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Optimization of heating and cooling system for a passive house equipped with heat pump and heat storage

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MASTER THESIS

for

Student
Damian Mindykowski

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Optimization of the heating and cooling system for a residential passive house
equipped with heat pump and energy storage

*Optimalisering av oppvarming og kjøling av et passiv hus for bolig formål
utstyrt med varmepumpe og energilager*

Background and objective

The goal of authorities, both in Norway and worldwide, is to reduce energy consumption in buildings and its impact on the environment. This is stimulated both by use of appropriate administrative measures and by encouraging sustainable practices in the design and operation of buildings. Few years ago, Norwegian authorities introduced national passive house standards both for residential and for office buildings, and they promote a broader use of renewable energy.

Heat pump is a promising technology that use the free environmental energy and is able to supply both heating and cooling. However, the great shortcoming of heat pumps is that their operation is not efficient at peak demands. Consequently, the heat supply system must contain an additional source. The energy demands vary with user activity and the outdoor conditions. An energy storage could be a convenient solution to keep the energy available for the occurring demands as well as to give space for utilization of varying energy prices for powering of heat pump.

The objective of this work is to evaluate the applicability challenges for and optimize energy supply systems intended for residential building designed in compliance with the Norwegian passive house standard, containing heat pump and heat storage.

The following tasks are to be considered:

1. Accomplish a literature search and present results of a literature study regarding mathematical modelling for estimation of performances of energy supply systems containing heat pump, appropriate top lad source and energy storage.
2. Analyze applicability of different simulation tools (e.g. IDA ICE, EnergyPlus, Trnsys and others) and select a suitable tool for further analysis work. Create a workable simulation model for the considered energy supply system.
3. Consider and select appropriate optimization criteria and perform optimization of the considered energy supply system. Analyze results and give some broader recommendations

for design of energy supply systems based on use of heat pump and energy storage for residential buildings.

4. Make a draft proposal (6-8 pages) for a scientific paper based on the main results of the work performed in the master thesis.

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Within 14 days of receiving the written text on the master thesis, the candidate shall submit a research plan for his project to the department.

When the thesis is evaluated, emphasis is put on processing of the results, and that they are presented in tabular and/or graphic form in a clear manner, and that they are analyzed carefully.

The thesis should be formulated as a research report with summary in both English and Norwegian, conclusion, literature references, table of contents etc. During the preparation of the text, the candidate should make an effort to produce a well-structured and easily readable report. In order to ease the evaluation of the thesis, it is important that the cross-references are correct. In the making of the report, strong emphasis should be placed on both a thorough discussion of the results and an orderly presentation.

The candidate is requested to initiate and keep close contact with his/her academic supervisor(s) throughout the working period. The candidate must follow the rules and regulations of NTNU as well as passive directions given by the Department of Energy and Process Engineering.

Risk assessment of the candidate's work shall be carried out according to the department's procedures. The risk assessment must be documented and included as part of the final report. Events related to the candidate's work adversely affecting the health, safety or security, must be documented and included as part of the final report. If the documentation on risk assessment represents a large number of pages, the full version is to be submitted electronically to the supervisor and an excerpt is included in the report.

Pursuant to "Regulations concerning the supplementary provisions to the technology study program/Master of Science" at NTNU §20, the Department reserves the permission to utilize all the results and data for teaching and research purposes as well as in future publications.

The final report is to be submitted digitally in DAIM. An executive summary of the thesis including title, student's name, supervisor's name, year, department name, and NTNU's logo and name, shall be submitted to the department as a separate pdf file. Based on an agreement with the supervisor, the final report and other material and documents may be given to the supervisor in digital format.

☐ Work to be done in lab (Water power lab, Fluids engineering lab, Thermal engineering lab)
☐ Field work

Department of Energy and Process Engineering, 13. January 2016

Olav Bolland
Department Head

Vojislav Novakovic
Academic Supervisor

Preface

This master thesis was written at the Department of Energy and Process Engineering at NTNU during the spring semester of 2016, as. It makes up 30 ECTS credits for the last semester of my 1,5 - year master studies began at GUT (Gdańsk University of Science and Technology) in Poland, later continued at NTNU (Norwegian University of Science and Technology) in Norway.

I would like to honestly thank all people who supported and helped me during writing this thesis. Special thanks go to my supervisors - prof. Vojislav Novakovic and prof. Janusz Cieśliński for taking time to answer all my questions, for guidance throughout the semester and for constructive support, which they never refused me. I would like to also thank Natasa Nord for help with IDA ICE simulation environment, Michael Bantle from SINTEF who presented me Dymola software, Eugen Uthaug who shared me the licence for Dymola and PhD candidate Daniel Rohde for answering all my questions related with Dymola. Moreover, I would like to thank Salvatore Carlucci who shared me the licence for IDA ICE and Michael Bodmann from TLK-Thermo company for free provision of the licence on TIL data library and DaVE program, related with Dymola. Furthermore, I would like to thank Ingo Frohböse from TLK-Thermo for help and answering my few questions related with Dymola.

Damian Mindykowski,

Trondheim, 10.06.2016

Summary

During considerations included in this master thesis re - design of previously developed low energy building destined to be located in Oslo was performed using IDA ICE simulation software. After some construction modifications, retrofit of this building to the passive standard was carried out. Analysis of the previously designed heating system (consisting of transcritical air - based heat pump, thermally - stratified hot water storage tank and top - up electric heaters) for low energy building was conducted, followed by re - design and modification of the heat pump using Dymola with TIL extension and storage tank utilizing IDA ICE. In addition, cooling system was designed and implemented into the passive house in IDA ICE, because there was no cooling system in previous considerations of analyzed building. Subsequently, optimization of both heating and cooling system was performed using mentioned simulation tools. Afterwards, some broader recommendations for design of modern energy supply systems based on use of heat pump and hot water storage tank for residential buildings were listed. Finally, draft proposal for scientific paper based on the main results of the work performed in this master thesis was created and placed in the appendix 1.

Retrofit of analyzed building from its low energy state in previous considerations to the passive standard performed in IDA ICE resulted in 31% drop of its total energy demand (from 40979 kWh down to 28177 kWh annually). The main difference was embodied in energy required for covering space heat demand of the building, therefore it is clearly visible that properly designed passive house is much better insulated building than low energy, although both buildings achieve good performance. In turn, ground source heat pump designed in Dymola achieved SPF of 2,40 in basic configuration (without optimization, including water pump energy consumption), therefore its utility in the passive house resulted in 40% decrease of total electricity supplied to the building, compared to usage of 100% efficient electric heating system (from 28177 kWh to 16955 kWh annually). Furthermore, thanks to performed optimization of both transcritical heat pump, hot water storage tank and cooling system further drop of total electricity supplied to the building was possible - a drop of 30% (from 16955 kWh down to 11931 kWh) was achieved. In case of the heat pump optimization, it resulted in increase of its SPF from 2,40 up to 3,27. For comparison, SPF of previously designed air - based heat pump co-operated with top - up electric heaters was equal to 2,51, what constitutes a value similar to 2,40, knowing that electricity consumption of fans supplying the air flow to the evaporator was not included in calculations.

In opinion of the author of this master thesis energy supply system which he developed for the passive building can be characterized by relatively realistic design and good performance.

Optimization of this system conducted afterwards resulted in significant improvement of the heating and cooling system operation, essentially decreasing electricity consumption of analyzed passive house. However, many issues and cases which were desired to be considered as integral parts of this master thesis had to be neglected, mainly due to time restrictions. As the passive house achievements are very good and the building was re - designed to be more realistic, these issues were mostly related with either more accurate and realistic design of the heat pump, implementation and adjustment of other thermal energy storage methods utilized in residential building or with further optimization of considered energy supply system, from viewpoint of other criteria depicted in the thesis.

Sammendrag

Under hensyn i denne masteroppgaven re - design av tidligere utviklet lavenergibygg forutbestemt til å være lokalisert i Oslo ble utført ved hjelp av IDA ICE simuleringsprogram. Etter noen bygge modifikasjoner, ble ettermontering av denne bygningen til passiv standard utført. Analyse av tidligere utviklet varmesystemet (bestående av trans luft - basert varmepumpe, varme - stratifisert varmtvannstanken og toppen - opp elektriske varmeovner) for lavenergibygg ble gjennomført, etterfulgt av re - design og modifikasjon av varmepumpen bruker Dymola med tIL forlengelse og lagertank utnytte IDA ICE. I tillegg ble kjølesystemet utviklet og implementert i passivhus i IDA ICE, fordi det var ingen kjølesystem i tidligere betraktninger analysert bygningen. Deretter optimalisering av både varme og kjøling systemet ble utført ved hjelp av nevnte simuleringsverktøy. Etterpå ble noen bredere anbefalinger for utformingen av moderne energiforsyningssystemer basert på bruk av varmepumpe og varmtvannstank for boligbygg oppført. Til slutt ble utkast til vitenskapelig artikkel basert på de viktigste resultatene av arbeidet utført i denne masteroppgaven opprettet og plassert i vedlegg 1.

Ettermontering av analysert bygning fra sin lave energitilstand i tidligere betraktninger til passivhusstandard utført i IDA ICE resulterte i 31% nedgang i det totale energibehovet (fra 40979 kWh ned til 28177 kWh årlig). Den største forskjellen er nedfelt i energien som kreves for å dekke plass varmebehovet i bygningen, derfor er det tydelig at riktig utformet passivhus er mye bedre isolerte bygninger enn lav energi, selv om begge bygningene oppnå gode resultater. I sin tur, bergvarmepumpe utformet i Dymola oppnådd solfaktor 2,40 i grunnkonfigurasjonen (uten optimalisering, inkludert vannpumpe energiforbruk), derfor sin nytte i passivhus resulterte i 40% reduksjon av totale elektrisitet levert til bygningen, i forhold til bruk av 100% effektiv elektrisk varmesystem (fra 28177 kWh til 16955 kWh årlig). Videre, takket være utført optimalisering av både transkritisk varmepumpe, varmtvannslagertank og kjølesystem videre fall av den totale strøm som tilføres til bygningen var mulig - en reduksjon på 30% (fra 16955 kWh ned til 11 931 kWh) ble oppnådd. Ved varmepumpen optimalisering, det resulterte i økning av dens SPF fra 2,40 opp til 3,27. Til sammenligning solfaktor tidligere designet luft - basert varmepumpe samarbeidet med topp - opp elektriske varmeovner var lik 2,51, hva som utgjør en verdi lik 2,40, vel vitende om at elektrisitetsforbruket til fans som forsyner luftstrømmen til fordamperen ble ikke tatt med i beregningene.

I oppfatning av forfatteren av denne masteroppgaven energiforsyning system som han utviklet for passivhus kan være preget av relativt realistisk design og god ytelse. Optimalisering av dette systemet utført etterpå resulterte i betydelig forbedring av varme- og kjølesystemet drift, i

hovedsak å redusere strømforbruket av analysert passivhus. Men mange saker og saker som ble ønsket å bli betraktet som en integrert del av denne masteroppgaven måtte bli neglisjert, hovedsakelig på grunn av tidsbegrensninger. Som passivhus prestasjoner er veldig god, og bygningen ble re - designet for å være mer realistisk, ble disse problemene stort sett beslektet med enten mer nøyaktig og realistisk design av varmepumpen, gjennomføring og justering av andre termisk energi lagringsmetoder benyttes i bolighus eller med ytterligere optimalisering av regnet energiforsyning system, fra synspunktet til andre kriterier som er avbildet i avhandlingen.

Podsumowanie

Podczas rozważań zawartych w niniejszej pracy magisterskiej zostało przeprowadzone odtworzenie uprzednio zaprojektowanego budynku niskoenergetycznego przeznaczonego dla Oslo. W tym celu użyto programu IDA ICE. Po wprowadzeniu pewnych modyfikacji konstrukcyjnych budynku przeprowadzono renowację budynku do standardu pasywnego. Analiza poprzednio zaprojektowanego systemu ogrzewania (składającego się z transkrytycznej powietrznej pompy ciepła, zbiornika ciepłej wody użytkowej ze stratyfikacją termiczną oraz grzałek elektrycznych, pokrywających szczytowe zapotrzebowanie na energię do ogrzewania) dedykowanego uprzednio do budynku niskoenergetycznego została przeprowadzona, następnie odtwarzając i modyfikując pompę ciepła w programie Dymola, wykorzystując dodatkową bibliotekę plików TIL, a zbiornik c. w. u. w IDA ICE. Dodatkowo, system chłodzenia został od nowa zaprojektowany i zaadoptowany do domu pasywnego w IDA ICE, ponieważ we wcześniejszych rozważaniach rozważanego budynku (wtedy jeszcze jako niskoenergetycznego) systemu tego nie było. Następnie stworzony system ogrzewania i chłodzenia budynku został zoptymalizowany, wykorzystując wspomniane programy. W dalszej kolejności zostały przedstawione ogólne rekomendacje dotyczące projektowania nowoczesnych systemów ogrzewania i chłodzenia (bazujących na pompach ciepła i zbiornikach c. w. u.) dla budynków mieszkalnych. Po ukończeniu zasadniczej części pracy została przygotowana propozycja artykułu naukowego bazującego na jej treści i wynikach. Szczegóły dostępne są w załączniku 1 pracy.

Wynik renowacji analizowanego budynku wykonanej w IDA ICE, przeprowadzonej ze stanu niskoenergetycznego do pasywnego to 31% spadek całkowitego zapotrzebowania budynku na energię (od 40979 kWh do 28177 kWh rocznie). Spadek ten został spowodowany głównie zmniejszeniem się potrzeb grzewczych budynku pod kątem ogrzewania pomieszczeń. Widać więc, że choć obie wersje budynku (tj. wykonanego w standardzie niskoenergetycznym i pasywnym) odznaczają się stosunkowo niskim zapotrzebowaniem energetycznym, to dobrze zaprojektowany budynek pasywny jest wyraźnie lepiej zaizolowany. Z kolei, gruntowa pompa ciepła zaprojektowana w Dymola osiągnęła współczynnik SPF rzędu 2,40 w podstawowej odmianie (tj. bez optymalizacji, z uwzględnieniem energii zużywanej przez pompę wodną w obiegu dolnego źródła ciepła pompy). W porównaniu z elektrycznym systemem ogrzewania i chłodzenia o przyjętej sprawności 100% użycie tej pompy ciepła w budynku pasywnym poskutkowało obniżeniem się jego zapotrzebowania na energię elektryczną aż o 40% (z 28177 kWh do 16955 kWh rocznie). Ponadto, dzięki przeprowadzonej optymalizacji zarówno pompy ciepła, zbiornika c. w. u. ze stratyfikacją oraz systemu chłodzenia możliwe było dalsze obniżenie zapotrzebowania budynku na energię elektryczną. Łącznie w wyniku optymalizacji uzyskany został jej spadek o kolejne 30% (z 16955 kWh do zaledwie 11931 kWh). Co do pompy ciepła, dzięki optymalizacji jej współczynnik SPF wzrósł z 2,40 do 3,27. Dla porównania, SPF powietrznej pompy ciepła współpracującej z grzałkami elektrycznymi pokrywającymi szczytowe zapotrzebowanie, zaprojektowanej w rozważaniach poprzedzających tą pracę magisterską wynosił 2,51, co jest wartością podobną do 2,40 wiedząc, że zapotrzebowanie na energię elektryczną wentylatorów dostarczających strumień powietrza do parownika pompy ciepła nie zostało uwzględnione w obliczeniach.

W opinii autora tej pracy magisterskiej system energetyczny, który został przez niego odtworzony i znacząco zmodyfikowany charakteryzuje się względnie rzeczywistym funkcjonowaniem i dobrymi osiągami. Z kolei przeprowadzona w tej pracy optymalizacja systemu zaowocowała istotną poprawą jego działania, zmniejszając zapotrzebowanie domu pasywnego na energię elektryczną w znaczący sposób. Jednakże, wiele pomysłów i kwestii, które chciano poruszyć w niniejszej pracy dyplomowej musiało zostać zaniechanych, głównie z powodu czasowych ograniczeń. Jako że osiągnięcia budynku pasywnego są bardzo dobre, a budynek został zaprojektowany w sposób bardziej rzeczywisty, pomysły te dotyczyły głównie jeszcze bardziej dokładnego i rzeczywistego zaprojektowania pompy ciepła, zastosowania innych metod magazynowania energii cieplnej w budynku mieszkalnym, oraz były związane z dalszą optymalizacją rozważanego systemu energetycznego, bazującą na podstawie innych kryteriów optymalizacyjnych opisanych w pracy.

Nomenclature and abbreviations

A_{EV} (A_{GC}) - heat exchange area of evaporator (gas cooler) [m^2],

A_v - opening area of the valve [m^2],

c - specific heat capacity (specific heat) [$\frac{J}{kg \cdot K}$],

C_v - orifice coefficient [-],

C_{min} - minimum heat capacity rate of [$\frac{J}{K \cdot s}$],

C_{ra} - heat capacity ratio [-],

$C_{sf,EV}$ - heat capacity rate of the secondary fluid (e. g. air, water or glycol mix) in evaporator [$\frac{J}{K \cdot s}$],

COP - coefficient of performance [-],

COP_{max} - maximum achievable COP of the conventional heat pump [-],

COP_{LZ} - maximum achievable COP of the transcritical heat pump [-],

E - energy savings [%], [kWh],

E_{HP} - total heat annually delivered by the heat pump [kWh],

E_T - total electrical energy supplied to heat pump in considered period [kWh],

f - frequency of the compressor work, providing information about its running speed [Hz],

h_1 - specific enthalpy of refrigerant at suction state [$\frac{kJ}{kg}$],

h_2 - specific enthalpy of refrigerant at discharge state after actual compression [$\frac{kJ}{kg}$],

h_{2s} - specific enthalpy of refrigerant at discharge state after isentropic compression [$\frac{kJ}{kg}$],

$\dot{m}_{CO_2}$ - mass flow rate of CO_2 [$\frac{kg}{s}$],

N - number of gas cooler (or evaporator or SLHX) subsections [-],

NTU - number of transfer units [-],

p_{crit} - critical pressure [bar], [MPa],

P_C - electric power of the compressor [W],

P_{HP} - maximum power of the heat pump [kW],

P_{max} - maximum heating capacity of the heat pump at design conditions [kW],

$p_{GC,opt}$ - optimal gas cooler presure [bar],

P_T - peak heating load of the building [kW],

Q_C - heat removed from the cold reservoir [W],

Q_H - heat supplied to the hot reservoir [W],

Q_{HP} - total annual heating demand covered by the heat pump [kWh],

Q_T - total annual heating demand of the building [kWh],

Q_s - total heat delivered from heat pump in considered period [kWh],

$\dot{Q}_{GC}$ - heat flux released from CO_2 flow, in the gas cooler of transcritical heat pump [kW],

$\dot{Q}_{EV}$ - heat flux absorbed by refrigerant in the evaporator [kW],

SPF - seasonal performance factor of the heat pump [-],

t_{crit} - critical temperature [^{0}C],

T_C (T_0) - temperature of the cold reservoir (evaporation temperature) [^{0}C],

(T_H) - temperature of the hot reservoir [^{0}C],

T_m - average temperature of heat rejection in the gas cooler [^{0}C],

ΔT_{SH} - superheat temperature value before compression [^{0}C],

$T_{sf,I,GC}$ ($T_{sf,O,GC}$) - secondary fluid (water) inlet (outlet) temperature for gas cooler analysis [^{0}C],

$T_{r,EV}$ - constant temperature of the refrigerant (CO_2) in evaporator [^{0}C],

$T_{avg,r}$ - average temperature of refrigerant during heat rejection process in gas cooler [^{0}C],

$T_{r,j}$ - inlet temperature of refrigerant (CO_2) in currently analyzed gas cooler subsection [^{0}C],

$T_{r,j-1}$ - outlet temperature of refrigerant (CO_2) in currently analyzed gas cooler subsection [^{0}C],

$T_{r,GC,o}$ ($T_{r,GC,I}$) - temperature of the refrigerant flowing out (flowing in) of gas cooler [^{0}C],

V - displacement volume of the compressor [m^3],

$\dot{V}$ - displacement rate of the compressor [$\frac{m^3}{s}$],

VRC (VHC) - volumetric refrigeration (heating) capacity [$\frac{kJ}{m^3}$],

UA_{EV} (UA_{GC}) - heat transfer factor of the evaporator (gas cooler) [$\frac{W}{K}$],

U_{EV} (U_{GC}) - U-value of evaporator (gas cooler) [$\frac{W}{m^2 \cdot K}$],

W - total electric power supplied to the heat pump [W],

W_C - work of the compression, taking heat losses into account [W],

β - heat loss factor (compressor) [-],

ΔE - relative energy savings [%], [kWh],

Δh_{evap} - specific enthalpy of vaporization [$\frac{kJ}{kg}$],

Δp - pressure difference between gas cooler (high pressure) and evaporator (low pressure) side [bar],

ΔP - power coverage factor [-],

ΔQ - energy coverage factor [-],

η - efficiency of alternative heating system [-],

η_{LZ} - Lorentzen efficiency of transcritical heat pump cycle [-],

η_C - carnot efficiency of conventional heat pump cycle [-],

η_{vol} - volumetric efficiency of the compressor [-],

η_{is} - isentropic efficiency of the compressor [-],

ρ_v - vapour density [$\frac{kg}{m^3}$],

ρ - density of the refrigerant [$\frac{kg}{m^3}$],

τ - equivalent operating time [h],

ε - effectiveness of a heat exchanger [-],

$\frac{H}{D}$ - high to diameter ratio of hot water storage tank [-];

VAV - Variable Air Ventilation system,

CAV - Constant Air Ventilation system,

PCM - Phase Change Material,

GWP - Global Warming Potential,

ODP - Ozone Depletion Potential,

SH - Space Heating,

DHW - Domestic Hot Water,

LMTD - Logarithmic Mean Temperature Difference.

Table of contents

1. Introduction

From above the last two decades a design of residential buildings in Europe (especially in well - developed countries) is getting more and more ecological meaning. That resulted in the necessity of extreme reduction in overall energy consumption of the buildings, especially from viewpoint of the building heating needs[1], in order to limit emission of CO_2 and other green house gases to the atmosphere. Hence, highly insulated buildings - low energy, passive and even zero energy buildings[2] are becoming more and more popular, not only in commercial appliances (as it was at the beginning of their development) but also in residential sector. Today many building companies offer their customers complete projects of residential buildings which are well - insulated and on the market exist many technical solutions describing how to retrofit the older buildings to current requirements. One could therefore say, that potential of further reduction related with the building energy consumption is already significantly lower. It is true - at least from viewpoint of the building construction. Now, when modern buildings are characterized by clearly lower net energy consumption, people are trying to develop new ideas increasing energy efficiency of the buildings energy supply systems, which would allow for supplying even less energy covering already decreased heating (and cooling) demand.

Current situation on the European building market described above may justify increased development of modern HVAC-R technologies (such as transcritical heat pumps and thermal energy storage applications), observed in the last few years. The name „transcritical" derives from operating conditions of heat pumps utilizing CO_2 as the refrigerant - evaporation process occurs in subcritical circumstances (i. e. in the range below CO_2 critical point) but heat release (not condensation!) process occurs in gas cooler in supercritical conditions (where working pressures exceed the value of 85 bar[3] and may reach even 125 bar [8]). Nowadays the terms „transcritical" and „CO_2" heat pump can be used alternatively, because there is no other substance (currently discovered) which could be both used as the heat pump refrigerant in a wider scale and would require to operate in transcritical conditions. Such transcritical heat pumps have some advantages compared to heat pumps based on standard refrigerants[4], i. a. higher efficiency (in terms of achieved COP, SPF) but only when low - temperature space heating system is installed in the building[5] and more environmentally - friendly

[1] Heating needs of the buildings built in accordance with so-called old building requirements (in Norway old building code concern all building law regulations released before TEK10) constituted the main part of their overall energy consumption, therefore there was the highest potential in reduction of energy consumed in the building sector.

[2] In terms of annual net energy consumption of the building covered by external energy sources, e. g. electricity from the grid.

[3] Minimum recommended value of pressure which can be maintained on high – pressure side of transcritical heat pump to ensure its stable operation is equal to 85 bar [8].

[4] Working fluids typically used as refrigerants in subcritical heat pumps are: R-134a, R-410a, R-407c, R-717 and R600 [14].

[5] In transcritical heat pumps CO_2 releases sensible heat in gas cooler, instead of latent heat released in condenser of typical heat pump. It means that its temperature decreases while rejecting the heat and COP of such heat pump is dependent on this temperature drop

character due to CO_2 utilization which (at least in short time perspective) has significantly less destructive influence on the climate (GWP $= 0$ and ODP $= 1$) than other utilized refrigerants.

In case of thermal energy storage solutions destined for residential buildings the most popular were and still are hot water storage tanks, both utilized as buffers (co-operating with space heating loop) and accumulators (used for domestic hot water storage). However, their development progresses all the time, especially in case of thermally - stratified accumulators. There are many benefits of the stratified DHW tanks utility compared to standard, not stratified tanks, e. g. higher stability of desired water temperatures, reduced heat losses from tank to the ambient and less lime scaling inside the tank. Furthermore, stratified tanks can be smaller to cover similar DHW needs, hereby the investment cost of tank decrease. They also allow for increased utilization of solar collectors (if they are used in the building heating system), so their efficiency rises. Another method allowing for thermal energy storage in residential buildings (clearly less popular than hot water storage tanks) is utilization of PCMs (phase change materials), operating almost exclusively in form of either special heat exchangers filled with PCMs, special packs containing PCMs or building materials where PCMs occur as supplements (e. g. special PCM - based plasters which can be applied on internal layers of walls). Less popularity of PCMs utility in residential buildings (compared to hot water tanks) results from the fact that PCM - based products are expensive and the technology is relatively new. It is however almost sure that this popularity will be rising in the nearest future, due to many advantages resulting from their utility (e. g. lower daily variations of the indoor air temperature and utilization of natural heat sources, such as sun, people or equipment emitting heat inside the building).

In the first part of this master thesis broad theory was provided, related both with heat pumps and thermal energy storage methods applicable in modern residential buildings. Afterwards, various programs used for estimation of the building and its energy supply system performance will be compared, in order to select the program which is the most appropriate from viewpoint of this master thesis goals. Afterwards, specification of considered building in previous and current state will be depicted, with many other relevant information. In the next step, energy supply system of the building (also in state before and after modifications performed in this thesis) will be described. Subsequently, optimization of the energy supply system will be conducted, with conclusions and suggestions for further work located afterwards.

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value. If so-called return temperature of the water from space heating loop is too high, then CO_2 cannot be cooled hard enough, and COP of transcritical heat pump decreases.

2. Heat pump theory

Many literature sources indicates different date assuming as the beginning of the heat pump development. It is obvious that heat pump technology derives from refrigeration systems (basic refrigerator and heat pump consists of the same components, working in the same but reversed cycle), therefore it seems very genuine that it was a year of 1748 [12, 34], when William Cullen presented first artificial refrigeration system. After around 100 years, in 1852 Lord Kelvin concluded that reversed refrigeration cycle could be used for heating and device working in such conditions would need less amount of primary energy than other devices known in that period, due to gaining the heat from ambient [76]. Hence some sources indicates Kelvin as the inventor of the heat pump, although he did not demonstrate the concept [26].

The spread of electricity at the beginning of XX century allowed for the rapid development of both refrigeration and heat pumps systems. Although in 1928 the first system for heating the house based on ammonia compressor unit was launched [77], the first commercial heat pump installation appeared in Portland, Oregon in 1948 [26]. Despite the huge potential of heat pumps for energy savings, they have never been dominated form of heating systems applied in buildings, probably due to relatively high investment cost compared to another space heating systems and uncertainties about its reliability, albeit this tendency seems to be changing since few decades. As an example, the number of heat pumps utilized in Austria between 1980 and 2013 can be seen on the figure 2.1.

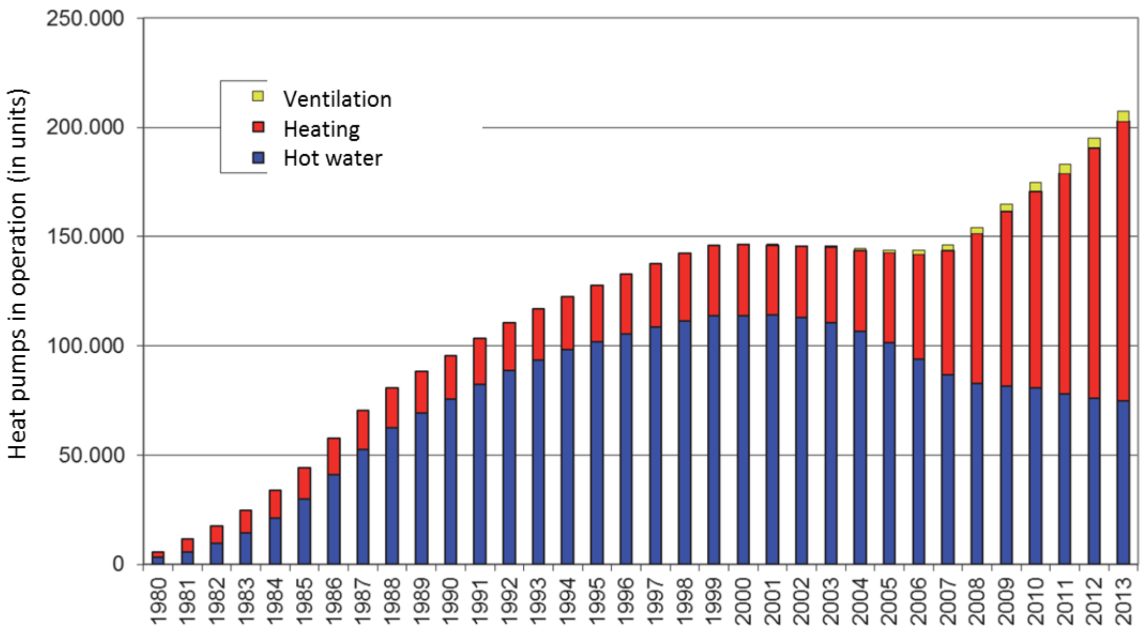

Figure 2.1. Heat pump units used in Austria for both space and water heating (red label), water heating only (blue label) and ventilation purposes (yellow label) during past three decades [35]

At the beginning of this chapter, a description of the basic heat pump work principle will be presented, together with overall depiction and comparison of the most common refrigerants used in heat pumps, in order to select the most appropriate working fluid for modern residential buildings, similar to the passive building considered in this thesis. Afterwards, various types of heat pumps will be briefly characterized, with divisions based on i. a. different heat (bottom) sources and working conditions[6], to compare them from viewpoint of their utilization in residential buildings. Important factors utilized in description of the heat pumps performance will be presented. Different heat distribution systems which can work in co-operation with heat pumps in buildings will be described, with information about their main components. Subsequently, overall description of transcritical heat pump operation will be provided, together with pros and cons of this heat pump in comparison with conventional one.

2.1. Heat pump - work principle and utilized refrigerants

In general, heat pumps can be used either for heating or cooling purposes, though definitely more often they are used for heating. In this mode, heat pump is a device designed to transport thermal energy in a direction opposite to spontaneous heatflow, i. e. it absorbs the heat from a cold environment and moves it to a warmer one. The name „pump" derives from the analogy to traditional pumps, which can transport various fluids, usually water, although the heat pump construction is more complicated.

Basic, conventional (i. e. subcritical) heat pump consists of four main components (compressor, condenser, expansion valve and evaporator), working together in a refrigerating cycle (early developed by Carl von Linde). All parts are connected together via insulated pipes, inside which the *refrigerant* flows. This is a fluid, which plays role of a medium for transporting heat in the heat pump system, which is schematically presented on the figure 2.2.

A refrigerant is a uniform substance (or a mixture of few), usually a fluid, which is used in heat pump and refrigeration cycles. It normally undergoes phase transitions during the cycle (from liquid to gas state and back again). **Excellent refrigerant** [12, 36] should be characterized by excellent thermodynamic properties (e. g. normal boiling point below 0°C or appropriate for application, relatively large heat of evaporation) and do not result in corrosion of the installation pipes and other mechanical components. Its required flow rate per unit of cooling or heating provided, in refrigeration (or air conditioning or heat pump) system should be as low as possible in order to minimize the charging quantity and the size of compressor. It should be safe for the people and environment (in case of toxicity and flammability), chemically stable in operating conditions and do not cause both ozone depletion and global warming. It should be easily detectable in case of leakage.

[6] Dependent on properties of refrigerant used in heat pump installation, one can distinguish heat pumps working in subcritical or transcritical conditions.

Moreover, perfect refrigerant should be compatible with common construction materials, relatively inexpensive to produce and easy to recycle or destruct after utility.

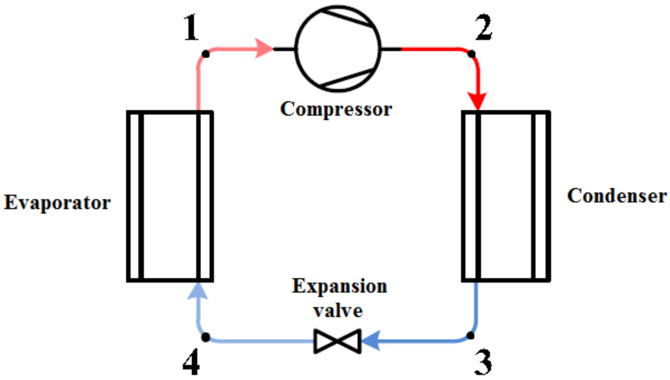

Figure 2.2. Scheme of the basic heat pump circuit [9]; each number among 1 ÷ 4 represents different states of refrigerant circulating in the system[7]; 1 - low pressure vapour, 2 - high pressure vapour, 3 - high pressure liquid, 4 - low pressure liquid

Among **widespread refrigerants** used nowadays, we can distinguish [27, 31]:

- CFC (Chlorofluorocarbons) and HCFC (Hydrochlorofluorocarbons) - also known as *freons*, consist of a group of synthetic working fluids, presently displaced by HFCs in developed countries due to high *ODP* (Ozone Depletion Potential) and *GWP* (Global Warming Potential), although freons are non-toxic and non-flammable. The main ingredient resulting in high ODP of freons was chlorine, which is not present in HFCs any more. The most popular CFC was R12 (currently displaced by R134a - tetrafluoroethane) and one of the most common HCFC was R22. In nowadays some people name as freons also some HFCs which replaced old CFC and HCFC;
- HFC (Hydrofluorocarbons) - these synthetic working fluids are the most common refrigerants in heat pump installations over the world. The most popular HFCs are: R407C[8], R410A[9] and R134a. They are characterized by $\mathrm{ODP} = 0$ since they do not contain fluorine, however due to relatively high GWP they are gradually displaced by natural working fluids;
- Natural working fluids - both organic and inorganic. Common organic natural fluids utilized in heat pumps are hydrocarbons (e. g. R600 - butane, R600a - isobutane, R1150 - ethylene), and important inorganic refrigerants are R717 (ammonia) and R744 (carbon dioxide), with still rising popularity. Natural working fluids seem to be the best environmental friendly choice, mainly due to $\mathrm{ODP} = 0$ and $\mathrm{GWP} \approx 0$.

First, one should know that there is no refrigerant, which is characterized by all the properties of perfect refrigerant described above. Therefore a selection of the suitable working fluid for certain

[7] Each state of refrigerant in the heat pump cycle is usually heterogeneous, e. g. before compressor (point 1 on fig. 2.2) there still may be a liquid part of working fluid, what can be destructive for compressor work. Hence, there are undertaking various actions to avoid it.

[8] Zeotropic blend of R32, R125 and R134a, mixed in % by mass of 23/25/52 [27].

[9] Zeotropic blend of R32 and R125, 50/50 [27].

case is always individual choice. In opinion of the European Heat Pump Association, most commonly used refrigerants are placed in the table 2.1.

Table 2.1. Most commonly used refrigerants in few different types of heat pumps [14]

Heat pump type	Refrigerant
Air-air	R-410a, R-407c
Air-water	R-134a, R-407c, R-410a, R-290, R-744
Exhaust air	R-134a, R-290
Brine water	R-134a, R-407c, R-404a, R-410a

In order to try to indicate which refrigerant could be the most appropriate for utilization in heat pump installation in modern residential buildings, a comparison and closer insight into few common used working fluids (i. e. R410A, R134a, R600, R717) has to be made. In addition, R744 will be taken into account, as its popularity is still rising.

There is no doubt that physical properties of refrigerant have big influence on the system operation. Comparison of them for considered working fluids is presented in the table 2.2.

Table 2.2. Physical properties of chosen refrigerants [12]

	Triple point Temp. °C	Boiling point Temp. °C	Critical temp °C	Critical pressure MPa	Critical density kg/m^3	Min. temp °C	Max. temp °C	Max Pressure MPa	Max density kg/m^3
R-134a	-103.30	-26.17	101.06	4.059	511.9	-103.3	180	70.0	1592
R-410A	-------	-------	70.17	4.770	551.9	-------	-------	-------	-------
Butane	-138.29	-0.54	152.01	3.796	227.8	-138.3	226.9	70.0	735.3
NH_3	-77.66	-33.33	132.25	11.333	225.0	-77.7	426.9	1000.0	732.9
CO_2	-56.57	-78.40	31.06	7.384	466.5	-56.6	167.0	40.0	1223

Among considered refrigerants, two belong to HFC group, i. e. **R134a** and **R410A**. They were both introduced to replace more environmentally harmful CFCs - R134a to replace R12 and R410A to replace R22. Main difference between them is that R410A is a zeotropic blend[10] of two other refrigerants and R134a is homogeneous working fluid. Both fluids are characterized by relatively good thermophysical properties, non-toxicity and non-flammability. A noticeable difference between them lays in critical temperature value - for R134a t_{crit} is clearly higher than for R410A (tab. 2.2), for similar critical pressures. That indicates that vapour density at certain (vaporization or condensation[11]) temperature is lower for R134a than for R410A, e. g. for 0°C[12] ρ_v is around two times lower [27]. In connection with similar specific enthalpy of evaporation (heat rejection[13]) for

[10] Temperature glide of R410A is very small - 0,2 K [78], so R410A can be assumed as homogeneous fluid in some cases.
[11] Dependent on whether a cooling or heating device is considered.
[12] On ITS-90 scale - International Temperature Scale of 1990 is an equipment calibration standard for making measurements on the Kelvin and Celsius temperature scales.
[13] In terms of heating devices, such as the heat pump.

both refrigerants, to achieve similar (theoretically) **capacity** of evaporation[14] (condensation), the mass flow rate (or rather **refrigerant charge**[15]) of R134a should be around two times higher than the one of R410A. In other words, **VRC[16]** (VHC) is around two times lower for R134a than for R410A. Looking forward, it would lead to higher compressor size in case of R134a, what is undesirable. Apart from this, relevant disadvantage of both R134a and R410A are relatively high GWP factors, what can be seen in the table 2.3.

Table 2.3. Global Warming Potential for considered refrigerants [12, 38, 39]

Name of refrigerant	Atmospheric lifetime	GWP 100[17]
R134a	13,8 years	1430
R410A	17 years	2088
R600	12 ± 3 years	4
R717	max. one week	0
R744	uncertain (29300 ÷ 36100 years)	1

Another refrigerant taken into account in conducting analysis is **butane** (R600), which belongs to organic hydrocarbons (HC). R600 is a natural refrigerant, which is environmental friendly - its $\mathrm{ODP} = 0$ and $\mathrm{GWP} = 4$ (tab. 2.3). Butane is characterized by relatively low critical pressure and very high critical temperature (tab. 2.2). Compared to previously described HFCs, R600 has few times lower vapour density than R134a, e. g. for vaporization temperature of 0ºC, $\rho_{v,R600} \approx 0{,}18\rho_{v,R134a}$ [27]. On the other hand, the specific heat of vaporization for butane at 0ºC is equal to $\Delta h_{evap,R600} \approx 385\,\frac{kJ}{kg}$ and for R134a, $\Delta h_{evap,R134a} \approx 199\,\frac{kJ}{kg}$ [27]. As a result, VRC for R600 at 0ºC will be two and half times lower than for R134a, and five times lower than for R410A. It indicates that to achieve similar capacity of evaporation, it is necessary to charge butane inside the circuit in amount approx. two and half times higher than for R134a, or around five times higher than for R410A. Thus the mass flow rate of refrigerant would have to be clearly increased, what would result in bigger components of the heat pump, including compressor. Apart from this issue, although R600 is environmental friendly and non-toxic working fluid, it is highly flammable, what can be noticed in the table 2.4.

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[14] Capacity of evaporation [kW], equal to mass flow rate of refrigerant through evaporator $[\frac{kg}{s}]$ multiplied by Δh_{evap} $[\frac{kJ}{kg}]$. It can be related with condensation if heating device is considered, although capacity of evaporation and capacity of condensation have not the same values.

[15] Total weight of refrigerant in considered refrigeration (heating) system [kg].

[16] Volumetric Refrigeration (Heating) Capacity $[\frac{kJ}{m^3}]$ is calculated as the multiplication of vapour density and specific enthalpy of evaporation (heat rejection), dependent on whether a refrigerating or heating device is considered. $\mathrm{VRC} \neq \mathrm{VHC}$, though it has similar tendency.

[17] GWP calculated over time interval of 100 years.

Next working fluid which will be considered is R717. As can be easily seen in tab. 2.2, **ammonia** has very broad range of temperatures in which it can be utilized, therefore it is very popular refrigerant in many industrial refrigeration systems. In addition, its popularity results from other features, like higher **efficiency**[18], lower environmental impact, relatively high availability and low price compared to analyzed HFCs, i. e. R134a, R410A (tab. 2.4). Moreover, NH_3 has $GWP = 0$ (tab. 2.3). However, these advantages are occupied by not less important drawbacks, among which the most dangerous are its high flammability and toxicity, especially in higher concentrations in the air. Although it has a specific smell and can be easily detected (in case of eventual leakage), ammonia seems to be not really suitable refrigerant for the residential building and people living there. In case of critical temperature, t_{crit} for R717 is around 20ºC lower than for butane but its critical pressure is much more higher (tab. 2.2). Vapour density in temperature of 0ºC is few times lower compared to R134a, although it is higher than for R600 ($\rho_{v,R717} \approx 0{,}25\rho_{v,R134a} \approx 1{,}4\rho_{v,R600}$ [27]). In turn, specific enthalpy of evaporation is much more higher for ammonia than for any other considered working fluid, i. e. $\Delta h_{evap,R717} \approx 1260\,\frac{kJ}{kg}$ for 0ºC [27]. Therefore, despite of lower vapour density than compared HFCs, due to very high heat of vaporization of ammonia, to achieve similar capacity of evaporation there would have to be approx. one and half times lower mass flow rate of R717 than R134a in the circuit, but in comparison with R410A - the mass flow rate of ammonia would have to be higher. Apart from NH_3 drawbacks mentioned above in this paragraph, its another disadvantage is that in presence of moisture it tends to become corrosive to copper and other non-ferrous materials, which is often applied material used for pipes in refrigerant systems.

