. Gabel, and I. Andersen. 1993. A toolkit for secondary HVAC System Energy Calculations. Published for ASHRAE by Joint Center for Energy Management, University of Colorado at Boulder.
30. Wang, G., M. Liu, and D. Claridge. 2007. Decoupled modeling of Chilled-Water cooling coils, ASHRAE transaction 113(1).
31. Oklahoma State University. 1988. Closed-Loop/Geothermal Heat Pump Systems: Design and Installation Standards, Stillwater, OK: Prepared for the National Rural Electric Cooperative Association and distributed by the International Ground-Source Heat Pump Association
32. Ingersoll, L.R., O.J. Zobel, and A.C. Ingersoll. 1954. *Heat conduction: With engineering and geological applications*, 2d ed. McGraw-Hill.
33. Carslaw, H.S., Jaeger, J.C. (1947). Conduction of Heat in Solids. Oxford.
34. Cooper, L.Y. 1976. Heating of a Cylindrical Cavity. International Journal of Heat and Mass Transfer, Vol. 19, pp. 575-577.
35. Sfeir, A., Bernier, M., Million, T., Joly, A. 2005. A methodology pumping energy consumption in GCHP systems. ASHRAE Transactions 111(2): 714-729
36. Bernier, M. And Lemire, N. 1999. Non-dimensional pumping power curves for water loop heat pump systems. ASHRAE Transactions 105(2): 1226-1232.
37. Bernier, M. And Bourret, B. 1999. Pumping energy and variable frequency drives. ASHRAE Journal 41(12): 37-40.
38. Chapont, C., & Delandre, M. Air Conditioning by heat pumps on a water loop. ASHRAE Transactions: Research.
39. Bourdouxhe P, Grodent M, Lebrun J. Cooling-tower model developed in a toolkit for primary HVAC system energy calculation—part 1: model description and validation using catalog data, Proceedings of the fourth international conference on system simulation in buildings, Liège, December 1994.
40. Lebrun J, Aparecida Silva C, Trebilcock F, Winandy E. Simplified models for direct and indirect contact cooling-towers and evaporative condensers. 6th International conference on System Simulation in Buildings, Liège, December 2002.
41. Bolher, A., Casari, R., Fleury, E., Marchio, D., Millet J.R., Morisot O. 1999. Méthode de calcul des consommations d'énergie des bâtiments climatisés « ConsoClim » Présentation de la méthode,ENEA/CVA-n°98.162R, CSTB.
42. Marchio, D., & Morisot, O. (1999). Modelisation simplifiée d'un aéroréfrigerant dans l'optique d'un calcul des consommations d'énergie d'une installation de climatisation dans un bâtiment tertiaire. ENEA/CVA-n°98.162R, CSTB.
43. Morisot, O., Marchio, D. 1998. CCSIMOL : A few inputs model of variable flows cooling coil. Proceedings of the fifth International Conference on System Simulation in Buildings, Liège, Belgium.
44. Braun JE, Klein SA, Mitchell JW. Effectiveness models for cooling towers and cooling coils. ASHRAE Transactions 92(2):164–174.

45. ADEME. 2007. Guide des facteurs d'émissions – calcul des facteurs d'missions et sources bibliographiques utilisées. Bilan carbone version 5.0.

46. Andre, P., Bertagnolio, S., Caciolo, M., Franck, P.-Y., Rogiest, C., Sarrade, L., et al. (2008). *Analysis of building heating and coolign demands in the purpose of assessing the reversibility and heat recovery potentials.* Annex 48 of the International Energy Agency, Energy conservation in buildings and Community Systems Programme.

47. Bernier, M. (2000). *A Review of the Cylindrical Heat Source Method for the Design and Analysis of Vertical Ground-Coupled Heat Pump Systems.* 4th International Conference on Heat Pumps in Cold Climates, p. 1-14.

48. Bernier, M. (2006). *Closed-Loop Ground-Coupled Heat Pump Systems.* ASHRAE Journal 48(9): 12-18.

49. Bernier, M. (2001). *Ground-coupled heat pump system simulation.* ASHRAE Transactions 107(1): 605-616.

50. Bernier, M., & Bourret, B. (1999). *Pumping energy and variable frequency drives.* ASHRAE Journal 41(12): 37-40.

51. Bernier, M., & Lemire, N. (1999). *Non-dimensional pumping power curves for water loop heat pump systems.* ASHRAE Transactions 105(2): 1226-1232.

52. Bernier, M., Chahla, A., & Pinel, P. (2005). *Long-term ground temperature changes in geo-exchange systems.* ASHRAE Transactions 114(2).

53. Bernier, M., Kummert, M., & Bertagnolio, S. (2007). *Development and applications of test cases for comparing vertical ground heat exchanger models.* Beijing, China: Proceeding of Building Simulation 2007.

54. Bernier, M., Labib, R., Pinel, P., & Paillot, R. (2004). *A Multiple Load Aggregation Algorithm for Annual Hourly Simulations of GCHP Systems.* HVAC&R Research 10(4): 471-487.

55. Bertagnolio, S., Caciolo, M., Corgier, D., & Stabat, P. (2008). *Review of heat recovery and heat pumping solutions.* Annex 48 of the International Energy Agency, Energy conservation in buildings and Community Systems Programme.

56. Bolher, A., Casari, R., Fleury, E., Marchio, D., & Millet, J. (1999). *Méthode de calcul des consommations d'énergie des bâtiments climatisés « ConsoClim » Présentation de la méthode.* CSTB.

57. Cane, D., Clemes, B., & Morrison, A. (1998). *Operating experiences with commercial ground-source heat pumps- Part 1.* ASHRAE Transactions.

58. Casari, R., Fleury, E., Marchio, D., Millet, J., & Bohler, A. (1999). *Méthode de calcul des consommations d'énergie des bâtiments climatisés « ConsoClim » Présentation de la méthode.* CSTB.

59. Chapont, C., & Delandre, M. *Air Conditioning by heat pumps on a water loop.* ASHRAE Transactions: Research.

60. Cooper, W. (1994). *Operative experience with water loop heat pump systems.* ASHRAE Transactions:NO-94-26-1.

61. Consoclim. 2002. Cahier des Algorithmes – version 2.3. In-french.

62. EN 15603-2007. Energy performance of buildings – Overall energy use and definition of energy ratings.

63. EN 14511-2-2008 Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling – Part 2: test conditions.

64. Energy Efficiency and Certification of Central Air Conditioners (EECCAC). (2003). ARMINES.

65. Geothermal Bore Technologies. (2007). Retrieved from http://www.geoclip.com/

66. Hydeman, M., Taylor, S., & Winiarski, D. (2002). *Application of component models for standard development.* ASHRAE Transactions 108(1).

67. Hydeman, M., Webb, N., Sreedharan, P., & Blanc, S. (2002). *Development and testing of a reformulated regression-based electric chiller model.* ASHRAE Transaction 108 (2).

68. ASHRAE. 2007. ANSI/ASHRAE/IESNA Standard 90.1-2007. Energy standard for building except low-rise residential buildings. American Society of Heating, Referigerating and Air Conditioning Engineers, Atlanta.

69. Jeffrey, J., Splitler, D., Fisher, D., & Xu, X. (2006). *Simulation of Hybrid Ground Source Heat Pump Systems and Experimental Validation.* Liège: 7th International Conference on System Simulation in Buildings.

70. Kavanaugh, S. (1998). *A Design Method for Hybrid Ground Source Heat Pump.* ASHRAE TRANSACTIONS, V. 104, Pt. 2.

71. Kavanaugh, S., & Rafferty, K. (1997). *Ground-Source Heat Pumps: Design of Geothermal Systems for Commercial and Institutional Buildings.* ASHRAE.

72. Lebrun, J. (2007). *Introduction aux Machines et Systèmes Thermiques. Notes de cours.* Université de Liège, Belgium.

73. Lebrun, J. (2004). *Thermodynamique appliquée et Intrduction aux machines thermiques.* Liège: Université de Liège, Département de thermodynamique.