Table 2.4. Functional aspects of R744, R600 and R717 in comparison with considered HFCs (R134a, R410A). Different numbers represent relations with HCFs, as follows: 1 - better than for HFCs; 0 - similar to HFCs; -0,5 - little worse than HFCs; -1 - worse than for HFCs [37]

Name of the property	**R744**	**R600**	**R717**
Capacity of evaporation (condensation); VRC (VHC)	1	-0,5	0
Efficiency	0	1	1
Pressure[19]	-1	0	0
Environmental impact	1	1	1
Flammability	1	-1	-1

[18] In terms of specific enthalpy of evaporation (heat rejection) [$\frac{kJ}{kg}$].

[19] Pressure levels in which devices utilizing refrigeration circuit (i. e. refrigerator, heat pump, air conditioning) work. Higher pressures have more destructive influence on the system operation and may increase operation costs, hence they are treated as a drawback.

Table 2.4. Continuation

Name of the property	R744	R600	R717
Toxicity	-0,5[20]	1	-1
Availability of refrigerant	-0,5	1	1
Cost of refrigerant	1	1	1
Availability of components	-0,5	1	1
Average cost of the system	-1	-0,5	-1

The last substance taken into account in conducted comparison, used in refrigeration systems (and currently more and more often in heat pump systems for heating) since mid-nineteenth century [40] is **carbon dioxide**. Its environmental impact is very low in the short scale - $\mathrm{ODP}_{\mathrm{CO_2}} = 0$ and $\mathrm{GWP}_{\mathrm{CO_2}} = 1$ (tab. 2.3), although $\mathrm{CO_2}$ in the atmosphere affects global warming in a little different way. The problem is that its atmospheric lifetime is very long (tab. 2.3) and all $\mathrm{CO_2}$ emitted to the atmosphere since pre-industrial times (since 1750 [41]) is still cumulating and do not disappear, what is leading to still rising average global air temperature. Today $\mathrm{CO_2}$ is treated as the most important greenhouse gas and various methods to lower its further emissions to the atmosphere are currently applying, e. g. CCS[21]. On the other hand, $\mathrm{CO_2}$ utilized in HVAC-R[22] systems is closed inside installation pipes and it is not allowed to emit to the atmosphere (eventual leakages are undesirable). Furthermore, $\mathrm{CO_2}$ can be directly stored (e. g. in the ground) after disposal of any HVAC-R device using R744 as refrigerant. Thus, according to the author of this thesis, considerations about the total impact of atmospheric carbon dioxide on global warming in aspect of systems using refrigeration circuit is redundant. Apart from this, R744 is chemically neutral, non-flammable and non-toxic (tab. 2.4). What is more important, carbon dioxide has rather unusual thermodynamic properties in comparison with other considered refrigerants in this chapter. Its critical temperature is very low - $\mathrm{t_{crit}} \approx 31^{\mathrm{o}}\mathrm{C}$ (tab. 2.2) and critical pressure is relatively high[23] $\mathrm{p_{crit}} \approx 7{,}38\ \mathrm{MPa}$. It results in high vapour density at 0ºC, which is approx. 28 times higher than vapour density of R717 and seven times higher than vapour density of R134a [27]. Meanwhile, the heat of vaporization for $\mathrm{CO_2}$ is similar to analyzed HFCs, i. e. $\Delta \mathrm{h_{evap,R744}} \approx 231\,\frac{\mathrm{kJ}}{\mathrm{kg}}$ for 0ºC [27]. As a result, one of the most important advantages of R744 appears, i. e. its high VRC (VHC). It means that to achieve similar capacity of evaporation, mass flow rate of $\mathrm{CO_2}$ in the system would have to be at least few times lower

[20] R744 is non-toxic in direct meaning of this word but it is asphyxiant.
[21] Carbon Capture and Storage is the process of capturing waste $\mathrm{CO_2}$ (mainly from industrial processes), transporting it to a storage site and depositing it where it will not enter the atmosphere, normally in underground geological formations [42].
[22] Heating, Ventilation, Air Conditioning and Refrigeration.
[23] In overall, reasonable operating pressures are preferred in order to keep costs at minimum.

compared with other refrigerants. It allows to minimize the size of all the components (including compressor) of the heat pump or another system utilizing carbon dioxide as refrigerant. However, significant drawback of CO_2 is a disability of its utilization in conventional heat pumps - due to low critical CO_2 temperature it cannot be used in conventional heat pumps, due to its very low maximum condensation temperature (caused by its low critical temperature) which is too low from viewpoint of needs of heating systems applied in residential buildings. Comparison of temperatures and pressures achieved by different refrigerants can be noticed on the figure 2.3

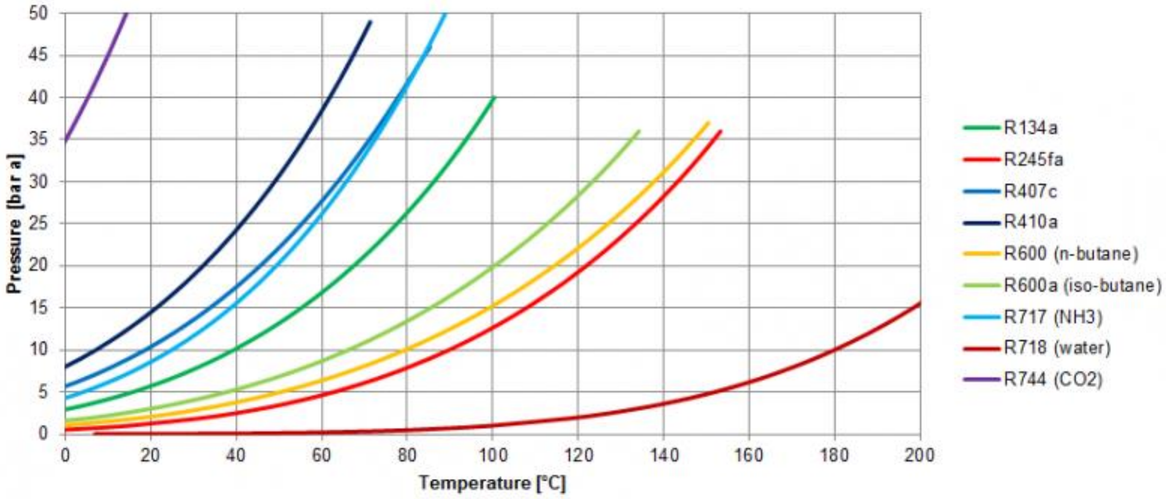


Figure 2.3. Temperatures in a function of pressures for different refrigerants [50]

To sum up, after review of few common refrigerants, author of this work believes that the most appropriate refrigerant for utility in heat pump system in modern, well - insulated residential buildings is carbon dioxide (R744), due to several reasons:

- huge reduction in refrigeration charge in case of the heat pump system, thanks to the high vapour density of R744 and moderate specific enthalpy of heat rejection (evaporation) - relatively high VHC, what allows to minimize the size of heat pump installation (including main components and pipes) in the residential building, where space available for placing such systems is usually limited;
- non-flammable nature of CO_2, what is relevant due to people living in the building, spending there even more time than in case of non-residential buildings;
- environmental impact is very low (if to ignore its long atmospheric lifetime and further considerations with global warming);
- low price of refrigerant and, what is more important, low energy embodied in its aqcuisition cycle (as it is a component of the air, it does not have to be recovered from any industrial process), what seems to be ecological.

2.2. Classification of heat pumps. Indicators describing heat pump performance

Heat pumps can be classified in case of **various criteria**, among which the most significant are:

- Dependent on **type of propulsion and principle of operation** (or temperature level of working fluid and way to pressurize it), following heat pump categories can be distinguished [79, 80]:
  - *compression heat pumps* - currently the most common group of heat pumps used for heating and cooling purposes in buildings, mainly due to lower prices and higher efficiencies compared to other types. This type of heat pump is mechanically driven by compressor (supplied by electricity), which pressurizes refrigerant in the system. They can be divided into *subcritical* (conventional) and *transcritical* heat pumps, depended of working parameters (e. g. pressure, temperature) of certain refrigerant;
  - *sorption heat pumps* - they can be divided into *absorption* and *adsorption* heat pumps. In absorption process, certain fluid (called absorbent) absorbs (i. e. pearmeats or dissolves) another fluid or gas (absorbate), meanhwile in adsorption process particles of special fluid (adsorbate) cling to a solid surface (adsorbent) in adhesion process. Compression process in sorption heat pumps is carried out thermally. They are often used in large industrial plants to increase the energy potential of waste heat;
  - *thermoelectric heat pumps* - they are driven electrically and their operation relies on Peltier's thermoelectric effect, treating that when a current flows through a junction between two (different) semi-conductors, heat may be generated (or removed) from the junction. As a result, both heating and cooling effects appear simultaneously, although the utility of this kind of heat pumps in heating and cooling systems in buildings in the nearest future will be very small or none. Thermoelectric heat pumps are usually applied when there is a need for continuous and efficient heat removal from small objects;
  - *Vuilleumier's heat pumps* - this type of heat pump employs the cycling of various volume devices. An example can be an inner cylinder, and an outer cylinder surrounding the inner one. An annular space is created between walls of the inner and outer cylinder, where heat exchangers are placed on few different levels [28]. Utilized working fluid inside the cylinder can be either high pressure helium or hydrogen gas. Unfortunately, this type of heat pumps is characterized by low efficiency (e. g. due to presence of dead volumes between cylinders and heat exchangers). Another problem of this heat pumps is price - they are expensive to produce since they require many steps to manufacture. Due to these reasons, they have not been applied in heating and cooling systems of buildings in wider scale.

- Dependent on **working conditions of the heat pump**, succeeding heat pumps (utilized in residential buildings) can be differentiated [32]:
  - *subcritical* (*conventional*) *heat pumps* - working in subcritical conditions, i. e. when thermodynamic state of utilized refrigerant circulating in the heat pump cycle is always below the critical point. There are many refrigerants which can be used in subcritical heat pumps, i. a. R134a, R717, R600;
  - *transcritical heat pumps* - working in transcritical conditions, i. e. when thermodynamic state of applied refrigerant is sometimes subcritical (below the critical point) and sometimes supercritical (over the critical point). The only transcritical heat pumps currently used in residential buildings are working with R744 (carbon dioxide) as refrigerant. This technology was developed by Norwegian professor Gustav Lorentzen and his team in 1988 - 1991 [49].
- Depended of **type of the bottom heat source**, heat pumps can be divided into [32]:
  - (*ambient*) *air heat pumps*, where the heat source is normally an ambient air. Advantages of the air as a heat source are its common availability, low maintenance costs, low investment cost of the heat pump (due to simple construction of the bottom heat source - no need to drilling or preparation of bottom heat source installation pipes for working fluid circulation) and easy implementation in case of refurbishment of old heating systems in buildings. However, ambient air has also many drawbacks, especially low heat transfer coefficient (hence air is good insulator), what forces application of bigger heat exchange surfaces (thus bigger heat exchangers). Another disadvantage is instability of air temperature during the day and year, what results in worse performance of the heat pump - especially during the winter, when heat source temperature is low, COP[24] decreases and more electricity need to be supplied to the heat pump. In addition, below ambient air temperature of -20ºC ÷ -15ºC [80, 82] heat pump needs to work in bivalent mode with additional heat supply system (heat pump alone is then not able to cover 100% of the building peak heating load), what generates additional operating costs. Thus air source heat pumps are more suitable to regions where the climate is moderate (with absence of extremely low ambient temperatures). Further disadvantage of air - based heat pumps is relatively loud operation of fans placed inside its external (and internal, in case of air to air heat pumps) module;
  - *water heat pumps*, where the heat source is water, typically obtained from a well, lake, river, sea, ground or an artificial reservoir, specially prepared for the heat pump installation. Water - based heat pumps are much more efficient than air heat pumps, mainly due to higher and much more stable temperature of the water throughout the year, especially in case of ground water

[24] In terms of $\mathrm{COP_{heating}}$

(temperature varies between 8 ÷ 10 °C [81]) or sea / lake water, with little less stable temperature values (usually 7 ÷ 13 °C [80]). However, this big advantage is occupies by many drawbacks, like high investment cost (more expensive bottom source installation, especially for ground water usage where two deep wells have to be drilled), high maintenance costs (water is usually polluted and have to be filtrated, and elements of installation pipes transporting working fluid[25] have to be changed periodically, due to corrosion caused by constant contact with water) and limited availability (not all places have direct access to water resources and not every water source is suitable for usage - water needs to have certain physiochemical properties to allow for bottom heat source utilization);

- *ground heat pumps*, where the heat source is ground, e. g. a soil or bedrock. Heat is gained in direct heat transfer between the ground and fluid (normally a glycol - based fluid with some additives) circulating in the bottom heat source installation pipes. In comparison with water heat pumps, ground heat pumps are characterized by slightly lower efficiency (heat transfer between ground and working fluid is less efficient, though temperature of the ground is similar to most of surface water sources, i. e. 7 ÷ 13°C [80] in case of vertical probes utility or 0 ÷ 8°C [81] in case of horizontal collectors) but higher availability and reliability, because closed bottom source installation is insensitive on pollution or changes of hydrogeological conditions. Although temperature of the ground is similar to most of surface water sources (example ground temperature variations in a function of depth are depicted on the figure 2.4), heat transfer between ground and working fluid is less efficient and bottom source installation of the ground heat pump need to have much more bigger heat exchange surface. As a result, ground bottom heat source installations are more expensive compared to water - based or air - based heat sources, especially when vertical probes are utilized, drilled even up to 200 m below the ground surface [83], although the reliability of such source is the highest and they require relatively small area, therefore they can be applied even for small estates. Sometimes more but shallower boreholes are drilled, what can be cheaper and may be executed without necessity of having various permissions;
- *solar heat pumps*, where solar collectors are used to warm up water or air (in indirect system), which afterwards releases thermal energy to refrigerant in evaporator of the heat pump. In direct system, a refrigerant is directly warming up by solar radiation incident on the solar collector surface. Performance of solar heat pumps is similar to air - based heat pumps (with little higher

[25] In water - based heat pumps, water is often utilized as a fluid transporting the heat to evaporator, due to properly high temperature of water in the source, what keeps the water transporting heat not frozen during all the year.

efficiency), due to relatively unstable temperature and solar radiation conditions throughout the year. Solar heat pumps are little more expensive than air heat pumps, due to utilization of solar collectors as a part of bottom heat source installation;

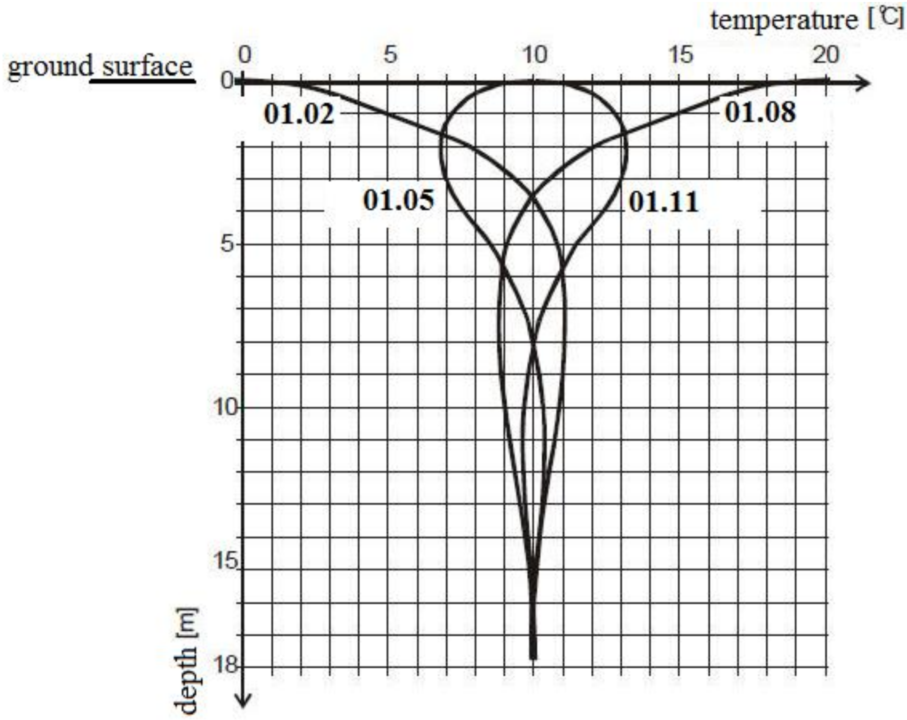


Figure 2.4. Average ground temperature variations in Poland, dependent on the season [86]

- *other heat pumps*, utilizing sources of waste heat, e. g. ventilation exhaust air or waste water.

- Depended of a **relation between the bottom heat source type and the warming medium type** in the building heat distribution system, one can distinguish:
  - *ground to water* heat pumps;
  - *ground to air* heat pumps;
  - *water to water* heat pumps;
  - *air to water* heat pumps;
  - *air to air* heat pumps;
  - *water to air* heat pumps.

- With reference to the previous criterion, due to **different way of obtaining heat and its transfer** to the building inner space, the following division of heat pumps can be made [81]:
  - *brine / water* heat pump - brine (or glycol - based fluid) is circulating in the ground heat exchanger, uptaking heat from the ground and transporting it to the evaporator where the heat is transferred to refrigerant in the heat pump circuit. Afterwards, refrigerant releases heat in condenser to water - based heat distribution system in the building;

- *direct evaporation / water* heat pump - the ground heat exchanger is simultaneously an evaporator of the heat pump, where refrigerant circulates and evaporates. Next, the heat is transferred from the refrigerant to water based heating system in the building. Thanks to elimination of glycol - based fluid (or other intermedium), such a system gains higher efficiency;
- *direct evaporation / direct condensation* heat pump - the ground heat exchanger is simultaneously an evaporator of the heat pump and the heat distribution system in building is a condenser. Both in ground exchanger and in heating system there is no intermedium in case of heat transfer, and due to this fact an efficiency of such heat pump is the highest, although this heat pump has another disadvantages[26];
- *water / water* heat pump - the refrigerant in evaporator of the heat pump is supplied by heat transferred from water, typically drawn from a well, river or lake. Next, working fluid releases heat in condenser to water - based heat distribution system in building;
- *air / water* heat pump - evaporator of the heat pump is supplied by a stream of the ambient air. Later the heat is transferred in condenser from refrigerant to water - based heating system;
- *air / air* heat pump - evaporator is supplied by heat from ambient air. Afterwards, refrigerant releases heat in condenser and thus warms the air stream blown to the inner space by the fan;
- *water / air* heat pump - refrigerant is warmed by heat from the water in evaporator, and next it releases the heat in condenser to the air blown into the inner space.

• Dependent on a **type of heat pump operation in the building heating system**, one can distinguish [3, 84]:

- *monovalent* heat pumps - heat pump working in this mode is the only heating device in the building heating system, therefore covering 100% of annual heating demand of the building. This kind of heat pump installation is mostly recommended to ground - based or water - based heat pumps, due to their higher stability and reliability throughout the year. In case of air heat pumps (due to relatively big variety of ambient air temperatures during the heating season) bivalent modes are recommended, because in this way heat pump will not overestimate the heat demand;
- *bivalent - alternative* heat pumps - before certain outdoor temperature value[27] is reached, heat pump is the only heating device working in the building heating system (similarly like in monovalent mode). After reaching this temperature, heat pump stops on a current level of power and further part of heating demand is covered by additional heating device (normally direct electric heating, gas or oil boiler), what can be noticed on the figure 2.5;

[26] The biggest problems are lower reliability compared with brine / water heat pumps and harder control of the heat pump capacity.
[27] This temperature value is the so-called bivalent point temperature

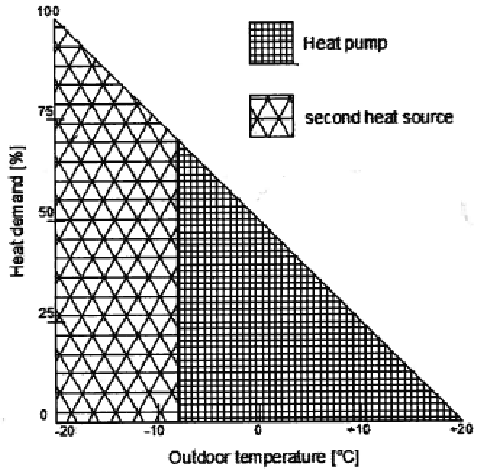


Figure 2.5. Bivalent - alternative mode of heat pump operation [3]

- *bivalent - parallel* heat pumps - in this mode heat pump works in the building heating system in co-operation with additional heating device. Depended of the outdoor temperature (and thus certain heating demand), after reaching its certain value the additional heating device is turning on by the heat pump regulator and afterwards both devices are working together, trying to cover all heating needs. It is presented schematically on the figure 2.6;

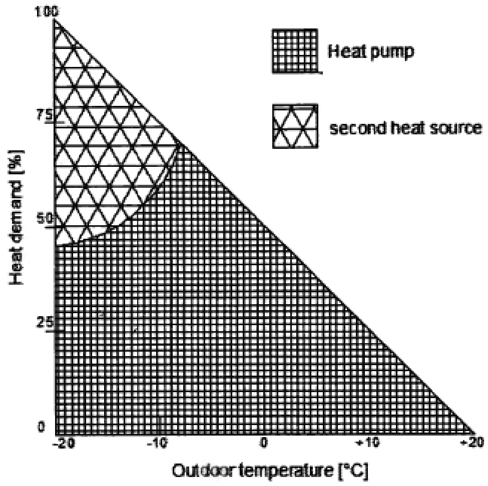


Figure 2.6. Bivalent - parallel mode of heat pump operation [3]

- *bivalent - partially parallel* heat pumps - this mode is similar to bivalent - parallel heat pump work but here after reaching bivalent point temperature, during further lowering of the outdoor temperature the disconnection of heat pump is increasing and after some time only additional heating device works.

There are many different **indicators describing heat pump performance**, among which the most common used are:

- *COP - Coefficient of Performance* is a ratio of heating or cooling provided to work required. In case of compression heat pumps (such as the heat pump which will be described and analyzed later in this thesis) the required work equals to electricity power supplied to the compressor (and eventually to other facilities working in the heat pump installation, e. g. circulation pumps). Instead of just converting work to heat like other heating systems in buildings, heat pumps pump use additional heat from renewables, thus $\mathrm{COP} > 1$ (100%). In overall, two different COP can be distinguished, i. e. $\mathrm{COP_{cooling}}$ (when it is wanted to check how well the heat pump cools) and $\mathrm{COP_{heating}}$ (to check how well the heat pump heats). Mathematically, they can be calculated using the following equations:

$$\mathrm{COP_{cooling}} = \frac{Q_C}{Q_H - Q_C} = \frac{|Q_C|}{W}, \tag{2.1}$$

$$\mathrm{COP_{heating}} = \frac{|Q_H|}{W} = \frac{|Q_C + W|}{W} = \mathrm{COP_{cooling}} + 1\,, \tag{2.2}$$

where:

Q_C - heat removed from the cold reservoir [W];

Q_H - heat supplied to the hot reservoir [W];

W - total electric power supplied to the heat pump [W].

However, one should know that COP is highly dependent on operating conditions, especially on temperature of the bottom heat source and so-called temperature lift[28] (what is shown on the figure 2.7). Therefore its value is changing almost all the time throughout the year and it is not appropriate indicator which treats about seasonal heat pump efficiency. COP values for various compression heat pumps amount to [3, 83]:

+ air to air heat pumps: $\mathrm{COP} = 1{,}5 \div 3{,}9$;

+ water to air heat pumps: $\mathrm{COP} = 2{,}0 \div 4{,}0$;

+ water to water heat pumps: $\mathrm{COP} = 3{,}0 \div 6{,}0$;

+ ground to water heat pumps: $\mathrm{COP} = 3{,}0 \div 5{,}0$.

For a heat pump operating at **maximum** theoretical efficiency (i. e. Carnot efficiency), it can be proved that:

$$\frac{Q_C}{T_C} = \frac{Q_H}{T_H} \rightarrow Q_C = \frac{Q_H \cdot T_C}{T_H}, \tag{2.3}$$

where:

T_C - temperature of the cold reservoir [K];

T_H - temperature of the hot reservoir [K].

Therefore, COP at maximum theoretical efficiency can be equal to:

[28] A temperature difference between hot and cold reservoir, i. e. $\Delta T = T_H - T_C$. Better performance of the heat pump can be achieved if temperature lift is lower.

$$\mathrm{COP_{cooling,max}} = \frac{T_C}{T_H - T_C} \tag{2.4}$$

$$\mathrm{COP_{heating,max}} = \frac{T_H}{T_H - T_C} = \mathrm{COP_{cooling,max}} + 1 \tag{2.5}$$

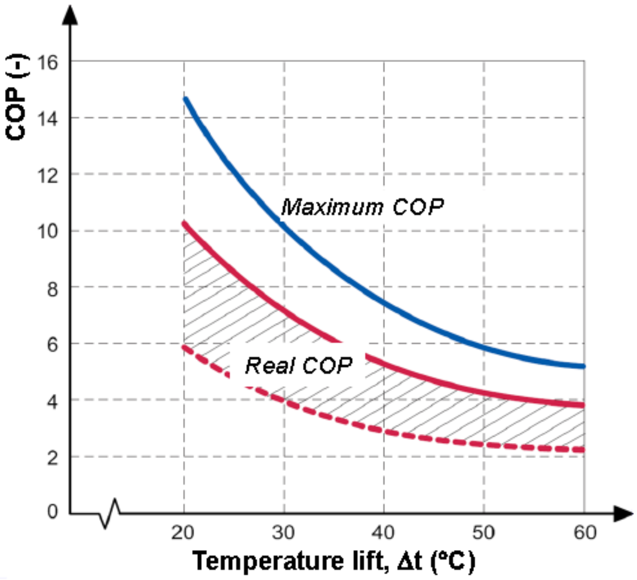


Figure 2.7. Coefficient of Performance as a function of temperature lift of the heat pump. Maximum COP values are referred to maximum (temperature) efficiency of the heat pump cycle [32]

- *Energy savings* (*E*) - savings expressed in % or kWh, depended of the COP value of the heat pump. In case of heating, *E* can be calculated as follows:

$$\mathrm{COP_{heating}} \cong \mathrm{COP} = \frac{Q_H}{Q_H - Q_C} \rightarrow Q_C = Q_H - \frac{Q_H}{\mathrm{COP}} = Q_H \cdot (1 - \frac{1}{\mathrm{COP}}) \tag{2.6}$$

$$\frac{Q_C}{Q_H} = \mathbf{1} - \frac{\mathbf{1}}{\mathbf{COP}} = \mathbf{E} \tag{2.7}$$

If COP is higher, it is equate to lower operating costs and simultaneously to higher energy savings. Presentation of dependency between E and COP can be noticed on the figure 2.8.

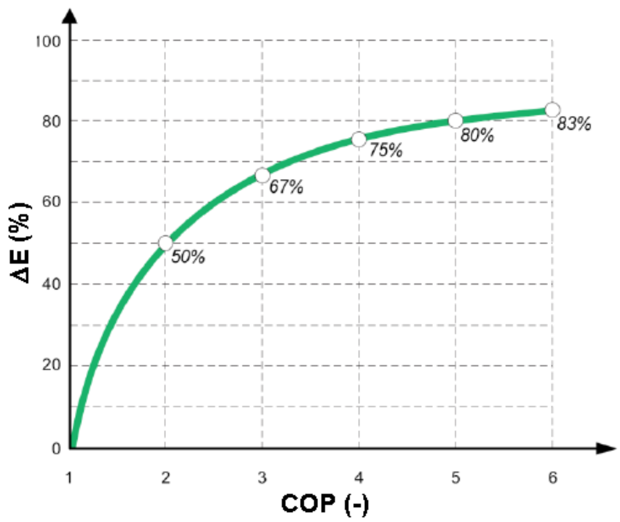


Figure 2.8. Energy savings as a function of COP of the heat pump [32]

- *SPF - Seasonal Performance Factor* is a measure of the heat pump performance over a period of time, e. g. over a full year period. It is defined as:

$$\mathrm{SPF} = \frac{\mathrm{Q_s}}{\mathrm{E}}, \tag{2.8}$$

  where:

  $\mathrm{Q_s}$ - total heat delivered from heat pump in considered period [kWh];

  E_T - total electrical energy supplied to heat pump in considered period [kWh].

  In practice, SPF is used i. a. to calculate the renewable portion of total heat generated by the heat pump, when calculating payments [44]. In United Kingdom, all heat pumps (to be eligible) must have a minimum SPF value of 2,5 [44].

- *Coverage factors* - there exist **two** basic factors which can describe the heat pump utility in the building heating system:

  - ΔP - *power coverage factor* - it means a ratio between peak heating load of the building covered by the heat pump ($\mathrm{P_{HP}}$ [kW]) and total peak heating load of the building ($\mathrm{P_T}$ [kW]):

$$\Delta \mathrm{P} = \frac{\mathrm{P_{HP}}}{\mathrm{P_T}} \tag{2.9}$$

    It is expressed in % and its typical values for heat pumps utilized in residential buildings are usually equal to $40 \div 70\%$ [1];

  - ΔQ - *energy coverage factor* - it is a ratio (expressed in %) between annual heating demand covered by the heat pump ($\mathrm{Q_{HP}}$ [kWh]) and total annual heating demand of the building ($\mathrm{Q_T}$ [kWh]):

$$\Delta \mathrm{Q} = \frac{\mathrm{Q_{HP}}}{\mathrm{Q_T}} \tag{2.10}$$

    Energy coverage factor is dependent on many other agents, including [9]:

    + climate conditions at the building site;
    + building usage pattern;
    + heat pump performance at different operating conditions;
    + operational limitations of the heat pump;
    + type of the bottom heat source ($\Delta \mathrm{Q}$ is normally higher for water and ground based bottom sources than for utilization of ambient air as a heat source) [32];
    + size of investment and operating costs.

    For typical values of power coverage factor, energy coverage factor normally equals to $60 \div 95\%$ [1].

- *Relative Energy Savings* (ΔE) - they express energy savings (in % or kWh) of considered heat pump system (with defined SPF) compared to alternative heating system with certain efficiency. Mathematically:

$$\Delta \mathrm{E} = \frac{\Delta \mathrm{Q}}{\eta} - \frac{\Delta \mathrm{Q}}{\mathrm{SPF}}, \tag{2.11}$$

where:

ΔQ - energy coverage factor, characterizing fraction of annual heat demand covered by the heat pump [%];

η - efficiency of alternative heating system [%].

On the figure 2.9 the relative energy savings are presented as a function of heat pump SPF, for three different values of energy coverage factor (i. e. 70%, 90%, 100%). As an alternative heating system, direct electric heating system (with efficiency of 100%) is assumed. Moreover, ΔE is strictly connected to energy coverage factor (equation 2.11). Graphically it is presented on the figure 2.10, for few different SPF values.

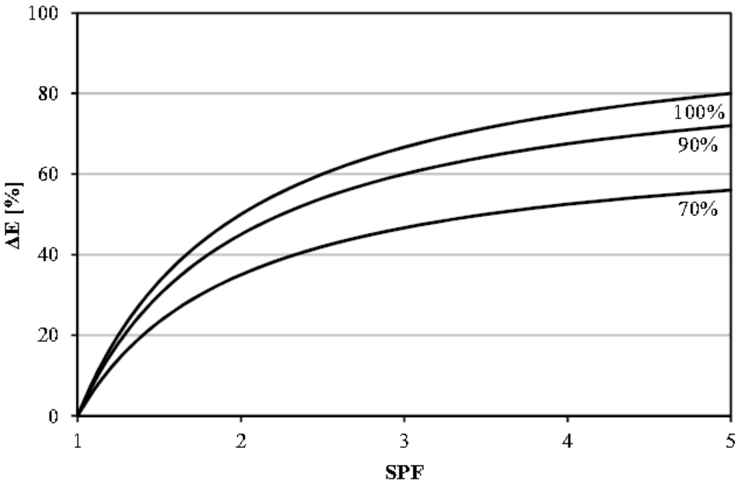


Figure 2.9. Relative energy savings as a function of SPF for the heat pump system covering 70%, 90% and 100% of annual heating demand of the building [9]

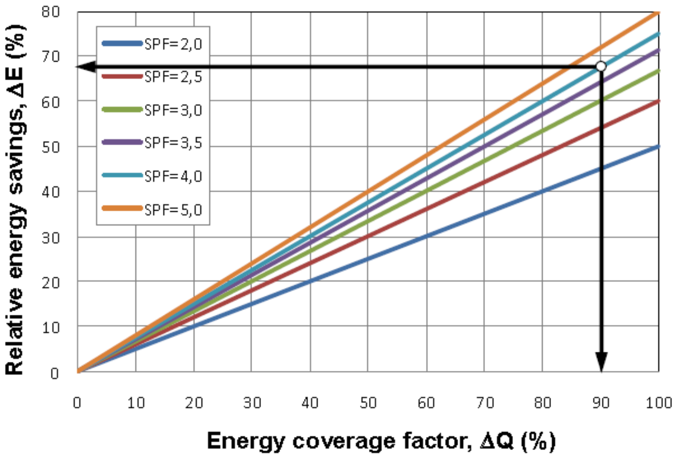


Figure 2.10. Relative energy savings as a function of ΔQ for six different SPF values [32]

- *Equivalent operating time* (τ) - it is defined as the time (in hours) during which the heat pump has to run at full capacity in order to cover the yearly heat production of the heat pump [2]. Mathematically [30]:

$$\tau = \frac{E_{HP}}{P_{max}}, \tag{2.12}$$

where:

E_{HP} - total heat delivered from the heat pump during certain time period (normally one year) [kWh];

P_{max} - maximum heating capacity of the heat pump at design conditions [kW].

Equivalent operating time gives some information about how well the heat pump capacity is being utilized. Its low value may (related to one year period) may indicate that heat pump often operates at part load or does not work at all. As the heat pump economy is considered, τ is one of the most important factors taken into account, especially that heat pump is characterized by high investment cost in comparison with alternative heating systems, using fossil fuels or direct electricity. In Norway, where electricity (gained mainly from hydropower and nuclear energy) is really cheap, direct electric heating systems are real competitor for heat pumps. On the figure 2.11 it is shown how simple payback period[29] of the heat pump system is dependent on main factors, compared to direct electric heating system (electricity price is assumed as 0,40 NOK/KWh).

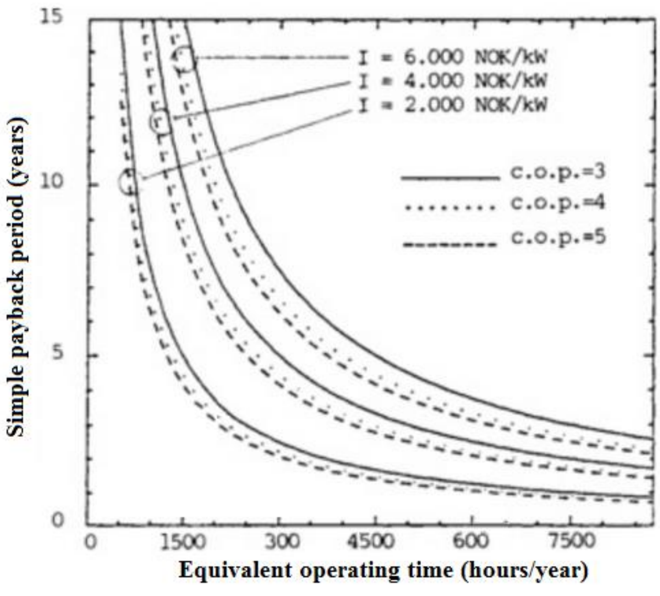


Figure 2.11. Simple payback period of the heat pump system as a function of τ and COP, for different investment costs [2]

Typical values of equivalent operating time for heat pumps in residential buildings are often in the range of 2000 ÷ 4000 hours [30].

[29] The period of time (expressed in years) required to recoup the funds expended in an investment.

2.3. Comparison and evaluation of different heat distribution systems

Nowadays, there are various heat supply systems utilized for heating purposes in residential buildings. Each heat supply system consists of two main parts, i. e. heat production and heat distribution side, including also miscellaneous solutions to improve its efficiency (e. g. heat storage). Since **air to water heat pump** is assumed as a heat production system in **passive** residential building in further considerations contained in this master thesis, presented heat distribution systems are limited to such solutions which can work in co-operation with this type of heat pump, assuming passive character of the building[30]. Thus, **the following heat distribution systems** can be distinguished:

- *Underfloor water heating* - warm (hot) water from thermal storage tank (or directly from heat exchanger of the heat pump) is pumping into copper pipes laid under the floor, placed in certain patterns, controlling the flow of water through each tubing loop by using distribution valves. There exist two kinds of such heating installation, i. e. wet, older type (pipes are embed in a solid form, like thick concrete foundation slab) and dry type (tubes are usually put in air space between joists). In some modern dry installations an additional plywood subfloor can be built - pipes are then laid between two subfloors, with tubing grooves and aluminum heat diffuser plates placed between pipes, which allow for significant reduction of the installation loop length in comparison with traditional dry underfloor heating loops [46]. Main advantages of this heat distribution system are evenly distributed air temperature field inside the building, high reliability and low maintenance costs (if properly installed and tested[31]). Moreover, underfloor heating is very comfortable due to low and comfortable vertical air temperature gradient in heated space (the air is the warmest near the ground, where human body cools faster and cooler in the upper parts of the rooms, what can be seen on the figure 2.12), so-called „warm feet" feeling and lack of convective drafts[32] (which can occur in case of forced air heating systems). Thanks to utility of radiant heat transfer via infrared waves, heat penetrates to objects and surfaces surrounding heated space, afterwards warming the air. As a results, both air temperature and mean radiant temperature of inner surfaces in the building are on appropriate level, thereby allowing to avoid radiant drafts[33]. Another advantages of underfloor heating are flexibility in locating furnitures and equipment where it is desired (all floor area can be utilized, because there is a lack of visible heaters), and relatively higher indoor air quality than in case of compact heaters usage

[30] In passive buildings, low temperature heating is preferred, with water supply temperature values equal to even $20 \div 35^{0}C$ [9, 45].
[31] Pressure testing should be made in order to check a tightness of installation pipes and detect potential leakages.
[32] Convective drafts appear as unpleasant, too fast air movements, resulting in local temperature and / or humidity differences.
[33] Heat radiant drafts exist due to radiation from warm human bodies against cold surfaces or when heat radiation asymmetry is too great.

(convection currents, occuring both in convectors and radiators (though with lower share) can spread much more airborne dust and other pollutants than in case of underfloor heating). However, these all advantages of underfloor heating system are occupied by its certain drawbacks, i. a. high investment cost and limited range of water temperatures (utility of too high water temperatures in underfloor heating is not very healthy and may cause slight radiant draft feeling), although this drawback is less important in case of considered building, due to relatively small heating demand in passive buildings. Furthermore, temperature regulation and steering in underfloor heating is more difficult compared to other heat distribution systems (i. a. due to large thermal inertia of the system and limited range of allowed floor temperatures, not higher than 29 ÷ 33ºC [45] in places of permanent human habitation and less than 35ºC [45] in so-called boundary zones, i. e. near windows and doors). In case of pipes distributing warm water potential problems (though possibility of their occurence is low) are hard to find, because they are placed under floor and they may be expensive to repair (i. e. high maintenance cost if problems occur). Moreover, similarily as other radiant low temperature heating distribution systems, underfloor heating can be slower in warming up the living space compared to forced air heating systems.

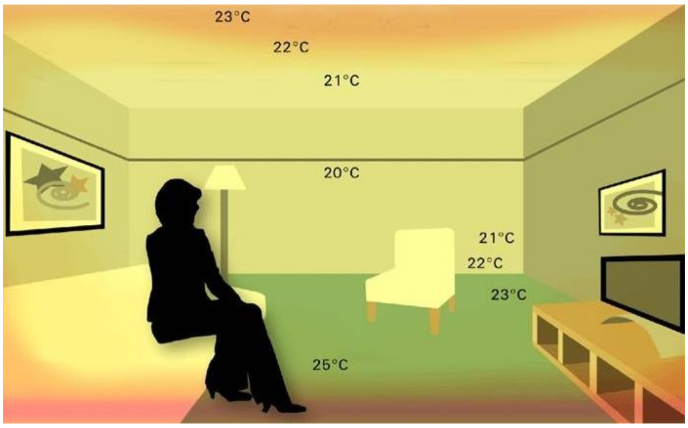


Figure 2.12. Vertical air temperature distribution in heated room, assuming heat gains from people and appliances [85]

- *Radiant water heaters* - compact radiators, though emitting heat similarily like underfloor heating (i. e. via electromagnetic infrared waves), they releases heat to the space pointwise, though simultaneously in all directions with good efficiency (especially in modern radiators, where the surface of heat transfer is increased). Share of convection heat transfer in total heat emission is higher than in case of underfloor heating, although is still relatively small[34]. Recommended temperature values of water supplied low temperature radiators[35] are between 40 ÷ 50ºC [48] (when supply water temperature is 45ºC, then its return temperature is normally around 35ºC [48]). Main advantages of radiators compared to underfloor heating is lower investment cost of

[34] In comparison with traditional convectors.
[35] Utilized in low energy and passive buildings.

installation, shorter period of warming up the inner space (radiators are placed inside heated space and have higher surface temperature, though lower heat transfer area), easier regulation and steering (thanks to much lower thermal intertia of radiators, wider temperature range of allowed water temperatures and compactness of each radiator, what allows for easier steering of each heater separately) and prevention of draughts from windows more effectively (due to higher share of convection heat transfer). Due to easier regulation, radiators can adapt to current indoor heat gains better than floor heating (their aluminum[36] surface temperature can both warm up and cool off quickly). Radiators are also practically maintenance - free. However, air temperature field in heated zones is less equable than in case of underfloor heating and vertical temperature gradient is little less comfortable from viewpoint of human health (comparison of vertical air temperature profiles for different heat distribution systems is presented on the figure 2.13). Another drawbacks of radiators compared with underfloor heating are space limitation (they need some space in room to be hung), little lower air quality due to spread more airborne dust and more noisy work, especially during warming up period (cracks or ticks can be heard).