74. Lebrun, J., Aparecida Silva, C., Trebilcock, F., & Winandy, E. (December 2002). *Simplified models for direct and indirect contact cooling-towers and evaporative condensers.* Liège: 6th International conference on System Simulation in Buildings.

75. Marchio, D., & Morisot, O. (1999). *Modelisation simplifiée d'un aéroréfrigerant dans l'optique d'un calcul des consommations d'énergie d'une installation de climatisation dans un bâtiment tertiaire.* ENEA/CVA-n°98.162R, CSTB.

76. Morisot, O., & Marchio, D. (1998). *CCSIMOL : A few inputs model of variable flows cooling coil.* Liège, Belgium: Proceedings of the fifth International Conference on System Simulation in Buildings.

77. Morisot, O.; Marchio, D. (2002). *Simplified Model for the Operation of Chilled Water Cooling Coils Under Nonnominal Conditions.* HVAC&R Research 8(2): 135-158.

78. Pietsch, J. (1990). *Water-loop heat pump systems assesement.* ASHRAE Trans. 96, pp. 1029–1038.

79. Sfeir, A., Bernier, M., & Joly, A. (2005). *A methodology pumping energy consumption in GCHP systems.* ASHRAE Transactions 111(2): 714-729.

80. Stabat, P., & Marchio, D. (2004). *Simplified model for indirect-contact evaporative cooling-tower behavior.* Applied Energy 78: 433-451.

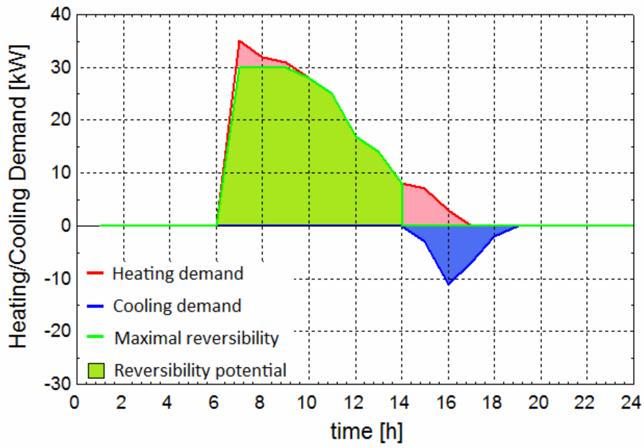
81. Remund, C.P. (1999) Borehole thermal resistance: laboratory and field studies. ASHRAE Transactions 105 (1)

82. Oklahoma State University (1988). Closed-Loop/Geothermal Heat Pump Systems: Design and Installation Standards, Stillwater, OK: Prepared for the National Rural Electric Cooperative Association and distributed by the International Ground-Source Heat Pump Association.

APPENDIX 1: Recovery and reversibility potentials

Reversibility potential

The reversibility potential (REV_{pot}) 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 ($I_{rev,pot}$).



These two quantities are calculated hour by hour, 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[5], the supplementary demand is assumed to be covered by the boiler.

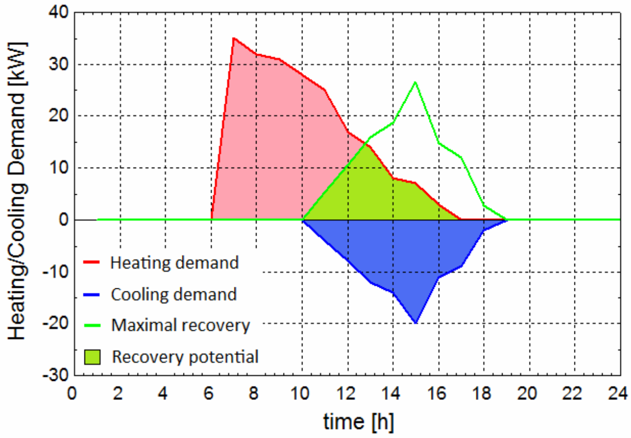
No consideration on the emitter temperature levels required is taken into account at this stage.

Recovery potential

The recovery potential (REC) 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 Hot Domestic water and/or humidification).
The recovery potential is calculated hour by hour 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.

[5] 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.



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.

APPENDIX 2: Components modeling

Heat pumps

The heat pump model used in this work is known as the "DOE-2" method and has been described in details in [6] [66] [67]. Its aim is to accurately determine the energy consumed by the heat pump given some fundamental inputs which could be easily measured by the user. The choice of this method has been motivated by the fact that the error prediction in terms of RMS is limited compared to other methods in which an error which could reach to higher errors [6] [66] [67]. The sequence of this method is in 3 steps:

1) Firstly determining the **CAPFT curve**. Such curve represents the available capacity as a function of evaporator and condenser temperatures thanks to six coefficients and two independent variables. The method which will follow is described in the case of a chiller where these variables are the entering condenser temperature and the leaving evaporator temperature. However in the case of a heat pump variables are defined in terms of leaving condenser temperature and supply evaporator temperature. The CAPFT curve is calculated thanks to, at least, six full-load performance data points as follows:

$$CAPFT_i = \frac{\dot{Q}_i}{\dot{Q}_{ref}}$$

where $\dot{Q}_i$ is defined as the cooling capacity at the evaporator in certain full load conditions. The value of $\dot{Q}_{ref}$ is defined as the rating capacity in nominal conditions [63]. With these six points the regression law can be established

Cooling mode: $CAPFT = a_1 + b_1 t_{su,cd} + c_1 t_{su,cd}^2 + d_1 t_{ex,ev} + e_1 t_{ex,ev}{}^2 + f_1 t_{ex,ev} t_{su,cd}$

Heating mode: $CAPFT = a_1 + b_1 t_{ex,cd} + c_1 t_{ex,cd}^2 + d_1 t_{su,ev} + e_1 t_{su,ev}{}^2 + f_1 t_{su,ev} t_{ex,cd}$

2) Secondly determining the **EIRFT curve**. Such curve represents the full-load efficiency as a function of the evaporator and condenser temperatures. The principle is exactly the same as in the case of the CAPFT curve but absorbed electrical power is now considered.

$$EIRFT_i = \frac{\dot{W}_i}{\dot{W}_{ref} CAPFT_i}$$

The value of $\dot{W}_{ref}$ is the power consumption of the unit in nominal conditions [63]. With the same six points the regression law can be established:

Cooling mode: $EIRFT = a_2 + b_2 t_{su,cd} + c_2 t_{su,cd}^2 + d_2 t_{ex,ev} + e_2 t_{ex,ev}{}^2 + f_2 t_{ex,ev} t_{su,cd}$

Heating mode: $EIRFT = a_2 + b_2 t_{ex,cd} + c_2 t_{ex,cd}^2 + d_2 t_{su,ev} + e_2 t_{su,ev}{}^2 + f_2 t_{su,ev} t_{ex,cd}$

3) Finally determining the **EIRPLR curve**. Such curve represents the efficiency as a function of the unloading percentage. It could be found using several part load points in order to define:

$$EIRFPLR_i = \frac{W_i}{W_{ref}^{\cdot}\ CAPFT_i\ EIRFT_i}$$

A law for $EIRFPLR$ could be drawn as a function of the PLR using several data points at part load.

$$EIRPLR = a_3 + b_3 PLR + c_3 PLR^2$$

Where PLR is defined as:

$$PLR = \frac{\dot{Q}}{Q_{ref}^{\cdot}\ CAPFT}$$

The shape of this curve is mainly determined by the part load control system and the type of the compressor used.

Thanks to this method it is possible to calculate the consumption of the chiller (or heat pump) for a certain capacity (which determines the PLR) and under certain operating conditions thanks to the following equation:

Cooling mode:

$$\dot{W} = W_{ref}^{\cdot}.\ CAPFT\ (t_{ex,ev}\ .t_{su,cd}).\ EIRFT\ (t_{ex,ev}\ .t_{su,cd}).\ EIRFPLR\ (t_{ex,ev}\ .t_{su,cd})$$

Heating mode:

$$\dot{W} = W_{ref}^{\cdot}.\ CAPFT\ (t_{su,ev}\ .t_{ex,cd}).\ EIRFT\ (t_{su,ev}\ .t_{ex,cd}).\ EIRFPLR\ (t_{su,ev}\ .t_{ex,cd})$$

Having the consumption the EER (cooling mode) or the COP (heating mode) of the unit can be determined. As the CAPFT, EIRFT and

Dual condenser heat pump

This type of heat pump is only used in the dual condenser system which will be described later. The heat pump has two different condensers and is always in a chiller mode but with two different modes:

- When there is no heat demand the air condenser works like in any air-to-water chiller.
- When there is a heat demand the main condenser becomes the water condenser and the system works in the recovery mode. In such case there are two possibilities:
    - The heat recovered at the condenser is lower than the heat demand and the water flow takes as much energy as it can;
    - Or the heat recovery is higher than the heat demand. In that case one will assume that both condensers are functioning together with the air condenser rejecting the difference of the heat recovered and the heat demand.

In this work a perfect control on the condensers is assumed so that the utilisation of the condensers exactly matches the demand. This system will be called heat pump D.

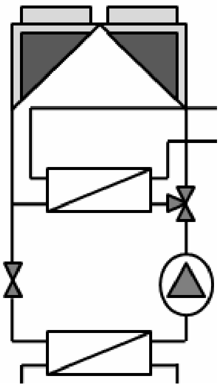


Figure 45: Schematic and real view of dual condenser heat pump

Calculating the consumption of this type of heat pump has been somewhat more complex than in the other cases. With no heat demand on water condenser side the model provides the consumption and obviously the EER as the system is considered as a simple water cooled chiller.

When the water condenser is used (recovery mode) the regression laws for heating mode are used. In this case, CAPFT and EIRFT curves are expressed in terms of exhaust evaporator temperature and exhaust condenser temperature. The evaporator load is an input of the model and allows determining the corresponding heat available on condenser side by means of the following equation:

$$\dot{Q}_{cd} = \dot{Q}_{ev} \cdot \frac{EER + 1}{EER}$$

The system is now considered as a water cooled chiller with the difference that the possible fan consumption should be taken into account when the air condenser is used to condense a part of the refrigerant flow. An easy way to compute this auxiliary consumption was to consider that the fan consumption is proportional to the load of the air condenser. This constant ratio has been identified based on the nominal capacity and power.

$$\mathbf{C} = \frac{\dot{W}_{fan}}{\dot{Q}_{cd}} = 0.02$$

It is so possible to determine the additional amount of energy consumed by the fan when the system operates in a recovery mode and finally the COP of the system.

Reversible water-to-air heat pump (system 4 – WLHP)

The model used in the present work is based on the work of [8]. For water-to-air heat pumps, the regression laws have to take into account the possible condensation on evaporator side in cooling model. So, in cooling mode, the operating conditions influencing the performances of the unit will be: the condenser supply temperature, the evaporator drybulb temperature and the evaporator wetbulb temperature. The coefficients of the regressions laws described below can be easily identified based on manufacturer data, as manufacturers provide performance data points for different water, wetbulb and drybulb temperatures.

In heating mode, only sensible heat exchanges are considered and the model is similar to the DOE-2 model.

Part load degradation is not taken into account in the present model. Because only "equivalent" (aggregated) cooling and heating loads are used in the present work (no matter how many units are used to supply these demands), it is not possible to consider the influence of part load operation. So, no EIRFPLR curve is used in the present model and units are supposed to have the same performance at part load and full load operation.

The auxiliary fan consumption is also computed and is supposed to be proportional to the load of the unit

In cooling mode, the total cooling capacity, the sensible cooling capacity and the EER are expressed as functions of the condenser supply fluid temperature, the evaporator supply wetbulb temperature and the evaporator supply drybulb temperature:

$$CAPFT_{tot} = A_1 + B_1 * \frac{(t_{cd,su} + 273.15)}{t_{ref}} + C_1 * \frac{t_{ref}}{(t_{wb,ev,su} + 273.15)}$$

$$CAPFT_{sens} = A_2 + B_2 * \frac{(t_{cd,su} + 273.15)}{t_{ref}} + C_2 * \frac{t_{ref}}{(t_{wb,ev,su} + 273.15)} + D_2 * \frac{t_{ref}}{(t_{db,ev,su} + 273.15)}$$

$$EERFT_{tot} = A_3 + B_3 * \frac{(t_{cd,su} + 273.15)}{t_{ref}} + C_3 * \frac{t_{ref}}{(t_{wb,ev,su} + 273.15)}$$

Where t_{ref} is the reference temperature equal to 283K.

The sensible and total cooling rates at full load and the corresponding EER (defined in terms of total cooling rate) are given by:

$$\dot{Q}_{tot,FL} = \dot{Q}_{tot,n} * CAPFT_{tot}$$

$$\dot{Q}_{sens,FL} = \dot{Q}_{sens,n} * CAPFT_{sens}$$

$$EER_{FL} = EER_n * CAPFT_{sens}$$

The part load ratio is defined in terms of (useful) sensible rates:

$$0 \leq PLR = \frac{\dot{Q}_{sens,ac}}{\dot{Q}_{sens,FL}} \leq 1$$

At part load operation, the ratio between the total and sensible cooling rates and the EER are supposed to stay constant (no part load degradation of the EER)

$$\dot{Q}_{tot,ac} = PLR * \dot{Q}_{tot,FL}$$

$$EER_{ac} = EER_{FL}$$

Finally, the compressor electricity consumption is given by:

$$\dot{W}_{cp,ac} = \frac{\dot{Q}_{tot,ac}}{EER_{ac}}$$

The corresponding fan consumption is supposed to be proportional to the load of the unit and is given by:

$$\dot{W}_{fan} = PLR * \dot{W}_{fan,n}$$

The amount of heat rejected on condenser side is given by:

$$\dot{Q}_{cd,ac} = \dot{Q}_{tot,ac} \frac{(EER_{ac} + 1)}{EER_{ac}}$$

In heating mode, the full load heating capacity and the COP are expressed as functions of the supply fluid temperature (evaporator side) and the supply air drybulb temperature (condenser side):

$$CAPFT = A_1 + B_1 * \frac{(t_{ev,su} + 273.15)}{t_{ref}} + C_1 * \frac{t_{ref}}{(t_{db.cd.su} + 273.15)}$$

$$COPFT = A_2 + B_2 * \frac{(t_{ev,su} + 273.15)}{t_{ref}} + C_2 * \frac{t_{ref}}{(t_{db,cd,su} + 273.15)}$$

The heating rate at full load and the corresponding COP are given by:

$$\dot{Q}_{cd,FL} = \dot{Q}_{cd,n} * CAPFT$$

$$COP_{FL} = COP_n * COPFT$$

The part load ratio is defined in terms of (useful) condenser rates:

$$0 \leq PLR = \frac{\dot{Q}_{cd,ac}}{\dot{Q}_{cd,FL}} \leq 1$$

At part load operation, the COP is supposed to stay constant (no part load degradation)

$$COP_{ac} = COP_{FL}$$

Finally, the compressor electricity consumption is given by:

$$\dot{W}_{cp,ac} = \frac{\dot{Q}_{cd,ac}}{COP_{ac}}$$

The corresponding fan consumption is supposed to be proportional to the load of the unit and is given by:

$$\dot{W}_{fan} = PLR * \dot{W}_{fan,n}$$

The amount of heat absorbed on evaporator side is given by:

$$\dot{Q}_{ev,ac} = \dot{Q}_{cd,ac} \frac{(COP_{ac} - 1)}{COP_{ac}}$$

Selected heat pumps

Description	**Air Cooled Chiller**
System	0
Nominal Capacity	399100 W
Compressor Power	139300 W
Fan Power	6240 W
Compressor type	Reciprocating
Refrigerant	R 407C

Description	**Water cooled chiller**
System	0
Nominal Cooling Capacity	34700 W
Compressor Power	7500 W
Compressor type	Scroll
Refrigerant	R 410A