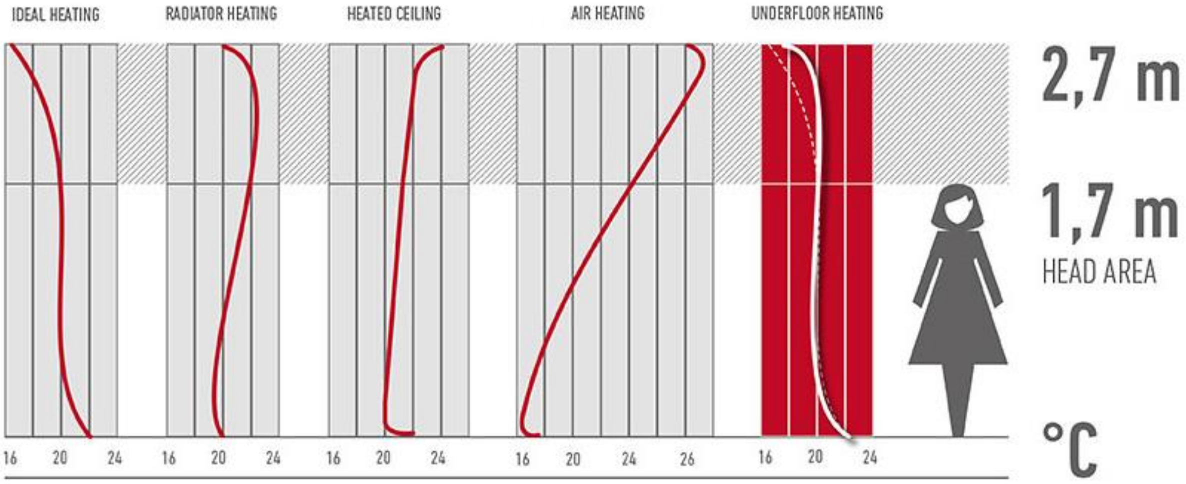


Figure 2.13. Vertical air temperature profiles in heated room for different heat distribution systems, in comparison with ideal profile, excluding heat gains from people and appliances [47]

- (*Forced*) *air heating* - it is another way of heat distribution (which can co-operate with air - water heat pump), recommended only in well - insulated buildings (like passive houses), mainly due to low thermal capacity of the air and limited temperature of the air warmed in ducts (air is not allowed to be heated over 50ºC [45]). In contrast to previous heat distribution systems, air heating relies on convection heat transfer. The air flowing in ventilation ducts can be warmed up by water - based heating coils placed there, whilst water in coils is heated by the heat pump. One of the biggest advantages of air heating compared to underfloor heating (and in less extent to radiators) is quicker warming up of the building. Furthermore, air heat distribution system seems to be less

[36] Most common low temperature radiators are made from aluminum.

expensive in case of investment cost than underfloor system, due to the fact that in modern passive buildings ventilation is needed anyway, thus investor needs to pay only for water - based heating coils, pipes connecting them with thermal storage tank (or with heat pump directly), additional water pumps and eventually more expensive fans[37]. Another advantage of air heating is the fact that it can be easily combined with cooling mode (e. g. additional water cooling coils), with relatively low increase of investment cost. Unfortunately, air heating system has many disadvantages compared with radiant heat distribution systems, i. a. lower efficiency of the system, due to air heat losses in ducts (even if they are properly insulated) and unwanted intermittent operation of the heat pump, warming up the water in heating coils (warmed air cannot be stored anywhere, and so relatively frequent intermittent work of heat pump is very probable in order to keep appropriate level of air temperature in heated space and compensate its fluctuations). Moreover, in order to ensure certain temperature level in rooms, supply air (heated or cooled, whatever) has to be much warmer (or colder) during outflow from ventilation duct. It results in less comfortable vertical air temperature profile in heated space (fig. 2.13) - the air is often too warm in its upper part and too cold in its lower part. Apart from this, another drawbacks of forced air heating are less equable air temperature field, more intensive spread of airborne dust (even though clean air comes from ventilation ducts, it later mixes with not such clean air from heated space), relatively noisy operation (caused by air pressure difference between building interior and ambient), which can be additionaly increased by fans operation. Maintenance costs of air heating system are also higher (periodic change of air filters, ducts cleaning and other maintenance actions), ventilation ducts used for heating have to be bigger and thus need more space, and regulation and steering of air temperatures in rooms is difficult if creation of different temperature zones is desired (temperature control is usually centralized via one thermostat, hence some rooms may be too cold or too warm).

- *Ceiling water heating* - it is inverted version of floor heating. Since the ceil heating installation is put in the highest place of heated space, there may be some uncertainties if such heating is efficient, because warm air is going up. Meanwhile, it is not such important, because ceil heating system relies on heat radiation in clearly higher extent than on convection. In this way emitted infrared waves warm the air in room (losing a part of their thermal energy) and reach the floor surface (as the surface placed opposite to the ceiling), which afterwards reheats the air. As a result, in the lower part of heated space, air temperature is relatively constant (excluding zone close to the floor level) but in its higher part - temperatures are higher (fig. 2.13). Thus, in rooms

[37] For heating or cooling purposes clearly higher airflow in ventilation ducts is needed than for only hygienic purposes, thus faster and bigger fans may be required.

where ceiling heating is applied the air temperature can be few degrees lower than in case of other heat distribution systems without a lack of thermal comfort, due to apparent feeling of the heat. This is a big advantage, because it allows for minimization of operation costs of ceiling system (supply water temperature can be lower than in case of underfloor heating). Another significant advantage of this system is that electromagnetic waves can easier reach all objects in heated space - practically the whole surface of ceiling is able to emit the heat effectively, what is not possible in case of underfloor heating (many objects are usually placed on the floor, thereby disturb the heat emission). Moreover (like other radiant - based heating systems) ceil heating system does not result in dehumidification of the inner air. However, ceiling system has also many drawbacks. To be able to warm up a part of space near the floor to desired level, the space under the ceiling needs to be heated up to even higher temperature, because heat is emitted from ceiling. Thus there are some considerations if the air over the head of average person is not too warm (what may result in headache and other uncomfortable feelings) or if the air above the floor is not too cold (while keeping desired optimal operative temperature[38] in upper part of the room). Hence, vertical air temperature profile in heated space is worse (than in case of underfloor system) from viewpoint of the heating needs of human body. Another disadvantage of the ceiling heat distribution system is high investment cost (also due to the fact that this heating system is not popular).

- *Wall water heating* - radiant heating system, which (in case of construction) is similar to underfloor and ceiling heating systems, installed under the surface of (both external, and internal, though less often) walls. This system is characterized by similar pros and cons as for underfloor heating system but there are some differences. Main advantage of wall system compared to underfloor installation is a lack of insulation layer - in case of external walls it is not required, because such walls in passive buildings are well - insulated anyway and in case of internal walls insulation is not needed, because heat is not lost through these walls but released to other rooms. As a result, wall heat distribution systems are characterized by lower investment cost than underfloor systems. They are clearly more often built in wet type (pipes are embed in concrete), what makes an investment cost is even lower but simultaneously it is more difficult to get to installation pipes if they need to be repaired. Pipes are usually mounted horizontally (one above the other), because it allows for reaching vertical air temperature profile more similar to ideal profile (fig. 2.13). In comparison with underfloor heating, it seems that horizontal air temperature field in room heated by wall heating is less equable and is more similar like a field reached by

[38] Its value is dependent on actual (dry bulb) air temperature and mean radiant temperature of surfaces surrounding considered part of space. Optimal operative temperature can be equated to felt temperature.

compact radiators (normally not the whole wall surface is covered by installation pipes but only a part of it). Another drawback of wall heating system is a special attention which have to be kept while mounting something on walls in places where installation tubes were put, in order to not damage the installation.

- *Underfloor water heating combined with radiant water heaters* - it is so-called hybrid heat distribution system, used in order to reach even better adaptation to current heating demand of the residential building. It combines some advantages of both underfloor heating and compact radiators, which in low temperature heating systems are recommended to place under the windows. Underfloor heating system has higher thermal inertia (warming up period of the floor is slower but afterwards it stays warm for longer) than radiators, which in turn allow for faster warming up of the heated space, when heating demand of the building is not such high. In combined mode, underfloor heating and radiators can work in co-operation or separately. This combined system, compared to underfloor heating alone has few advantages, i. e. it is more energy efficient, allows for faster response on changes in thermal balance of the building and investment cost of such installation (two different heat distribution installations have to be installed but then the area of underfloor heating pipes is usually clearly smaller) seems to be slightly lower. It is also easy to combine radiators with underfloor tubes into one circuit, because in low temperature heating supply water temperatures are more or less the same for both systems. Furthermore, it seems that vertical air temperature profile in heated space is less similar to its ideal shape for combined mode, compared to underfloor heating alone. Similarly, horizontal temperture field seems to be less homogeneous.

After wide explanation of possible heat distribution systems which can co-operate with air to water heat pump in passive residential buildings, their evaluation has to be made in order to indicate the most appropriate system, which will be taken into account in further considerations associated with the topic of this master thesis. During rating, distribution systems will be compared with respect to **the following criteria**:

- investment cost - including both installation components prices and expenses for labourers;
- operation costs - related to energy efficiency of the heat distribution in the building, with possibilities for energy savings due to easier regulation (less energy is wasted);
- maintenance costs - which have to be paid in order to maintain heat distribution system at the appropriate level of quality;

- thermal quality of the heated space - both the quality of vertical air temperature profile (compared to its ideal curve from viewpoint of human needs) and homogeneity of horizontal air temperature field in warmed space are considered;
- influence on air quality - influence of heat distribution system on deterioration of the heated air quality (propagation of airborne contaminants, appearance of both convection and radiant drafts, local changes in air humidity) is taken into account;
- regulation and steering - abilities of the system for quick adaptation to changing thermal balance of the building and for warming its inner space up to few different air temperature levels (i. e. creation of different thermal zones in the building);
- inner space limitation - inner space in the building which have to be allocated to heat distribution system components;
- reliability - the risk of problem occurence and average lifetime of the installation are considered.

In the table 2.5, evaluation results of previously described heat distribution systems are presented. Grades were numbered from 1 (the worst grade) to 10 (the best grade), with a step of 1. Each criterion has its own weight, dependent on its importance (total weight of criteria is equal to 1). Evaluated systems were named (in tab. 2.5) by numbers (I - underfloor water heating, II - radiant water heaters, III - (forced) air heating, IV - ceiling water heating, V - wall water heating, VI - underfloor water heating combined with radiant water heaters).

Table 2.5. Evaluation of different heat distribution systems from viewpoint of usage in passive buildings

No.	**Criterion**	**Weight of the criterion [Σ = 1]**	**Evaluated heat distribution system**			
			I	**II**	**III**	
1	Investment cost	0,19	4	7	6	1 - high; 10 - low
2	Operation costs	0,11	6	8	3	1 - high; 10 - low
3	Maintenance costs	0,08	7	9	4	1 - high; 10 - low
4	Thermal quality of the heated space	0,21	9	6	4	1 - low; 10 - high
5	Influence on air quality	0,15	9	7	4	1 - big; 10 - small
6	Regulation and steering	0,11	5	8	3	1 - difficult; 10 - easy
7	Inner space limitation	0,10	10	5	6	1 - big; 10 - small
8	Reliability	0,05	7	9	5	1 - low; 10 - high

Table 2.5. Continuation

No.	**Criterion**	**Weight of the criterion [$\Sigma = 1$]**	**Evaluated heat distribution system**			
			IV	**V**	**VI**	
1	Investment cost	0,19	3	5	5	1 - high; 10 - low
2	Operation costs	0,11	6	6	7	1 - high; 10 - low
3	Maintenance costs	0,08	7	7	8	1 - high; 10 - low
4	Thermal quality of the heated space	0,21	6	7	7	1 - low; 10 - high
5	Influence on air quality	0,15	9	9	8	1 - big; 10 - small
6	Regulation and steering	0,11	5	6	6	1 - difficult; 10 - easy
7	Inner space limitation	0,10	10	9	7	1 - big; 10 - small
8	Reliability	0,05	7	7	8	1 - low; 10 - high

Based on conducted analysis, comparing heat distribution systems gained final grades presented in the table 2.6.

Table 2.6. Final grades of evaluated heat distribution systems

Number of evaluated system	**I**	II	III	IV	V	VI
Assigned grade	**7.12**	7.07	4.41	6.30	6.90	6.79

As it can be noticed in tab. 2.6, heat distribution system with the highest grade (**7.12**) assigned based on conducted evaluation is **underfloor water heating**. It gained such high score due to the best thermal properties of heated space inside the building which can be reached using this system. However, in evaluation process the most important criteria (i. e. with the highest weights) were related with thermal comfort (criteria no. 4, 5), not with costs (criteria no. 1, 2, 3). In practice, operation, maintenance and especially investment cost are usually more important and therefore radiant water heaters (2nd place in tab. 2.6, with score of 7.07) are still more often utilized in modern passive buildings as a heat distribution system.

Another heat distribution systems with slightly lower grades are wall water heating (6.90) and combined system (6.79). In practice they can also be taken into account, as a kind of average solutions between underfloor heating (excellent thermal performance but expensive) and radiant water heating (relatively low thermal performance but cheap).

Moreover, author of this master thesis believes that ceiling water heating (6.30) is not really appropriate heat distribution system to use in passive buildings. Though it is characterized by no space limitation, very low influence on warmed air quality and equal horizontal temperature field in heated space, this system is very expensive and not really healthy from viewpoint of human needs. Also forced air heating system (the worst grade in conducted evaluation, i. e. 4,41) is not really good choice, though it is easy to be combined with cooling mode. Air heating is not such expensive to buy but expensive in exploitation. Furthermore, it has big influence on the air quality inside the building - its users may experience convection drafts and vertical air temperature profile is far away from the ideal profile (fig. 2.13).

2.4. Principles of transcritical heat pump cycle

In conventional heat pump system the whole cycle occurs in subcritical conditions (i. e. below critical point of utilized refrigerant). In transcritical cycle heat pump works in different way - subcritical conditions are maintained on so - called low pressure side of the cycle (i. e. after expansion and before compression), however on high pressure side of the cycle supercritical conditions (i. e. above critical point of the refrigerant) exist. The necessity of transcritical working conditions maintenance in CO_2 heat pumps utilized in residential buildings applications results from low upper limit of condensation temperature value in subcritical CO_2 cycle - only about 28°C [29], what is too low from viewpoint of the building heating needs.

The main difference between conventional and transcritical heat pumps is therefore embodied in different way of heat rejection - in conventional cycle heat is rejected in subcritical condensation process at practically constant temperature and pressure but in transcritical cycle heat is released at clearly higher, theoretically constant pressure but with varying temperature. As the result, in conventional heat pump heat is released in condenser and in transcritical heat pump - in the gas cooler. Comparison of both heat pump cycles is visible on p - h diagram located on the figure 2.14.

Thermodynamic transformations included in transcritical heat pump cycle, occuring between points marked on fig. 2.14 are:

- 4 ÷ 1 - heat absorption at constant temperature in the evaporator;
- 1 ÷ 2s - isentropic compression (1 ÷ 2 - taking heat losses into account) in the compressor;
- 2s ÷ 3 - heat rejection at constant pressure in the gas cooler;
- 3 ÷ 4 - isenthalpic expansion in the expansion valve.

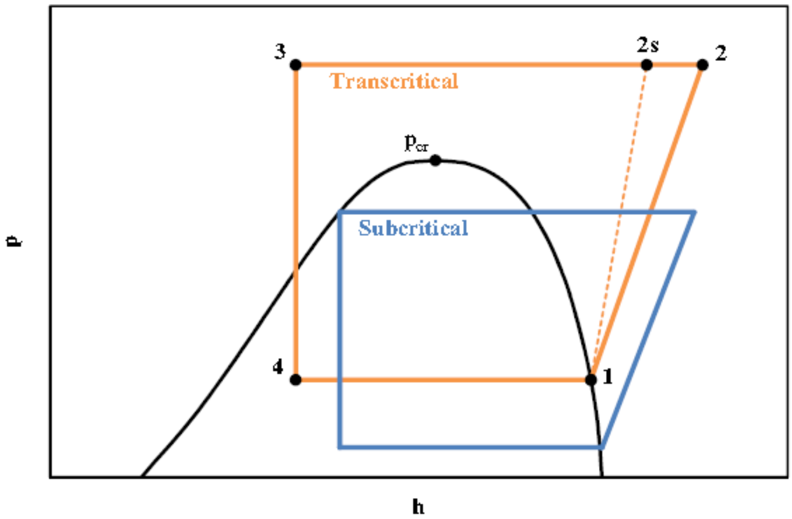


Figure 2.14. Comparison between ideal transcritical and conventional heat pump cycle on p - h diagram [9]

In reality all processes deviate from those depicted on fig. 2.14, mainly due to mechanical and heat losses. For instance, real compressor can experience various inefficiencies leading to mechanical and thermodynamic losses. Exergy analysis of transcritical heat pump suggest that compression losses constitute the largest part of the all losses in the cycle [8]. On the other hand, another studies indicate that expansion valve is the largest contributor of heat pump cycle losses [8].

COP of transcritical heat pump is calculated in analogous way as for conventional heat pumps. In terms of heating, it is defined as the ratio between heat released in the gas cooler and electricity supplied to the compressor. However, there is a slight difference in nomenclature: fundamental limit on conventional heat pump cycle efficiency is put by the Carnot efficiency (ratio between COP and COP_{max} of certain heat pump) but in case of transcritical heat pump such limit is put by the Lorentz efficiency (η_{LZ}) in the following way:

$$\eta_{LZ} = \frac{COP}{COP_{LZ}}, \tag{2.13}$$

where COP_{LZ} is the maximum achievable COP of the transcritical heat pump.

Similarly as COP_{max} for conventional heat pumps is calculated based on constant condensation and evaporation temperature values, in transcritical cycle COP_{LZ} is calculated in the following way:

$$COP_{LZ} = \frac{T_m}{T_m - T_0}, \tag{2.14}$$

where:

T_m - average temperature of heat rejection in the gas cooler [K];

T_0 - evaporation temperature [K].

As one can realized (analyzing equation 2.14), COP achieved by transcritical heat pump is strongly dependent on average temperature of CO_2 in the gas cooler. Temperature changes of the refrigerant during process of heat release in the gas cooler are generally not linear but have rather non - linear duration. In order to achieve high COP it is desirable to keep T_m as low as possible. Due to the fact that outlet secondary fluid temperature and inlet R744 temperature are usually set values, it is the inlet secondary fluid temperature which determines T_m in the highest extent. Hence, when temperature difference of the secondary fluid uptaking heat in the gas cooler is relatively low, then COP of transcritical heat pump is normally lower than COP which conventional heat pump could achieve, because condensation temperature can be lower than T_m (what is illustrated on the fig. 2.15.a). In turn, when temperature glide of the secondary fluid is high, then T_m can be usually lower than condensation temperature, therefore higher COP would be achieved by transcritical heat pump (fig. 2.15.b).

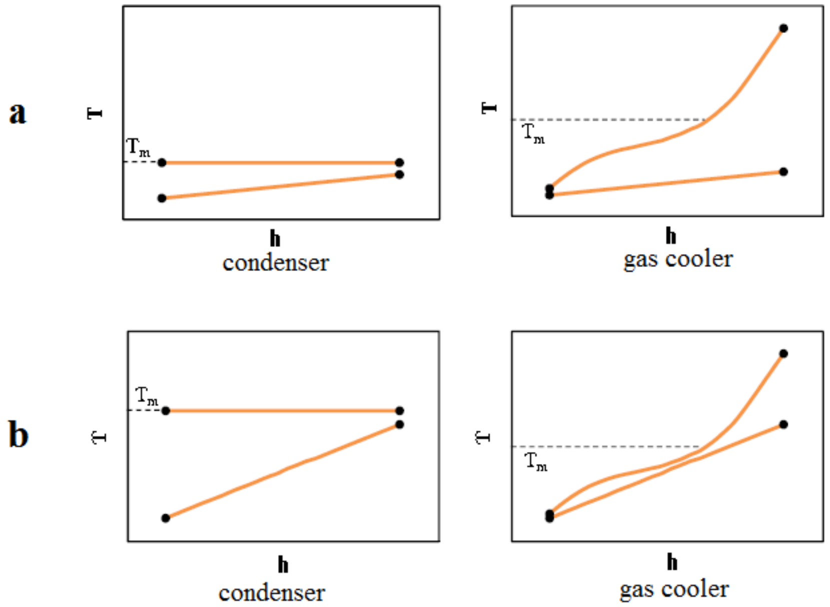


Figure 2.15. Temperature changes of the refrigerant during heat release process, depicted on T - h diagram; a - in condenser; b - in gas cooler [9]

Non - linear character of the CO_2 temperature drop is caused by its specific heat variations, especially near the critical point. Changes of the specific heat for different supercritical pressures are shown on the figure 2.16.

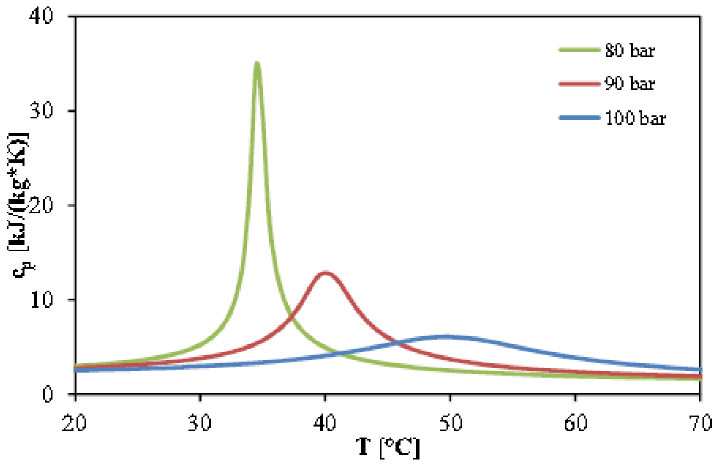


Figure 2.16. Dependency between specific heat of CO_2 and its temperature, for different supercritical pressures [9]

Based on fig. 2.16 one can notice that specific heat variations are smaller in case of higher pressures. Strongly varying properties of CO_2 make the heat rejection process occuring in transcritical heat pump totally different than condensation process in conventional heat pumps. The special care is therefore needed during the gas cooler modelling.

To sum up, among the main **advantages** of transcritical CO_2 heat pump utility compared to conventional heat pumps one can distinguish [8]:

- higher efficiency (in terms of SPF, COP) but only if the following requirements are fulfilled:
  - CO_2 heat pump covers the entire DHW heating demand and the annual heat delivered for DHW production is minimum 25 ÷ 30% of the total annual heat delivered from the heat pump;
  - main operation mode of the heat pump is combined, i. e. with simultaneous space heating and DHW preparation. In comparison with conventional heat pumps, transcritical heat pumps achieve higher COP in combined mode and DHW mode but lower in space heating mode only. Hence, the larger the annual DHW heating demand, the higher the SPF of CO_2 heat pump;
  - return water temperature in waterborne underfloor heat distribution system is equal to ~30^{o}C or lower. The lower return water temperature result in further increase of heat pump SPF;
  - water from the mains, supplying DHW tank has a temperature about 10^{o}C or lower;
  - thermodynamic losses in DHW storage tank are low, what means very low mixing losses and minimum conductive heat transfer between hot and cold water layers during charging and discharging phases;
- more environmentally - friendly character due to CO_2 utilization which (at least in short time perspective) has significantly less destructive influence on the climate ($GWP = 0$ and $ODP = 1$) than other utilized refrigerants.

Among transcritical heat pump **drawbacks** are its higher investment cost, more complex construction and automatics and low efficiency when the heat pump is combined with high - temperature space heating system (e. g. convectors, radiators).

3. Energy storage theory

There are many different kinds of energy which can be stored, however in most cases either thermal or electric energy storage is considered. The main goal of the energy storage is its saving during periods when energy production is higher than its consumption and its later utilization when energy demand is higher than energy which can be produced, thereby allowing for lower operation costs of considered energy system and for more efficient management of previously produced energy. Energy storage in nowadays is almost exclusively maintained by large energy companies - the biggest energy magazines in the world are hydropower stations, where energy is accumulated in incredibly large volumes of water in the lake (formed either naturally or artificially). Based on EPRI[39] report from 2011, 99% of total energy stored in the world is stored in pumped - storage hydropower plants [87]. Total power of these power plants was higher than 100 GW, whilst overall power of the all other energy storages was lower than 1 GW [88].

In this chapter the author will focus on thermal (heat) energy storage only, presenting various methods allowing to do that in the first section. Afterwards, two selected heat storage solutions, which can be utilized in the modern building will be wider described, as the topic of this master thesis is related with i. a. thermal energy storage in the passive house.

3.1. Thermal energy storage methods

Accumulation of the thermal energy is mostly associated with storage of water vapour or hot water. In overall it can be also related with storing of cold water or ice.

Heat storage can be executed utilizing various media which are able to store large amounts of heat due to relatively large heat capacity of substances. However, in general heat storage can exist for the wide range of their heating capacity and in a wide range of source temperatures. Heat storage methods can be mainly divided from viewpoint of the following criteria [3]:

- temperature of the medium, into:
  - low temperature heat storage (with storage temperatures of no more than 100 ÷ 120°C);
  - medium temperature heat storage (temperatures in the range of 120 ÷ 500°C);
  - high temperature heat storage (temperatures above 500°C);
- time period of heat storage:
  - short - term heat storage (the heat is accumulated for hours or days);
  - long - term heat storage (accumulation of the heat for months, seasons or years);
- type of the medium:

[39] Electric Power Research Institute.

- heat storage in water;

- heat storage in compressed or liquid air;

- heat storage in oils or liquid metals, etc.;

- mechanism of heat storage:

- heat storage using specific heat;

- heat storage based on phase transitions;

- heat storage by use of chemical reactions.

From viewpoint of different mechanisms of heat storage, the simplest one is storage using **specific heat** of the medium. The amount of heat which can be stored in the battery is mostly dependent on three factors - magnitude of the material heat capacity (if higher capacity, then more energy can be stored), amount of the medium storing the heat and temperature level in which the energy is stored. Due to relatively high specific heat (4189,9 $\frac{\mathrm{J}}{\mathrm{kgK}}$ [89]) and widespread availability water is almost always utilized in the low temperature storage (i. e. below 120°C). In turn, for the heat storage in the medium temperature range (120 ÷ 500°C) low boiling oils are used. Besides, in case of temperatures above 500°C heat is mainly stored in liquid metals, what however is very often unefficient, due to relatively low heat capacity and thermal conductivity of metals, thereby requiring larger heat exchange areas and special constructional solutions, compared to heat storage in lower temperatures. Due to this fact, inorganic solids are often utilized instead of liquid metals. Heat storage can be therefore relatively cheap and often with many years of failure free operation. For instance, specific heat of granite in temperature of 700°C is equal to 1000 $\frac{\mathrm{J}}{\mathrm{kgK}}$. Advantages of heat storage methods using specific heat are thus relatively low price and reliability (compared to heat storage methods based on either phase transitions and chemical reactions).

In case of the heat storage systems based on **phase transitions**, the most important factor is enthalpy of phase transition of the material - if it is higher then the material can absorb (and later release) larger amount of heat. The overall name of the group of substances which can play the role of media for heat storage based on fusion heat is <u>PCM</u> - Phase Change Materials. Among the most popular phase transitions which are utilized in heat storage are melting - coagulation and evaporation - condensation (applied clearly less often than melting - coagulation phase change, due to large volume changes in the system during phase transition), which occur endothermically (uptaking of the heat from the environment - melting) or exothermically (rejecting the heat to the environment - coagulation). Disadvantage of heat storage systems based on phase transitions is a variability of thermal and chemical properties of PCMs in a function of time. In turn, advantage of batteries using PCMs is the absorption and rejection of the heat in very narrow temperature range.

The „ideal" substance utilized as a heat storage carrier in the heat storage mechanism related with its fusion heat[40] should have the following features [3]:

- its heat of melting should be as large as possible;
- its volume change during phase transition should be small;
- thermal conductivity and specific heat at high working temperatures should be maximally high;
- its chemical reactivity in the whole given operating temperature range should be minimal, allowing for minimization of the damage caused to the installation over many years of operation;
- it should be non toxic, non flammable and should be characterized by minimum risk of explosion;
- it should be cheap and widely accessible;
- changes of physical properties of its liquid phase in the range of exploitive changes of operation temperature should be small;

The third mechanism in which the heat can be stored relies on **chemical reactions**. This heat storage method is characterized by usually lower efficiency and higher cost than previously depicted methods, especially if reversible reactions of the solid - gas type are taken into account. Storage systems using the heat of such chemical reactions usually involve very complicated engineering. Example of reversible chemical reaction used in the process of heat storage can be a synthesis of ammonia ($2\mathrm{NH_3} \leftrightarrow \mathrm{N_2} + \mathrm{H_2O}$), during which change in enthalpy of chemical reactions occurs and is equal to 92,2 $\frac{\mathrm{kJ}}{\mathrm{mol}}$, keeping the temperature level of 466 K (193ºC).

3.2. Chosen heat storage methods in residential buildings

3.2.1. Water storage tanks

Typical water storage tank is a cylindrical block usually made of steel, which is destined for storing of the heated water, hereby supporting the whole heating system of the building (space heating and domestic hot water simultaneously) or only one of its part - tank can play a role of the <u>buffer</u> tank for space heating system (increasing the water amount of the circuit) or the <u>accumulation</u> tank for domestic hot water storage, preparation and distribution. Cylindrical shape of the tank is not accidental, especially when bigger tanks (with volume of 1 m^3 or 2 m^3) are considered. Amounts of water such as 1 or 2 tons can sometimes generate really serious pressure forces. Cylindrical construction allows for equal distribution of these forces, thereby preventing potential damages to the tank, even though there is usually a low pressure increase or no pressure increase inside the

[40] In further part of this chapter, phase transition of any PCM is considered as melting - coagulation processes.

heating installation loops. Moreover, cylindrical shape of the tank allows to decrease heat losses from the tank in a positive manner. Besides, analysis of operation of the tanks with spherical shape (theoretically allowing for storing the water at higher pressures) was conducted, however it was found that variable shape of the tank boundary layer during charging and discharging phases resulted in decrease of the tank operation efficiency [7].

Water storage tanks for utilization in residential buildings can be divided into two main categories (wider described separately in this section):

- tanks for domestic hot water storage - <u>accumulators</u>;
- tanks destined for space heating water storage - <u>buffers</u>.

Withing the last years both buffers and accumulation tanks are becoming more and more popular, often in combination with solar panels. In case of commercial users, tanks can be exploited in central heating (or CHP - combined heat and power) stations, supplying the heat to many houses or blocks at the same time. They can be mounted either directly or indirectly (parallely), reaching volumes even above 10 000 m^3 [71]. For private users living in single family buildings typical volumes of the tanks depend on its destiny.

3.2.1.1. Tanks for domestic hot water

Tanks destined for domestic hot water storage (accumulators) can be divided into few main types, dependent on the construction:

- sheathed tank. Its construction relies on placement of smaller tank inside the second one which is bigger. Space placed between both tank layers is fulfilled with hot water supplied by heating device, which warms domestic hot water in the smaller tank. Insulation layer is mounted to the external shell of the tank;
- monovalent tank - standard tank (one shell) filled by water, equipped with one heating coil drowned inside its volume, through which hot water or another energy carrier flows and thus warming up DHW inside the tank;
- bivalent tank - tank with either sheathed or standard construction, equipped with two or three heating coils. Bivalent tank is therefore destined for the cases where more than one energy source is used in heat generation, e. g. through the one coil the heat is supplied from the boiler and via the second coil solar panels provides heat to the storage tank. The user can define the selection order (strategy) of heating devices operation, i. e. in the first approach solar collectors will try to meet the heating demand without involving boiler but when the water heated by solar collectors does not reach desired temperature, boiler provides an additional amount of energy. Such solution

seems to be more efficient than two previously described types of tanks, however higher investment cost is required due to solar panels and few heating coils mounted inside the tank, instead of one (or any). Despite the higher cost, development and popularity of bivalent tanks within the last years seems to rise faster than in case of other tanks.

Irrespective of the construction type, hot water storage tank is anyway relatively expensive element of the heating system. However, its utilization has many advantages, compared to the heating system not equipped with tank:

- possibility of gaining the constant water flow rate with stable high temperature;
- allowed simultaneous water usage from few water intakes (faucets or showers), without clearly noticeable decrease of either flow rate and temperature of outflowing water independenly (in certain degree) on how many people use the water at the same time;
- capability of reaching hot water in any water intake immediately after opening of the valve (if water circulation is applied);
- heating device can work with lower maximum power, because it does not have to heat up desired volume of water immediately (as it does in current water heating mode without tank utility). It can be done in the longer time period, therefore max. power value can be decreased. As the result, heating device can be cheaper.

However, these advantages are occupied by the following drawbacks:

- investment cost of the water tank which in case of the solid tank is not small. Price of the tank is mostly dependent on:
  - size of the tank - bigger tanks are more expensive;
  - material of the tank coat. Water storage tanks can be made of either enamelled carbon steel, stainless steel, copper or synthetic material. Stainless steel is more expensive than enamelled steel but it is also more durable;
  - material and thickness of the tank insulation layer. Typical material used for water tank insulation is a soft polyurethane foam, with thickness of 8 ÷ 10 cm [90];
  - amount of heating coils integrated with the tank - if more coils are attached, the tank is more expensive;
  - type of anode placed inside the tank as additional protection against corrosion[41]. For storage tanks made by carbon steel or enamelled steel the most popular and cheaper is magnesium anode (which needs to be replaced one time per year) and the more expensive is titanium anode, which does not have to be replaced.

[41] Due to water temperature changes inside the tank, steel expands and contracts alternately, what results in microcracks of the tank coat, hereby leading to corrosion.

- higher annual energy consumption from viewpoint of DHW needs, caused by heat losses through the water storage tank envelope;
- water storage tank is not small and therefore it requires some space to be installed, slightly decreasing usable floor area of the house.

Hot water tanks destined for domestic hot water storage (accumulators) are utilized all the year, as DHW is needed every day. Specific feature of such tanks is cyclic mode of their operation - charging and discharging phase of the tank are progressing alternately. DHW tanks are recommended to be placed in vertical position, thus they are normally longer than wider, however in practice storage tanks can be also mounted horizontally, especially if due to some reasons they cannot be located vertically.

Size of the accumulator depend on the water consumption level by the occupants. If the water usage is relatively small, daily storage of 15 ÷ 30 litres of hot water per person is usually enough. In turn, it the water usage in the building is high one should predict 30 ÷ 60 litres of hot water per occupant stored in the tank [91]. Hence, for 5 - people family total volume of the assumed storage tank should vary between 75 and 300 litres. Results of the research performed by EST (Energy Saving Trust) in 2008, concerning on measurements of DHW consumption in residential dwellings [18] seems to confirm these values. They indicate that domestic hot water consumed by the building occupants is decreasing together with increase number of the occupants in the house, what is depicted on the figure 3.1. Taking these results into account, one can notice that average daily water consumption of 5 - people family equalled ~175 $\frac{l}{\mathrm{day}}$,

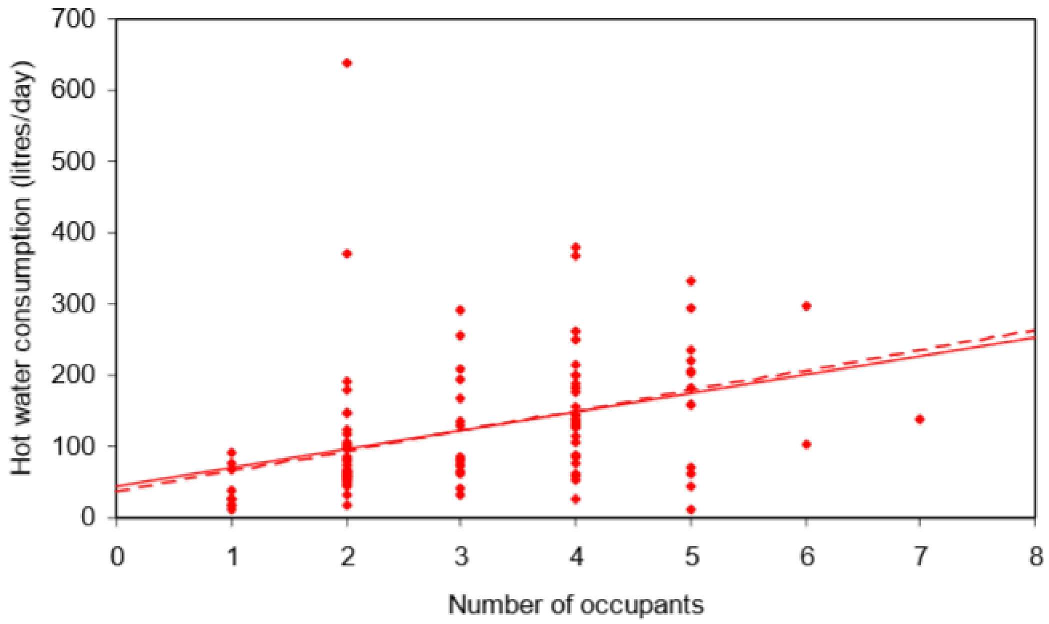


Figure 3.1. Domestic hot water consumption as a function of the building occupants, based on conducted measurements [18]

Temperature of domestic hot water stored in the tank should be maintained on appropriate level, in order to avoid legionella disease and other contaminations. It is mostly recommended to keep water temperature in the tank above 60°C, with water temperature flowing to faucets or showers not lower than 50°C [51]. Potential risk of the legionella growth or spread for different DHW temperatures maintained in the tank can be seen in the table 3.1.

Table 3.1. Risk of the legionella occurence dependent on different DHW temperatures in storage tank [51]

DHW tank temperature value	Behaviour of the Legionella bacteries
70 ÷ 80°C	Disinfection range
66°C	Legionellae die within 2 minutes
60°C	Legionellae die within 32 minutes
55°C	Legionellae die within 5 to 6 hours
Above 50°C	They can survive but do not multiply
35 ÷ 46°C	Ideal growth range
20 ÷ 50°C	Legionellae growth range
Below 20°C	Legionellae can survive but are inactive

Beside a division conducted at the beginning of this subsection, DHW tanks can be also either <u>non</u> <u>stratified</u> (where temperature of water stored in the tank is constant - most of currently utilized tanks) or <u>thermally</u> <u>stratified</u> (water temperature changes in a function of the tank height). Thermal stratification is physical phenomenon resulting from density difference between hot and cold water. Mechanism of thermal stratification in the tank begins from the state in which hot water flowing into the tank is moving upwards (or staying there, dependent on which height level it flows into the tank) due to buoyancy forces and cold water falls down, because is heavier. Two layers of water are then separated by intermediate layer, in which rapid change of the temperature occurs, because of intensive heat exchange between hot and cold water layers. This interlayer is called as <u>thermocline</u> and it states an indicator of correctly designed storage tank. Thickness of thermocline should be as small as possible to maintain good stratification inside DHW tank by minimization of mixing process between cold and hot water masses. The outlook of temperature profile in typical stratified storage tank is schematically presented on the figure 3.2. Ideal thermal stratification occurs when hot water and cold water zones are totally separated and temperature gradient of thermocline (layer in which water temperature changes) is infinite.

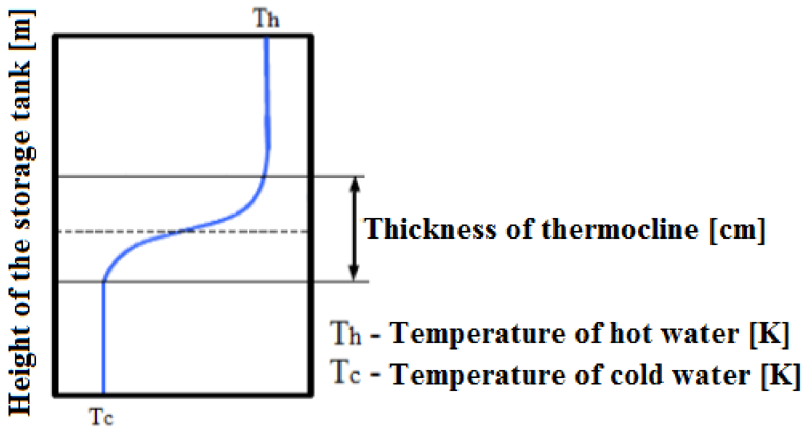


Figure 3.2. Water temperature profile in thermally stratified DHW tank [73]

In practice, complex physical phenomena occurs inside the tank filled with water, having direct influence on mixing inside the tank and hereby on duration of its thermal stratification. Factors which have direct influence on creation and maintenance of thermal stratification inside the tank are specified below:

- slimness of the tank - expressed as the ratio between its height and diameter, slimness has direct influence on the thickness of thermocline, which have great impact on self creation mechanism of the water layers with different temperatures inside the tank. Researchers who performed experiments in order to determine the optimal value of slimness suggest that slimness value between 3 and 4 allows for achievement of maximum stratification [74], however exact value of this parameter can be different in each case. It is also worth to remember that together with increase of $\frac{H}{D}$ total area of the tank is also getting higher. That may result in increase of heat losses from tank to the ambient [71, 74], what is unwanted;
- temperature difference between hot and cold water - the higher the difference is, the more accurate thermal stratification can be achieved;
- velocities of water flows at inlets and outlets of the tank - different ways of charging and discharging of the water placed inside the tank were analyzed [25, 52], in aim to find a way causing the smallest disturbances which result in the smallest mixing of the water. Various inlet and outlet diffusors[42] were tested, mounted in a set of few similar pipes, pipes with different shape, flat plates and perforated plates. Moreover, both cold and hot water streans were supplied to the tank on different heights. It was indicated that the lowest influence on water mixing inside the tank exists when cold water inlet supplying the tank is placed as low as possible and is separated with the plate partition from the rest of the tank volume. Similarily, hot water should be supplied to the tank through the inlet placed as high as possible and separated with the partition

[42] Diffusors (instead of normal inlets) are used in thermally stratified tanks, in aim to decrease velocity and kinetic energy of the water flow supplying and flowing out of the storage tank.

from rest of the tank. Intakes arranged in such way allows for relatively flat introduction of both cold and hot water streams to the inner of the tank. In turn, outlets of water streams (supplying heating device and the building hot water distribution grid) should be located on similar heights as intakes. Duration of charging and discharging phases in correctly designed thermally stratified storage tank is presented on the figure 3.3;

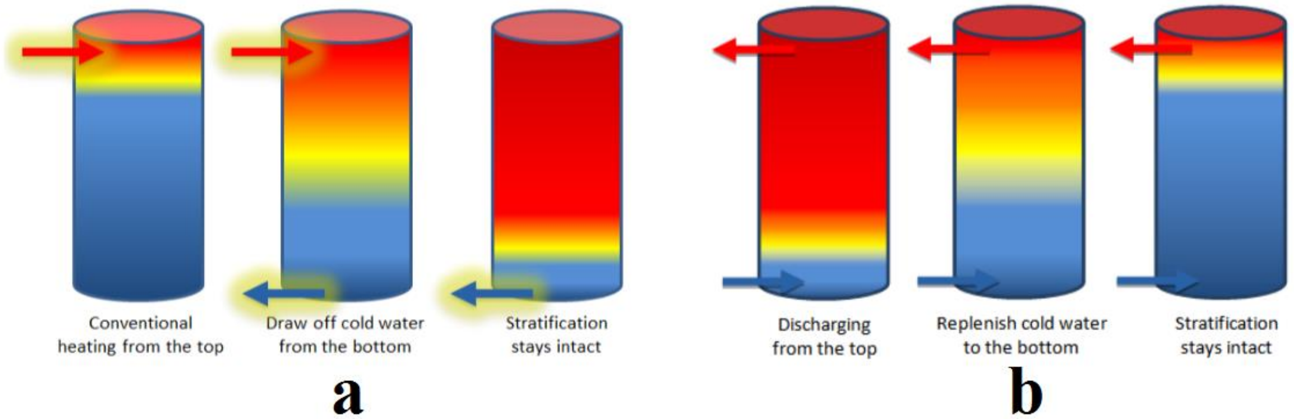


Figure 3.3. Thermal stratification maintained in water storage tank during charging and discharging phase [19]; a - charging phase; b - discharging stage

- material of the tank shell and insulation of the tank - due to economic and strength reasons, hot water tanks are mostly made of steel, with thermal conductivity of 50 ÷ 60 $\frac{W}{mk}$ [73]. Walls of the tank with higher conductivity increase the magnitude of temperature gradient appearing in their nearby. For instance, in aluminium tank (206 $\frac{W}{mk}$) vanishing period of thermal stratification was 10 times shorter than in the same tank but made of glass (0,754 $\frac{W}{mk}$) [70]. Hence, lower thermal conductivity of the tank coat is desired to keep the water thermally stratified for a longer time. Simultaneously, insulation layer of the tank should be appropriately thick (at least 5 cm of mineral wool or material with equivalent conductivity [73]), to minimize heat losses to surroundings and temperature gradient near the tank walls. Construction of the tank insulation is also important - the best option is to insulate the tank from inside and additionally from the outside, while the external insulation layer should be as tight as possible, hereby minimizing thermal bridges.