Description	Air Cooled Chiller
System	1
Nominal Cooling Capacity	451400 W
Compressor Power	153900 W
Fan Power	11200 W
Nominal Heating Capacity	467400 W
Compressor Power	159400 W
Fan Power	12600 W
Compressor type	Scroll
Refrigerant	R 407C

Description	Water-to-Water HP
System	2
Nominal Cooling Capacity	34700 W
Compressor Power	7500 W
Nominal Heating Capacity	37900 W
Compressor Power	9300 W
Compressor type	Scroll
Refrigerant	R 410A

Description	Dual condenser HP
System	3
Nominal Capacity-cooling	264700 W
Compressor Power	107200 W
Fan Power	6960 W
Nominal Capacity-recovery	377400 W
Compressor Power	94000 W
Compressor type	Screw
Refrigerant	R 134a

Description	Reversible water/brine-to-air HP
System	4
Nominal Total Cooling Capacity	1941 W
Nominal Sensible Cooling Capacity	1441 W
Compressor Power	550 W
Fan Power	190 W
Nominal Heating Capacity	2656 W
Compressor Power	590 W
Fan Power	100 W
Compressor type	Rotary
Refrigerant	R 407C

Description	Reversible brine-to-water HP
System	5A
Nominal Cooling Capacity	30100 W
Compressor Power	6400 W
Nominal Heating Capacity	32500 W
Compressor Power	8000 W
Compressor type	Scroll
Refrigerant	R 410A

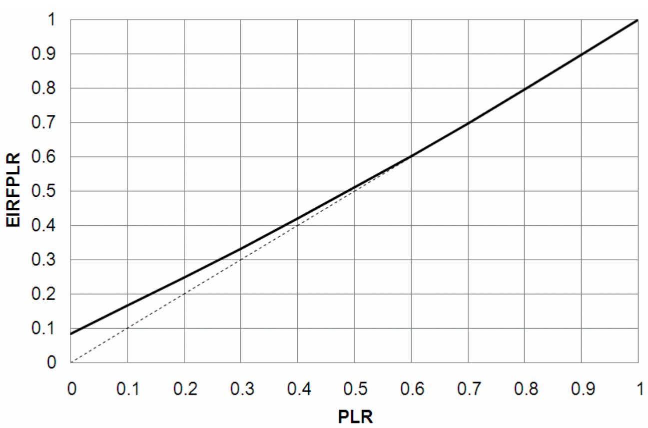
Description	Brine-to-water HP
System	5B
Nominal Cooling Capacity	34700 W
Compressor Power	7500 W
Nominal Heating Capacity	37900 W
Compressor Power	9300 W
Compressor type	Scroll
Refrigerant	R 410A

System	Operation	Curve	a	b	c	d	e	f
S0 – AC	Cooling	CAPFT	1.05550427	-0.00481977	-8.2038E-05	0.04001648	0.00027692	-0.00032612
		EIRFT	0.702679468	-0.01120397	0.000712429	-0.00011601	0.000581	-0.00085964
S0 - WC	Cooling	CAPFT	0.893940694	-0.00085242	-0.00012214	0.041953074	-1.10151E-05	-0.00022746
		EIRFT	0.928705663	-0.01973348	0.001092968	-0.00089144	0.001462216	-0.00192248
S1	Cooling	CAPFT	1.05550427	-0.00481977	-8.2038E-05	0.04001648	0.00027692	-0.00032612
		EIRFT	0.702679468	-0.01120397	0.000712429	-0.00011601	0.000581	-0.00085964
	Heating	CAPFT	0.99303839	-0.0064081	5.60458E-05	0.032544029	0.000474272	-0.00021713
		EIRFT	0.407882006	0.008229753	0.000203749	-0.00620837	0.00026067	-0.00051274
S2		CAPFT	0.893940694	-0.00085242	-0.00012214	0.041953074	-1.10151E-05	-0.00022746
		EIRFT	0.928705663	-0.01973348	0.001092968	-0.00089144	0.001462216	-0.00192248
S3	Cooling	CAPFT	0.314725038	0.035201426	-0.00059149	0.068475852	-0.000205556	-0.00111255
		EIRFT	0.140634566	0.02227573	0.000187482	0.016506201	0.000278095	-0.00109024
	Heating	CAPFT	1.06985521	-0.00322286	-6.8555E-05	0.036575848	0.000395915	-0.00019432
		EIRFT	0.754263096	-0.01827651	0.000627211	0.020785843	0.000502885	-0.00118634
S5A	Cooling	CAPFT	0.958073935	-0.004226	-7.6446E-05	0.040247989	0.000303559	-0.00030926
		EIRFT	0.921512458	-0.02599998	0.001270112	0.014066791	0.001380502	-0.00227738
	Heating	CAPFT	0.813160106	-0.00104774	-2.0251E-05	0.037045212	0.000281194	-0.00029367
		EIRFT	0.50261952	0.001182571	0.000394173	-0.00678142	0.000472362	-0.00070255
S5B	Cooling	CAPFT	0.893940694	-0.00085242	-0.00012214	0.041953074	-1.10151E-05	-0.00022746
		EIRFT	0.928705663	-0.01973348	0.001092968	-0.00089144	0.001462216	-0.00192248
	Heating	CAPFT	0.775249033	0.000710048	-4.5948E-05	0.039635683	-2.85194E-05	-0.00023253
		EIRFT	0.64369656	-0.00395322	0.000448164	-0.00921557	0.000636769	-0.00075009

System	Operation	Curve	a	b	c	d
S4 A,B & C	Cooling	CAPFT total	10.7990919	-2.22930629	-7.66088842	
		CAPFT sensible	-20.8187858	-1.40894426	89.6341798	-67.3449067
		EERFT	14.7641367	-6.67782528	-6.8113622	
	Heating	CAPFT	-5.40937976	5.40817464	0.836302185	
		COPFT	-10.0117138	4.65460526	6.43410827	

As manufacturer data in part load regime were not available for each unit, a typical EIRFPLR curve is proposed. Of course, this curve can be easily replaced or adapted if actual part load data are available.

	a	b	c
EIRFPLR	0.0847548284	0.819563444	0.086598514



Water pump

As a first approach, only constant flow pumps are considered. Pumps power consumption is directly linked to the fluid flow rate by defining a SPP ("Specific Pump Power") in W/kg/s.

This model uses ratios based on standard values and existing installations reported in the literature. Depending on the type of systems studied, the pump used in the hydraulic network has to be distinguished from the ones used in a water loop system or to supply a ground heat exchanger, as this latter systems use much larger hydraulic networks.

ASHRAE directly provides values of SPP for classical hot and cold water networks [68]. Additional references have been used to estimate the value of the SPP for WLHP and GCHP systems.

Type of hydraulic network	SPP	Source
Cold water network	350 W/(kg/s)	[68]
Hot water network	300 W/(kg/s)	[68]
Water loop system	550 W/(kg/s)	[1,2,3,4]
Ground coupled Water Loop	800 W/(kg/s)	[5,6]

Table 11: Hydraulic network pump consumption correlations

Nominal hot and chilled water mass flow rates are determined based on design heating and cooling powers and chosen temperature variations (5, 10 or 20K). Water flow rate in WLHP system is generally included between 0.03 and 0.05 l/s/kW of installed cooling capacity.

Boiler

According to [9], the supply water temperature should not be below a certain point defined close to 60°C for a usual boiler. This limit highly depends on the type of boiler used and can decrease in the case of a condensing boiler. However in the case of a hot hydraulic network considered in the work the temperature needed is in the range of 40-50 °C. This problem will be addressed with the assumption that a mixing hydraulic network is placed before entering the boiler and can perfectly control the valves in order to maintain sufficient temperature difference and water flow. This hydraulic network can be described as followed:

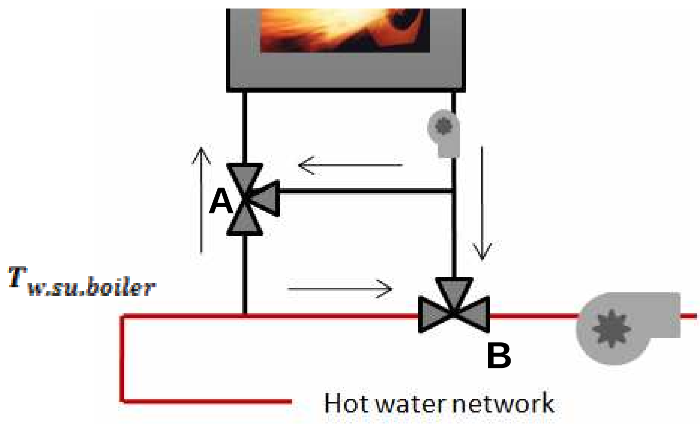


Figure 166: Boiler regulation network

In this network the left valve (A) controls the incoming water flows in order to have a supply temperature high enough not to damage the boiler. The right valve (B) perfectly controls both incoming water flows in order to have the right exhaust temperature $t_{w,ex,boiler}$.

Simplified boiler model

Four types of water boilers are currently available: Classical On-Off burners, High-Low-Off burners, Modulating burners and condensing modulating boilers. All these models uses performances curves generated by means of detailed validated and calibrated reference models [10]. Figure 17 shows the normalized efficiency curves for classical (non-condensing) boilers.

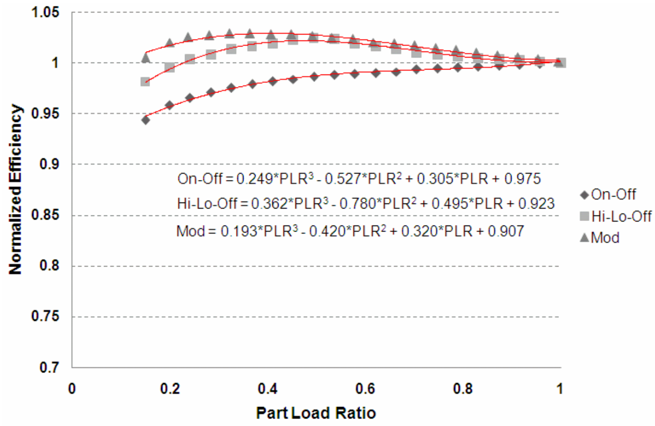


Figure 177: Default part load performance curves for water boilers

Firstly the capacity of the boiler is limited to the rating capacity defined by the user or by the automatic pre-sizing procedure. This maximal capacity corresponds to a maximal water exhaust temperature. Secondly, the actual load of the boiler (corresponding to the actual water exhaust temperature set point) is used to determine the part load ratio (PLR):

$$PLR = \frac{\dot{Q}_{boiler}}{\dot{Q}_{nom,boiler}}$$

The PLR value allows determining the value of the normalized efficiency by means of the regression laws mentioned above. The actual efficiency is then computed as:

$$\eta_{actual} = \eta_{nom} \bullet \eta^{*}(PLR)$$

Where η_{nom} is the nominal (full load) efficiency of the boiler defined by the user.

Finally, the actual efficiency allows determining the related fuel consumption.

The dependence of the boiler efficiency to the water temperature is not taken into account in the present model.

Cooling tower & Dry Fluid Cooler

The model involves three steps.

1. a sizing procedure is used to determine the heat transfer coefficient;
2. the full load capacity of the cooler is computed using the previously calculated heat transfer coefficients and as a function of the actual operating conditions (water and outdoor environment conditions) ;
3. the actual energy consumption of the cooler (fans and pumps) is computed as function of the load of the cooler. Regression laws are used to simulate the part load operation

Dry cooler sizing

The first part of the sizing step is to determine the heat transfer coefficient related to some nominal data. Using the logarithmical temperature difference it is possible to establish the total coefficient of transfer AU_{DC}.

$$\Delta T_{lm} = \frac{\left((t_{w,ex} - t_{a,su}) - (t_{w,su} - t_{a,ex})\right)}{LN\left(\frac{t_{w,ex} - t_{a,su}}{t_{w,su} - t_{a,ex}}\right)}$$

while

$$AU_{DC,n} = \frac{\dot{Q}_{dc,n}}{\Delta T_{lm}}$$

The following repartition key is used to estimate the heat transfer coefficients on air and water sides:

$$AU_{w,n} = (n+1).AU_{DC,n}$$
$$AU_{a,n} = \frac{n+1}{n}.AU_{DC,n}$$

with $n = 1.89$ as a repartition key [26].

Nominal air flow rate and corresponding fan power are established based on regression curves extracted from manufacturer data (Figure 188 and Figure 199).

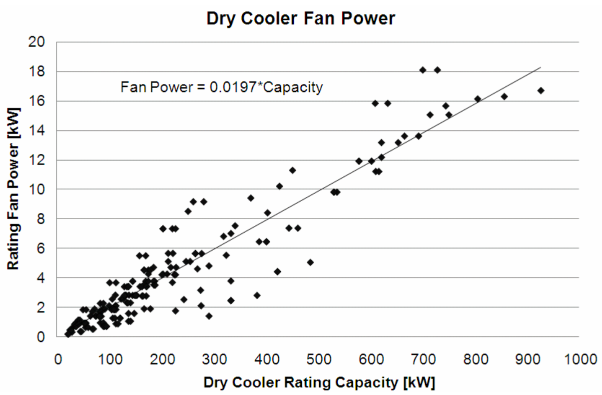


Figure 188: Dry Cooler Fan Power

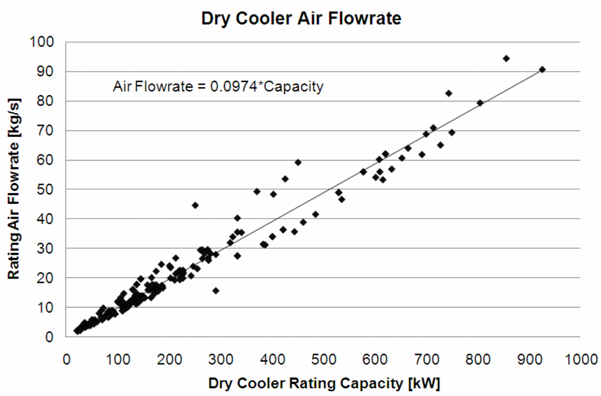


Figure 199: Dry Cooler Air Flowrate

Cooling tower sizing

The first part of the sizing procedure aims to determine the nominal heat transfer coefficient of the unit. First, the air flow rate and the corresponding fan power are determined through regression laws based on manufacturer data (50, Figure).

Using the logarithmical temperature difference it is possible to identify the global heat transfer coefficient AU_{CT}. To simplify the calculation, the air is considered as a fictitious fluid characterized by its wet bulb temperatures.

$$\Delta T_{lm} = \frac{\left((t_{w,ex} - t_{wb,su}) - (t_{w,su} - t_{wb,ex})\right)}{LN\left(\frac{t_{w,ex} - t_{wb,su}}{t_{w,su} - t_{wb,ex}}\right)}$$

While

$$AU_{glob} = \frac{1}{R_{glob}} = \frac{\dot{Q}_{CT,n}}{\Delta T_{lm}}$$

The specific heat of this fictitious fluid is defined as:

$$c_{p,af} = \frac{(h_{a,ex} - h_{a,su})}{(t_{wb,ex} - t_{wb,su})}$$

The repartition of the global heat transfer coefficient into air-side and water-side heat transfer coefficients is done by means of the following equations:

$$R_a = \frac{R_{glob}}{\left(\frac{c_{p,a}}{c_{p,af}} + \frac{1}{n}\right)}$$

$$R_w = \frac{R_a}{n}$$

Where n is the key repartition factor. As a first approximation, it is proposed to consider a value of $n = 4$.