In reality, when storage tank is fully loaded (i. e. full of heated water) and only heat losses through the tank walls and valves to the ambient occurs (without any additional supplying and discharging of the water), water located inside the tank is naturally aspiring to be thermally stratified, what is depicted on the figure 3.4[43].

[43] In further description of the mechanism of water temperature changes in the tank horizontal temperature differences in each layer of the tank (in reality number of layers is infinite) were neglected.

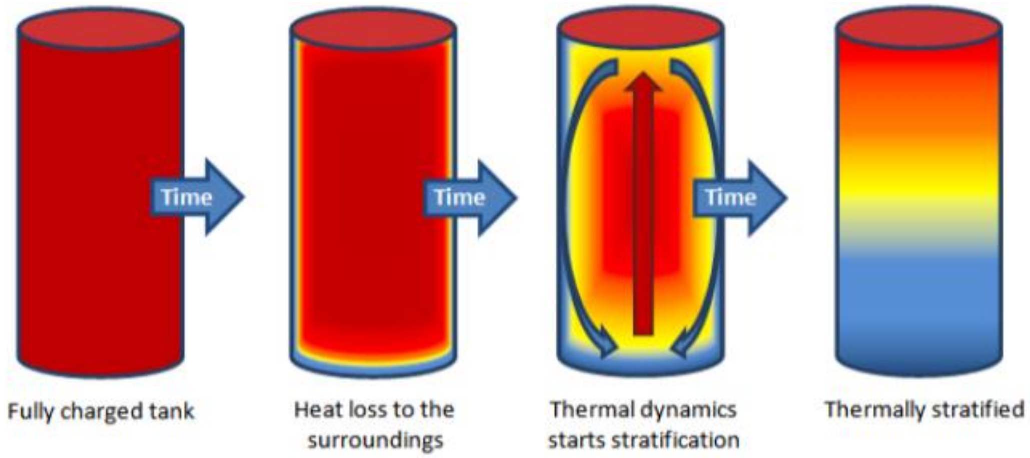


Figure 3.4. Further steps of water temperatures changes in the storage tank without disturbances (only heat losses from the tank) [19]

Together with lapse of the time water temperature in all tank layers is decreasing[44], whilst thickness of thermocline is rising (fig. 3.5).

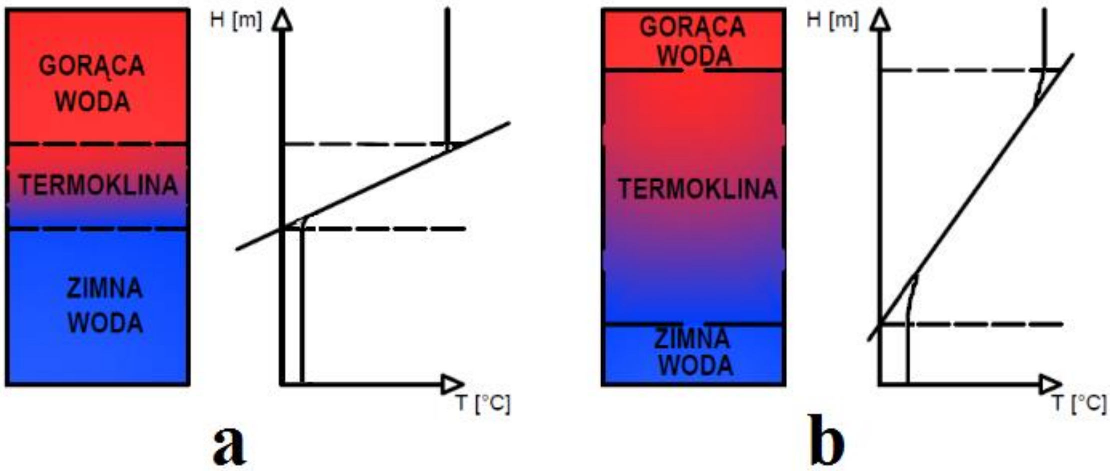


Figure 3.5. Part of the process of water temperature changes in the storage tank [73]; a, b - further steps, presenting how thickness of the thermocline increases during the time. In reality, temperature curve in the middle part of thermocline has non-linear character

After certain time period, thickness of thermocline is equal to height of the tank, with maximum water temperature (equal to the initial temperature after heating) in the highest layer (with infinitely small thickness) near the top of the tank and minimum temperature (equal to the air temperature in surroundings) in its bottom part. Next, thickness of water layer with minimum temperature equal to the ambient air outside of the tank is increasing, therefore thermocline is simultaneously getting narrowed and its thickness is decreasing. In the end, water temperature in the whole tank is uniform and equal to the ambient air temperature.

To sum up, maintenance of the thermal stratification in domestic hot water storage tanks has few important advantages (compared to water tanks without thermal stratification):

[44] Water temperature is decreasing in all layers until the whole tank will reach uniform temperature, equal to the ambient air.

- reduced heat losses from the tank to ambient - due to the fact that water in the bottom part of the tank is lower than temperature of the utilized water (stored in its upper part) overall heat losses to the surroundings are lower, thus decreasing annual energy consumption of heating system from viewpoint of DHW preparation. Furthermore (if solar collectors are installed), heat losses from pipes connecting solar panels with the tank are also lower, because average water temperature in the pipes is lower than in case of the tank without stratification (colder water in the bottom part of tank is supplied to solar collectors to be heated, thus decreasing its mean temperature being somewhere between water temperature value before solar panels and after flowing through them);
- higher stability of desired water temperatures - when storage tank is utilized, domestic hot water flowing out of faucets or showers in the house are usually obtained by mixing hot water from top of the tank and cold water from the mains (what is schematically depicted on the figure 3.6). In non-stratified tank the average water temperature in upper part of the tank without thermal stratification is generally lower than in comparable non - stratified tank [19], what may sometimes result in disability of supplying maximum desired temperature to the house faucets or showers. The comfort temperatures are therefore more often available in a stratified tank. Moreover, if a top-up heater is installed near top of the tank (supporting main heating device when it is not able to meet the demand on its own), the additional energy consumption is lower in case of thermally stratified tanks;

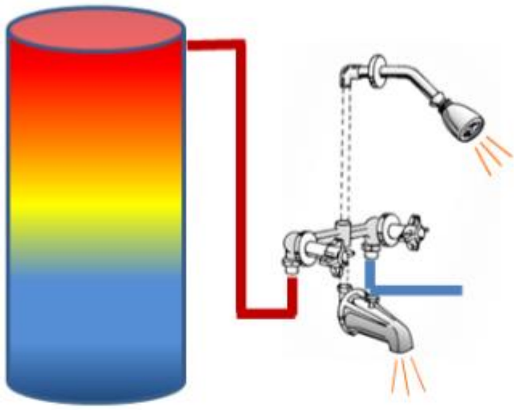


Figure 3.6. A scheme presenting mixing of hot water from the tank with cold water from the mains, resulting in desired water temperature used in faucets or showers [19]

- smaller volume of tank can be used - in the lifetime of stratified DHW tank, total water volume circulated through the tank is smaller compared to a mixed tank, because average temperature of the hot water flowing out of the tank from its upper part is higher than in non-stratified tank. As the result, less volume of hot water is needed to cover DHW demand when stratified tank is used, therefore a smaller tank can be bought instead of the bigger non-stratified one, what decrease the investment cost;

- less lime settling - water, before flowing through appropriate filter, always contains some lime. When less volume of the water is flowing through the tank during a certain period then less amount of lime is gathering inside of the tank. Research performed in this case indicates that the lime scaling in thermally stratified tank can be even 2,5 times less than in comparable non-stratified tank [25];
- increase utilization of solar collectors is possible - tank water which can be heated using solar collectors is firstly supplied to them from the bottom part of the tank. Next, solar collectors are starting to warm up the water when their temperature becomes higher than the inlet water temperature. Temperature achieved by solar collectors normaly vary throughout the day (solar collector temperature curve for typical sunny day is shown on the figure 3.7). In thermally stratified tanks the inlet water temperature is lower, therefore solar collectors can start to warm up the water earlier and finish their work later in the day, compared to non-stratified tanks. Consequently, utilization of solar panels is more efficient and operation costs of the heating system from viewpoint of DHW preparation decrease;

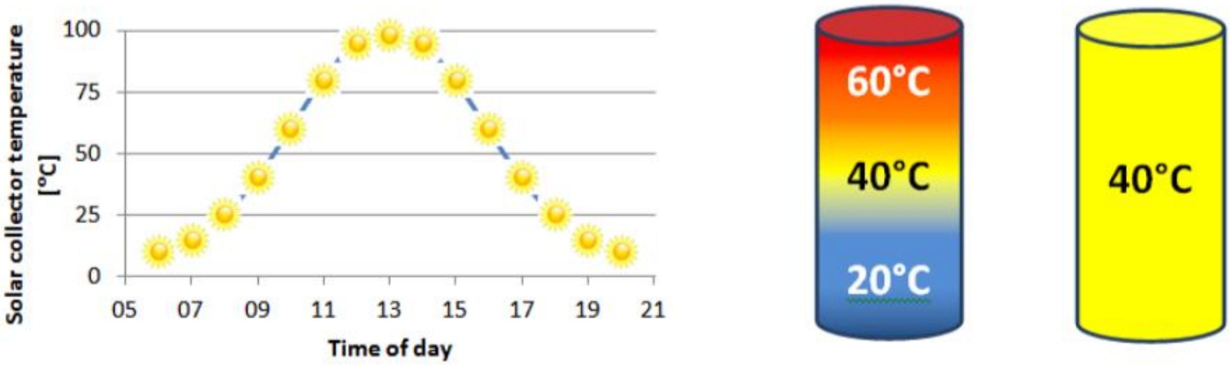


Figure 3.7. Temperature curve presenting temperatures achieved by solar collectors during sunny day [19]. On the right side of the illustration both stratified and mixed tank is depicted in order to present typical water inlet temperatures to solar collectors in both cases

3.2.1.2. Tanks for space heating systems

Tanks co-operating with space heating loop (buffers) are almost always non - stratified, because it is useless for their operation. The aim of their utility is an increase of the total water volume in space heating system, allowing for its more efficient work. They are mostly recommended to heat generation systems which work irregularly and / or can warm technical water to only high temperatures[45], however in practice almost every heating system has some benefits resulted from buffer tank utility. For instance:

- in (especially older) heating systems based on solid fuels (e. g. wood or coal), where heating device (boiler) have to either operate with maximum power or be turned off, during periods of

[45] Temperatures higher than these which are desired in the building heat distribution system.

low heating demand of the building too much heat is generated, what decreases the efficiency of heating system. If the buffer tank is applied, a surplus of produced heating energy can be stored there for its later utility, hence the tank plays a role of equalizer. Another advantage is less time spending in the boiler room - one portion of solid fuel supplied to the boiler by its user allows for keeping the house heated for longer period than without buffer tank utility.;

- in modern systems (condensing gas boilers, heat pumps) where heating device is turning on and off very often in response to the building heating demand varying all the time (detected by more precise steering of the system) buffer tanks increase water volume in the circuit, thus decreasing the number of turn on / off cycles, extending lifetime of the device and increasing the overall system efficiency (during period when heating device is turning on it requires little more energy at the beginning to get warmed).

Drawbacks of buffer tanks utility are similar to DHW tanks. The main difference may be in annual energy consumption, which is not always higher compared to the system without buffer tank. Of course, there are heat losses through the tank envelope but heat losses from heating device (when big part of produced energy is not utilized) may be sometimes higher.

Buffer tanks are utilized mainly during the heating season, as they co-operate with space heating system. Assumed tank volumes are related with types and maximum powers of both heating device and heat distribution system. For boilers using solid fuels (wood, wood pellets, wood chips, coal, coke, sawdust or straw with appropriate humidity) it is recommended to assume the tank volume of 40 ÷ 80 l per each kW of heating power [92]. In case of heating devices allowing for more precise adjustment of heating power to actual demand (e. g. modern heat pumps) volumes of buffer tanks can be smaller. Meanwhile, when radiators are used as heat distribution system buffer tanks usually increase water volume in the system more significantly and should be bigger than in case of floor heating system, where more water is included in the circuit. To sum up, while overall water volume in heat distribution system is rising, meaning and profitability of the buffer tank utility falls down.

3.2.2. Phase change materials (PCMs)

The name of these materials comes from the fact that their principle of operation is based on cyclic - repetitive phase change (melting - coagulation), while these changes occur due to differences in ambient air temperature during the day and night, regardless of the season. PCMs are substances which are able to absorb, accumulate and release a large amount of thermal energy (melting - heat absorption, solidification - heat rejection). Both absorption and rejection of the heat occurs in constant temperature of the phase change (which is dependent on the type of PCM), because during phase change only latent heat[46] is absorbed or emitted to the surroundings. However, during the whole cycle of PCM operation, the sensible heat exchange also exist. The work principle of PCMs is schematically depicted on the figure 3.8.

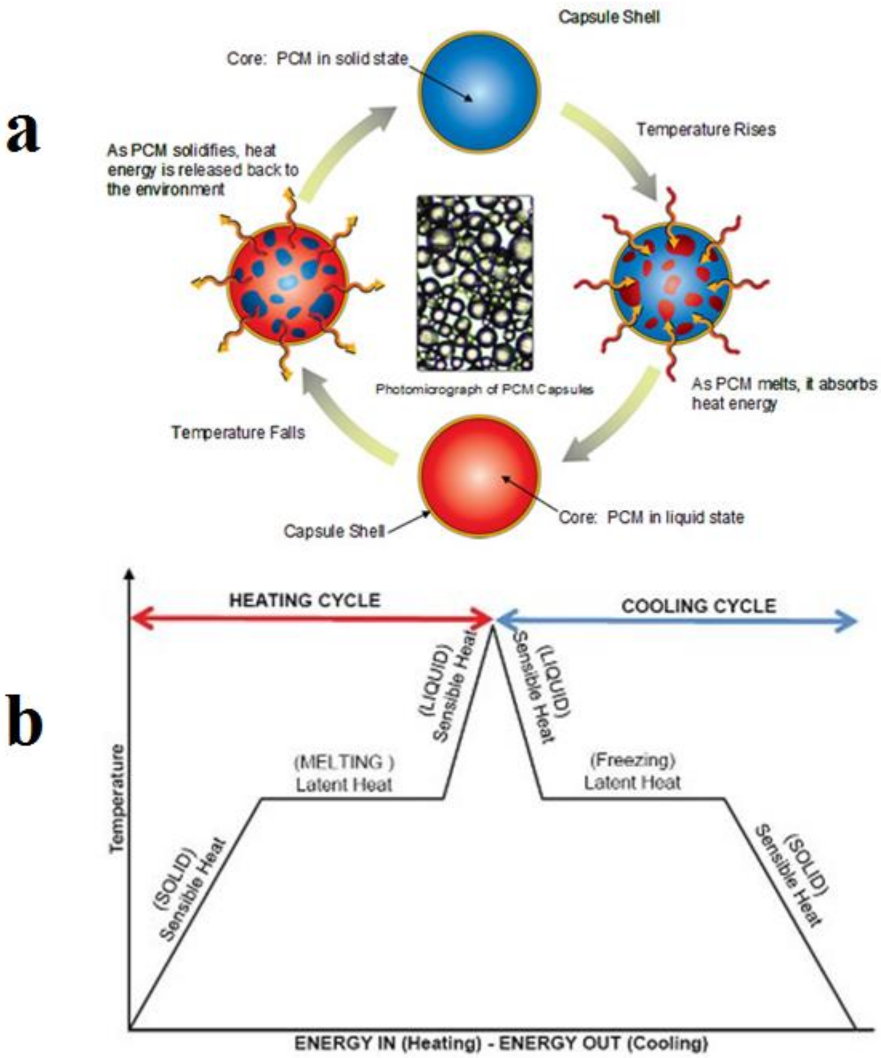


Figure 3.8. Illustration describing how PCMs (in form of microcapsules) work; a - scheme of operation, b - curve describing PCM temperature changes [52, 53]

PCMs are characterized by large heat capacity, therefore they are intended to increase overall heat capacity of the building. In turn, the greater the heating inertia of the building is, the more stable indoor temperature is. Hence, thanks to PCMs utility inside the building its users can experience clearly smaller daily temperature fluctuations in individual thermal zones (rooms), than without their application. PCMs operation is possible thanks to occurence of natural heat sources (i. a. solar radiation during the day or occupants present in the building) and natural cold sources (at night, when outdoor temperature values are smaller than during the day). Thanks to their utilization in the

[46] Emission or absorption of latent heat does not cause temperature change of the medium, instead of sensible heat, whose emission or absorption is always related with temperature changes.

building applications, less overheating during summer months and less overcooling during the winter season will occur in its thermal zones.

Nowadays, PCMs used in the building industry exist in a form of supplements to another building materials (e. g. they can be added to plaster used as external layer of indoor walls) or as the filling of special heat exchangers (designed for PCM storage), which can be mounted in free spaces of the building (e. g. over the suspended ceiling, in construction of the floor[47] or even suspended below the ceiling in a form of tray). As the example, outlook and work mechanism of PCM mounted over the suspended ceiling can be noticed on the figure 3.9.

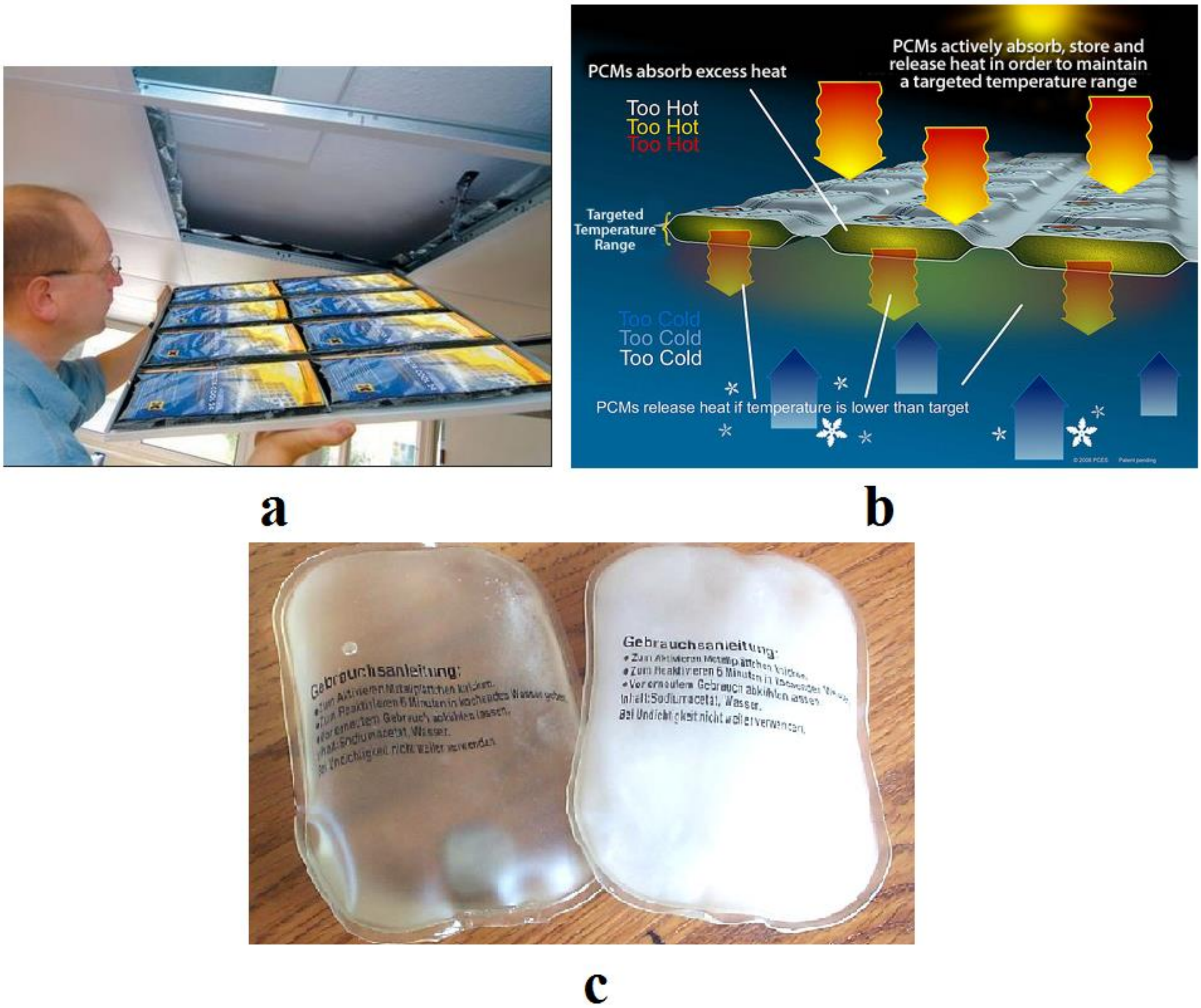


Figure 3.9. Utility of PCM in the area above suspended ceiling [53, 55]; a - mounting of PCM in practice, b - scheme presenting principles of PCM operation; c - example PCM - a sodium acetate solution (phase change occurs during its crystallization)

Among specific properties of PCMs, which should be considered by the user before selection of certain PCM are:

[47] It seems that application of appropriate PCM into the gaps below the floor surface (when underfloor water heating is applied as the heat distribution system) could be very profitable.

- heat capacity - heat storage capacity of phase change material is mainly dependent on its specific enthalpy (latent heat) of phase transition. For PCMs destined to be used in the building industry this parameter varies between ~100 $\frac{\mathrm{kJ}}{\mathrm{kg}}$ (for building products in which PCMs are present in form of closed capsules, where material of the capsule reduces the effective heat capacity of PCM) and ~250 $\frac{\mathrm{kJ}}{\mathrm{kg}}$ (for homogeneous PCMs). In overall, PCMs with significantly higher heat capacities exist, however these are high - temperature materials, useless in the building sector. Important factors for the heat capacity of PCM are also specific heat values for the liquid and solid phases. Temperature of the phase change for certain PCM is always clearly defined but in working conditions occuring in practice PCM is sometimes overheated and overcooled with respect to the phase transition temperature - temperature differences may reach even several degrees (in the energy balance enthalpy change related with these processes must be taken into account);
- temperature of the phase transition - as it was mentioned previously, PCM absorb or release the heat mostly during the phase transition, therefore it is very important to select PCM with temperature of phase transition which value is located in the range of temperatures which influence PCM directly during its operation in the building. These temperatures are affected by outdoor air temperatures during the year[48] (determined by climate location of the building) and by indoor air temperatures, maintained inside the building. In practice, evaluation of the applicability of particular PCM in specific location of the building must be always performer individually, because dependent on where in the building structure PCM is integrated, its operating conditions (in terms of mentioned temperature range) may be influenced by indoor and outdoor temperatures in a different way. If PCM is arranged in internal part of the walls or ceilings it is usually assumed that temperature of its phase change should be $1 \div 3^{\mathrm{o}}\mathrm{C}$ higher than mean indoor air temperature of the room. In turn, when PCM is desired to be placed inside the heated floor construction, then the type of underfloor heating system also influence temperatures sensed by PCM, therefore PCMs supporting floor heating systems are usually characterized by temperature of phase transition which may exceed even $40^{\mathrm{o}}\mathrm{C}$;
- thermal conductivity - in PCMs this parameter should be appropriately high, to allow for effective heat absorption and rejection. If certain PCM is not characterized by high conductivity, some other materials with higher conductivity (usually graphite or metal chips) are added, hereby creating a mix. Conductivity of PCM can be also improved by increase of the heat exchange area,

[48] In terms of maximum ambient air temperature during the day and minimum temperature at night.

if PCM is utilized in a form of special heat exchanger (e. g. by using ribbed surface instead of flat);

- degree of overcooling during coagulation - in some PCMs solidification starts not when the material reaches its melting point[49] but after supercooling of the liquid phase to temperature level below its melting temperature (magnitude of overcooling may reach even several degrees). Thus it is important to always analyze if the temperature of PCM after overcooling will not be lower than the lowest indoor air temperature which is desired to be kept inside the building. Otherwise, to allow PCM for its correct operation a decrease of indoor air temperature below its minimum desired value will have to be done. Sometimes a degree of overcooling during the phase change is the main factor which excludes possibility of the PCM utility in certain application, because the operating conditions (in terms of temperature range) does not allow for correct duration of the phase transition. Fortunately, degree of overcooling can be artificially minimized by adding substances which play a role of nucleating embryos;
- stability in many melting - solidification cycles - in building applications one melting - coagulation cycle lasts normally one day. PCM with a good (high) stability should be able to retain its properties from few up to several thousands of such cycles. Unfortunately, many PCMs are not such stable as they should be (even if they are characterized by high heat capacity).

In reality there exist also other important factors, which decide about the quality of PCMs and may determine their suitability for certain applications:

- low saturation vapour pressure (too high vapour pressure enforces necessity of PCM utility in a form of pressurized heat exchangers);
- small volume changes during melting process - compensation of PCM volume changes requires a very wide dispersion of the material (e. g. into microcapsules) or special construction of heat exchanger (what can easily increase the investment cost of PCM heat exchanger);
- chemical stability;
- compatibility with building materials, like cement, plaster, metals or plastics;
- non - toxicity and non - flammability;
- low investment cost;
- possibility of PCM regeneration.

As it was depicted above, there are many parameters which should be taken into account while evaluating certain PCM, therefore there is no PCM which fulfill at least most of the requirements.

[49] Temperature of the phase change.

Hence in practice phase change material is selected based on its heat capacity, temperature of the phase change and price.

To sum up, among the most relevant advantages of PCM usage in residential buildings one can distinguish:

- lower daily variations of the indoor air temperature, what positively influence a sense of thermal comfort experienced by the occupants (figure 3.10);

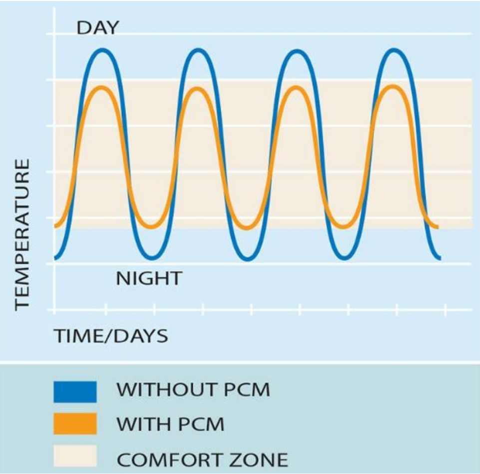


Figure 3.10. Comparison of the indoor air temperature variations in the building with and without PCM utility [53]

- utilization of the natural heat sources (sun, people, equipment), therefore PCM is an ecologic product[50];
- capability of the latent heat utilization, what turns out to be very profitable from viewpoint of operation costs of the building;
- universality - if properly selected, PCM can be applied in many places in the building.

Naturally, PCMs have also some drawbacks (e. g. occurence of supercooling during coagulation), however most of them can be reduced or totally eliminated, applying various physical treatments or designing PCM - based products accurately, e. g. by creating appropriate shapes and dimensions of heat exchangers containing PCMs, ensuring proper granulation of PCMs when they are mixed with plaster or concrete and optimizing the arrangement of PCM elements in the building structure. Obviously, significant disadvantage of PCMs which cannot be changed (at least in the next few years) is their high investment cost, resulting from their uniqueness and low popularity.

[50] Utilization of typical heating or cooling device instead of PCM (in the same goal) would result in CO_2 emission to the atmosphere, either directly or indirectly.

4. Building performance assessment - overview of common simulation programs

In the first part of this chapter the most common and relevant programs for the building performance simulation will be described, in order to select the most appropriate tool(-s) for further analysis conducted in this master thesis. Afterwards, simulation tools will be compared and evaluated from viewpoint of actual needs and the best solution will be chosen.

4.1. Depiction of the building simulation tools

4.1.1. CASAnova

CASAnova was developed at Siegen University in Germany by prof. F. D. Heidt. It is an educational software allowing for calculations of energy and heating demand in the building. It also predicts its solar heat gains and potential overheating risk. All calculations performed by this program are in accordance with the European norm EN 832. Although this program is relatively simple, it can be used to get intuitive understanding of the relations existing between the building geometry, orientation, thermal insulation, glazing, solar heat gains and its heating (cooling) demand. It is not possible to design energy supply system inside CASAnova but the purpose for development of this tool was only to illustrate dependencies between the outdoor climate and the building envelope parameters.

Like all other programs of this type, CASAnova needs some input data introduced by the user and after performed simulation it generates some output data. The user of the program can adjust and change more than 20 different parameters and observe their influence on the building energy balance. All parameters which can be implemented into CASAnova can be divided into 6 groups (i. e. building geometry, windows, insulation, building in overall, climate and energy). In turn, the most relevant output data are:

- monthly and yearly energy balances for ventilation and transmission heat losses, usable solar and internal heat gains, as well as total heating demand and primary energy demand of the building,
- solar heat gains and transmission losses of windows for each facade,
- number of overheating hours,
- energy flow diagram for the yearly energy balance,
- emission of carbon dioxide regarding the primary energy demand of the building,
- characteristics of the building design, i. a. A/V value, thermal inertia and others.

After each simulation the results are shown in diagrams and tables. The newest version of the program is CASAnova 3.3 and it is available in English. Program is licence-free, available in English and it can be easily downloaded from the network. Like most of programs it has its own interface which is presented on the figure 4.1. It can co-operate with a package of other programs related with building energy balances developed also at Siegen University - **IDEA** [54].

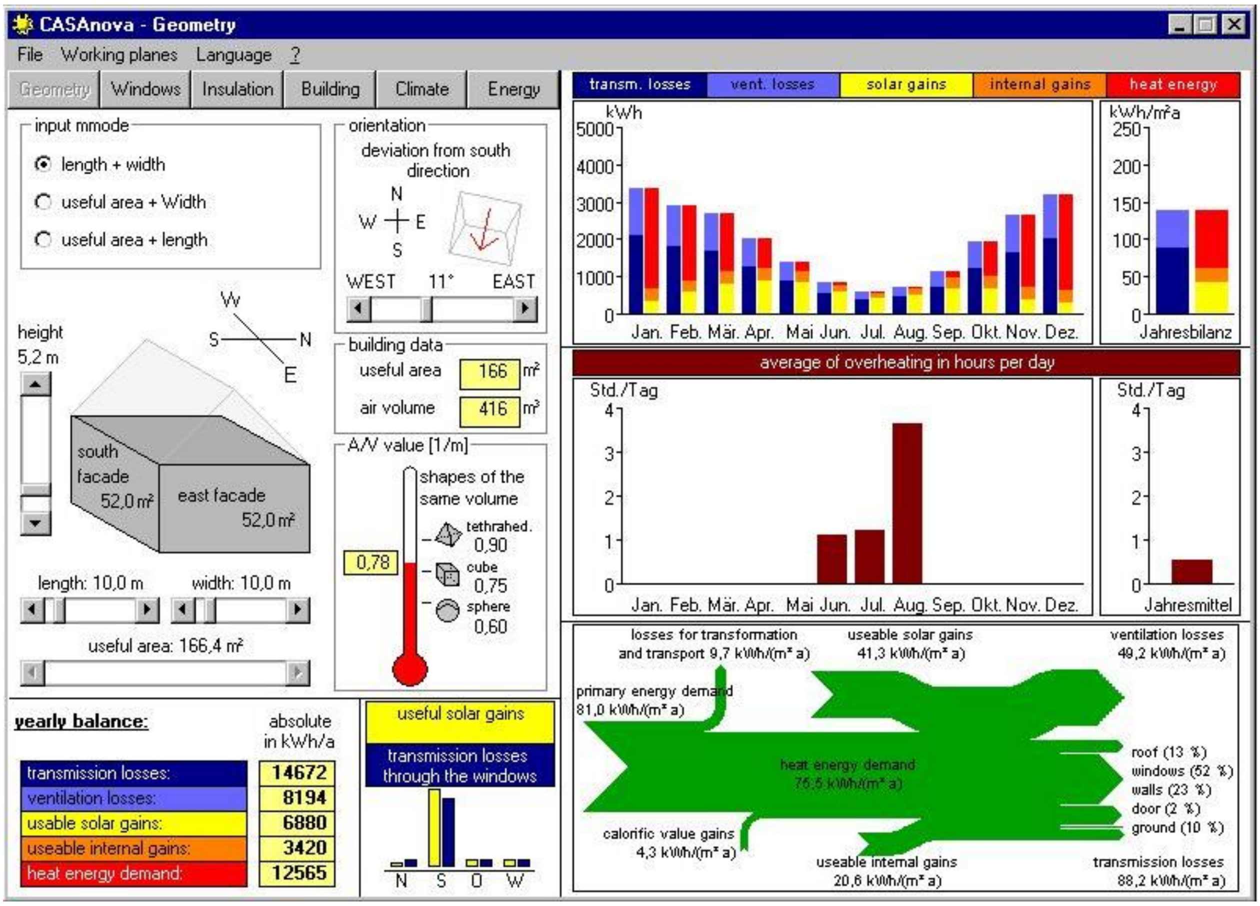


Figure 4.1. Outlook of the CASAnova interface [53]

4.1.2. SIMIEN

SIMIEN is another program for the building performance assessment during the certain period. It was developed by ProgramByggerne (eng. Program Builders) company in Norway in 2013. This program is clearly more advanced than the one previously described (CASAnova), offering a possibility of implementation more precise description of the building and its HVAC systems as the input data. Interface of the SIMIEN can be noticed on the figure 4.2.

SIMIEN, excepting calculations of the power consumption, heating or cooling demand in the building, allows for relatively detailed analysis of the indoor climate, including indoor air quality. In case of climate data a desired city in Norway has to be chosen in SIMIEN interface. Afterwards, it is possible to define various factors influencing the building energy balance, i. a.:

- internal heat gains and their intensity (people, lighting, domestic hot water, electric equipment),
- materials, U-values and orientation of the building facades, windows, window curtains, doors, etc.,
- internal walls, ceilings and floors, ground floor and roof specifications,

- setpoints for desired indoor temperature range in case of ventilation system. Parameters of heating, cooling coils and recovery heat exchanger. SPF for fans in ventilation ducts,
- separated heating and cooling systems, described by i. a. maximum power, desired water inlet and outlet temperatures (if water-based heating or cooling is desired), SPP for water pumps.

Moreover, it is possible to define and manage energy supply sources and/or solutions which are desired to be utilized in the building. The user can select one or few sources among 7 available options (electricity, oil, gas, district heating, biofuel, sun, heat pump) or even define them onself. It is also possible to define the price of energy unit provided in certain way, therefore allowing for cost analysis of designed building and energy supply system. Hence it is further possible to compare different solutions from viewpoint of their profitability. For instance, one can consider two different types of windows (with different U-values) applied in certain building, by comparing their investment cost with the benefits resulted from their utility (lower heat losses and therefore energy costs).

Simulations in SIMIEN are based on dynamic model of the building defined by the user, which changing throughout desired time period with a constant time step of 15 minutes. It is possible to conduct winter, summer or annual simulation. Each type of simulation is usually performed for different aims:

- winter simulation - usually used to find the necessary power for space heating and heating coils in the ventilation system,
- summer simulation - conducted to validate the indoor climate at during summer period and to design of ventilation and space cooling system properly,
- yearly simulation - performed to calculate net energy consumption and energy supplied to the building[51].

Apart from these three basic types of simulation, one can conduct a special simulation in order to check if the building fulfills indicated law regulations. Since SIMIEN is a Norwegian program, it can be done comparing designed building with either Norwegian Building Code (TEK07 or TEK10) or Norwegian Standard for energy efficient buildings (NS 3700:2010 or NS 3701:2012). In newer versions of SIMIEN a comparison of the building with newer versions of these regulations will probably be possible.

Although SIMIEN is more advanced than CASAnova, it is still less precise than newer simulation tools currently available on the market. Another defect of SIMIEN is that this program is available only in Norwegian language. However, that should not be a big problem if the user has

[51] Equalled to net energy consumption plus energy losses inside HVAC systems in the building.

knowledge related with buildings and HVAC systems. In turn, the advantage of SIMIEN is the fact that it is licence - free software.

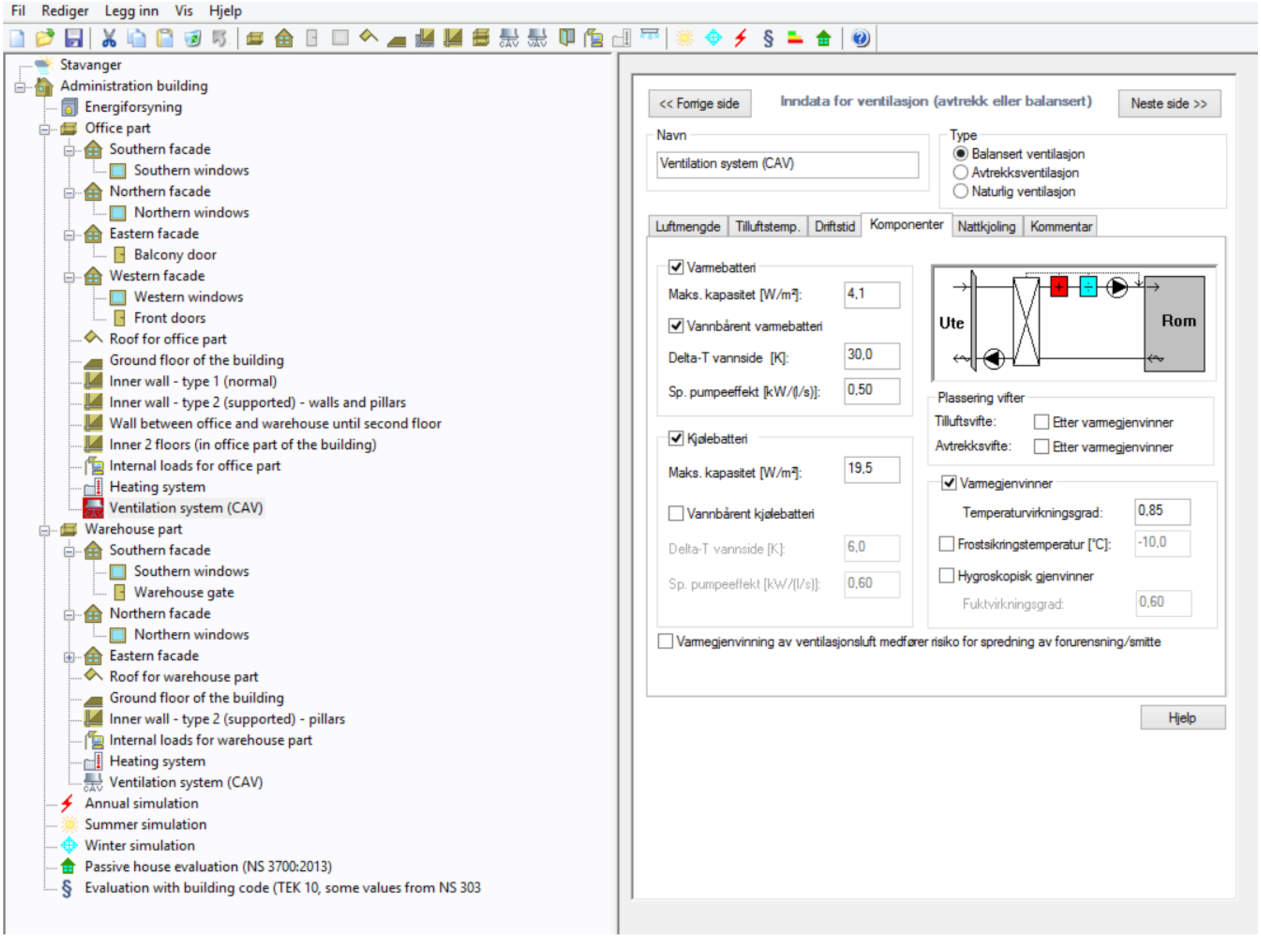


Figure 4.2. Outlook of the SIMIEN interface

4.1.3. EnergyPlus

EnergyPlus is a whole building energy simulation program that engineers, architects and researchers can use to model both energy consumption (for HVAC systems) and water use in buildings. It is actually a stand - alone simulation engine without a user - friendly graphical interface. EnergyPlus reads input data and converts it to output as text files. It was first developed in 1997 by BTO (Building Technologies Office) in USA. A big advantage of EnergyPlus is that it is an open - source software, which theoretically means infinite possibilities of this program in energy efficient buildings area. The program was written in Fortran 90 programming language, however the new versions of EnergyPlus (since version 8.2.0) are written in C++ [56]. Some noteworthy EnergyPlus features include [57]:

- heat balance - based solution of radiant and convective effects that produce surface temperatures, thermal comfort and condensation calculations,
- combined heat and mass transfer model that accounts for air movement between zones,

- sub-hourly, user - definable time steps for interaction between thermal zones and the environment, with automatically varied time steps for interactions between thermal zones and HVAC systems,
- advanced fenestration models, including controllable window blinds, electrochromic glazings and layer - by - layer heat balances that calculate solar energy absorbed by window panes,
- component - based HVAC structure which supports both standard and modern system configurations,
- illuminance and glare calculations for reporting visual comfort and driving lighting controls.

Although EnergyPlus does not have its own interface, there exists **IDF** (Input Data File) **Editor**, which allows for edition of EnergyPlus files (the outlook of IDF Editor is depicted on the figure 4.3). It is possible to use only IDF Editor in the whole building design process, however due to a lack of interface it may be difficult. Hence there exists many programs which co-operate with EnergyPlus.