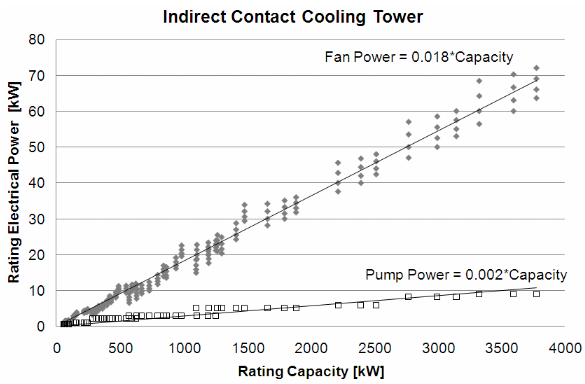


Figure50: ICCT (Centrifugal) Fan and Auxiliary Pump Powers

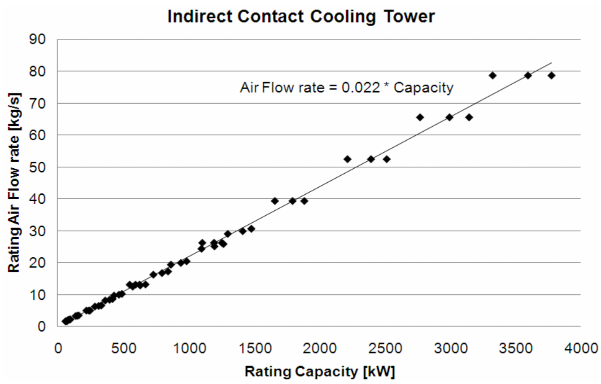


Figure51: ICCT Air Flow rate

Cooler simulation

Both dry fluid cooler and cooling tower are simulated as counterflow heat exchangers by means of classical ε-NTU method. The effectiveness of the cooling tower considered as a heat exchanger is calculated according to the following equations:

$$\varepsilon = \frac{1 - exp(-NTU * (1 - \omega))}{1 - \omega * exp(-NTU * (1 - \omega))}$$

$$\omega = \frac{\dot{C}_{min}}{\dot{C}_{max}}$$

$$NTU = \frac{AU}{C_{\min}}$$

$$\dot{Q} = \varepsilon * C_{\min} * (t_{w,su} - t_{wb,su})$$

Energy balances on air and water sides allow computing the water and air exhaust conditions.

$$\dot{Q} = \dot{M}_w * c_{p,w} * (t_{w,su} - t_{w,ex})$$

$$\dot{Q} = \dot{M}_a * (h_{a,ex} - h_{a,su})$$

No iteration is needed for simulating the dry fluid cooler (dry operation without evaporation). An iterative loop is needed in the case of the cooling tower to compute the fictitious fluid specific heat ($c_{p,af}$) and the corresponding air exhaust wet bulb temperature.

Cooling coil model

The (condensing) cooling coil model is used to simulate the heat source of the exhaust ventilation air heat pump system. Like for other components, the sizing and the simulation part are present.

Software developers and researchers developed and validated many water cooling coil models during the last decades. The most commonly used water cooling coil models are:
- Single zone epsilon-NTU models [44] [27] [76] [77].
- Variable boundary models [29]
- Finite element [30] and lumped models.

The model used in the present work has been developed by Morisot [76] [77].The author proposed a combination of the "Toolkit" [29] model and the Braun's method [44] [27]. The characteristics of this third model are:
- The wet regime is described in the same way than the "Toolkit" model (using air enthalpies to compute the sensible and latent heat exchanges).
- The Braun's method is used to determine the operation regime of the coil and consists in determining simultaneously the cooling capacity by assuming the coil completely dry or completely wet and to choose the regime leading to maximal heat transfer rate.

Cooling coil sizing

The main parameters of the considered cooling coil model are the nominal heat transfer coefficients on air and water sides and the corresponding air and water flow rates. In order to simplify the model, default values of the heat transfer coefficients are directly determined according to manufacturer's data. This parameters identification method allows reducing the number of inputs needed by the system to only describe the air mass flow rate. According to the manufacturer data the following laws have been established under nominal conditions:

$$AU_w = -1.962 * 10^3 + 3.650 * 10^3 * \dot{M}_a$$
$$AU_a = -7.826 * 10^2 + 1.501 * 10^3 * \dot{M}_a$$
$$\dot{M}_w = -1.822 * 10^{-1} + 6.055 * 10^{-1} * \dot{M}_a$$

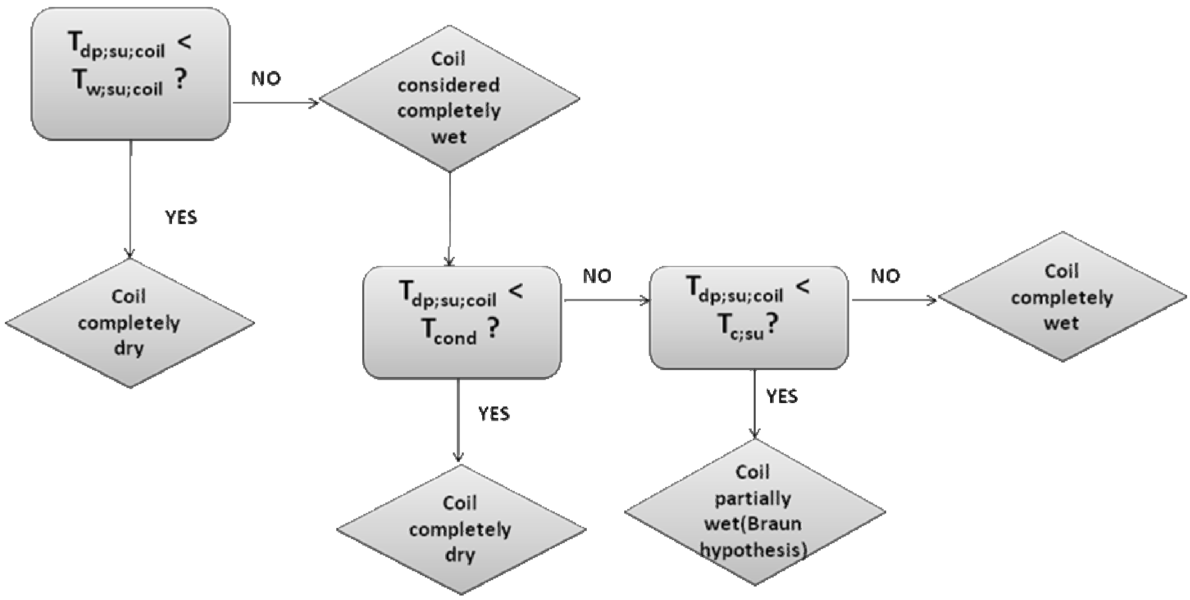
Cooling coil simulation

Three operating modes can be identified during the simulation:

- in totally dry regime, the dry portion covers the whole heat transfer area and the wet portion area is reduced to zero;
- in totally wet conditions, the dry regime portion area is reduced to zero;
- in partially wet conditions, the Braun's method is used to determine the part of heat exchange which is latent and the exhaust air conditions.

The implemented solving procedure includes the following steps:

1. To compare $T_{dp;su;coil}$ (supply dew point temperature) and $T_{w;su;coil}$ (supply water temperature).
2. Obviously, if $T_{dp;su;coil}$ is inferior to $T_{w;su;coil}$, the coil is completely dry.
3. If $T_{dp;su;coil}$ is superior to $T_{w;su;coil}$, the coil is considered completely wet first.
4. The fourth step is to compare $T_{dp;su;coil}$ to the condensated water temperature.
5. If the condensated water temperature is superior to $T_{dp;su;coil}$ then the cooling coil is completely dry.
6. The sixth step is to compare $T_{dp;su;coil}$ to the supply cooling coil contact temperature
7. If the supply cooling coil contact temperature is inferior to $T_{dp;su;coil}$ then the cooling coil is completely wet.
8. In the opposite case, the cooling coil is partially wet and the Braun's hypothesis is applied.



Under dry conditions, exhaust air and water conditions can be determined using standard heat effectiveness relationships (classical ε – NTU method) based on the overall heat transfer coefficient (AU_{dry} in W/K).

In wet regime, enthalpies are preferred instead of temperatures to determine the cooling capacity of the coil because they are including latent effects. In this model, water enthalpies are replaced by "fictitious fluid" enthalpies, defined as the enthalpy of saturated air at the temperature of the water. Heat transfer in the coil is calculated by the following relationships:

$$\dot{Q}_{coil;wet} = \dot{C}_{a;wet} \cdot (h_{a;su;coil} - h_{a;ex;coil})$$

$$\dot{Q}_{coil;wet} = \dot{C}_{w;wet} \cdot (h_{w;sat;su;coil} - h_{w;sat;ex;coil})$$

with

$$\dot{C}_{a;wet} = \dot{M}_a$$

$$\dot{C}_{w;wet} := \dot{M}_w \cdot \frac{c_{p;w}}{c_{p;sat}}$$

where the variable $c_{p;sat}$ is an effective specific heat of saturated air, defined by the following relationship:

$$c_{p;sat} = \frac{h_{dp;su;coil} - h_{w;sat;su;coil}}{t_{dp;su;coil} - t_{w;su;coil}}$$

It is important to notice that the rigorous determination of $c_{p;sat}$ requires the knowledge of the surface temperature at the entrance and at the exit of the wet part of the coil. In order to avoid iterations, the dewpoint temperature of the entering air and the supply water temperature are preferred.

The air-side and water-side heat transfer coefficients are given by:

$$AU_a = AU_{a,n} * \left(\frac{\dot{M}_a}{\dot{M}_{a,n}}\right)^{0.67}$$

$$AU_w = AU_{w,n} * \left(\frac{\dot{M}_w}{\dot{M}_{w,n}}\right)^{0.8}$$

where the indice "n" stands for "nominal values".

The overall enthalpy transfer coefficient can be related to conventional internal and external heat transfer resistances by the following relationship:

$$AU_{wet} := \frac{1}{c_{p;sat} \cdot R_w + c_{p;a} \cdot R_a}$$

where $R_w = \frac{1}{AU_w}$ and $R_a = \frac{1}{AU_a}$.

Knowing the overall enthalpy transfer coefficient, the air exhaust enthalpy and the water exhaust enthalpy can be determined by the classical ε – NTU method (as example for a counterflow heat exchanger):

$$\varepsilon_{wet} := \frac{1 - \mathbf{exp}(-NTU_{wet} \cdot (1 - \omega_{wet}))}{1 - \omega_{wet} \cdot \mathbf{exp}(-NTU_{wet} \cdot (1 - \omega_{wet}))}$$

The leaving air conditions (dry bulb temperature and humidity) are determined by considering a fictitious semi isothermal heat exchanger. One of the two fluids is the air and the other is a fictitious fluid with an infinite capacity rate at the condensate temperature.

Ground heat exchanger

The model used in this work is directly based on a model developed of Bernier [12] [13] [14] [15] [16]. Many assumptions have been made in order to develop the model:

- The thermal gradient along the borehole is not taken into account. Only the mean temperature of the surrounding ground is considered. The effect of the ambient air temperature affecting the ground temperature is neglected. This effect is negligible after 5-10 meters deep.

- The undisturbed ground temperature is supposed to remain constant on the depth of the borehole.

- An analytical solution of the infinite cylindrical heat source was used to simulate the influence of the heat rejected or extracted on the surrounding ground temperature (“global heat transfer”).

- Dynamic effects are neglected in the calculation of the “local heat transfer” (heat transfer in the borehole, between the borehole’s wall and the fluid). Only static convective and conduction thermal resistances of, respectively, the fluid and the pipe and the grout are used.

- Long term effects are neglected in the calculation of the “global heat transfer” (heat transfer in the surrounding ground): three dimensional heat transfers and the interaction between two boreholes of the same borefield (temperature penalty) are not taken into account. These effects are negligible for a simulation on a period shorter than a few years.

- An aggregation algorithm is used to calculate the thermal history of the borehole.

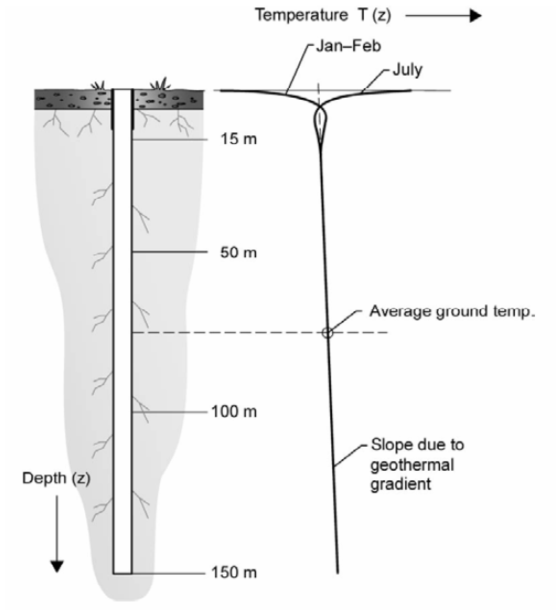


Figure52: Ground temperature gradient

The solving procedure of the model involves the following steps:

1. Calculating the temperature of the inner surface of the borehole (global heat exchange) by means of the Cylindrical Heat Source method and an improved time superposition algorithm (Multiple Load Aggregation Algorithm).

2. Calculating the average fluid temperature by considering only static heat transfer in the borehole between the borehole’s wall the fluid (local heat exchange)

3. Calculating the exhaust fluid temperature

The borehole thermal resistance Rb is given by the following equation:

$$R_b = R_{pp} + R_g$$

where R_{pp} is the thermal resistance between the fluid and the pipe external surface, [m-K/W];
R_g is the thermal resistance of the grout (between the pipe surface and the borehole wall), [mK/W].

The thermal resistance of the pipes is given by

$$R_{pp} = \frac{R_{pipe}}{2}$$

Where R_{pipe} is the sum of the thermal resistance of the pipe wall and the convective film,

$$R_{pipe} = \frac{\ln\left(\frac{D_{p,o}}{D_{p,i}}\right)}{2\pi k_p} + \frac{1}{\pi D_{p,i} h_i}$$

where $D_{p,o}$ is the outter diameter of the tubes, [m] ;
$D_{p,i}$ is the inner diameter of the tubes, [m] ;
k_p is the thermal conductivity of the piper material, [W/mK] ;
h_i is the convective heat transfer coefficient of the fluid, [W/m²K].

The Nusselt number in the tubes and the corresponding convective heat transfer coefficient are computed by means of the following set of equations:

Procedure **Nusselt** ($c_{p,f}$, ρ_f, μ_f, $\mu_{f,w}$, k_f, $D_{p,i}$, L_b, $\dot{m}_{f,b}$: Nu#, Re, Pr)

Nusselt calculation

$$A_p := \pi \cdot \frac{D_{p,i}^2}{4}$$

$$V_f := \frac{\dot{m}_{f,b}}{\rho_f} \cdot \frac{1}{A_p}$$

$$Re := \rho_f \cdot V_f \cdot \frac{D_{p,i}}{\mu_f}$$ Reynolds Number

$$Pr := c_{p,f} \cdot \frac{\mu_f}{k_f}$$ Prandlt number

If (Re <= 2300) Then   Sieder and Tate's Correlation; Whitaker modification

valid for :
T_w = constant
0.48 < Pr < 16700
0.0044 ($mu_f/mu_{f,w}$) < 9.75

$$F_{whitaker} := \left[Re \cdot Pr \cdot \frac{D_{p,i}}{L_b} \right]^{(1/3)} \cdot \left[\frac{\mu_f}{\mu_{f,w}} \right]^{0.14}$$

If ($F_{whitaker}$ < 2) Then

$$Nu\# := 3.66$$

Else

$$Nu\# := 1.86 \cdot \left[Re \cdot Pr \cdot \frac{D_{p,i}}{L_b} \right]^{(1/3)} \cdot \left[\frac{\mu_f}{\mu_{f,w}} \right]^{0.14}$$