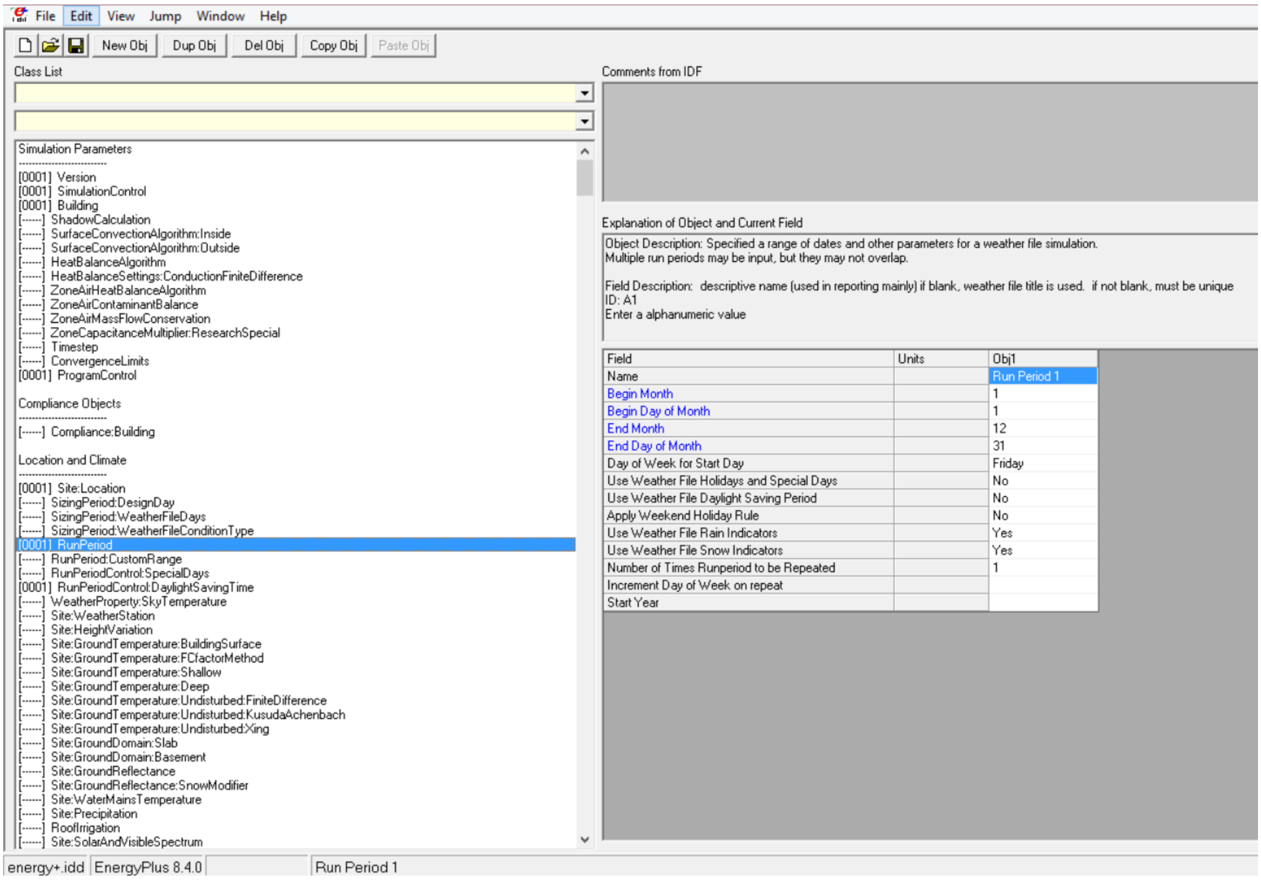
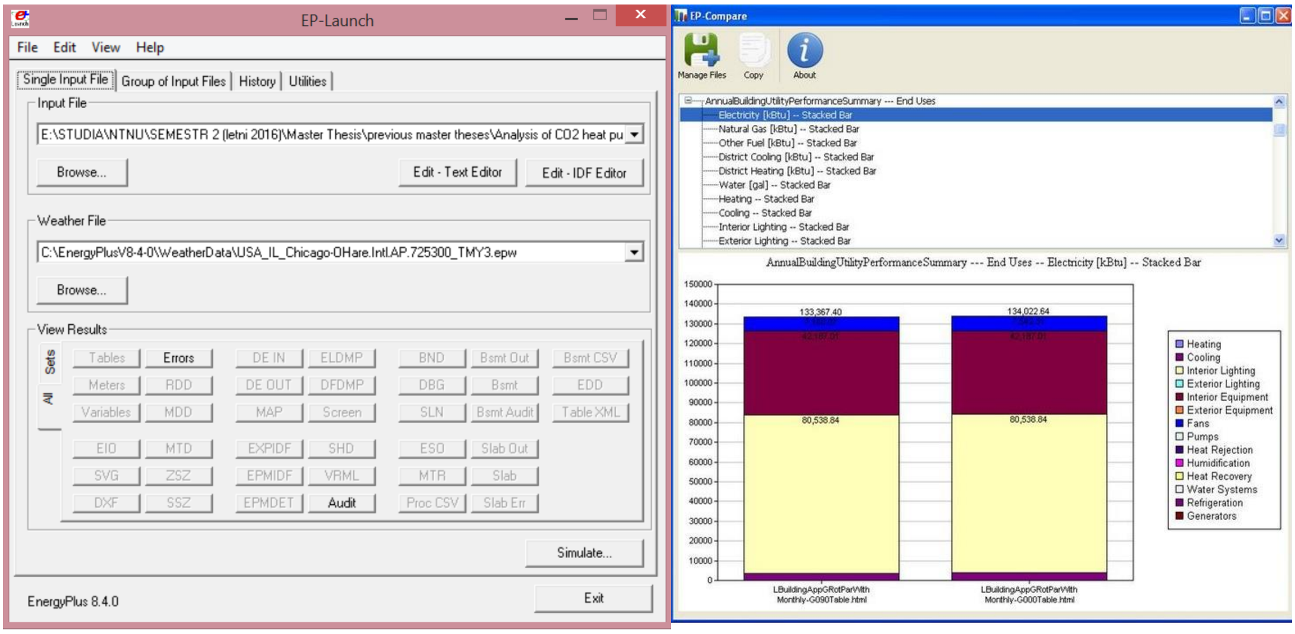


Figure 4.3. IDF Editor

In case of the building geometry description (and specification, to some extent), the most popular is **Google SketchUp**. It is a 3D modeling CAD program, which in overall can be used for many purposes. To allow for co-operation of EnergyPlus and SketchUp, the special **EnergyPlus SketchUp extension** need to be installed. In this plug-in it is also possible to simulate simple IDF files directly from SketchUp and open (or save) geometry of the building as IDF files (instead of SketchUp format files) for eventual further procedure. Beside of IDF Editor, there are also **EP-Launch** and **EP-Compare** extensions to EnergyPlus (their outlook can be seen on the figure 4.4). EP-Launch is for conducting simulations of IDF files. It allows the user to select the input file directly or from a list of recent or sample files. It also allows for easy selection of weather data files[52]. After the EnergyPlus run completes, EP-Launch reports if any errors or warnings occurred. It also presents various output data from performed simulation in diagrams. In turn, EP-Compare allows the user for direct comparison of the tabular results from multiple EnergyPlus simulations.



a **b**

Figure 4.4. Outlook of EnergyPlus extensions; a - EP-Launch, b - EP-Compare [59]

In nowadays EnergyPlus is one of the most popular softwares used throughout the world in the building performance simulation field. There are various programs currently available on the market (or under development) which utilizes EnergyPlus as the simulation engine. Two of them will be depicted in the further part of this chapter. Unfortunately, although EnergyPlus allows for very precise building and its HVAC systems description, in many cases it requires from the user a knowledge about Fortran 90 (or C++). As it was mentioned, EnergyPlus is an open - source software and therefore the user has to solve many issues independently. The program is licence - free and available in English.

4.1.4. OpenStudio

OpenStudio is a cross - platform[53] software to support building energy modelling using EnergyPlus and advanced daylight analysis using Radiance[54]. It was first released in 2008 by NREL

[52] Weather data files for many locations over the world are supported by ASHRAE and are utilized in EPW files.
[53] Able to be used on different types of computers, with different software packages.

in the USA. OpenStudio has a graphical interface, where the user can define various parameters of the building and its HVAC systems in step - by - step way. This interface can be noticed on the figure 4.5.

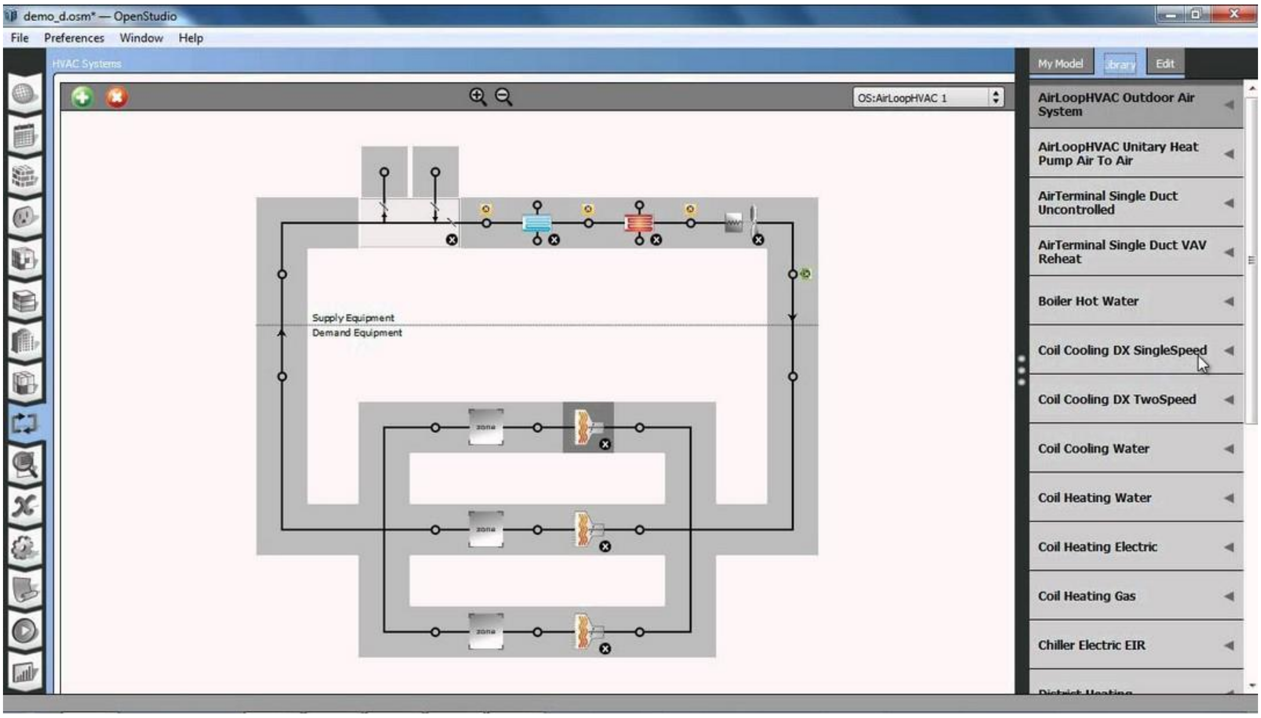


Figure 4.5. OpenStudio interface [53]

It is not possible to define a building geometry in OpenStudio and therefore it has to be loaded from another program. OpenStudio was designed to work with Google SketchUp, because many architects already use SketchUp for building designs. Similarly like for EnergyPlus, to allow for co-operation of SketchUp and OpenStudio it is necessary to install the **OpenStudio SketchUp Plug-in**, which is presented on the figure 4.6 and plays a role of extension to SketchUp. This extension allows users to export geometry[55] of the building (drawn in SketchUp) to OpenStudio or to IDF Editor (and even to import data from IDF editor back to SketchUp). Using this plug-in it is also possible to perform direct EnergyPlus simulation from SketchUp, as it was possible in the EnergyPlus SketchUp extension, therefore the OpenStudio extension is generally more advanced.

The simulation can be also performed in OpenStudio, after loading the SketchUp building geometry and definition of the building components and HVAC systems, utilizing EnergyPlus as the simulation engine. A comfortable feature of OpenStudio is that the user can easily increase its database, adding e. g. a new building insulation material or HVAC component, defined manually. OpenStudio utilizes the same weather data file as EnergyPlus (EPW file). Moreover, the program is licence - free and available in English.

[54] A suite of tools for performing lighting simulations.
[55] In overall, OpenStudio Plug-In allows to define not only geometry of the building but also its building / structure materials, internal and external heat loads, its schedules and HVAC-R systems.

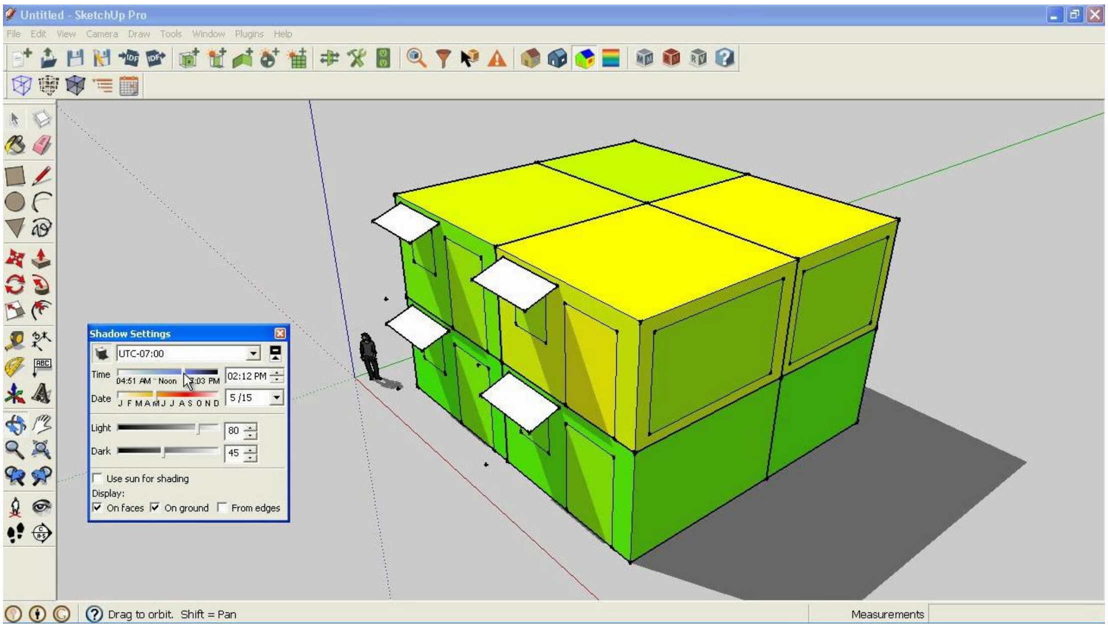


Figure 4.6. Outlook of the OpenStudio SketchUp plug-in [53]

4.1.5. BEopt

BEopt (Building Energy Optimization software) is an American, licence - free program developed in 2006 by the NREL (National Renewable Energy Laboratory), available in English. It was created in order to suport the U. S. Department of Energy in the Building America Program to develop energy efficient solutions for new and existing homes. Hence this simulation tool provides capabilities to evaluate residential buildings from viewpoint of energy efficiency (even up to zero net energy buildings) and to identify cost - optimal solutions. Although it is possible to use the program in analysis of new constructions, it is generally destined for existing building retrofits. Inside BEopt the user can perform detailed simulation - based analysis based on various input data which have to be provided first. In overall, three types of analysis are possible in BEopt, i. e. design, parametric or optimization analysis. In case of design analysis, the interface of the input data in BEopt is divided into three main parts, i. e.:

- building geometry - the user can draw the building using special field inside the program and see a 3D model of designed house in a preview. Unfortunately, only imperial units are available in the software. This part of the BEopt interface can be noticed on the figure 4.7.a,
- miscellaneous options - the user can customize various parameters of the building as it is desired. These options are related with i. a. desired indoor air quality, building orientation and its envelope components descriptions (e. g. thermal mass of walls, etc.), internal heat gains and HVAC systems. Each option regarding the HVAC solutions is depicted with estimated lifetime, investment cost and costs of operation. There are also some unusual options, e. g. the user can decide in which distance neighbours are placed from designed building - it influences the solar

heat gains significantly. Disadvantage of this part of the BEopt interface is that the user can select options among only these which are already specified in the BEopt database, what is shown on the figure 4.7.b,

- other settings - related with utilized climate data file at certain location, type of the terrain (suburban, etc.) and many economic parameters for cost analysis, also regarding to utilized type(-s) of energy sources. The user can choose among four different sources, i. e. electricity, natural gas, oil and propane. This part of the BEopt interface is presented on the figure 4.7.c.

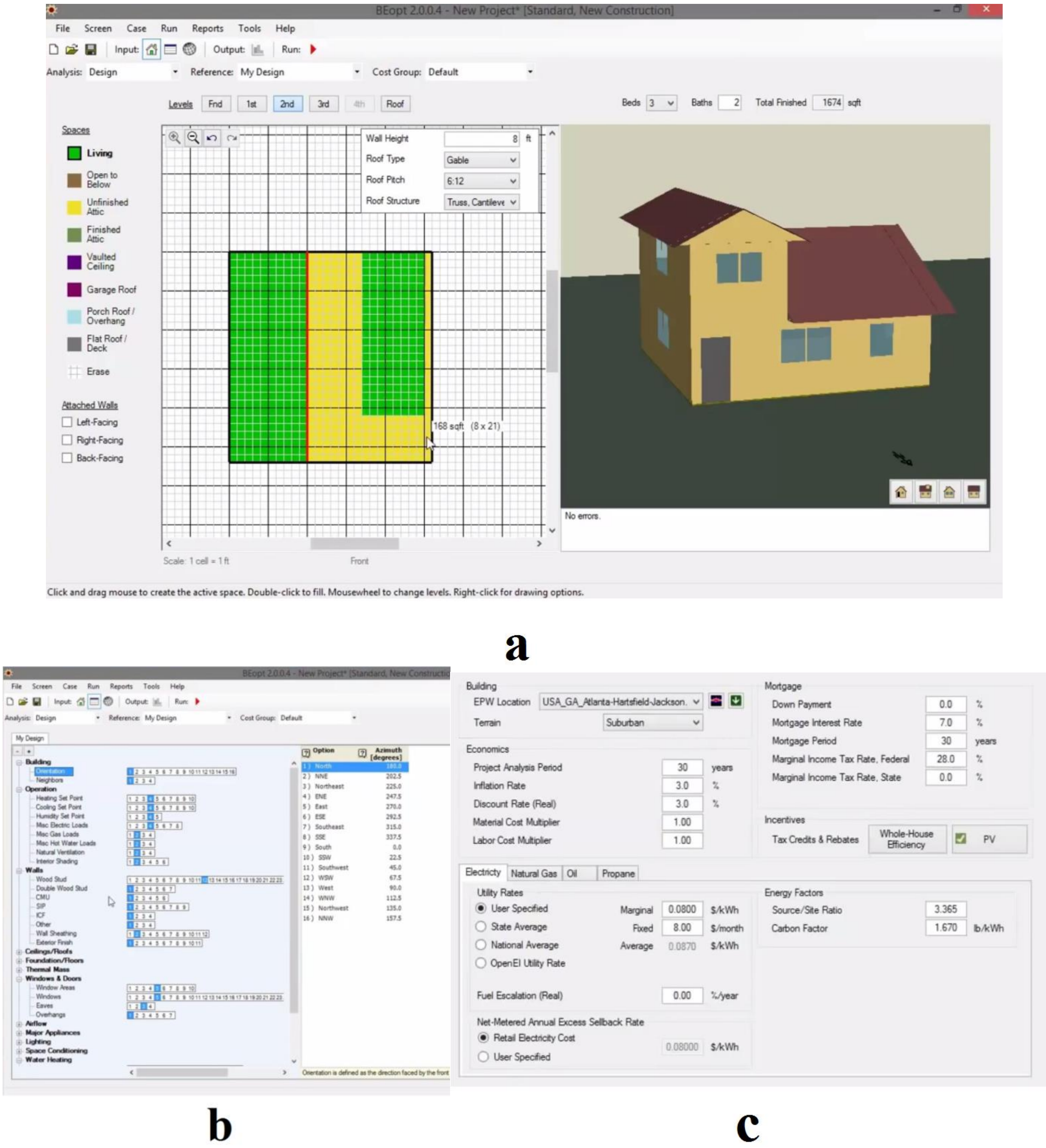


Figure 4.7. Outlook of the BEopt interface; a - house geometry part; b - building envelope components and HVAC options; c - cost and other settings part

The simulation engine which BEopt uses is **EnergyPlus** [58]. After beginning the simulation in BEopt, the program creates XML file first, which is later converted into an IDF file loading in EnergyPlus. Afterwards, the results are presented inside BEopt. For a custom building model, their outlook is presented on the figure 4.8.

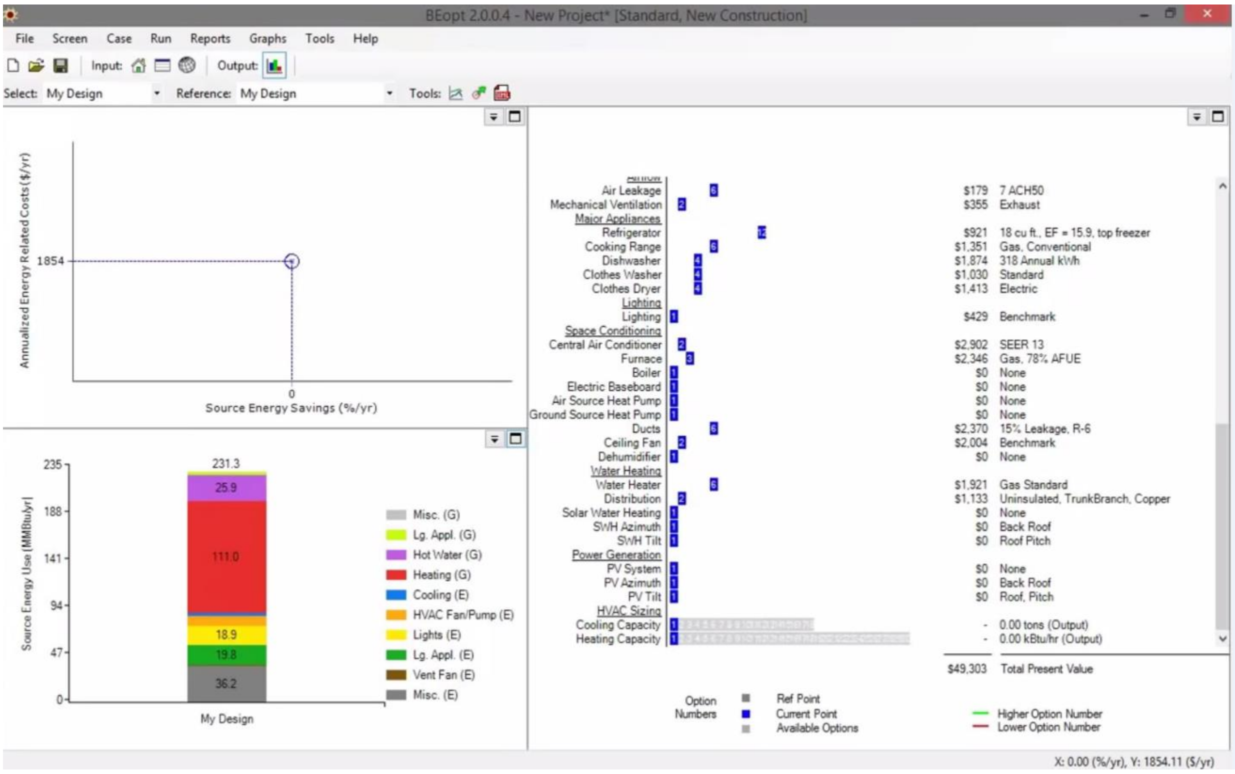


Figure 4.8. Results of performed simulation presented in BEopt

4.1.6. IDA ICE

IDA ICE (Indoor Climate and Energy) is a whole year detailed and dynamic multi - zone simulation tool, developed for the study of both indoor climate as well as the energy demand inside the buildings. The program is equipped with relatively intuitive user interface, however due to many options it may seems difficult. The software is licenced and available in English and few other languages. The basic panel in IDA ICE can be noticed on the figure 4.9.

Inside IDA ICE it is possible to build and simulate both simple and advanced cases. Accuracy of the simulation results is very high - the time step is not constant and it is changing dependent on the outdoor weather changes (in practice, there are approx. 20 ÷ 30 measurements per hour of simulation). The main advantage of IDA ICE is a precise modeling of the building, its systems and controllers, what ensures the lowest possible energy consumption and the best possible occupant comfort.

If to design an energy efficient building in IDA ICE, a design process begins from determination of the building geometry - it is possible to either import a CAD file into IDA ICE or to draw the geometry inside the program. The user must also select a wizard in which the building will be created, however the recommended one is ESBO (Early Stage Building Optimization) Wizard.

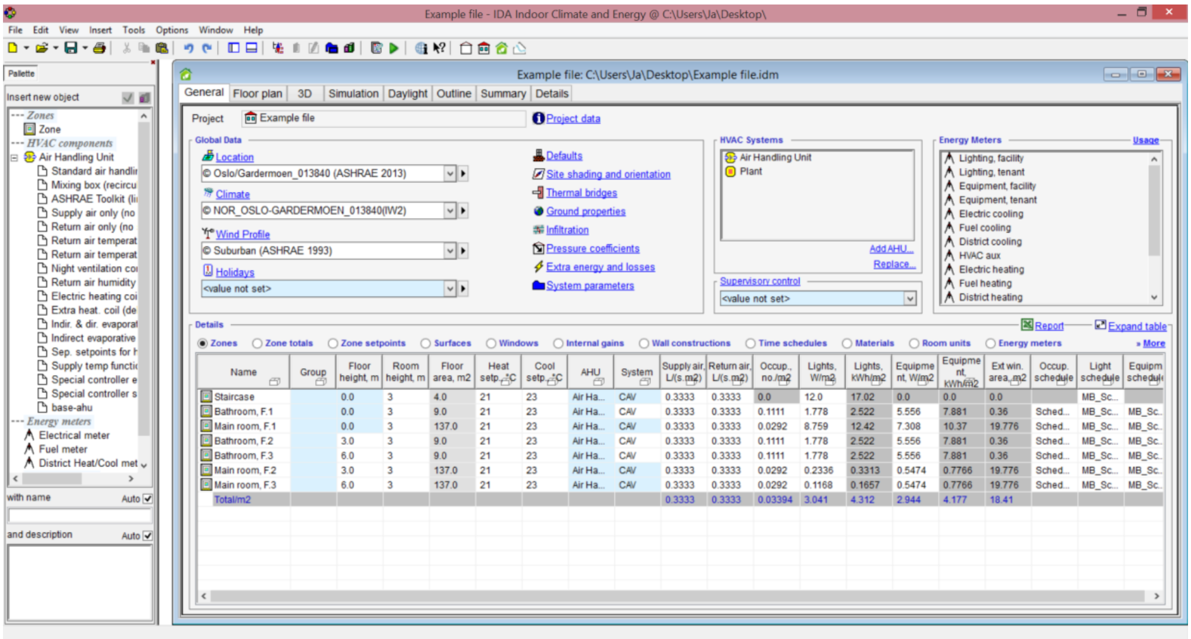


Figure 4.9. The primary window inside IDA ICE simulation tool

Moreover, a desired orientation of the building, side shading, ground properties, infiltration parameters, pressure coefficients, information about thermal bridges (linear or normalized), extra energy losses and many other factors have to be set by the user. Furthermore, IDA ICE has a capability of performing 3D movable visualizations of created building model and to conduct various animations (showing e. g. how the indoor air temperature changes during the set time period) after the simulation process is completed. Example 3D model of the building developed in IDA ICE is shown on the figure 4.10.

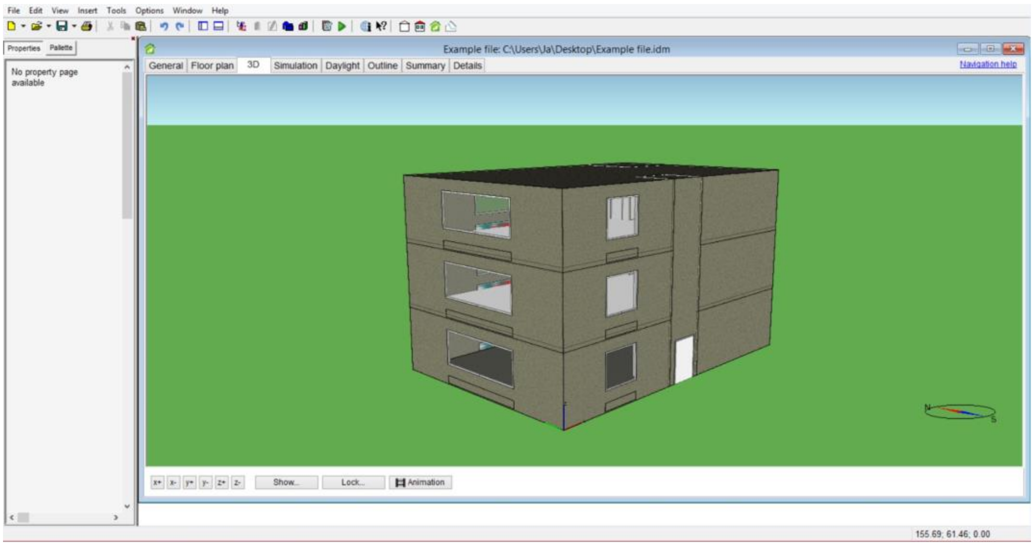


Figure 4.10. 3D preview of designed building model in IDA ICE

After the overall building specification and geometry are determined the user has to define each room in the building, which states a thermal zone. In each thermal zone various parameters can be defined, as it is depicted on the figure 4.11. In such zones the end elements of the heating (cooling) distribution system of the building have to be defined (e. g. underfloor water - based heating panel, water - cooled active beam, etc.).

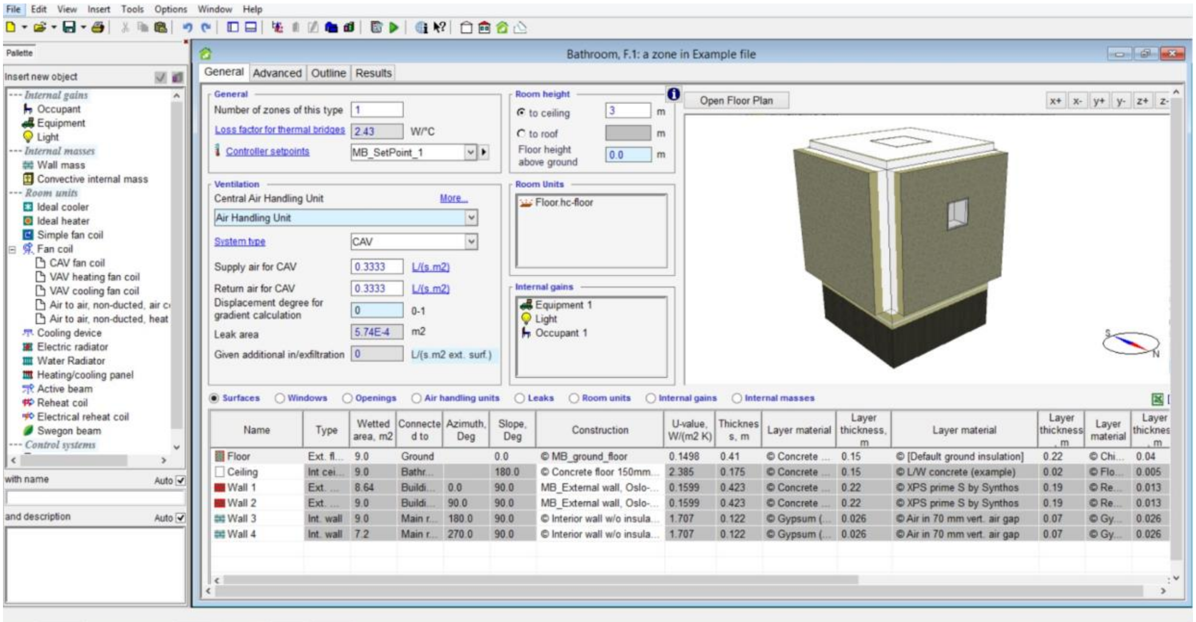


Figure 4.11. Overview of typical options for a typical thermal zone description

After all zones are defined, the energy supply system of the building (heating / cooling) and air conditioning (ventilation) system have to be determined. The user can select various types of air conditioning system and energy supply system and afterwards define their components or use the standard installation templates. In certain cases (to only check the building heating, cooling and electic energy demand) the user can can use default energy supply system and AHU - Air Handling Unit (depicted on the figure 4.12), customizing their parameters to individual needs. In more complicated cases it is possible to either develop the HVAC system from default plant or to use the advanced „ESBO" plant option, to generate complete energy supply system by dragging basic components from the pallette, what is shown on the figure 4.13.

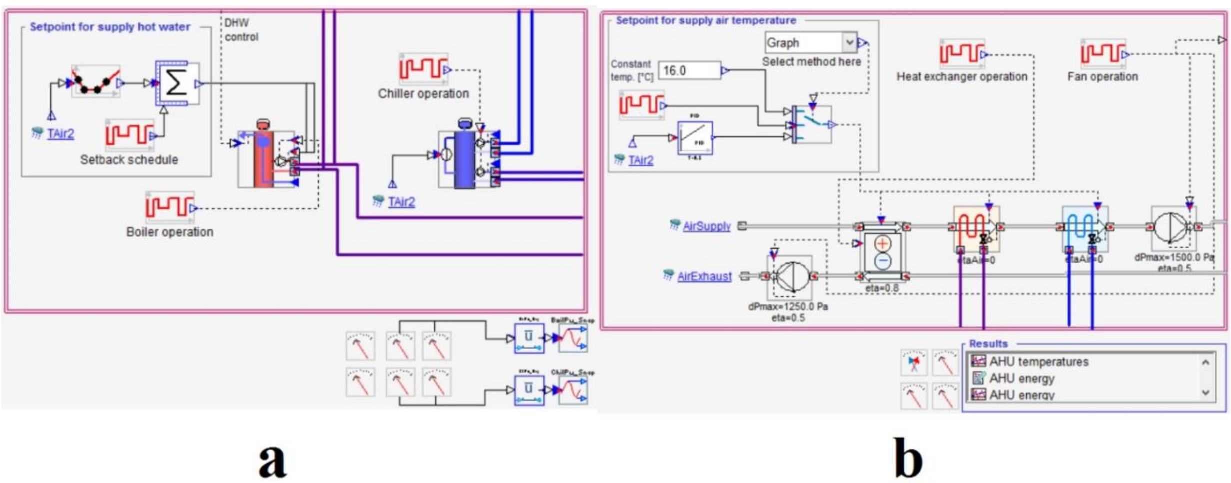


Figure 4.12. Default HVAC systems inside designed building; a - energy supply system; b - AHU

b

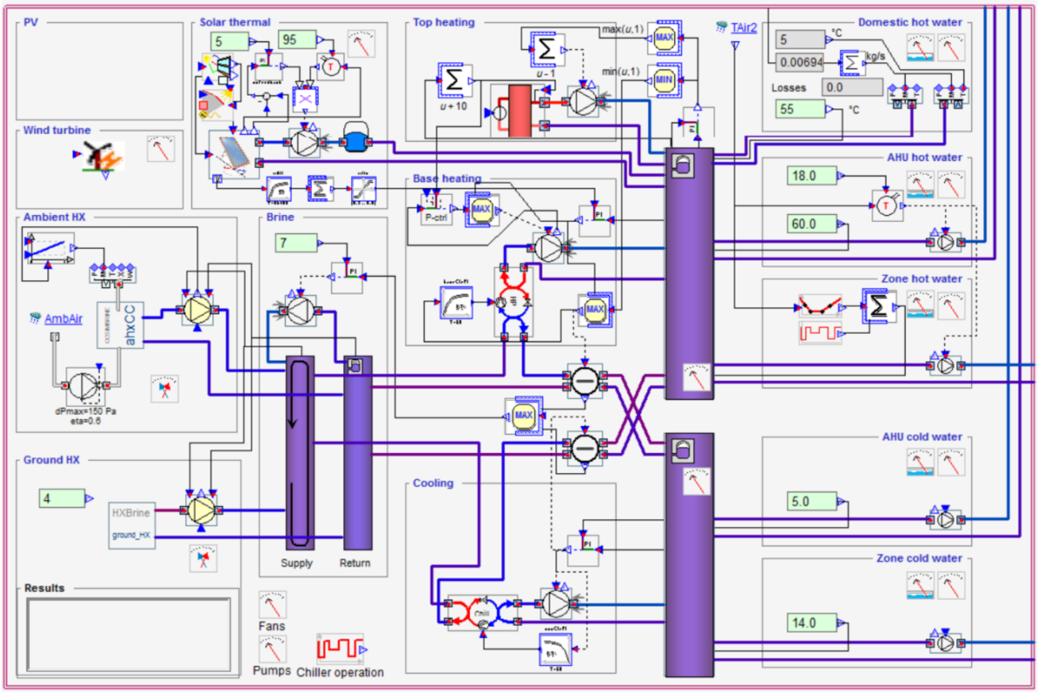
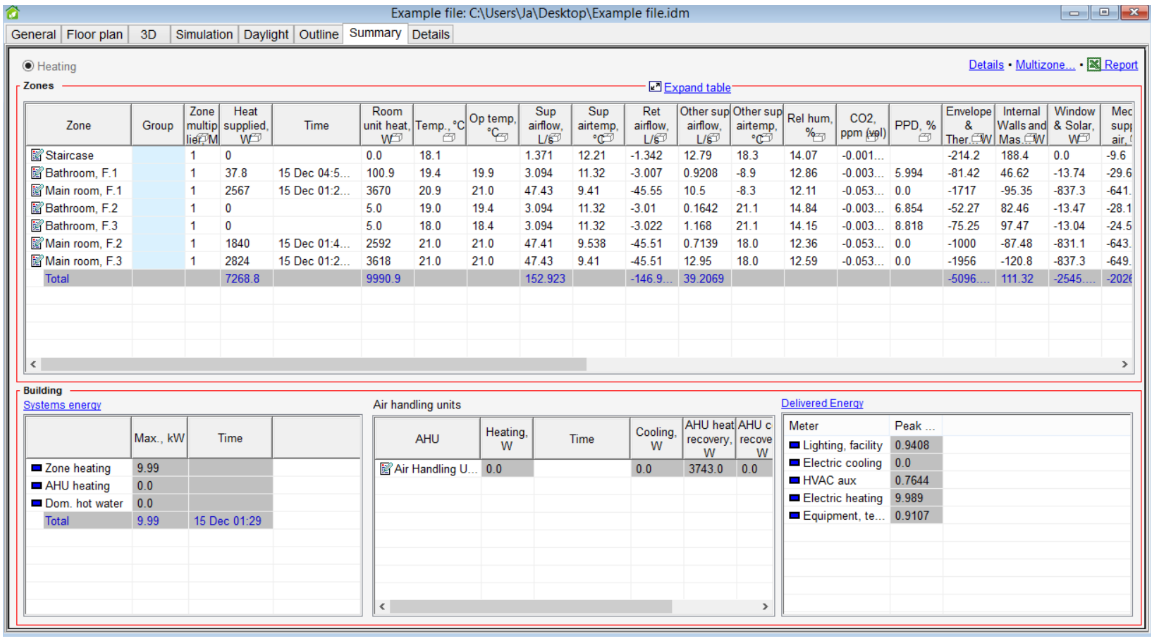
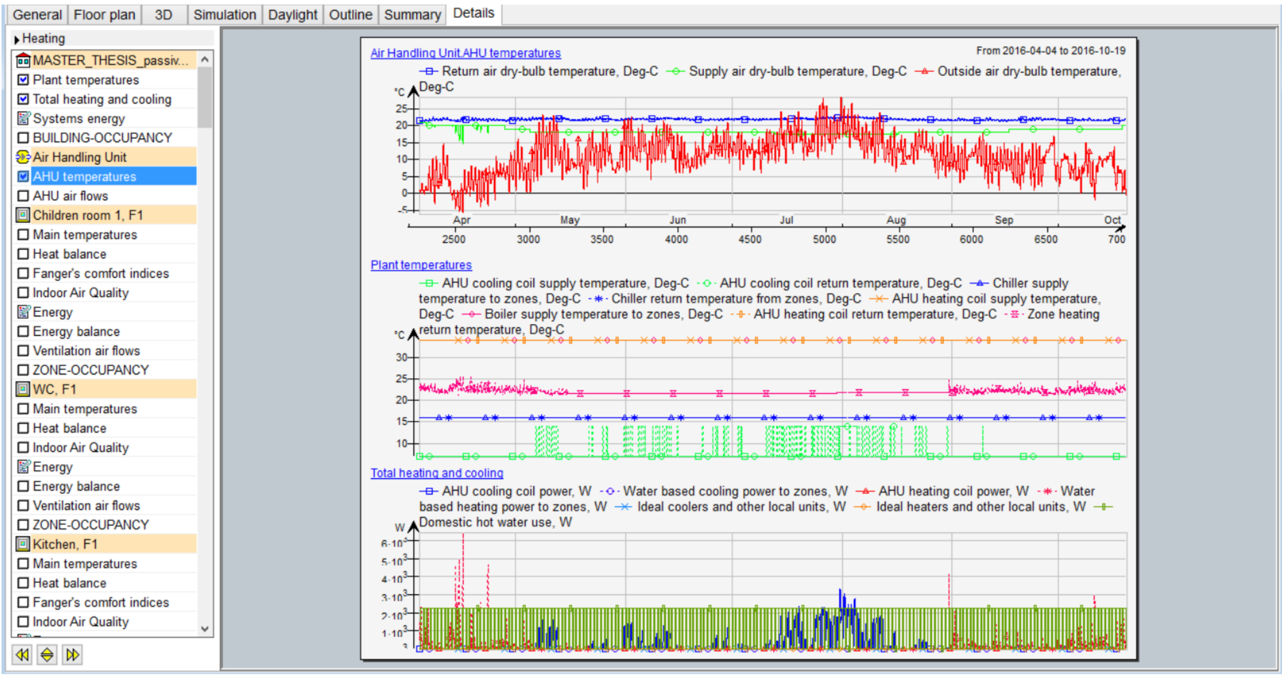


Figure 4.13. Outlook of the ESBO plant template after dragging selected components of energy supply system

When all parameters are properly set the user can start to think about simulation. IDA ICE offers few types of simulations - heating, cooling, advanced and custom. Each of the simulation offers different results, although the most often used is heating simulation. After the simulation is finished, IDA ICE shows miscellaneous diagrams, which can be also viewed in a tabular form, with desired time step (depended of the simulation period, the user can select hourly, daily, weekly or monthly - displayed data). Besides, after performed simulation IDA shows also summary in a tabular form. Example results can be noticed on the figure 4.14.



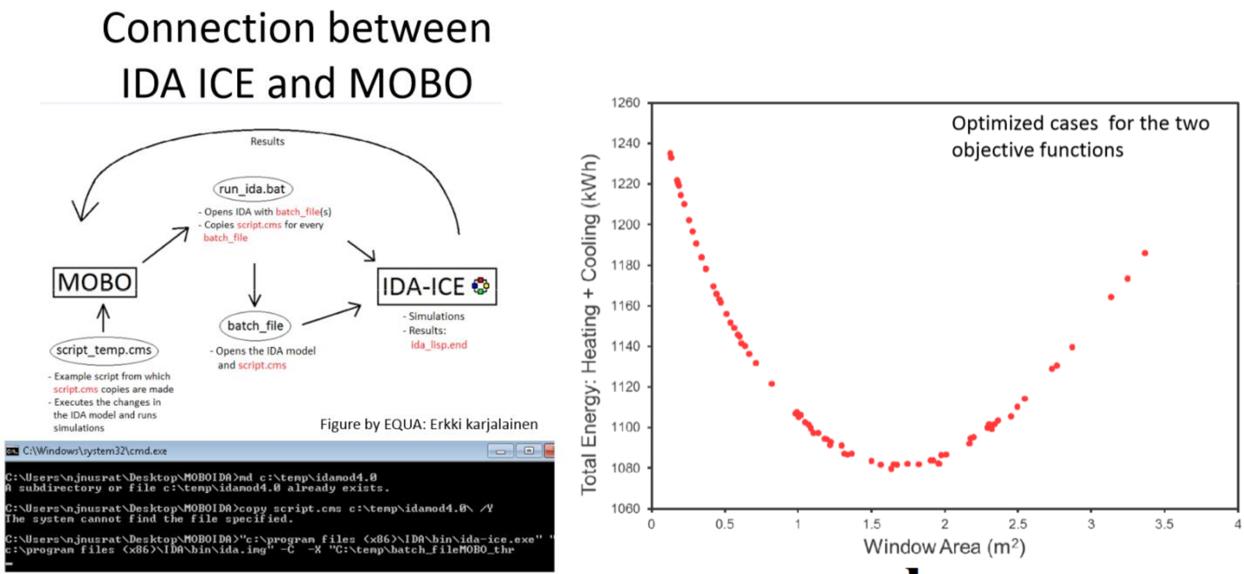
a



b

Figure 4.14. Overview of example results from the IDA ICE heating simulation; a - summary table, b - details

Apart from the basic IDA ICE capabilities, it is also possible to further optimize designed building in IDA ICE utilizing **MOBO** (Multi - Objective Building Optimization) software. For instance, using MOBO tool it is possible to predict the most appropriate window area in order to keep the energy consumption for heating and cooling of the building zone on minimum level. A possible coupling between MOBO and IDA ICE, as well as the example diagram being a result from MOBO optimization in IDA ICE are presented on the figure 4.15.



a b

Figure 4.15. Example relations between MOBO and IDA ICE; a - possible connection of the programs, b - typical result of the optimization

4.1.7. TRNSYS

TRNSYS is an extremely flexible graphically based simulation tool which has been developed in 1975 in the USA. The program is currently maintained by an international collaboration from the United States, France and Germany. Throughout the time 17 versions of the software have been released [60]. TRNSYS, similarly as EnergyPlus is an open - source code program which can simulate all fields of the energy system (including energy efficient buildings and their HVAC systems) except the transport sector. The DLL - based architecture of the program allows users and third – party developers to easily add custom component models, using all common programming languages (C, C++, Fortran, PASCAL, etc.), although the TRNSYS simulation engine is programmed in Fortran. As the name of the program suggests, it is developed mainly to simulate behaviour of transient systems. TRNSYS consists of two main parts:

- simulation engine (called as Kernel) - it reads and processes the input data file, iteratively solves the system, determines convergence and plots system variables;
- extensive library of components - each of the component included in the library models the performance of one part of designed HVAC system. Standard library includes around 150 models of i. a. pumps, multizone buildings, wind turbines, electrolyzers, weather data processors, basic HVAC equipment, etc. All models are developed in such a way that user can modify existing or implement own components.

The main visual interface of TRNSYS is **Simulation Studio** (also called as TRNSYS Studio), shown on the figure 4.16. Inside the studio a user can create models of various HVAC systems by dropping desired components from the library to the workspace, connecting them and setting the global simulation parameters. Simulation Studio saves the project information in the TRNSYS project file (.tpw). In case of the weather data, the program accepts many different format files. Besides Simulation Studio, there are few other relevant TRNSYS subprograms, i. e. [61]:

- TRNBuild - a graphical input program for describing multizone buildings,
- TRNEdit - an editor of the TRNSYS input files,
- Graphical online presentation of the simulation results.

A comfortable feature of TRNSYS is that its users can watch the value of any system variable on an online plot as the simulation progresses (temperatures, mass or volume flow rates, heat transfer rates, etc.). A typical outlook of the online plot containing simulation results from TRNSYS is depicted on the figure 4.17. Moreover, TRNSYS can be easily connected to many other applications which are useful for pre- or post processing of the simulation data (e. g. Microsoft Excel, Matlab, COMIS, etc.). The program is licenced (the user has to pay for its utility) and available in English.

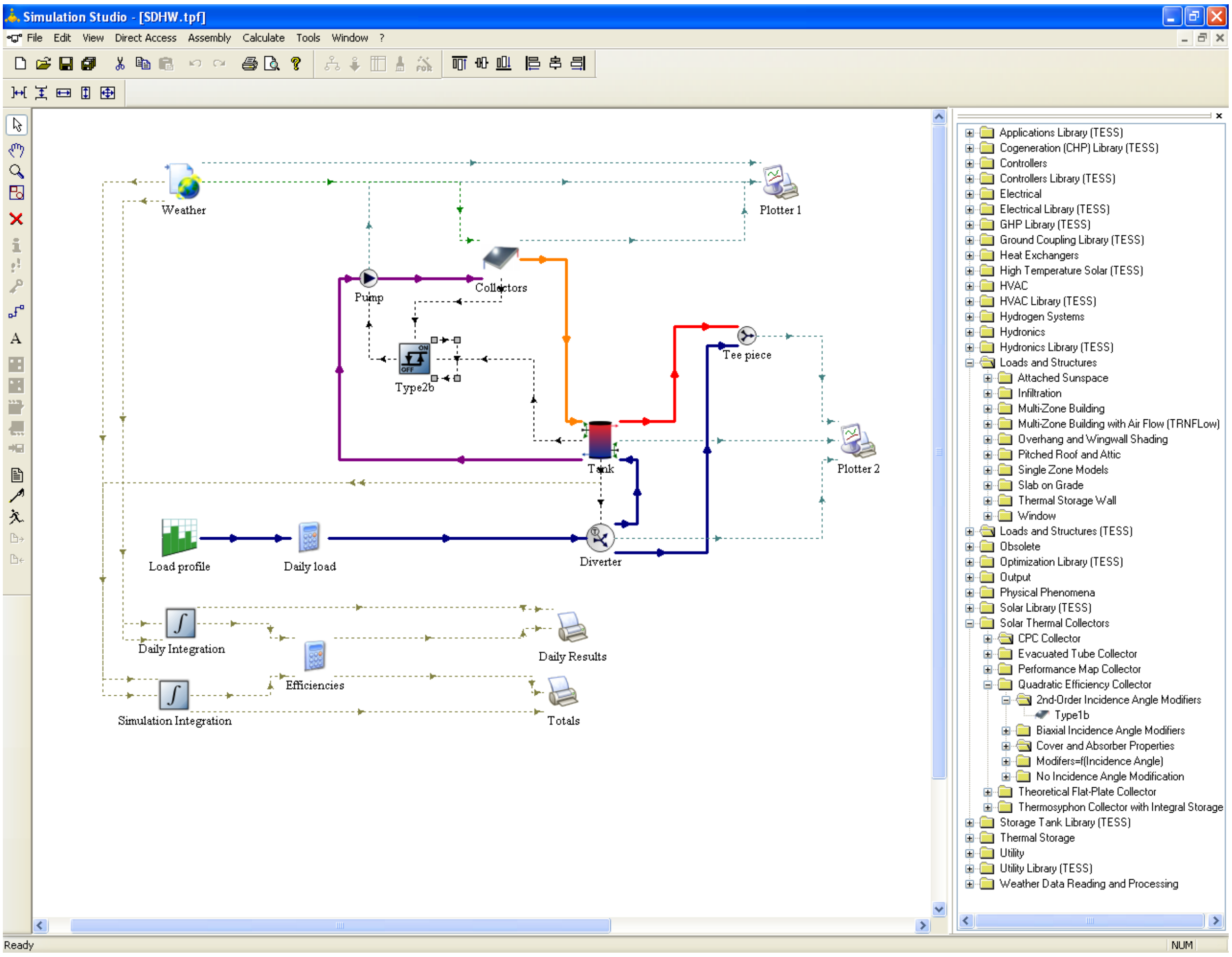


Figure 4.16. Outlook of the TRNSYS interface - Simulation Studio [53]

Figure 4.17. Typical TRNSYS simulation results presentation [53]

4.1.8. Dymola

Dymola is a commercial modeling and simulation environment based on the open Modelica programming language. Although its first version was initially designed in 1978 by H. Elmqvist for his PhD thesis [62], since 2006 Dymola is developed by Dassault Systemes company. The program is available alone or it can be integrated in the computer aided 3D interactive application - CATIA.
Dymola has unique multi - engineering capabilities, what means that models created in the software can consist of components from many engineering domains (data library of Dymola contains i. a. mechanical, electrical, control, thermal, pneumatic, hydraulic, power train, thermodynamic, vehicle dynamic and air conditioning components). The program is utilized mainly in automotive, aerospace, energy and robotic sectors but it can be also useful in many other industries. From viewpoint of the energy efficient buildings Dymola can be used for modeling of the energy supply systems and their components - both simple and non - standard configurations.
Apart from standard Dymola library, it is possible to add many extras containing new components or currently existing models but provided in a simpler way (i. e. their parametric determination can be easier). In modern buildings and their energy supply systems field, it is particularly worth to mention about **TIL Suite** package, developed by TLK-Thermo GmbH in Germany this year (2016). The package contains models of thermal components, systems and thermophysical properties of fluids. It is destined for the stationary and transient simulation of freely configurable thermodynamic systems. Thanks to the substance property library - **TILMedia** (a TIL Suite module) - system simulations can be performed extremely quickly and accurately. Another useful tool developed by TLK-Thermo GmbH is DaVE - a software for e. g. visualizing thermal systems designed in Dymola. For instance, the user can visualize the heat pump cycle on the p - h diagram and observe how the cycle changes throughout desired period of time, with customized speed. In overall, DaVE is a visualization and simulation environment that is suitable for both post-processing and online display of dynamic data sets, supporting various data sources. Hence, it can be also easily combined with other programs than Dymola. Example model dragged from the TIL library, depicting the interface of Dymola can be noticed on the figure 4.18. Besides, the outlook of DaVE can be found on the figure 4.19.

Dymola is a licence program (with English interface), therefore it is not free - available (only demo version can be downloaded easily). NTNU has bought a standard licence for Dymola. In turn, SINTEF has bought either standard licence with other extras described in this section.

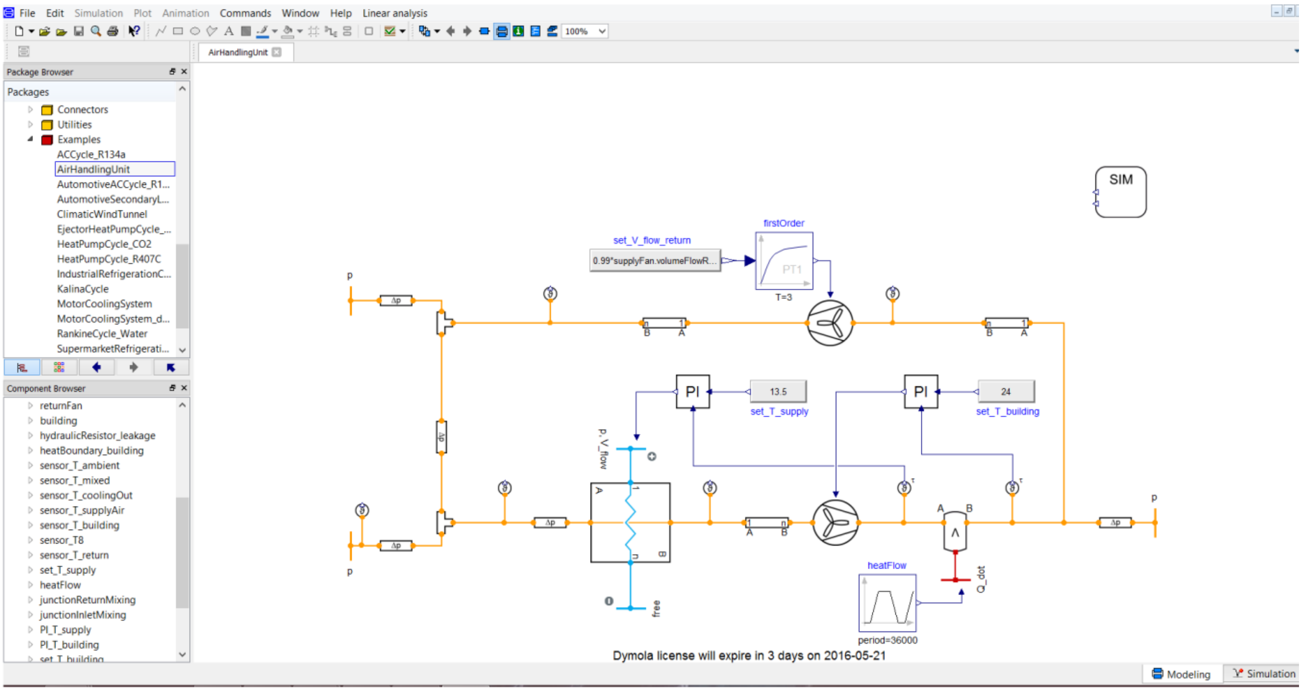


Figure 4.18. Example model of the air conditioner from TIL library in Dymola

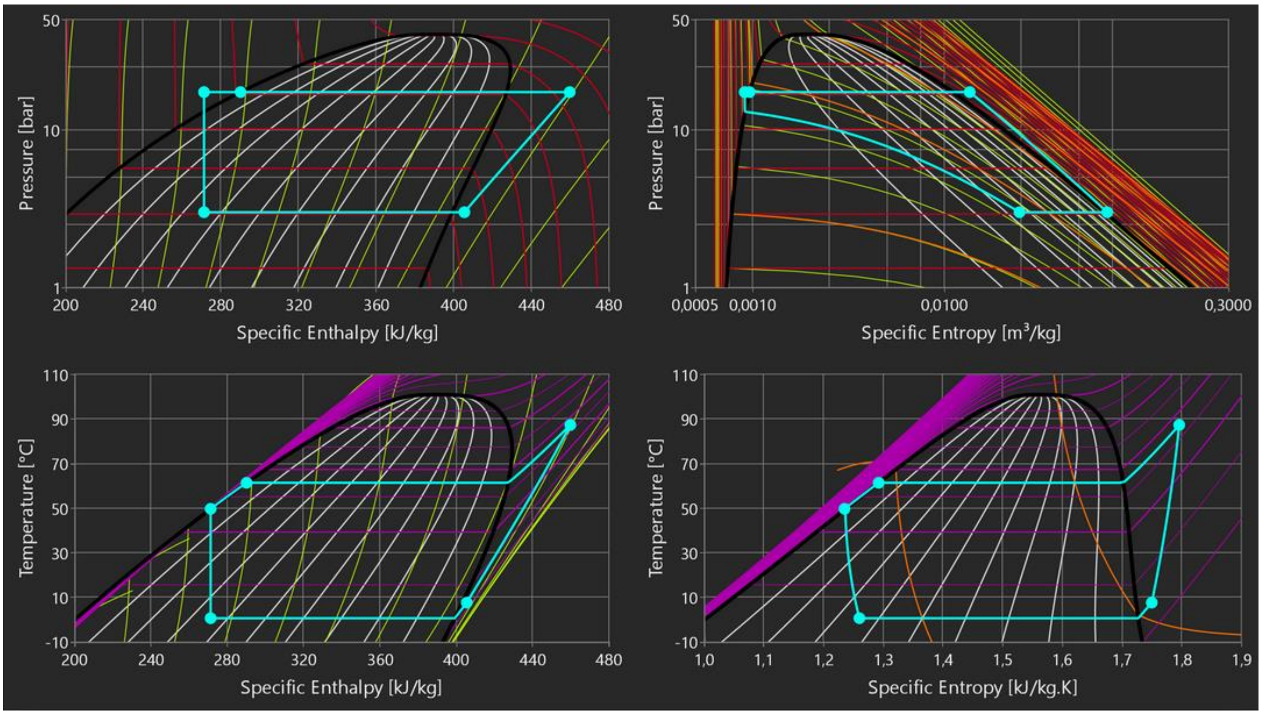


Figure 4.19. The interface of DaVE, currently depicting various state diagrams of a refrigeration circuit [53]

4.1.9. Audytor OZC

Audytor OZC is a Polish licenced software developed mainly for creating energy audits (certificates) of the buildings. It allows for evaluation of the annual building energy (heating) demand and the seasonal heating demand of each zone inside the building. Calculations conducted by the program are in accordance with Polish norms related with required thermodynamic parameters of buildings (i. e. required indoor air temperatures, U-values of fenestration and other building envelope components, etc.). Many Polish companies working in the modern buildings field utilizes Audytor OZC customized to their offerts (e. g. data library of the program is increased by models of devices or components which are sold by only these companies).

Audytor OZC is continuously developing and upgrading. The currently newest version of the program (6.7) allows for the graphical creation of the building model and its later observation in 3D preview in higher quality than previous versions of the program. Moreover, it contains other upgrades, i. a. the software allows for attaching own climate data files and it calculates usable living floor area (pol. PUM - Powierzchnia Użytkowa Mieszkalna [93]). Typical Audytor OZC interface outlook is depicted on the figure 4.20.

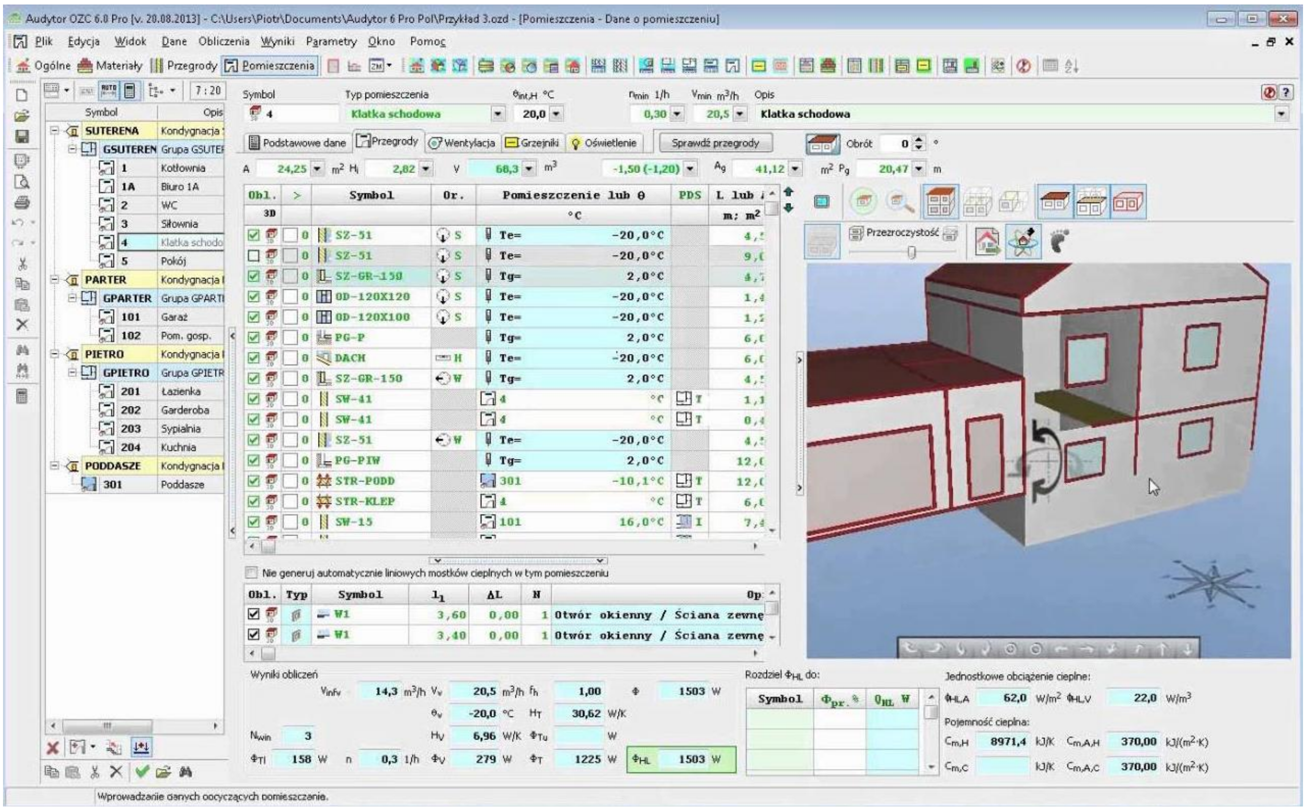


Figure 4.20. Interface of Audytor OZC 6.0 Pro [53]

Audytor OZC can be also used for the moisture analysis of the building envelope components. There is also another program which is similar and can be combined with Audytor OZC, i. e. **Audytor C.O**. It is destined for graphical support in either design of new central heating systems in buildings („C.O." in Polish means „Central Heating" in English) and for regulation of the currently existing installations (e. g. in well - insulated modern buildings). Audytor C.O., similarly as Audytor OZC, is licenced.

4.2. Evaluation of the building simulation tools

4.2.1. Comparison of the programs

The author of this master thesis is looking for a simulation software which fulfills at least 4 following requirements:

- allows for accurate design of the passive house, equipped with modern HVAC systems,
- enables for advanced design of the transcritical air (or ground) to water heat pump model, equipped with tri-partite gas cooler, heating up both domestic hot water and space heating water streams, connected with the thermal storage tank for DHW,
- allows for optimization of desired heat pump system (mainly from viewpoint of the energy optimization - possibilities of decrease of the required electric energy supplied to the heat pump compressor),
- is relatively simple to learn, contains intuitive interface and does not require a knowledge about programming language in which the program is written.

Softwares related with the building simulation performance depicted in the previous section of this chapter will be therefore compared based on criteria related with these requirements above. Moreover, other relevant features of the programs will be taken into account. The benchmarks are thus described below:

- complexity - the variety of options which a program offers, related with the passive house design,
- overall precision - the accuracy level of calculations performed by the software,
- heat pump model accuracy - the way of the heat pump system design in the program. The accuracy is high when a flexible, random construction of the heat pump coupled with water storage tank on a component level is available. Possibility of regulation of the heat pump components work also increases a precision of the model,
- optimization options - the variety of possible ways for energy optimization of designed heat pump and water storage tank,
- independency - if the program can reach its desired goals (conduct the whole building and energy supply system design process, simulation and optimization) alone, without installing other familiar programs (or subprograms),
- interface and 3D options - the intuity level and simplicity of the program interface. Visualization options of the program related with the building design and ability of performing various animations, either of the building and its energy supply system work.

- accessibility - if the program is licence - free or the user has to pay for its utility. If the software requires licence, the price is also taken into account (it is better if the price is low). Moreover, available language of the menu is considered (multi - language menu / English menu / menu in other language),
- reliability - if the program works fluently or it crashes, contains bugs, reports errors, etc.

In the table 4.1, evaluation results of simulation tools characterized in this chapter are presented. Similarly as during rating of different heat distribution systems in chapter 2, grades were numbered from 1 (the worst grade) to 10 (the best grade), with a step of 1. Each criterion has its own weight, dependent on its importance (total weight of criteria is equal to 1). Evaluated programs were named (in tab. 4.1) by numbers (I - CASAnova, II - SIMIEN, III - EnergyPlus, IV - OpenStudio, V - Beopt, VI - IDA ICE, VII - TRNSYS, VIII - Dymola, IX - Audytor OZC).

Table 4.1. Evaluation of simulation programs allowing for estimation of the building and its energy supply system performance

No.	**Criterion**	**Weight of the criterion [$\Sigma = 1$]**	**Evaluated simulation tool**			
			I	**II**	**III**	
1	Complexity	0,15	2	7	9	1 - low; 10 - high
2	Overall precision	0,15	4	6	8	1 - low; 10 - high
3	Heat pump model accuracy	0,17	0	0	8	1 - low; 10 - high
4	Optimization options	0,15	0	2	7	1 - low; 10 - high
5	Independency	0,12	10	10	3	1 - small; 10 - big
6	Interface and 3D options	0,15	6	7	3	1 - weak; 10 - great
7	Accessibility	0,07	10	7	10	1 - low; 10 - high
8	Reliability	0,04	9	9	8	1 - low; 10 - high
No.	**Criterion**	**Weight of the criterion [$\Sigma = 1$]**	**Evaluated simulation tool**			
			IV	**V**	**VI**	
1	Complexity	0,15	7	7	9	1 - high; 10 - low
2	Overall precision	0,15	7	7	8	1 - high; 10 - low

Table 4.1. Continuation

No.	Criterion	Weight of the criterion [Σ = 1]	Evaluated simulation tool IV	V	VI	
3	Heat pump model accuracy	0,17	5	4	7	1 - high; 10 - low
4	Optimization options	0,15	4	5	6	1 - low; 10 - high
5	Independency	0,12	2	6	8	1 - big; 10 - small
6	Interface and 3D options	0,15	5	7	9	1 - difficult; 10 - easy
7	Accessibility	0,07	10	10	5	1 - big; 10 - small
8	Reliability	0,04	7	8	7	1 - low; 10 - high

No.	Criterion	Weight of the criterion [Σ = 1]	Evaluated simulation tool VII	VIII	IX	
1	Complexity	0,15	8	4	8	1 - high; 10 - low
2	Overall precision	0,15	8	8	7	1 - high; 10 - low
3	Heat pump model accuracy	0,17	8	10	5	1 - high; 10 - low
4	Optimization options	0,15	7	7	3	1 - low; 10 - high
5	Independency	0,13	6	4	9	1 - big; 10 - small
6	Interface and 3D options	0,15	6	6	8	1 - difficult; 10 - easy
7	Accessibility	0,07	5	5	5	1 - big; 10 - small
8	Reliability	0,04	7	7	7	1 - low; 10 - high

4.2.2. Choice of the most appropriate software and justification

Based on performed comparison, analyzed simulation tools gained final grades presented in the table 4.2.

Table 4.2. Final grades of evaluated simulation programs

Number of the program	I	II	III	IV	V	**VI**	VII	VIII	IX
Assigned grade	4.16	5.43	6.75	5.50	6.33	**7.61**	7.05	6.53	6.52

As it can be seen in tab. 4.2, simulation tool with the highest grade (7.61) is **IDA ICE**. It gained the highest score mainly due to its wide capabilities in case of the passive house model development and very good precision of calculations during simulations (a time step of calculations is automatically adjusted to the actual changes of the ambient climate, therefore the time step is not constant). Furthermore, it has excellent (compared to other programs analyzed in the chapter) interface, which is intuitive, allows for 3D preview of designed building and for miscellaneous 3D animations of gained simulation results. Even though there are many options to be defined, the program is not confused and thus it is relatively easy to learn. Another relevant advantage of IDA ICE is its independency - the program does not require any other subprograms or tools to be installed, to reach its main goal, which is a design of the building and its HVAC systems. Furthermore, there exists one optional tool (MOBO) which can increase the optimization capabilities of IDA ICE.

Apart from IDA ICE, the second place with a score of 7.05 is taken by TRNSYS. An author of this master thesis agrees that selection of this software for further work in the thesis would be also a good choice, however due to more complicated interface, non licence - free nature of the program and limited time for preparation of the thesis, author finally gave up to use this program. Besides, the third place is taken by EnergyPlus, with a score of 6.75. However, due to the lack of interface and mainly due to a necessity of understanding the Fortran 90 programming language in order to reach goals desired in this master thesis, author finally resigned from this program too.

To sum up, from viewpoint of this master thesis needs, **IDA ICE** seems to be the best choice among the programs described in this chapter and therefore the author selected this program for further analysis work. However, the author felt too small possibilities of IDA ICE in case of desired heat pump model (with tri-partite gas cooler). Furthermore, IDA has not great possibilities from viewpoint of the heat pump work regulation and steering. Hence (although the stratified hot water storage tank can be modelled in IDA with good accuracy) the author decided to use also second program - **Dymola**, for more accurate development of the heat pump system.

5. Review of considered building

An object of considerations included in this master thesis is the passive residential building. Unfortunately, the building which is expected to be taken into account is already designed in low energy standard. Thus, first it has to be upgraded to passive house before optimization of the heating and cooling system contained in this building, what is the main goal of this thesis.

In the first part of this chapter, both low energy and passive building definitions will be depicted. Current requirements for low energy and passive residential buildings will be compared, in order to capture differences between these two standards. Law regulations in three different countries - Norway, Germany and Poland will be presented, as the author of this master thesis is from Poland and such comparison may be interested for both NTNU and author's home university (Gdańsk University of Technology). In the second section of the chapter, current (low energy) state of the analyzed building will be described, together with an overview of climate conditions (existing in its destination location), applied ventilation system and internal heat gains. Afterwards, detailed description of what should be done in aim to renovate existing low energy building to passive house will be placed. Next, aforementioned upgrade will be performed in IDA ICE and passive house state will be depicted. Afterwards, achievements of both previous state of low energy building, re - designed state of low energy building and developed passive house will be presented and compared.

5.1. Overview of chosen law regulations about low energy and passive residential buildings, in Norway, Germany and Poland

Low energy building (germ. Niedrigenergiehaus, pol. Budynek niskoenergetyczny) means energy efficient building, which have to fulfill certain requirements regarding to heat losses, energy demand and energy supply system of the building. There is no global definition for low energy house - national standards can vary considerably, therefore low energy developments in one country may not meet requirements in another. In Norwegian context, an approach called **passive energy design** has been proposed to fulfill low energy conditions [9]. In this approach five steps should be followed in the consecutive order:

- reduction of the heat loss from the building. This would include choosing low U-value building elements and heat recovery of the ventilation air;
- reduction of the electric energy consumption by choosing energy efficient appliances and lighting;
- location, orientation and design of the building and its facades in such a way to maximize free solar energy utility;

- choice of advanced control system that can allow the user for comprehensive and accurate control of the building energy demand and usage patterns;
- selection of appropriate energy source that matches the building energy demand, after all of the aforementioned steps have been taken.

Schematically, all steps can be shown in a form of pyramid, what is presented on the figure 5.1.

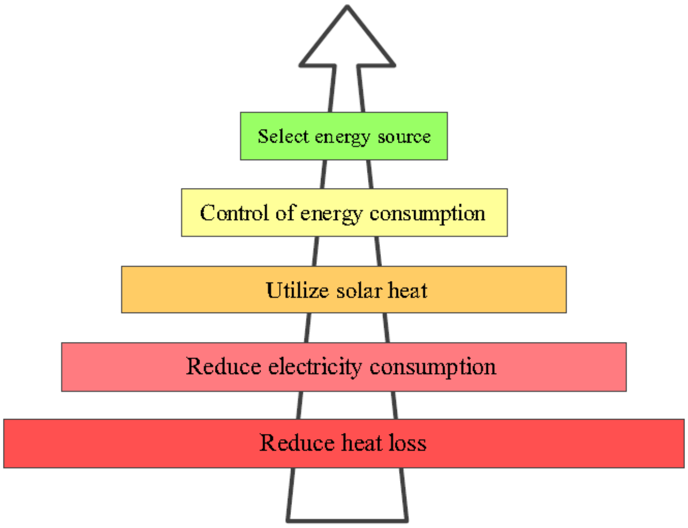


Figure 5.1. Passive energy design pyramid [9]

More energy efficient version of low energy residential building is **passive house** (germ. passivhaus, pol. dom pasywny), also called as ultra - low energy building. The main difference between low energy and passive buildings is that passive building does not require an active heating system to achieve a comfortable room temperature. From viewpoint of heating needs, reheating of the air from anyway needed ventilation system is sufficient, therefore passive buildings are not really more expensive than low energy houses[56]. In other words, passive house requires much less energy for heating and cooling purposes than low energy house. Although passive standard is mostly applied to new built modern buildings, sometimes it is also used for refurbishments of existing buildings.

In **Norway** the most relevant current standards for low energy and passive buildings are TEK 10 (Norwegian building code), NS 3700:2013 (Criteria for passive and low energy residential buildings) and NS 3031:2014 (Calculation of energy performance of buildings - methods and data). The new version of the building code - TEK 15 was supposed to be published in January 2015, however due to its complexity and desire to perform it more carefully, release date has been officially extended to January 2017. After introduction of TEK 15 on the market, all new buildings will have to be built in at least low energy standard.

[56] Construction and building materials for passive house may be more expensive than for low energy building but this is compensated by much more cheaper space heating system.

Nowadays, two types of low energy buildings can be distinguished in Norway, i. e. more advanced class 1 and less energy efficient class 2. Based on Norwegian TEK 10 and NS 3700:2013 requirements, total net energy needs of any residential building should not exceed 120 $\frac{\mathrm{kWh}}{\mathrm{m^2yr}}$. In turn, the maximum net energy supplied for space heating needs in low energy buildings should not be higher than 30 $\frac{\mathrm{kWh}}{\mathrm{m^2yr}}$ (for class no. 1) or 45 $\frac{\mathrm{kWh}}{\mathrm{m^2yr}}$ (class no. 2). Meanwhile, the total energy used for heating purposes in passive buildings cannot exceed 15 $\frac{\mathrm{kWh}}{\mathrm{m^2yr}}$ (10 $\frac{\mathrm{W}}{\mathrm{m^2}}$ [57]), what is two (three) times less than for low energy standard. As it was partly mentioned previously, low energy and passive buildings (as highly energy efficient buildings) have to follow strict requirements related with i. a. heat losses from the building. Thus, total U-values of the building envelope can be found in the table 5.1. Besides, U-values[58] for each part of the building envelope are placed in the table 5.2. Other important restrictions can be noticed in the table 5.3.

Table 5.1. Maximum allowed average heat transfer coefficients for the building envelope, based on NS 3700:2013

Type of the building		$U_{tr,inf}$ $[\frac{W}{m^2K}]$[59]		
		Residential building with A_{fl}[60] $< 100\ m^2$	Residential building with $100\ m^2 \leq A_{fl} < 250\ m^2$	Residential building with $A_{fl} \geq 250\ m^2$
Passive building		0,53	0,48	0,43
Low energy building	Class 1	0,70	0,65	0,55
	Class 2	0,93	0,83	0,68

Table 5.2. Allowed U-values for the building envelope parts, based on NS 3700:2013

Part of the building	U $[\frac{W}{m^2K}]$		
	Passive building	Low energy building	
		Class 1	Class 2
External wall	0,10 ÷ 0,12	0,15 ÷ 0,16	0,15 ÷ 0,16
Roof	0,08 ÷ 0,09	0,10 ÷ 0,12	0,10 ÷ 0,12
Ground floor	0,08	0,10 ÷ 0,12	0,10 ÷ 0,12
Window	≤ 0,80	≤ 1,20	≤ 1,60
External door	≤ 0,80	≤ 1,20	≤ 1,60

Table 5.3. Chosen relevant regulations for passive and low energy buildings, based on NS 3700:2013 and NS 3031:2014

[57] Maximum power of the space heating system which is allowed to be installed in the building, per square meter of the heated part of BRA.

[58] In overall, U-values of the building parts are often dependent on desired indoor air temperature. However, in this chapter all depicted U-values are correlated with indoor temperatures above 17^0C, as these tmeperatures are assumed in the residential building considered in this master thesis.

[59] Total U-value for transmission and infiltration heat losses, calculated by adding transmission losses through the building envelope (ground floor, external walls, roof) and to unheated zones, and infiltration losses.

[60] Heated part of BRA.

Feature	Passive building	Low energy building	
		Class 1	Class 2
Normalized thermal bridge, ψ_n	$\leq 0{,}03\ \frac{W}{m^2K}$	$\leq 0{,}05\ \frac{W}{m^2K}$[61]	$\leq 0{,}05\ \frac{W}{m^2K}$
Efficiency of recovery heat exchanger, η	$\geq 80\%$	$\geq 70\%$	$\geq 70\%$
SFP factor (ventilation)	$\leq 1{,}5\ \frac{kW}{\frac{m^3}{s}}$	$\leq 2\ \frac{kW}{\frac{m^3}{s}}$	$\leq 2{,}5\ \frac{kW}{\frac{m^3}{s}}$
Allowed air leakage, n_{50}[62]	$\leq 0{,}60\ h^{-1}$	$\leq 1{,}0\ h^{-1}$	$\leq 3{,}0\ h^{-1}$

In **Germany**, total energy needs for all residential buildings should not exceed $120\ \frac{kWh}{m^2yr}$ [66], similarly like in Norway. For low energy buildings, maximum allowed heating energy consumption depends on the type of building - $70\ \frac{kWh}{m^2yr}$ for detached houses and $60\ \frac{kWh}{m^2yr}$ for terraced houses [4]. This values are however based on German thermal performance standard (germ. Energieeinsparverordnung, EnEV), first published in 2002, so nowadays it is generally assumed that modern low energy houses use around half of energy mentioned in this standard. Thus, typical values of max. net heating energy are in the range of $20 \div 70\ \frac{kWh}{m^2yr}$ [65]. New version of EnEV (released in 2012) specify max. net heating energy to be approx. $35\ \frac{kWh}{m^2yr}$ [67]. In turn, for passive houses max. space heating energy demand equals $15\ \frac{kWh}{m^2yr}$ or $10\ \frac{W}{m^2}$ [66].

After introduction of EnEV standard in 2002, low energy standard was applied to all new emerging buildings. Another German standard for energy requirements of residential buildings is DIN V 18599, published in 2009 [67]. Passive House Institute in Darmstadt (where the first passive house in the world was built in 1990 [66]) distinguish all features, which a good passive house should have, e. g. criteria for thermal protection, dependent on the climate type. Average U-values for low energy residential buildings are presented in the table 5.4. Meanwhile, U-values and some other law regulations for passive buildings in both cold and cool - temperate climate zones[63] are shown in the table 5.5.

[61] For non-timber building envelopes another values are allowed (NS 3031:2014).

[62] Air tightness of the building, measured by air infiltrated through its envelope at pressure difference of 50 Pa between the building interior and ambient

[63] Both climate zones are characterized, because main part of Germany and Poland lays in cool - temperate climate but most of Norway lays in cold climate.

Table 5.4. Allowed U-values for the building envelope parts in low energy buildings, based on EnEV 2002

Name of the envelope component	U $[\frac{W}{m^2K}]$
External masonry wall	$< 0,25$; $120 \div 180$ mm insulation
External timber - framed wall	$< 0,20$; $200 \div 250$ mm insulation
Roof	$< 0,15$; $250 \div 300$ mm insulation
Internal wall between heated and unheated areas	$< 0,30$; $80 \div 120$ mm insulation
Window and external door	$< 1,30$; double glazing with inert gas (argon or crypton)

Table 5.5. Required heat transfer coefficients and other important factors for passive buildings in cold and cool - temperate climate zones [66]

Feature	Value	
	Cold climate	Cool - temperate climate
U-value for external wall	$\leq 0,12\ \frac{W}{m^2K}$	$\leq 0,15\ \frac{W}{m^2K}$
U-value for roof	$\leq 0,12\ \frac{W}{m^2K}$	$\leq 0,15\ \frac{W}{m^2K}$
U-value for ground floor	$\leq 0,12\ \frac{W}{m^2K}$	$\leq 0,15\ \frac{W}{m^2K}$
U-value for the wall window and external door	$\leq 0,65\ \frac{W}{m^2K}$	$\leq 0,85\ \frac{W}{m^2K}$
U-value for the roof window	$\leq 0,80\ \frac{W}{m^2K}$	$\leq 1,10\ \frac{W}{m^2K}$
Efficiency of recovery heat exchanger	$\geq 80\%$	$\geq 75\%$
Air tightness of the building frame, n_{50}	$\leq 0,60\ h^{-1}$	$\leq 0,60\ h^{-1}$
Linear thermal bridge, ψ	As low as possible ($\psi \leq 0,01 \frac{W}{mK}$)	

Requirement about maximum total net energy for any residential, single family building in **Poland** is similar to Norway and Germany (i. e. $120\ \frac{kWh}{m^2yr}$ [72]). In case of low energy residential buildings two types can be distinguished (similarly like in Norway), i. e. normal low energy building (pol. budynek energooszczędny) and more advanced, active low energy building (pol. budynek energooszczędny aktywny). Maximum net energy supply for heating needs should not exceed $70\ \frac{kWh}{m^2yr}$ (normal LE[64] buildings) and $30 \div 40\ \frac{kWh}{m^2yr}$ (active LE buildings) [75]. For passive houses, this value should be equal or lower than $15\ \frac{kWh}{m^2yr}$.

Currently used Polish Building Code (pol. Polskie Prawo Budowlane) was announced by the Government in 1994, although there are no special regulations for energy efficient buildings. The

[64] Low Energy.

actual requirements for low energy and passive residential buildings in Poland are controlled, introduced and announced by few instutions, i. a. Polish Institute of Passive Construction and Renewable Energy[65] (pol. Polski Instytut Budownictwa Pasywnego i Energii Odnawialnej), Institute for Sustainable Development (pol. Instytut na Rzecz Ekorozwoju) and National Agency for Energy Conservation (pol. Krajowa Agencja Poszanowania Energii). Besides, current and future requirements related with U-values of the building envelope and fenestation are depicted in the newest Regulation of the Minister of Transport, Construction and Maritime, announced in 2013 [72]. To sum up, current low energy and passive houses requirements in Poland are presented in the table 5.6. Meanwhile, U-values for the building parts and fenestration which have to be used for new emerging buildings in the nearest future are shown in the table 5.7.

Table 5.6. Current low energy and passive buildings requirements in Poland [75, 94, 95]

Feature	Type of the building		
	Passive	Low energy	Active low energy
U-value for external wall	$0{,}10 \div 0{,}15\ \frac{\mathrm{W}}{\mathrm{m^2K}}$	$\leq 0{,}30\ \frac{\mathrm{W}}{\mathrm{m^2K}}$	$0{,}20 \div 0{,}25\ \frac{\mathrm{W}}{\mathrm{m^2K}}$
U-value for roof	$0{,}10 \div 0{,}15\ \frac{\mathrm{W}}{\mathrm{m^2K}}$	$\leq 0{,}20\ \frac{\mathrm{W}}{\mathrm{m^2K}}$	$\leq 0{,}15\ \frac{\mathrm{W}}{\mathrm{m^2K}}$
U-value for ground floor	$\leq 0{,}15\ \frac{\mathrm{W}}{\mathrm{m^2K}}$	$\leq 0{,}35\ \frac{\mathrm{W}}{\mathrm{m^2K}}$	$\leq 0{,}30\ \frac{\mathrm{W}}{\mathrm{m^2K}}$
U-value for window and external door	$< 0{,}80\ \frac{\mathrm{W}}{\mathrm{m^2K}}$	$\leq 1{,}30\ \frac{\mathrm{W}}{\mathrm{m^2K}}$	$0{,}80 \div 1{,}10\ \frac{\mathrm{W}}{\mathrm{m^2K}}$
Efficiency of recovery heat exchanger	$> 75\%$	-	-
Type of required ventilation system	Mechanical with heat recovery and ground air heat exchanger	Hybrid (natural + mechanical)	Mechanical with heat recovery
Air tightness of the building frame, n_{50}	$< 0{,}6\ \mathrm{h^{-1}}$	$< 2{,}0\ \mathrm{h^{-1}}$	$< 1{,}0\ \mathrm{h^{-1}}$
Linear thermal bridge, ψ	As low as possible ($\psi \leq 0{,}01 \frac{\mathrm{W}}{\mathrm{mK}}$)		

Table 5.7. Maximum U-values of the building envelope and fenestration compulsory in Poland, today and within next years [72]

Type of the building component	Maximum allowed U-value $[\frac{\mathrm{W}}{\mathrm{m^2K}}]$		
	Since 01.01.2014	Since 01.01.2017	Since 01.01.2021
External wall	0,25	0,23	0,20
Internal wall, separating heated zone from staircase or corridor	1,00	1,00	1,00
Internal wall, separating heated zone from unheated zone	0,30	0,30	0,30

[65] The only institute in Poland accredited by Passive House Institute in Darmstadt to carry out certification of passive houses [94]

Table 5.7. Continuation

Roof	0,20	0,18	0,15
Ground floor	0,30	0,30	0,30
Internal ceiling (floor)	1,00	1,00	1,00
Internal ceiling (floor) over unheated zone or closed underfloor space	0,25	0,25	0,25
Wall window and balcony door	1,30	1,10	0,90
Diagonal and horizontal roof windows	1,50	1,30	1,10
Window in internal partition	1,50	1,30	1,10
External door and door separating heated and unheated zones	1,70	1,50	1,30

Apart from law regulations described above, relevant feature of the window controlled by standards both in Norway, Germany and Poland is a solar protection factor, which (in co-operation with artificial shading) determines a part of solar radiation coming through the window panes to the inside of the building. However, since residential building analyzed in this thesis is designed for Oslo (located in relatively cold climate), its values are not depicted in this section[66]. There are also some recommended values for assumption of air to water heat pump SPF (COP) factor, however in this master thesis heat pump was designed (not selected), thus its COP is one of the results and therefore these values are not presented in this section.