EndIf

If ((2300 < Re) and (Re <= 10000)) Then   Gnielinski's Correlation

valid for :
0.5 < Pr < 2000
2300 < Re < 5E6

$$f := (0.79 \cdot \ln (Re) - 1.64)^{-2}$$

$$Nu\# := \frac{\frac{f}{8} \cdot (Re - 1000) \cdot Pr}{1 + 12.7 \cdot \sqrt{\frac{f}{8}} \cdot (Pr^{(2/3)} - 1)}$$

If (10000 < Re) Then   Gnielinski's Correlation

valid for :
0.5 < Pr < 2000
2300 < Re < 5E6

$$f := (0.79 \cdot \ln (Re) - 1.64)^{-2}$$

$$Nu\# := \frac{\frac{f}{8} \cdot (Re - 1000) \cdot Pr}{1 + 12.7 \cdot \sqrt{\frac{f}{8}} \cdot (Pr^{(2/3)} - 1)}$$

End **Nusselt**

The concept of shape factors is used to evaluate the conduction heat transfer in the grout [Remund]. The grout thermal resistance is given by:

$$R_b = \frac{1}{S_b * k_g}$$

Where the shape factor S_b is given by

$$S_b = \frac{1}{\beta_0 \left(\frac{r_b}{r_p}\right)^{\beta_1}}$$

where the values of the parameters β_0 and β_1 are given in Table 12 as a function of the borehole configuration [Remund] and where r_b and r_p are, respectively, the borehole and the outer pipe radius (in m).

Table 12: Borehole configurations A, B and C [81]

Configuration	β_0	β_1
A	14.4509	-0.8176
B	17.4427	-0.6052
C	21.9059	-0.3796

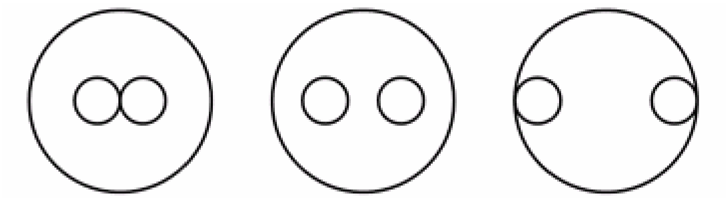


Figure53: Borehole configurations A, B and C [81]

The mean fluid temperature is given by the following equation:

$$T_w - T_{f,m} = \frac{\dot{Q}_b}{L} R_b$$

Where T_w is the borehole wall temperature at time t, in °C
$T_{f,m}$ is the mean fluid temperature, in °C
$\dot{Q}_b$ is the borehole load, in W
L is the borehole length, in m
R_b is the borehole thermal resistance, in m-K/W

The cylindrical heat source method [32] [33] is used to solve the "global heat exchange" in the surrounding ground and compute the borehole wall temperature:

$$T_w = T_g - \frac{q}{L}\frac{G(Fo)}{k_s}$$

Where T_w is the borehole wall temperature at time t, in °C
T_g is the undisturbed ground temperature, in °C
q is the borehole load, in W
L is the borehole length, in m
k_s is the soil thermal conductivity, in W/m-K
Fo is the Fourier number, defined as: $Fo = 4 * \frac{\alpha_s * t}{{d_b}^2}$
G is the cylindrical heat source function, [34]

```
Function  G (u)
   egam := 1.781
   p2 := 9.87
   tcrit := 6.124633
   ca := 1.128379
   c0 := - 0.5
   c1 := 0.2756227
   c2 := - 0.1499385
   c3 := 0.0617932
   c4 := - 0.01508767
   c5 := 0.001566857
   If ( u <= tcrit ) Then
      G := √u · [ca + c0 · √u + c1 · u + c2 · u^1.5 + c3 · u^2 + c4 · u^2.5 + c5 · u^3]
    Else
      z := ln[4 · u / egam]
      G := (2 · z · (8 · u · (1 + 2 · u) - 1 - 3 · z) + 16 · u + p2 + 3) / (64 · u^2)
   EndIf
   G := G / 6.283
End  G
```

The multiple load aggregation algorithm of Bernier et al. [14] is used for time superposition. Recognizing that the immediate short-term thermal history is more important than the past thermal history in the determination of the actual borehole wall temperature, the short-term thermal history (latest 12 hours) is kept intact while the past thermal history is averaged in four distinct terms with different durations (e.g. 48h, 168h, 360h and the remaining hours). This load aggregation technique is illustrated in Figure54.

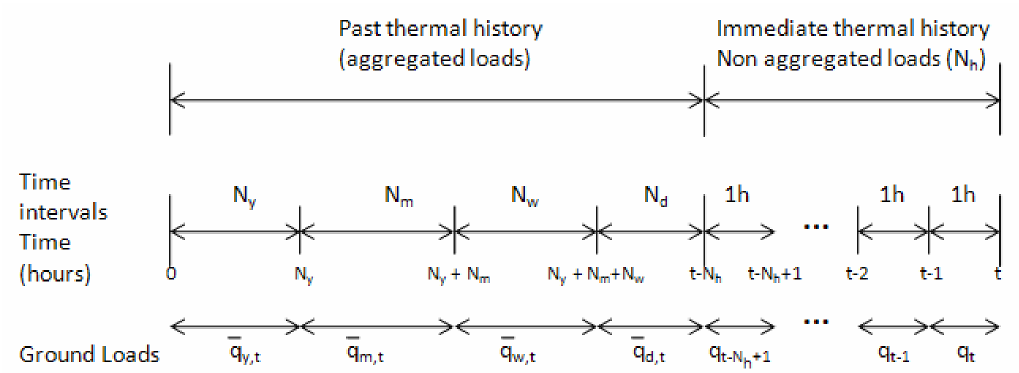


Figure 54: MLAA technique

Parameters	Value	Units
Borehole		
borehole diameter	0.15	[m]
borehole depth	100	[m]
borehole spacing	5 to 8	[m]
Grout		
thermal conductivity	0.7 (regular) to 2.1 (enhanced)	[W/m-K]
Pipe		
thermal conductivity	0.042	[W/m-K]
Interior diameter	0.0274	[m]
exterior diameter	0.0335	[m]
Glycol water (EG 25%)		
fluid specific heat	3795	[J/kg-K]
fluid density	1037	[kg/m^3]
Viscosity	0.00345	[Pa-s]
Conductivity	0.47	[W/m-K]

Table 13: Ground heat exchanger parameters – default values

	thermal conductivity	thermal diffusivity
	[W/m-K]	[m^2/day]
Heavy Soil Saturated	2.4	0.078
Heavy Soil Damp	1.3	0.055
Heavy Soil Dry	0.87	0.045
Light Soil Damp	0.87	0.045
Light Soil Dry	0.35	0.025

Table 14: Soil thermal properties [82]

Thermal storage and water loop inertia

The model is based on a first order differential equation. The perfectly mixed thermal storage is modelled as a unique capacitance equal to the thermal capacitance of the contained volume of fluid. Both water loop tanks and inertia of the water loop (WLHP systems) are modelled in a similar way.

The energy balance of the storage can be written as the following equation.

$$\dot{H} + \dot{Q}_{amb} = \dot{U}$$

where the enthalpy flow rates related to the fluid flow going through the storage tank is given by,

$$\dot{H} = \dot{M}_w * c_{p,w} * (t_{su,TS} - t_{ex,TS})$$

The thermal losses/gains are computed by means of the following equations:

$$\dot{Q}_{amb} = U * A * (t_{amb} - t_{w,in,TS})$$

The following integral should be computed to solve the differential equation:

$$\Delta U = \int_{\tau_1}^{\tau_2} \dot{U}\, d\tau$$

$$\Delta U = VOL * \rho_w * c_{p,w} * (t_{w,in,TS} - t_{init})$$

where t_{init} is the initial temperature of the storage and VOL, the volume of the tank. As the tank is simulated by a unique capacitance, the exhaust temperature is equal to the internal temperature of the tank. Table 15 provides typical values of loop volume and optimal sizes of water storage tanks for water loop systems.

Table 15: WLHP Specific Pump Powers

	Value	Source
WLHP loop volume	11 to 14 l/kW of installed cooling capacity	[18]
WLHP thermal storage volume	25 to 100 l/kW of installed cooling capacity	[18] [20]