To sum up, one can notice that Norwegian passive and low energy standards seems to be more demanding than German and Polish standards, however based on PHI indications (tab. 5.5) it is justified, because most of Norway lays in colder climate than Germany and Poland. Influence of the climate have a great impact on law regulations related with insulation properties of residential buildings. Apart from this issue, both Germany and Norway have their own standards and requirements related with passive and low energy buildings. In Poland, passive houses are something new (in second half of 2014, around 40.000 passive buildings existed in the world, whilst around half of them in Germany and only 30 in Poland [96]) and it seems that Polish institutions and companies designing such buildings are learning from other countries, mainly from their German neighbours.

[66] Solar factor of the window pane and artificial shading have great importance in warm climate, because their proper selection may protect the building against overheating during hot periods.

5.2. Previous state of the building - low energy standard

5.2.1. Overview of the construction parameters, weather data and ventilation system

Residential building which will be taken into account in this master thesis is the low - energy (class 1[67]) building, designed by two bachelor students of Natasha Nord[68] to be located in Oslo. The building has the total area of $450 m^2$ and contains three identical apartments, one per floor ($150\ m^2$ each). The outlook of the building is shown on the figure 5.2.

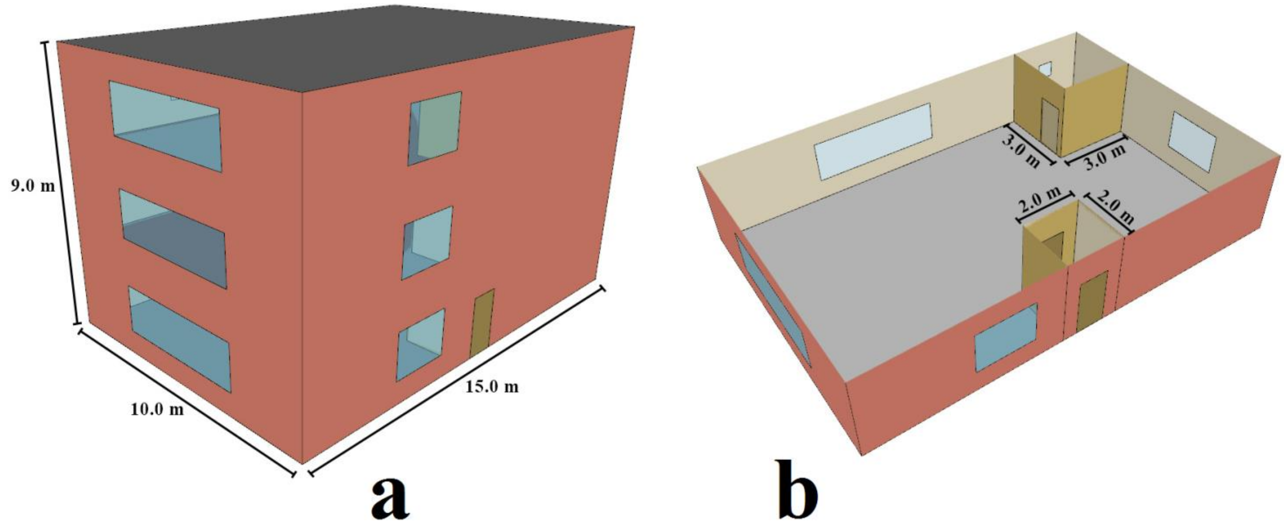


Figure 5.2. Overall outlook of considered building; a - geometry and dimensions of the building; b - plan of one floor (all floors are assumed to be the same). Easthern facade of the building is assumed as the facade where exterior door is placed. Presented views were made as 3D models in *Google SketchUp*[69] [9]

As the topic of this master thesis refers to not only residential building but also (or mainly) to its energy supply system, it is necessary to mention who created the system in previous considerations. It was Alexander Olsen Thoreby - student writing his master thesis with Natasa Nord supervision, who designed air - water R744 heat pump for the analyzed low energy building in 2013 (so one year after the building was created). An overview of both Thoreby's and Natasa Nord previous students master theses was made by the author of this master thesis and it was clearly noticeable that some of the building parameters were different. Therefore (although the shape of the building was similar in both theses) it was decided to present the low energy building description from Thoreby's thesis [9], completing only missing and necessary values with these from thesis of bachelor students [10].

Each apartment in considered building has three thermal zones, i. e. the main room (with area of $137\ m^2$), bathroom ($9\ m^2$) and unheated corridor ($4\ m^2$), which plays a role of the staircase. The height of each storey equals to 3 m. As it can be noticed on fig. 5.2, the building has only one

[67] According to Norwegian Standard - NS 3700:2013, two grades of low energy residential buildings can be distinguished, i. e. more energy efficient class 1 and class 2 (with regulations which are easier to fulfill).
[68] The building was designed by I. Rausand and Z. Iordache in their master thesis [10], written in Norwegian at NTNU in 2012.
[69] Software developed by *Last Software* company in 2001, bought by *Google Inc*. in 2006.

window (per each floor), the same for the northern and eastern facades, with area of $3\ \mathrm{m}^2$ (1,88 x 1,6 m). In case of western facade, there are two windows per floor, one bigger in the main room (with dimensions of $4{,}22 \times 1{,}6\ \mathrm{m}$ and area of $6{,}75\ \mathrm{m}^2$) and one smaller in the bathroom ($0{,}6 \times 0{,}6\ \mathrm{m}$; $0{,}36\ \mathrm{m}^2$). Except that, southern (the most sunny) facade is equipped with one big window per apartment, with area of $7\ \mathrm{m}^2$ (4,38 x 1,6 m). Generally, windows were placed in the building envelope in such a way to allow the daylight to get into the building effectively, thus making its inner space brighter.

Naturally, some requirements of the building were, and still they are based on Norwegian building code - TEK 10, because NS 3700 does not include all regulations but only these related to building energy demand for heating, heat losses and allowed internal heat gains. Thus, in TEK 10 there is requirement that total windows and external doors (collectively known as **fenestration**) area should cover max. 20% of BRA[70], i. e. usable floor area. In this building, the area of windows is equal to ~ $60{,}3\ \mathrm{m}^2$ and there are one front doors with area of $2{,}8\ \mathrm{m}^2$ (1,4 x 2 m), what in total constitutes 14,02% of BRA. Apart from it, each apartment in the building was equipped with two internal doors - one as the entrance to apartment from the staircase (1,2 x 2 m) and second as the bathroom door (0,9 x 2 m). In turn, all furniture inside the building was treated as convective internal mass, with convective heat transfer coefficient equal to $6\ \frac{\mathrm{W}}{\mathrm{m}^2\mathrm{K}}$ (radiation was assumed to be zero).

As considered building belonged to low energy standard (class 1), it was characterized by the following U-values of the building envelope (fulfilled by requirements of NS 3700:2010[71]), presented in the table 5.8. In case of thermal bridges, normalized thermal bridge value (referred to the square meter of BRA) was set to $0{,}05\ \frac{\mathrm{W}}{\mathrm{m}^2\mathrm{K}}$ (in pursuance of NS 3700:2010). Moreover, based on TEK 10 recommendation, **total net energy**[72] of the building should not exceed $120\ \frac{\mathrm{kWh}}{\mathrm{m}^2\mathrm{year}}$ per m^2 of the heated (cooled) usable floor space.

As it was mentioned previously in the text, desired location of analyzed building is Oslo. In case of **climate data**, the IWEC[73] hourly weather data (statistical aggregated values) for Oslo, Fornebu (airport) were assumed in previous considerations of the building (i. e. with considerations of the low

[70] Bruksarea (norw.) means usable floor area limited by enclosing building components inside. Surface of floor occupied by internal walls, doors and various devices are included, excluding only columns and other vertical structural elements with thickness over 0,3 m.

[71] During designing phase of analyzed building (in 2012), the newest Norwegian standard for residential energy efficient buildings - NS 3700:2013, was not available. Its previous version was NS 3700:2010.

[72] Total amount of energy which have to be supplied to the building (usually from the grid) in order to cover its whole energy demand, before conversion and losses in HVAC devices. Actual energy needed to fulfill the building needs is lower than net energy and is called as **usable energy**.

[73] International Weather for Energy Calculation; hourly weather data supplied by ASHRAE (American Society of Heating, Refrigerating and Air Conditioning Engineers).

energy building performed by Thoreby [9]). However, for purely informational purposes, typical climate conditions in Oslo are presented on the figure 5.3.

Table 5.8. Heat transfer coefficients of the building envelope [9]

Name of the building component	U-value $[\frac{W}{m^2K}]$
External walls	0,16
Roof	0,11
Ground floor	0,15
Windows	1,20
External door	1,20

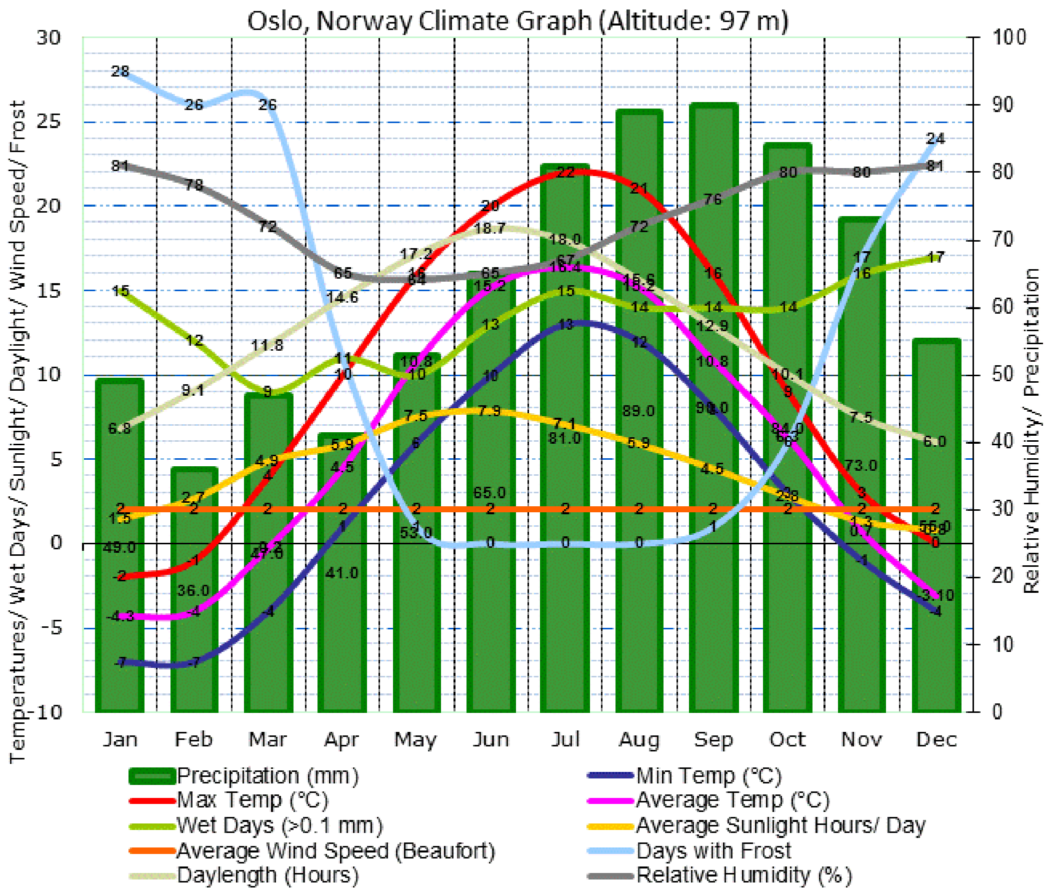


Figure 5.3. Average weather parameters in Oslo, at 59°56'N, 10°43'E [63]

The air infiltration rate in the current state of considered building was set to 0,5 h^{-1}, what fulfills NS 3700:2010 requirement. In case of **ventilation system**, a balanced CAV[74] system was assumed for living rooms and staircase, whilst an exhaust CAV system was chosen for bathrooms (a role of supply air was therefore taken over by outdoor air). Desired indoor air temperatures were controlled to be in the range of 21 ÷ 23°C (excluding staircase, where setpoint of 20 ÷ 23°C was desired). The

[74] Constant Air Ventilation - constant in terms of air supplied to the building in $\frac{m^3}{h}$.

air flow rate was set to $1{,}2\ \frac{\mathrm{m^3}}{\mathrm{m^2h}}$. Ventilation was mechanical, what means that fans were responsible for the ventilation air movement. In case of balanced system, two (intake and exhaust) fans with equal electric power and air flow rate were assumed to co-operate. Specific fan power factor (SFP) was set to $2{,}5\ \frac{\mathrm{kW}}{\frac{\mathrm{m^3}}{\mathrm{s}}}$. Moreover (for balanced CAV), the recovery heat exchanger - rotary wheel, with efficiency of 80% was assumed (in accordance with TEK 10 ans NS 3700:2010). Influence of energy losses from ventilaction ducts on the total energy balance of the building was neglected.

5.2.2. Internal heat gains

Internal heat gains in the building were assumed to not exceed their maximum values recommended by NS 3031:2007[75], i. e. $1{,}95\ \frac{\mathrm{W}}{\mathrm{m^2}}$ ($11{,}4\ \frac{\mathrm{kWh}}{\mathrm{m^2year}}$) for lighting and $1{,}8\ \frac{\mathrm{W}}{\mathrm{m^2}}$ ($10{,}5\ \frac{\mathrm{kWh}}{\mathrm{m^2year}}$) for electric equipment. In case of people, four occupants were assumed for each storey of the building. Heat gains from occupants were set by definition of their clothing and activity profiles. For domestic hot water, due to its very irregular, short periodic and complex mechanism of utility, NS 3031 does not specify the requirement for heat gains. Instead, it provides information about max. electric power which can be used for domestic water heating ($5{,}1\ \frac{\mathrm{W}}{\mathrm{m^2}}$ or $29{,}8\ \frac{\mathrm{kWh}}{\mathrm{m^2year}}$). However, due to the fact that in considered building heat pump system is responsible for DHW[76] preparation, only hot water temperature value was defined, as $45^0\mathrm{C}$[77]. Energy spent on DHW preparation was assumed to be as close as possible to maximum allowed value. DHW water was not treated as a typical internal heat gains source.

In practice both external and internal heat gains vary all the time. In aim to be closer to reality (thereby saving energy and money), internal loads can be controlled by <u>schedules</u>, describing their changes throughout the time. Furthermore, programs like EnergyPlus allow for <u>dynamical</u> calculations of energy demand and heat balance of the building within a certain period, therefore schedules of heat loads have to be defined. Although NS 3031 provides default time schedules for various internal heat gains, customized schedules can be also utilized if they are reasonable. In analyzed building such modified schedules were created for each internal heat gain. Outlook of the example schedule for lighting is presented on the figure 5.4. In turn, schedules used in the low energy building are shown on the figure 5.5.

[75] The newest Norwegian standard for calculation of energy performance of buildings - NS 3031:2014, was not available in 2012. Its previous version was NS 3031:2007.
[76] Domestic Hot Water.
[77] Temperature value of DHW finally utilized inside the building. It is not equal to the water temperature stored in DHW tank.

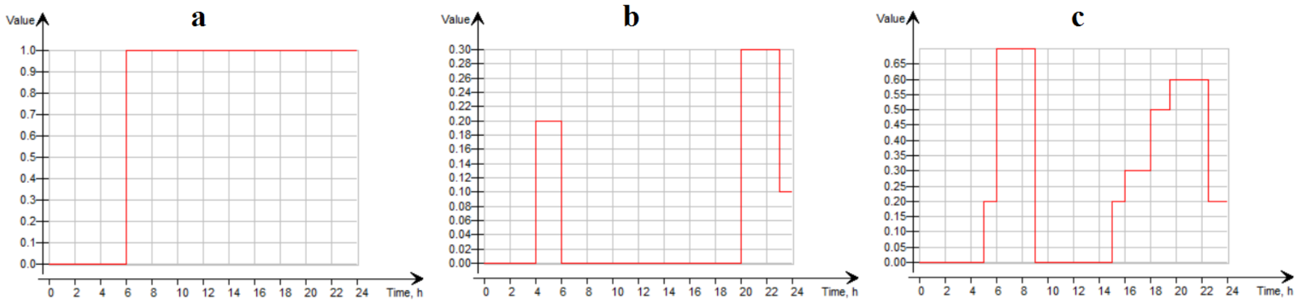


Figure 5.4. Comparison between few time schedules for lighting: a - schedule based on NS 3031:2007; b, c - custom schedules, respectively for summer and winter period. Vertical axes present current fractions (0 ÷ 1) of maximum electric power set for lighting

BELYSNING	VARMTVANN	UTSTYR	PERSONER	ACTIVITY
Fraction	Fraction	Fraction	Fraction	Any Number
Through: 12/31	Through: 12/31	Through: 12/31	Through: 12/31	Through: 12/31
For: AllDays	For: AllDays	For: AllDays	For: Weekdays	For: AllDays
Until: 05:00	Until: 06:00	Until: 08:00	Until: 07:00	Until: 07:00
0	0.0	0.20	1.0	70
Until: 07:00	Until: 07:00	Until: 12:00	Until: 09:00	Until: 17:00
0	0.8	0.20	0.5	70
Until: 08:00	Until: 08:00	Until: 13:00	Until: 14:00	Until: 20:00
0.3	0.8	0.20	0.0	95
Until: 17:00	Until: 12:00	Until: 17:00	Until: 16:00	Until: 24:00
0.9	0.0	0.20	0.5	70
Until: 18:00	Until: 14:00	Until: 18:00	Until: 18:00	
0.7	0.1	0.80	0.75	
Until: 20:00	Until: 17:00	Until: 20:00	Until: 24:00	
0.5	0.2	0.60	1	
Until: 22:00	Until: 18:00	Until: 22:00	For: Saturday	
0.3	0.8	0.50	Until: 11:00	
Until: 23:00	Until: 20:00	Until: 24:00	1	
0.1	0.6	0.40	Until: 16:00	

Figure 5.5. Constant schedules[78] for three internal heat gains (people, lighting and electric equipment) and hot water consumption [9]

Generally, not only internal heat gains can be controlled by schedules - they can be also set for i. a. ventilation, infiltration and thermostats for different places in the building. However, in previous considerations of analyzed building (as it was mentioned above) constant thermostat temperatures were assumed in zones. Meanwhile, ventilation and infiltration were set to function all the year without breaks.

5.3. Retrofit of the low energy building to passive standard - arrangements

5.3.1. Required changes

As it was widely described in section 5.2 and briefly depicted at the beginning of this chapter, considered building is low energy. In aim to upgrade this building to passive house (based on Norwegian requirements), the following features had to be changed:

[78] In overall, schedules can be either constant or variable (for certain period), e. g. schedule which is different for day and night periods is variable. Schedules can be defined for any needed time period, e. g. for a day, week, etc.

- U-values of the building envelope - their changes are depicted in the table 5.9;
- SFP factor of fans - reduction from 2,5 $\frac{\mathrm{kW}}{\frac{\mathrm{m^3}}{\mathrm{s}}}$ to 1,5 $\frac{\mathrm{kW}}{\frac{\mathrm{m^3}}{\mathrm{s}}}$, to fulfill NS 3700:2013 (tab. 5.3);
- Normalized thermal bridge - reduction from 0,05 $\frac{\mathrm{W}}{\mathrm{m^2K}}$ to 0,03 $\frac{\mathrm{W}}{\mathrm{m^2K}}$ (tab. 5.3);
- Maximum net energy supplied for space heating needs[79] - it had to be reduced from 30 $\frac{\mathrm{kWh}}{\mathrm{m^2yr}}$ to 15 $\frac{\mathrm{kWh}}{\mathrm{m^2yr}}$ (10 $\frac{\mathrm{W}}{\mathrm{m^2}}$), what enforced the change of a heat pump capacity;
- Air infiltration rate due to leakages at 50 Pa - reduction from 1 $\mathrm{h^{-1}}$ to 0,6 $\mathrm{h^{-1}}$.

Table 5.9. U-values of the considered building envelope, before and after renovation

Name of the building component	U-value [$\frac{\mathrm{W}}{\mathrm{m^2K}}$]	
	Before upgrade	After upgrade
External walls	0,16	0,10
Roof	0,11	0,08
Ground floor	0,15	0,08
Windows	1,20	0,80
External Door	1,20	0,80

Although passive house does not require an active heating system, underfloor water based heating system was decided to remain, due to excellent thermal comfort reached in the building inner space. Retrofit of the building will be performed based on NS 3700:2013, NS 3031:2014 and TEK 10.

5.3.2. Additional changes

Apart from necessary changes described in previous section, some other issues were altered (either due to program specification or a desire), among which the most important are depicted in the list below[80]:

- The staircase was re-constructed, in aim to be closer to reality. In the low energy building projected in 2012 staircase was assumed with the area of 4 $\mathrm{m^2}$ each, which (in the opinion of author of this master thesis) is too small to place there any stairs in a comfortable manner. Moreover, currently designed staircase was assumed as three separated zones (corridors) for each floor, what is unreal because all occupants of the building must be able to reach either ground, first or second floor by using the same staircase. Hence, instead of three zones with total height of

[79] Excluding renewable energy gained in bottom source of the heat pump. There is no option for changing the value of max. net energy in any program, so this change appears as the result of performed retrofit.
[80] Construction changes presented in this section are related to both low energy and passive building designed in IDA ICE.

3 m and area of 4 m^2 each, one staircase with total height of 9 m and area of 9 m^2 (3 x 3 m) was designed;

- All zones in the building were assumed to have a constant height of 2,6 m (with 0,4 m reserved for a partition, what in total gives mentioned 3 m). There is a lack of data how it was made previously in the low energy building;
- Thermal efficiency of the recovery heat exchanger in ventilation system was raised from 80% to 85%;
- Cooling system, consisting of cooling beams placed in the rooms was added[81], because author of this thesis believes that thermal comfort in the building should exist during all the year. At first it was desired to install overall ceiling cooling system (to achieve better thermal comfort inside the building), however due to relatively high investment cost and a lack of high cooling demand during the year it was hesitated;
- Layout of rooms per each floor (although still the same for all floors) was changed in order to be more realistic (e. g. increased presence of internall walls and doors, to the level as it occurs in real residential buildings). Instead of three rooms per floor (as shown on fig. 5.1.b), 10 thermal zones (excluding staircase, common for all floors) were designed, assuming a typical arrangement for 5 people (parents, two children and one guest or another participant of a family). Windows on external walls were divided and adjusted properly to new floor plan, keeping the same total window area as before refurbishment.

5.4. Current state of the building - passive standard

As it was told and justified in the chapter 4, **IDA ICE** simulation program was assumed in this master thesis for analysis of the considered building. After implementation of construction changes depicted in section 5.3.2 the building model is shown on the figure 5.6.

The temperature of indoor air was assumed to be the same in all zones, with a value between 21°C (setpoint for heating) and 23°C (setpoint for cooling). Besides, ventilation system (in the first approach[82]) was the same as in Thoreby's work. Desired U-values of the passive building envelope components (described in tab. 5.9) were successfully achieved, assuming appropriate building materials (e. g. extruded polystyrene with low thermal conductivity as an insulation material, wood instead of concrete). Lighting, electric equipment and people were defined as internal heat gains in IDA ICE. Although their parameters and schedules differ from these utilized previously in LE building, maximum values of heat gains determined by NS 3031:2014 were not exceeded. A LED

[81] In previous (low energy) state of analyzed building there was no cooling system.
[82] Before optimization of heating and cooling system described in the chapter 7.

lighting was applied in all rooms with the same schedule, area of 0,16 m^2, electrical power consumption of 16 W and convective fraction of 0,05[83] per one item and zone. In turn, a schedule and amount of heat generated by electric equipment vary dependent on a room type.

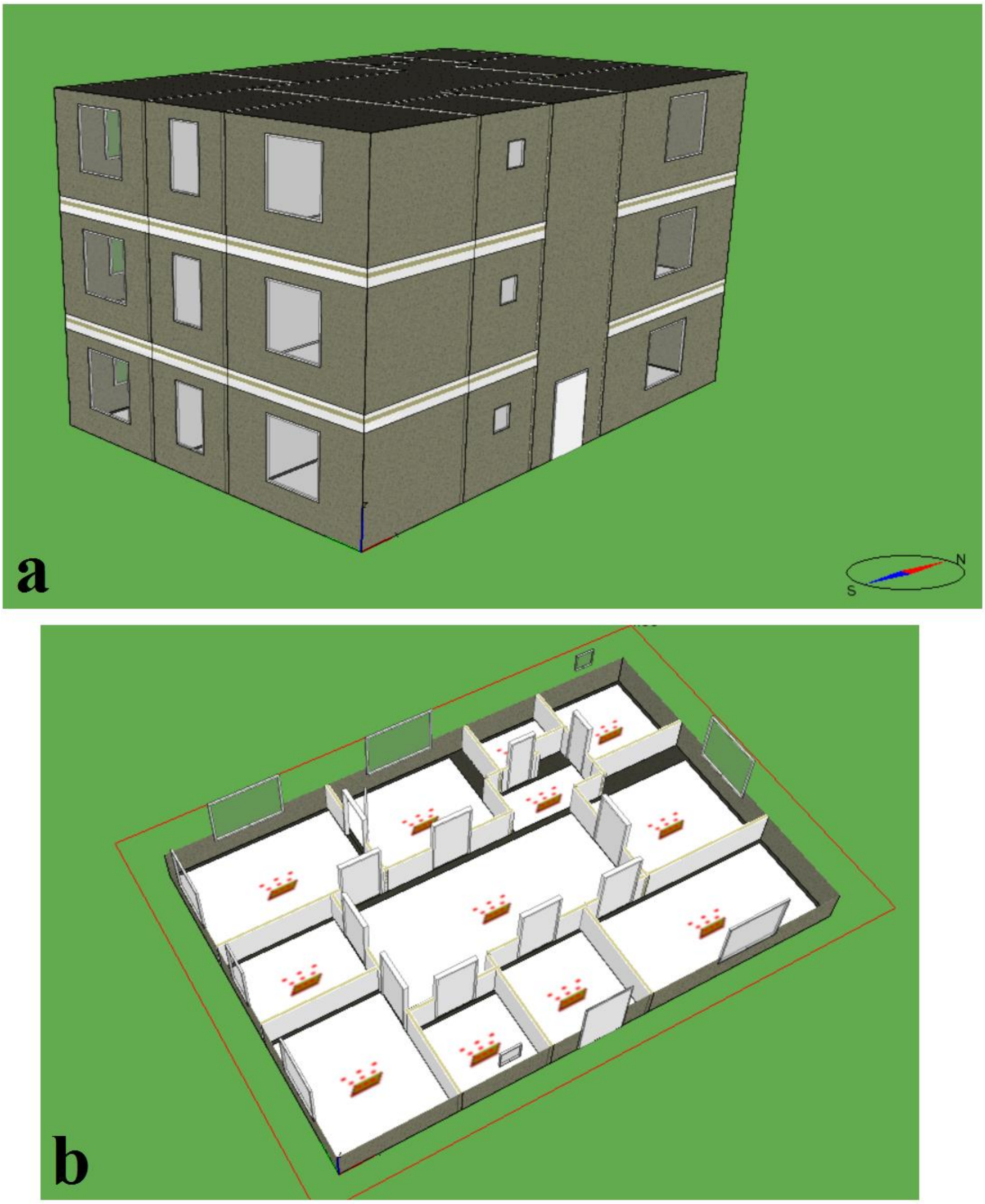


Figure 5.6. Outlook of the passive building energy model designed in IDA ICE 4.7; a - overall view; b - (first) floor plan

Besides, it was desired to model the people behaviour more accurately than it is usually done, therefore occupants have different schedules. Moreover, some of them were set to be in move between zones during the day. However, IDA ICE has limited capabilities related with a control of occupants behaviour, therefore their presence in each room had to be set separately (while checking

[83] Fraction of rated electric power emitted as a convective heat.

the building specification, the program shows that 21 people are living in the house - because of 21 schedules, although there are only 15 people inside). Activity level of the passive house occupants was set in range of 1 MET[84] ÷ 1,2 MET, depended of the room type. In turn, clothing of the people is always the same, and equals $0{,}85 \pm 0{,}35$ clo[85]. Definition of specific values (activity level, clothing) for the people in IDA ICE is not incidental - the program needs them for evaluation of the thermal comfort level in each of the building zones.

In case of domestic hot water, its share in internal heat gains of the building was neglected, due to very irregular periods of its usage by the people, different outlet water temperatures and flow rates which people can regulate and a lack of strict requirements related to allowed heat gains values.

Side orientation of the passive house is visible on the compass indication on fig. 5.6.a. Weather data necessary for simulations conducted in IDA ICE was implemented into IDA ICE from ASHRAE IWEC2 for Oslo - Gardermoen (in .prn format - IDA weather file). Furthermore, a suburban wind profile was adopted for calculations (based on ASHRAE 1993). In case of the ground, an ISO ground model was assumed (calculations of heat losses to the ground performed by the software are thus compatible with ISO-13370 standard).

5.5. Comparison of achievements between low energy and passive state of the building

Low energy building retrofitted by the author of this master thesis was recently analyzed by Thoreby, as it was mentioned in section 5.2. Hence, two comparisons have to be performed in this section:

- First comparison - related to heat load and energy consumption of the low energy building designed by Thoreby in EnergyPlus (he used also SketchUp and OpenStudio - all programs are characterized in the chapter X) with performance of the low energy building designed in IDA ICE by the author of this thesis;
- Second comparison - between low energy building and passive house, both created in IDA ICE.

All results presented in this section have a constant time step of 1 hour (or are based on calculation accuracy of 1 hour), although both EnergyPlus and IDA ICE are able to give more accurate results (in EnergyPlus the minimum time step is equal to 1 minute, and in IDA ICE - sometimes even less than 1 min). Various heat losses (e. g. heat distribution losses, plant losses or ventilation duct losses) were neglected in the whole analysis presented in this master thesis.

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[84] Unit describing the rate of heat emission from the human body, according to type of activity (1 MET = 100 W [33]). More details can be found in the attachment 6.

[85] Unit depicting the level of clothing insulation, dependent on the type of clothes. It can be expressed in similar way as the heat resistance of building components ($1\ \text{clo} = 0{,}155\frac{\text{m}^2\text{K}}{\text{W}}$). Further details are placed in appendix 6.

5.5.1. Low energy building - in Energy Plus and in IDA ICE

For evaluation of the building performance Thoreby assumed electric heating system with efficiency of 100%, thus the author of this master thesis did the same. Beside construction changes of the building (depicted in section 5.3.2) all parameters of low energy building created in IDA ICE were similar as in Thoreby's thesis.

Annual net energy consumption in the low energy building designed by Thoreby was equal to 40979 kWh (or 91,1 $\frac{\text{kWh}}{\text{m}^2\text{year}}$). To reach desired indoor air temperatures space heating and domestic hot water systems consumed 31031 kWh, whilst other electric equipment (fans, pumps and lighting) used 9948 kWh. Monthly distribution of total energy consumed in the building is depicted on the figure 5.7.

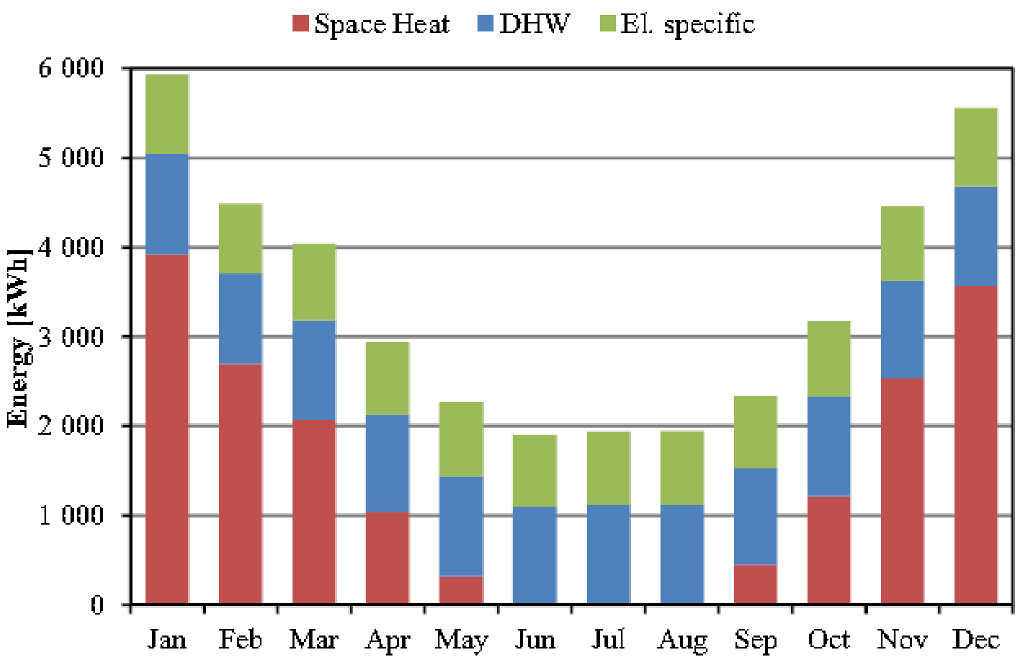


Figure 5.7. Monthly energy end-use for low energy building created in EnergyPlus [9]

Energy consumed for domestic hot water preparation was relatively constant throughout the whole year (approx. 1100 kWh per month) and the energy supplied to auxiliary electric equipment („El. specific" on fig. 5.7) varied between 780 and 890 kWh per month. In turn, energy utilized for space heating needs was maximum in January (3923 kWh) and minimum during three summer months (approx. 0 kWh in June, July and August). Apart from the monthly net energy consumption, total peak load of the building (including heating needs, pumps, fans and lighting) was found to be 17,4 kW, with the peak heating load being 15,7 kW. Based on this value Thoreby designed heat pump for the building, what will be described in the chapter 6. The heating load duration curve of the building is shown on the figure 5.8.

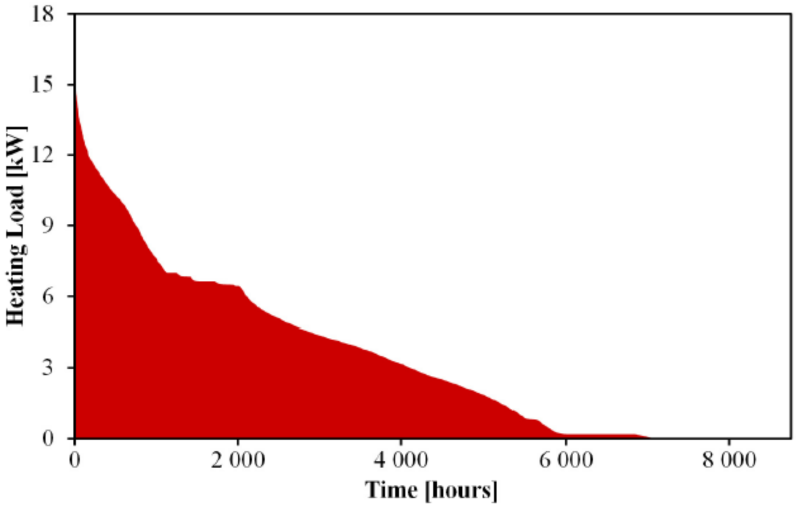


Figure 5.8. Heating load duration curve of the low energy building analyzed by Thoreby [9]

In turn, total net energy consumption of the low energy building designed in IDA ICE by the author of this master thesis is equal to 35008 kWh (or 81 $\frac{\text{kWh}}{\text{m}^2\text{year}}$) - 27450 kWh for heating needs (16012 kWh for space heating and 11438 kWh for DHW preparation) and 7558 kWh for auxiliary devices. Monthly distribution of the energy (electricity) consumption can be noticed on the figure 5.9. Energy spent on domestic water heating is constant for the whole year - approx. 30,7 kWh per day (~ 953 kWh per month). In turn, electricity supplied for space heating needs was the highest in January (the coldest month) - 3959 kWh and the lowest during summer months (~0 kWh). Besides, auxiliary energy needs varied between 587 and 664 kWh per month.

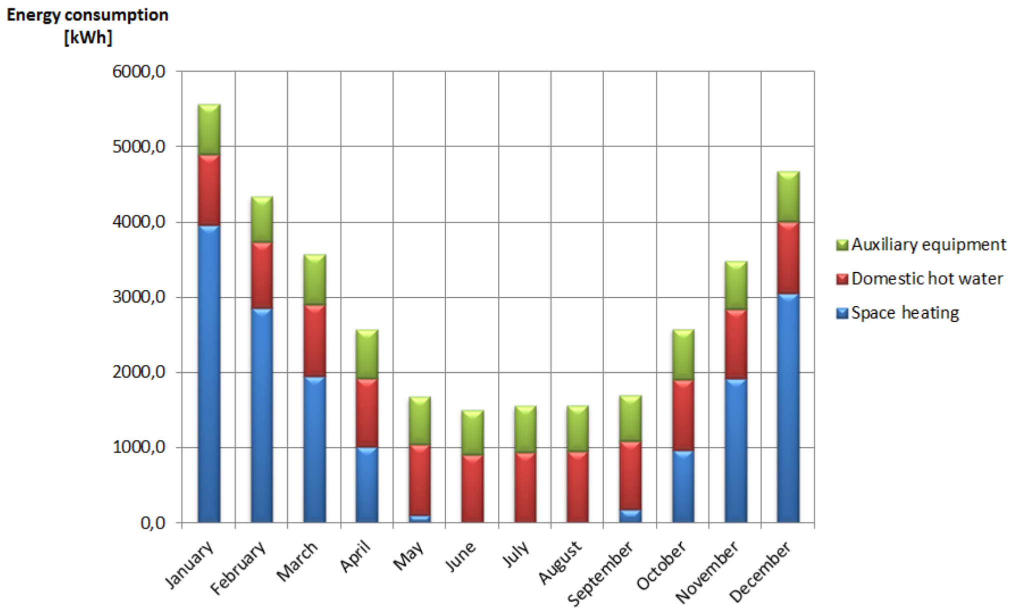


Figure 5.9. Energy consumption in low energy building designed in IDA ICE

Total peak load of the building was equal to ~18,1 kW, with the peak heating load being ~17,4 kW (including 15,84 kW for space heating and 1,57 kW for DHW preparation). Duration curve of the heating load for the year is placed on the figure 5.10.

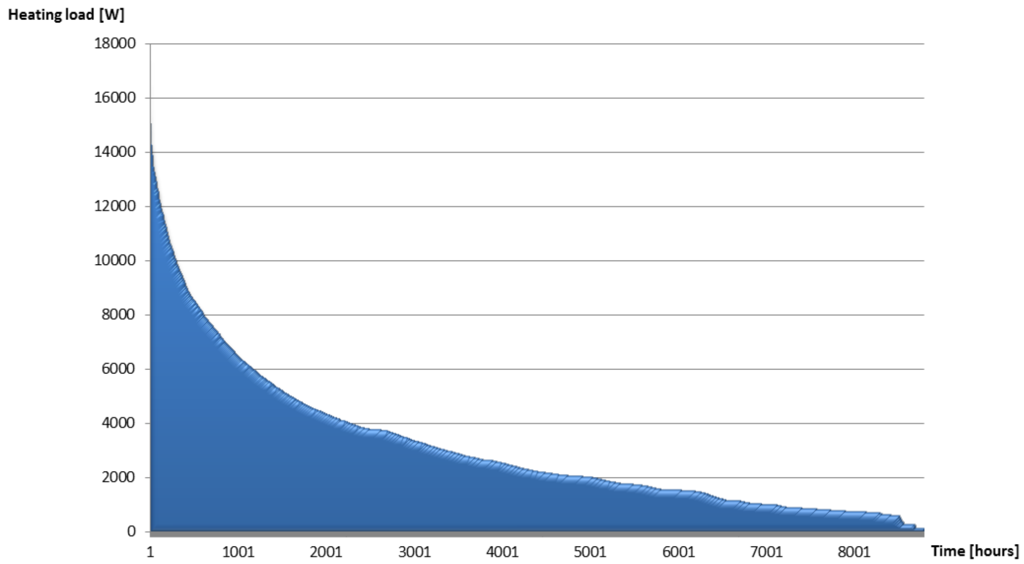


Figure 5.10. Heating load duration curve of the low energy building designed in IDA ICE

As it can be noticed in the current section of this chapter, there are some differences between energy demand and heating load of the low energy building analyzed by Thoreby and the author of this thesis. In case of the building designed in IDA ICE both total net energy consumption is higher than in Thoreby's thesis, probably due to differences in energy provided to space heating, domestic hot water preparation and lighting (in IDA ICE the author assumed LED lighting with clearly lower electricity consumption). In case of DHW, Thoreby assumed maximum allowed energy use close to $29{,}8\ \frac{\mathrm{kWh}}{\mathrm{m^2year}}$, but the author of this thesis assumed 33 litres of hot water per day and occupant (with a temperature of 60°C) based on [18, 64], what resulted in average energy usage of $26{,}5\ \frac{\mathrm{kWh}}{\mathrm{m^2year}}$. Comparing heating load curves of the building one can noticed that on fig. 5.8 the curve is sharper and shorter than on fig. 5.10, because Thoreby assumed faster but shorter period of domestic hot water heating during each day of the year. As a result, within more hours heating system was turned off but while it was working it needed more power. However, total peak heating load is higher in case of the building designed in IDA ICE. It may be caused by another weather data period assumed for simulations - author of this thesis assumed a year of 2016 but Thoreby assumed other year. Hence space heating needs of the building during the year may be different. Besides, all differences may be a consequence of different software applied for simulations (EnergyPlus and IDA ICE).

5.5.2. Low energy and passive house - IDA ICE

There was a clear dissimilarity in total energy consumption between passive and low energy building designed in IDA ICE. Thanks to better insulation of building envelope parts and other changes depicted in section 5.3.1 total net energy consumption of the passive building was equal to 28177 kWh (or $65{,}2\ \frac{\mathrm{kWh}}{\mathrm{m^2year}}$) - 20% lower than annual energy needs of the low energy building, despite the fact that a cooling system was additionally applied in the passive house. Overall energy spent on heating needs equalled 19368 kWh. DHW energy consumption was the same as for low energy building, therefore an overall 30% decrease in energy required for all heating needs was a result of over 50% decrease in energy provided for space heating only. In turn, the auxiliary equipment used 5141 kWh (same energy usage for lighting but lower for fans and pumps), what constitutes 32% lower consumption than electric devices in the low energy house. Besides, yearly cooling needs required 3668 kWh (energy consumption was the highest in July - 1079 kWh and lowest in a period of October ÷ March - 0 kWh). Comparison of monthly energy distribution between low energy and passive house during considered year is presented on the figure 5.11, while comparison of their heat load curves is shown on the figure 5.12.

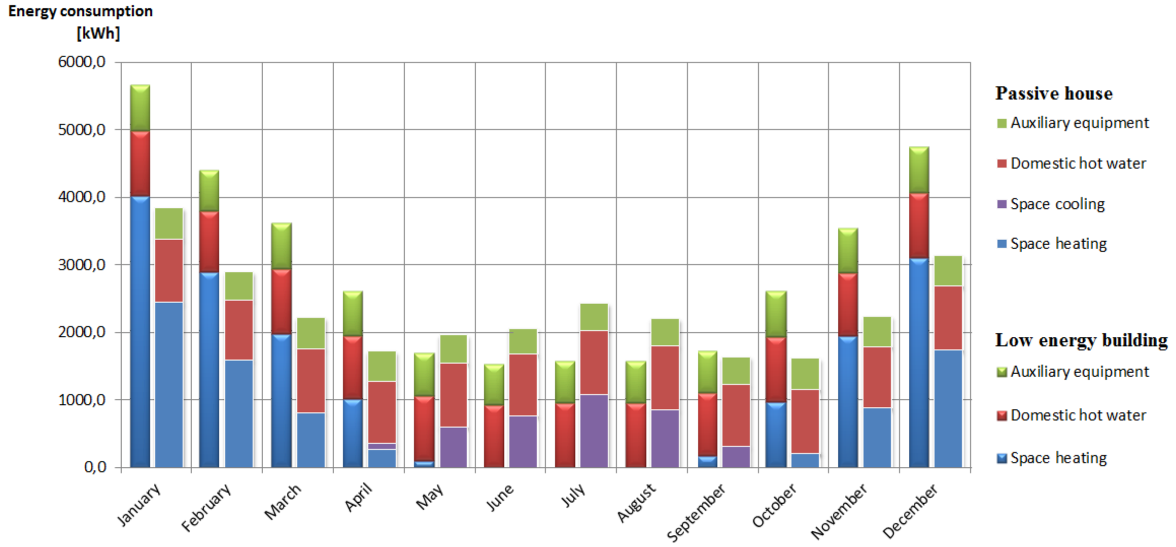


Figure 5.11. Comparison between energy consumption in low energy and passive house designed in IDA ICE during the reference 2016 year

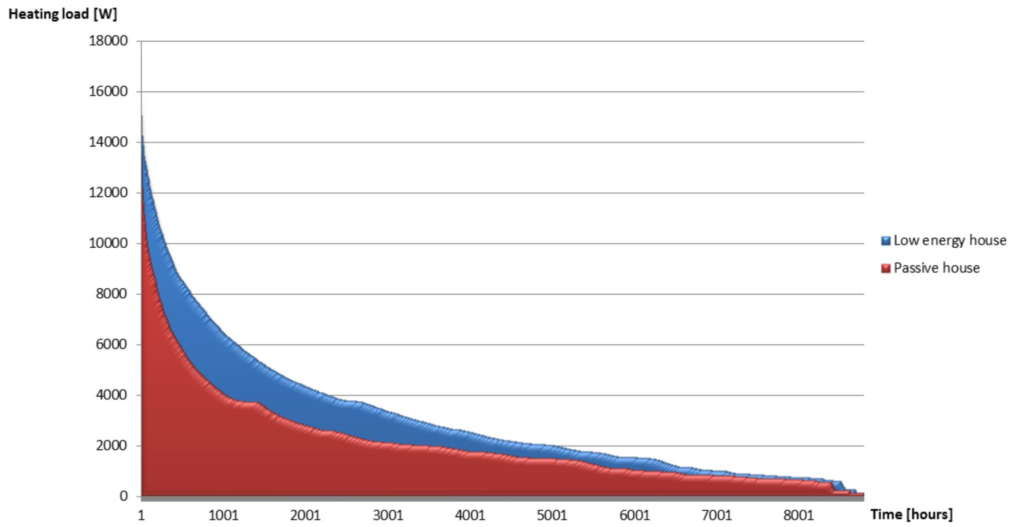


Figure 5.12. Comparison between heating load duration curves of low energy and passive house designed in IDA ICE

Total peak load of the passive house equals ~15,8 kW, whilst the peak heating load is equal to ~15,4 kW (including 13,87 kW for space heating and 1,57 kW for DHW heating). However, if to consider maximum heating power required for SH and DHW separately, these values are equal to 14,01 kW and 3,83 kW respectively. In turn, the peak cooling load is 4,41 kW. Moreover, a cooling load curve of the passive building can be found on the figure 5.13.

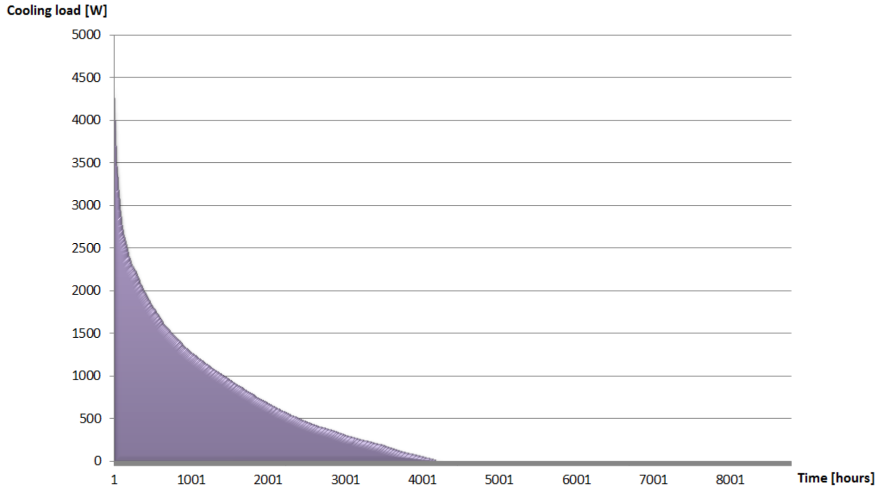


Figure 5.13. Cooling load duration curve of the passive house designed in IDA ICE

In case of the recovery heat exchanger placed in ventilation system, an increase of its efficiency resulted in only 0,52% higher amount of heating energy saved annually - from 20509 kWh (efficiency of 80%) up to 20615 kWh (85% efficiency), therefore one could conclude that a 5% change in the heat exchanger efficiency was not really profitable. Amount of energy saved by the heat exchanger seem to be high in comparison with annual energy demand of the building for heating needs, however it is worth to remember that ventilation system increases heat losses from the building significantly and without it its heating demand would be surely lower, even if there is a recovery heat exchanger installed. Comparison of both supply, return (indoor air) and outside air temperatures throughout considered 2016 year is presented for designed passive building on the figure 5.14, in order to allow for imagination how big ventilation heat losses from the building would be without recovery heat exchanger.

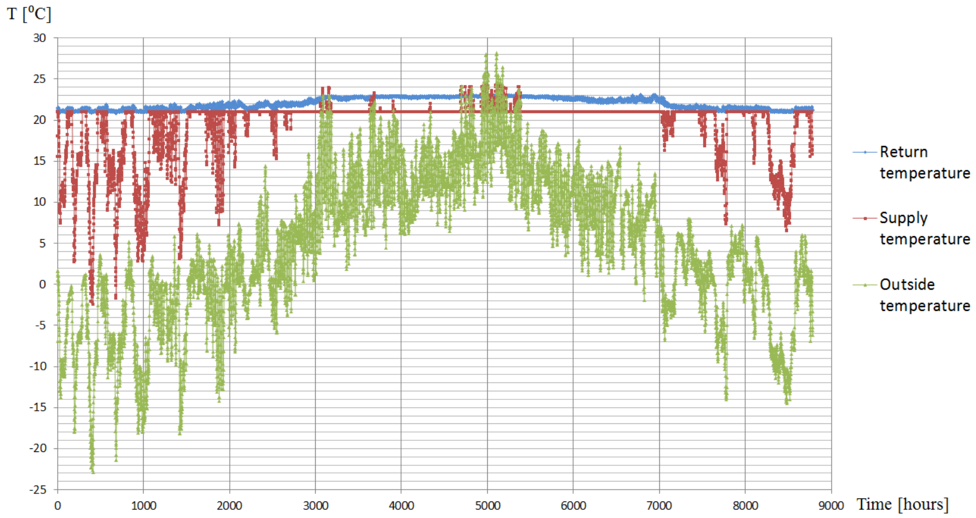


Figure 5.14. Ventilation air in the passive house designed in IDA ICE

6. Mathematical modelling and description of the energy supply system

In the first section of this chapter the energy supply system equipped with heat pump and water storage tank (designed by Thoreby in his master thesis in 2013) for the previous (low energy) state of considered building will be characterized. Afterwards, mathematical basics related with modeling of transcritical heat pump components will be provided. Based on most of them the author of this master thesis developed his own heat pump model, described in the next part of the chapter. Together with the heat pump depiction, selected energy storage system (hot water storage tank[86]) and cooling system models developed by the author of this work for the passive house (current state of analyzed building) will be defined. In the last part of the chapter a comparison between heat pump developed by Thoreby for the low energy building and heat pump developed by the author of this thesis for the passive house will be performed.

6.1. Heating, cooling and thermal energy storage system in low energy building

The energy supply system for low energy state of considered building was designed by A. O. Thoreby, therefore most of data in this section is taken from his master thesis [9]. In case of the **heating system**, air to water heat pump was applied for both domestic hot water and space heating. The topic of Thoreby's master thesis was related with analysis of $\mathrm{CO_2}$ heat pump in low energy residential building, thus the heat pump which he designed works in transcritical conditions, utilizing R744 as refrigerant. Advantages of $\mathrm{CO_2}$ usage in heat pump systems and work principles of the transcritical heat pump were described in chapter 2 (sections 2.1 and 2.4) of this master thesis.

The whole modeling process of the air - based transcritical heat pump in EnergyPlus Thoreby began with an initial guess of no heat transfer in gas cooler and evaporator (i. e. $\dot{\mathrm{Q}}_{\mathrm{GC}} = 0$ and $\dot{\mathrm{Q}}_{\mathrm{EV}} = 0$), therefore the evaporation temperature equalled to the heat source (ambient air) temperature. For the heat pump calculations the convergence criterion was set - iterations would continue until stable heat transfer conditions will be reached (changes in heat transfer rates of both evaporator and gas cooler will be lower than 0,1%). The algorithm depicting modeling process of the whole heat pump system created by Thoreby is shown on the figure 6.1.

In order to determine input parameters of the heat pump, an assessment of the peak heating load of the building had to be made. A simulation of the reference model of considered building (equipped with direct electric heating) was therefore performed in EnergyPlus[87]. Based on its results, Thoreby decided to implement heat pump with a heating capacity of 9 kW (what resulted in power

[86] At the beginning of writing this thesis, the author was wondering about utility of PCM (phase change materials) in analyzed building, however due to lack of programs allowing for relatively precise PCM modeling and lack of time he finally limited the energy storage system to DHW storage tank.
[87] Thoreby used EnergyPlus software for all needs of his master thesis.

coverage factor little lower than 60%). As heat pump capacity drops with decreasing outdoor temperature, it was decided that full capacity of 9 kW should be able to be delivered at outdoor temperature of -7ºC (based on technical report of J. Stene and T. Skiple [13]).

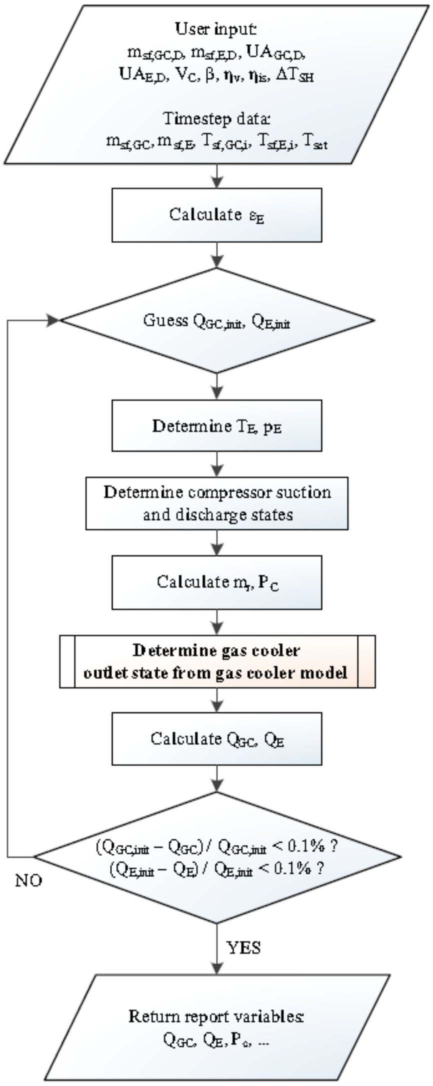


Figure 6.1. Calculation steps of the heat pump model [9]

After evaluation of different gas cooler configurations conducted by Thoreby in his master thesis, it was finally desired to model a system consisting of tripartite gas cooler, presented on the figure 6.2. Such solution is very profitable in co-operation with low - temperature underfloor water heating distribution system. First, superheated CO_2 releases heat in the reheat DHW gas cooler to domestic hot water, which has higher temperature than in case of other compared systems (various single and bipartite gas cooler configurations) [9], because it was earlier preheated in the preheat DHW gas cooler. Thanks to this, R744 temperature supplying the SH[88] gas cooler is warmer, what decreases a risk that SH water temperature will be too low. Thus tripartite gas cooler construction allows for both low temperature space heating (desired in low energy and passive buildings) and high temperature

[88] Space Heating

DHW, with relatively high probability that no top - up heater will be needed to cover the peak demand while working in combined mode. In general, tripartite gas cooler construction allows for three modes of heating, i. e. combined space and domestic water heating, SH only or DHW preparation only, when SH is not required (e. g. during summer period). Tripartite system is taking full advantage of the R744 temperature glide when cooling the CO_2. The minimum CO_2 outlet temperature equals the water mains temperature in combined and DHW modes, or the return temperature of the SH water in SH mode.

In overall, some of gas cooler units can be placed internally inside the water storage tank - it increases compactness of the system. However, due to the fact that such configurations have lower COP than external gas cooler systems (because the advantage of external gas cooler counterflow arrangement is lost) [9], internal gas cooler units were not analyzed any more.

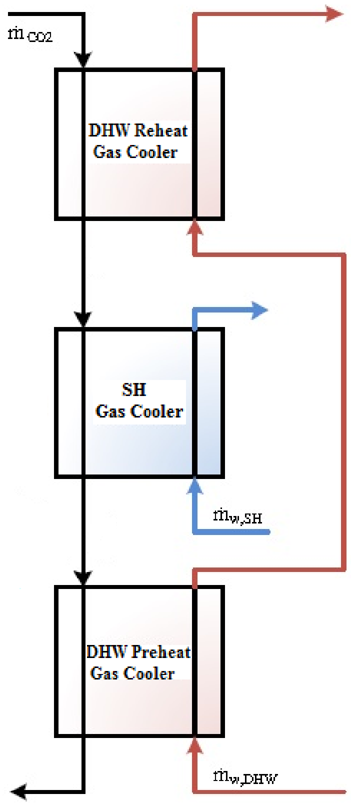


Figure 6.2. Scheme of tripartite gas cooler, by which heat is transferred from superheated R744 to water flowing in DHW and SH installation tubes [9]

The original EnergyPlus module for a conventional heat pump model existing in the program was made by Murugappan in 2002 [11] and it provided basics for the transcritical heat pump designed by Thoreby. Unfortunately, in EnergyPlus the heat pump could only be connected to one single loop on the gas cooler side. The system was therefore modeled with an intermediate water loop containing three ideal heat exchangers[89], what can be seen on the figure 6.3.

[89] Ideal means 100% efficient.

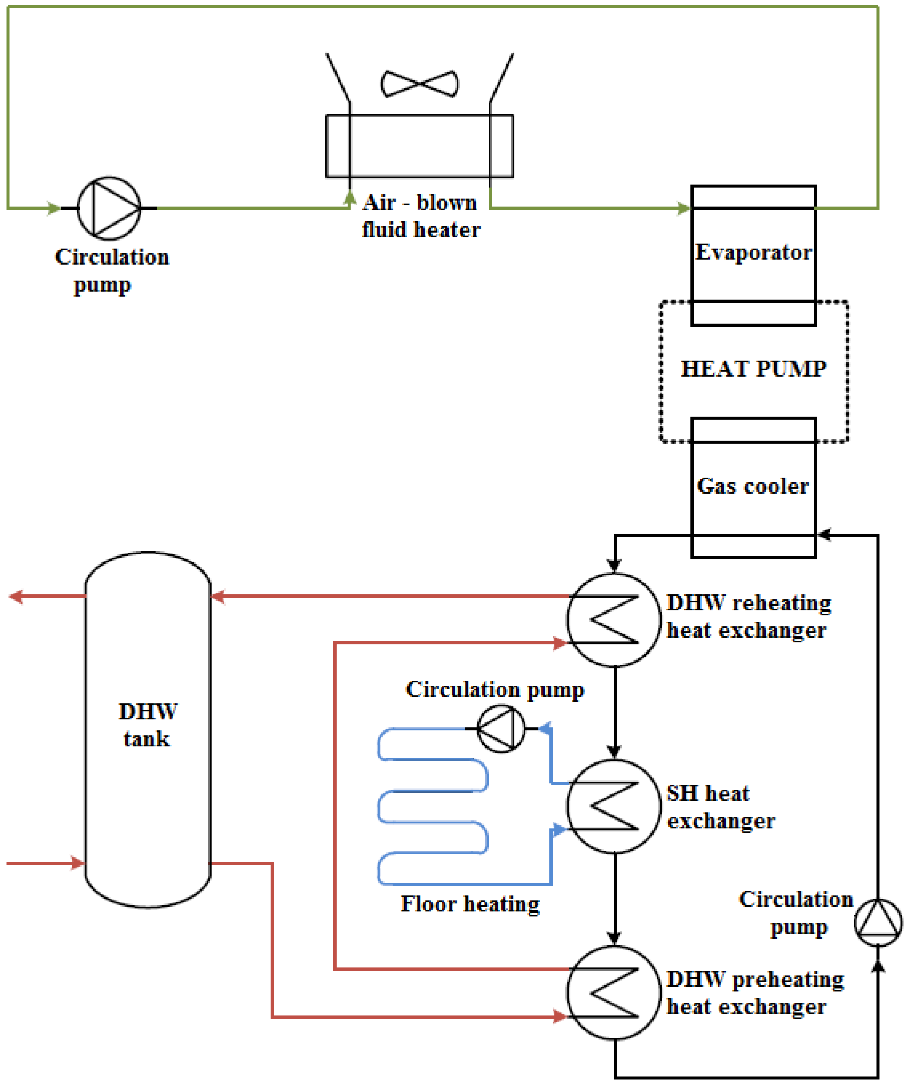


Figure 6.3. Scheme of the heating system finally designed by Thoreby in the low energy building [9]

System presented on fig. 6.3 is however not entirely consistent with desired configuration of gas coolers (fig. 6.2), mainly due to two following reasons:

- even if dimensions of single gas cooler in transcritical heat pump loop would be properly defined to replace three separate gas cooler units, the total heat transferred between R744 and water would not be the same, because water flowing in the intermediate loop has only one certain temperature while entering gas cooler. In case of tripartite gas cooler, CO_2 would release the heat to water streams at three different inlet temperature levels;
- even if water in the intermediate loop would be able to release the same (in total) amount of heat in three ideal heat exchangers as CO_2 would release in three separate gas cooler units, the percentage shares of the heat released to each (DHW or SH) water loop would be different in both cases, because water to water and R744 to water heat exchange mechanisms have different intensity.

Another significant limitation of EnergyPlus is related with the evaporator construction. The program does not allow for direct connection between air stream and CO_2 mass flow rate in the evaporator - it has to be done using an intermediate loop (fig. 6.3). Hence a glycol mix was assumed as the secondary fluid for evaporator. Glycol mix entering the evaporator held the same temperature as outdoor air at all times (Thoreby assumed constant ambient air temperature of 7°C for the whole year).

As it was mentioned previously and can be noticed on fig. 6.3, variable - flow hydronic underfloor heating was assumed for **space heating distribution system**. Relatively low supply and return water temperatures were selected - 35°C and 30°C. Meanwhile, additional top - up electric heater (with capacity of 9 kW) was installed in aim to cover the peak space heating load in the building. Influence of additional energy losses (e. g. leakages or heat losses from the heat pump, DHW and SH installation tubes) was neglected. Another numerical values which Thoreby assumed during designing phase of the heat pump are specified in the table 6.1. Some of them were taken into account by the author of this master thesis for further considerations (i. e. development of the new heat pump model for the passive house, what will be described later in this chapter).

Table 6.1. Constant values assumed for parametric study of the heat pump model

Name of the parameter	Symbol	Value and unit
Volumetric efficiency of the compressor	η_{vol}	0,7
Isentropic efficiency of the compressor	η_{is}	0,55
Heat loss factor (compressor)	β	0,1
Superheat temperature value before compression (temperature difference)	ΔT_{SH}	5°C
Number of gas cooler subsections	N	20 (50[90])
Heat transfer factor of the evaporator	UA_{EV}	$2000\ \frac{W}{K}$ [91]
Heat transfer factor of the gas cooler	UA_{GC}	$1300\ \frac{W}{K}$ [92]
Water inlet temperature range for the gas cooler	$T_{sf,I,GC}$	$10 \div 30^{o}C$
Water outlet temperature (setpoint) range for the gas cooler	$T_{sf,O,GC}$	$35 \div 65^{o}C$

Due to the fact that desired building location is Oslo, where climate is rather cold (based on investigation performed by Thoreby, only $10\ \div\ 20$ days per year would require cooling) Thoreby assumed **no cooling system** [9].

[90] In further analysis Thoreby finally assumed 50 subsections of the gas cooler [9].

[91] Results of the investigation conducted by Thoreby showed relatively small heat pump performance gain with increased evaporator UA-values over 2000 $\frac{W}{K}$, therefore this value was assumed.

[92] Based on investigation performed by Thoreby 1300 $\frac{W}{K}$ is a value giving the best overall heat pump performance for considered energy supply system.

A cylindrical DHW water storage tank was modeled as **thermal energy storage system** in the building, with height of 2 m and volume of 500 dm^3, what gives relatively small ratio between its diameter and height (~ 0,28). It should help in reduction of conduction and mixing inside the tank. For the thermal water stratification modeling, the tank was divided into ten parts (nodes) of equal height, what can be noticed on the figure 6.4. Temperature of the hot water which was supplied to the tank (after heating in preheat and reheat gas coolers of the heat pump). Heat loss coefficient for the tank envelope was set to 0,9 $\frac{W}{m^2K}$.

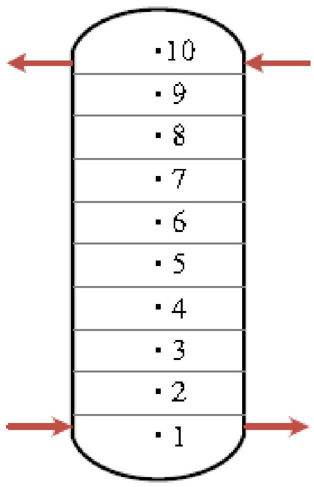


Figure 6.4. Scheme of DHW tank modeled in EnergyPlus [9]

In addition, a 7 kW electric heater was placed near the top of the tank (node no. 9) for supplementary DHW heating, in order to support the heat pump in peak heating demand periods. Top - up heater was set to begin heating the water when its temperature falls to 50°C. Time schedule for DHW tank utility resembles a typical pattern for residential building. Temperature of cold water from the mains was assumed to have a constant value of 7°C[93] before entering the DHW tank (in reality, its value would vary throughout the year due to weather changes). However the DHW inlet temperature reaching the DHW preheating heat exchanger was assumed as 10°C. Afterwards, hot water was heated up to 65°C (tab. 6.1) after flowing out of the reheater. Except that, the tap water temperature of 40°C was assumed, as the result of mixing between hot water from DHW tank and cold water from the mains.

6.2. Mathematical modelling of transcritical heat pump - basics

In the first part of this section a modeling principles of the main heat pump elements will be described. Afterwards, the case of optimal gas cooler pressure will be discussed.

[93] This temperature is a yearly average air temperature of the location for which considered building and its energy supply system are designed.

6.2.1. Components of the heat pump

6.2.1.1. Modeling of the expansion valve

There are several types of expansion valves which can be utilized for expansion process between gas cooler and evaporator - capillary tube, thermostatic expansion valve and electronic expansion valve are the most widely used. A mathematical model of the valve is dependent on its type. In order to evaluate dependency between refrigerant flow changes and corresponding opening areas of the valve an appropriate mathematical equation is necessary. When expansion valve is modelled as an orifice (electronically controlled), liquid refrigerant flowing through such expansion valve can be modelled using the following orifice equation [17]:

$$\dot{m}_{CO_2} = C_v \cdot A_v \cdot \sqrt{(\rho_3 \cdot \Delta p)}\,, \tag{6.1}$$

where:

C_v - orifice coefficient [-],

A_v - opening area of the valve $[m^2]$,

ρ_3 - density of the refrigerant at state before expansion (point 3 on fig. 2.14) $[\frac{kg}{m^3}]$,

Δp - pressure difference between gas cooler (high pressure) side and evaporator (low pressure) side [Pa].

In considerations included in this master thesis pressure drop of the refrigerant (CO_2) present in expansion valve was considered as isenthalpic (i. e. identical specific enthalpy of R744 before evaporator and after gas cooler).

6.2.1.2. Modeling of the compressor

For the compressor model included in this section the following issues were assumed by the author of this thesis:

- constant superheat temperature value before compression process,
- constant displacement rate of the compressor,
- heat losses during compression are taken into account via the heat loss factor,
- constant volumetric and isentropic efficiencies throughout the compressor range of operation.

Mass flow rate of the refrigerant flowing in the compressor (and hereby in whole heat pump cycle) can be therefore calculated using the following equation:

$$\dot{m}_{CO_2} = \eta_{vol} \cdot \rho_1 \cdot \dot{V}\,, \tag{6.2}$$

where:

η_{vol} - volumetric efficiency of the compressor [-],

ρ_1 - density of the refrigerant at suction state before compression (point 1 on fig. 2.14) [$\frac{kg}{m^3}$],

$\dot{V}$ - displacement rate of the compressor [$\frac{m^3}{s}$].

In turn displacement rate can be counted as:

$$\dot{V} = f \cdot V \,, \tag{6.3}$$

where:

f - frequency of the compressor work, providing information about its running speed [Hz],

V - displacement volume of the compressor [m^3].

An increased CO_2 outlet temperature from the gas cooler leads to higher evaporator inlet enthalpy and therefore lower enthalpy raise in the evaporator, what results in lower VRC[94] of the heat pump. That enforces the utility of bigger compressor (with higher displacement volume). Hence in calculations the worst case of the heat pump working conditions should be always considered (i. e. the case when temperature of CO_2 flowing out of gas cooler is the highest), to make sure that chosen compressor will not be too small.

If to assume that the compressor works at a single speed, then its displacement rate is constant. Enthalpy of the refrigerant at suction state before compression is known, thanks to the assumption of constant superheat temperature value before compression). Discharge state of the compressor (point 2 on fig. 2.14) can be therefore determined as follows:

$$h_2 = h_1 + \frac{h_{2s} - h_1}{\eta_{is}} \cdot (1 - \beta) \,, \tag{6.4}$$

where:

η_{is} - isentropic efficiency of the compressor [-],

h_1 - specific enthalpy of refrigerant at suction state [$\frac{kJ}{kg}$],

h_2 - specific enthalpy of refrigerant at discharge state after actual compression [$\frac{kJ}{kg}$],

h_{2s} - specific enthalpy of refrigerant at discharge state after isentropic compression [$\frac{kJ}{kg}$],

β - heat loss factor [-].

In theory compressor work is equal to total electric power input which should be supplied to the compressor. However, in practice (due to energy losses) power input must be higher:

$$W_C = P_C - \beta \cdot P_C = P_C \cdot (1 - \beta) \,, \tag{6.5}$$

where:

W_C - work of the compression, taking heat losses into account [W],

[94] Multiplication of the refrigenrat vapour density and its specific heat of vaporization. Higher VRC allows for lower compressor displacement volume for any given rate of heat transfer, hereby allowing for smaller compressor sizes.

P_C - electric power of the compressor [W].

On the other hand:

$$W_C = \dot{m}_{CO_2} \cdot (h_2 - h_1) = \dot{m}_{CO_2} \cdot \frac{h_{2s} - h_1}{\eta_{is}} \cdot (1 - \beta) \tag{6.6}$$

Total power input of the compressor can be therefore defined as:

$$P_C = \frac{W_C}{1 - \beta} = \dot{m}_{CO_2} \cdot \frac{h_{2s} - h_1}{\eta_{is}} \tag{6.7}$$

6.2.1.3. Modeling of the evaporator

Similarily as the compressor, evaporator model described in this section is also based on several assumptions:

- heat losses to surroundings are negligible,
- constant pressure and temperature of CO_2 throughout the whole heat exchanger,
- constant overall heat transfer coefficient of the evaporator (on both refrigerant and secondary fluid side)
- constant specific heats of both R744 and secondary fluid during the whole CO_2 evaporation process.

For calculations of the energy effectiveness and heat transfer rate in evaporator NTU method (wider described in the attachment 2) was utilized. Assuming counter-flow character of the evaporator and phase change occurence on the refrigerant side, the following equations were used:

$$NTU_{EV} = \frac{U_{EV} \cdot A_{EV}}{C_{sf,EV}} \rightarrow \varepsilon_{EV} = 1 - e^{-NTU_{EV}}, \tag{6.8}$$

$$\dot{Q}_{EV} = \varepsilon_{EV} \cdot C_{sf,EV} \cdot \left(T_{r,EV} - T_{sf,EV,i}\right), \tag{6.9}$$

where:

$C_{sf,EV}$ - heat capacity rate of the secondary fluid (e. g. air, water or glycol mix) in evaporator [$\frac{J}{K \cdot s}$],

$T_{r,EV}$ - constant temperature of the refrigerant (CO_2) in evaporator [^oC],

$T_{sf,EV,i}$ - inlet temperature of the secondary fluid in evaporator [^oC].

6.2.1.4. Modeling of the gas cooler

For gas cooler model the following issues were assumed:

- heat losses to surroundings are negligible,
- thermodynamic properties of fluids exchanging the heat in gas cooler (CO_2 and water) are constant: within its each subsection (CO_2) and for the whole heat exchanger (water),
- pressure drop through the gas cooler is neglected but temperature drop is considered,

- overall heat transfer coefficient of the gas cooler (on both refrigerant and secondary fluid side) is constant.

The gas cooler model presented in this section has common features with that of Yokoyama et al. (2007) [6]. It was chosen to define this model due to three main reasons:

- good agreement of the model with experimental results over a large range of operating conditions,
- Yokoyama's heat pump model was designed to co-operate with hot water storage tank, similarily as the heat pump in this master thesis,
- the model is relatively simple to be easily incorporated into the building simulation environment.

Gas cooler has to be first divided into N subsections with equal heat exchanger area and heat transfer factor of each subsection can be then calculated as:

$$UA_j \equiv U_j \cdot A_j = \frac{U_{GC} \cdot A_{GC}}{N}, \tag{6.10}$$

where U_j, A_j - parameters (U-value and heat exchange area) of the j - subsection ($j \in <1, N>$).

Pressure drop in the gas cooler is not taken into account, because it is anyway relatively low. Important difference between gas cooler modeled in master thesis and this in Yokoyama approach is that in his survey the LMTD (Logarithmic mean temperature difference) method was applied to determine heat transfer rate in gas cooler but due to the lack of necessary temperature values, in case of this thesis NTU (Number of Transfer Units) method should be utilized for that purpose (similar as in the evaporator case). Thanks to division of the gas cooler into subsections strongly varying properties of refrigerant during cooling process could be accounted for. Furthermore, modeled gas cooler takes transport properties into account, what complicates calculations of the model but gives more accurate results of simulation. Constant CO_2 properties for each subsection of the gas cooler are always defined for the arithmetic mean temperature, counted as below:

$$T_{avg,r,j} = T_{r,j-1} + \frac{T_{r,j} - T_{r,j-1}}{2}, \tag{6.11}$$

where:

$T_{r,j}$ - inlet temperature of refrigerant (CO_2) in currently analyzed gas cooler subsection [^oC],

$T_{r,j-1}$ - outlet temperature of refrigerant (CO_2) in currently analyzed gas cooler subsection [^oC].

Division of the gas cooler into subsections and presentation of the control volume of one subsection are schematically depicted on the figures 6.5 and 6.6 respectively.

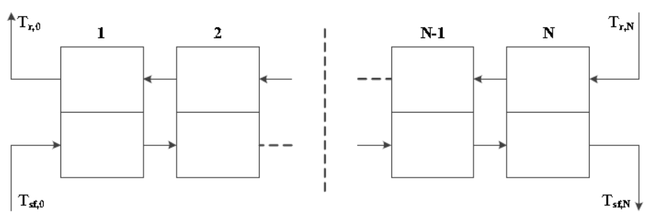


Figure 6.5. Illustration of the gas cooler subsection division [9]

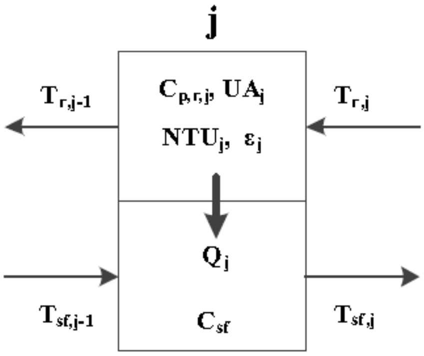


Figure 6.6. Control volume of each gas cooler subsection [9]

Based on NTU method principles the energy effectiveness of subsection *j* was calculated in the following way (assuming counter-flow arrangement of the gas cooler and no phase change):

$$\mathrm{NTU_j} = \frac{\mathrm{UA_j}}{\mathrm{C_{min,j}}} \rightarrow \varepsilon_j = \frac{1 - e^{-\mathrm{NTU_j}(1-\mathrm{C_{ra,j}})}}{1 - \mathrm{C_{ra,j}} \cdot e^{-\mathrm{NTU_j}(1-\mathrm{C_{ra,j}})}} \tag{6.12}$$

Next, heat transfer rate and outlet temperatures of fluids could be counted as:

$$\dot{Q}_j = \varepsilon_j \cdot \dot{Q}_{max,j} = \varepsilon_j \cdot C_{min,j} \cdot (T_{r,j} - T_{sf,j-1}) \tag{6.13}$$

$$T_{sf,j} = T_{sf,j-1} + \frac{\dot{Q}_j}{C_{sf}} \tag{6.14}$$

$$T_{r,j-1} = T_{r,j} - \frac{\dot{Q}_j}{C_{r,j}} \tag{6.15}$$

In turn, for the whole gas cooler the effectiveness, heat transfer rate and outlet temperatures of the secondary fluid can be calculated as below:

$$\mathrm{NTU_{GC}} = \frac{U_{GC} \cdot A_{GC}}{C_{sf,GC}} \rightarrow \varepsilon_{GC} = \frac{1 - e^{-\mathrm{NTU_{GC}}(1-C_{ra,GC})}}{1 - C_{ra,GC} \cdot e^{-\mathrm{NTU_{GC}}(1-C_{ra,GC})}} \tag{6.16}$$

$$\dot{Q}_{GC} = \varepsilon_{GC} \cdot \dot{Q}_{max,GC} = \varepsilon_{GC} \cdot C_{min,GC} \cdot \left(T_{r,GC,i} - T_{sf,GC,i}\right) \tag{6.17}$$

$$T_{sf,GC,o} = T_{sf,GC,i} + \frac{\dot{Q}_{GC}}{C_{sf,GC}} \tag{6.18}$$

The refrigerant outlet temperature from gas cooler must be determined as the outlet R744 temperature from the last subsection, i. a. due to different heat capacity rate of CO_2 in each subsection along the gas cooler.

6.2.2. Optimal gas cooler pressure

Due to the fact that heat transfer process in gas cooler of the heat pump analyzed in this thesis was assumed as isobaric and gas cooler pressure has direct influence on the heat pump performance, choice of the optimum gas cooler pressure is very important, therefore the author of this work decided to depict this issue in separate section.

Lorentzen in his work [20] suggested that for any operating condition there exists optimal gas cooler pressure, which results in the highest COP of the (transcritical) heat pump. In the past, many people were conducting experiments, trying to define empirical formulas for optimum gas cooler pressure which were based on results of their experiments. Liao, Zhao and Jakobsen [22] in their work showed that optimal heat rejection pressure of CO_2 heat pump is mainly dependent on the gas cooler CO_2 outlet temperature, the evaporation temperature and the compressor specification. If to assume constant volumetric and isentropic efficiencies of compressor, then optimum gas cooler pressure can be expressed as the function below:

$$p_{GC,opt} = f\,(T_{r,GC,o}, T_{r,EV}) \tag{6.19}$$

It was further proved that dependency between optimum pressure and $T_{r,GC,o}$ is stronger than between the pressure and $T_{r,EV}$. Moreover, it was indicated that influence of the evaporation temperature on the pressure decreases at lower evaporation temperatures. Chen and Gu [21] investigated experimentally the optimum heat rejection pressure for transcritical refrigeration system. They proposed a linear correlation between $T_{r,GC,o}$ and $p_{GC,opt}$. Their relationship predicted optimal gas cooler pressure with maximum deviation of 3,6% from the actual optimum [9] in evaporation temperature range of -10 ÷ 10^0C and CO_2 outlet temperatures of 30 ÷ 50^0C.

Thoreby in his considerations of the optimum gas cooler pressure accepted constant value of the ambient air (yearly average, i. e. 7ºC). He assumed a linear relation between the optimal gas cooler pressure and the secondary fluid inlet temperature, when the water outlet (setpoint) temperature was held constant, similar to the approach of Chen and Gu. Afterwards, a linear relation between optimum pressure and secondary fluid outlet temperature was considered, with a constant water inlet temperature. The proposed correlation for control of the gas cooler pressure was then determined by method of least squares, as:

$$p_{GC,opt} = 0{,}00504T_{sf,O,GC} \cdot T_{sf,I,GC} + 0{,}773T_{sf,O,GC} + 0{,}269T_{sf,I,GC} + 41{,}3\,, \tag{6.20}$$

where temperatures are expressed in [ºC] and pressure in [bar].

Secondary fluid (water) inlet and outlet (setpoint) temperature ranges taken into account in determination of $p_{GC,opt}$ are depicted in table 6.1. After definition of the optimum pressure Thoreby implemented its formula into EnergyPlus, as it is an open source software. He also presented a comparison between $p_{GC,opt}$ predicted from equation 6.20 and actual optimal pressures in temperature ranges of interest. This comparison can be seen on the figure 6.7.

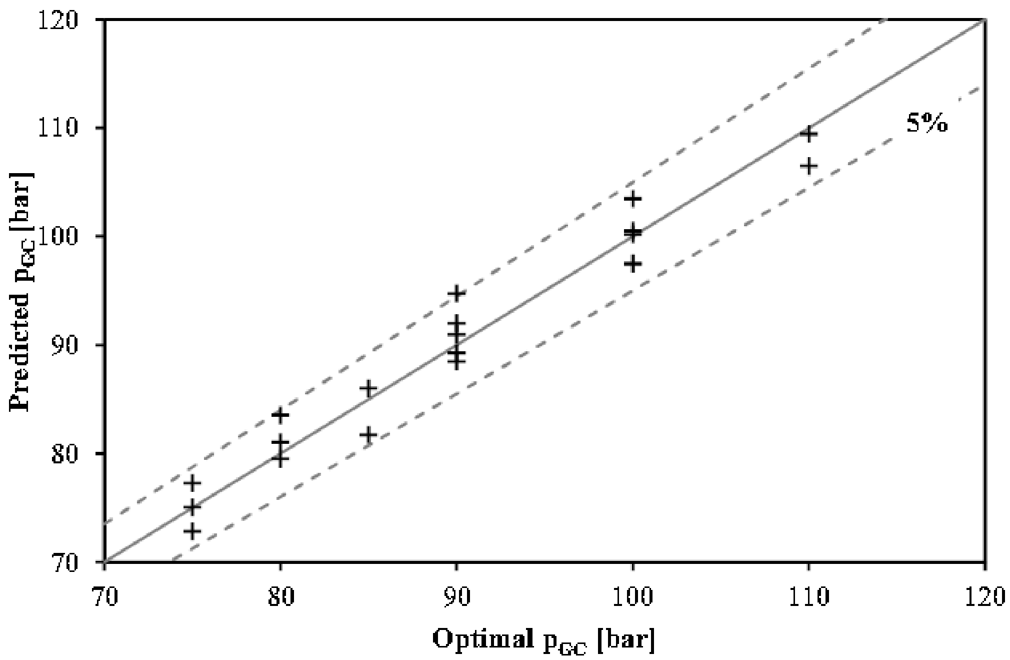


Figure 6.7. Comparison of optimum gas cooler pressures predicted by equation 6.20 and the actual optimum pressures determined in conducted experiments [21], assuming constant outdoor air temperature of 7^{0}C [9]

As it can be noticed on fig. 6.7 the predicted pressures generally stay within $\pm 5\%$ of the actual pressures in temperature ranges of interest[95]. However, it is worth to remember that deviations would probably be larger if to assume varying outdoor air temperature, what however was not analyzed.

6.3. Energy supply system in passive house

One of the main goals of this master thesis was to optimize the energy supply system, thus it had to be first designed from the beginning by the author of this work (a lack of Thoreby's heat pump project file unabled continuation of his work). Furthermore, new creation of the heat pump system was necessary, because heat pump had to be adjusted to the passive house requirements (previously it was designed for the same building but in low energy standard). It was also desired to project a heat pump with tri - partite gas cooler construction shown on fig. 6.2, which Thoreby could not develop using EnergyPlus, due to limited capability of the program.

After evaluation of different simulation tools conducted in chapter 4 it was finally decided to use two different programs (i. e. IDA ICE and Dymola) for further analysis of the energy supply system. However, in design process of the floor heating distribution system, hot water storage tank and cooling system IDA ICE was used. In turn, for the heat pump creation Dymola was applied. Heating demand of the building (from viewpoint of SH and DHW needs) and some other data necessary for dimensioning of the heat pump in Dymola were therefore taken from IDA ICE as a part of the results of performed simulations.

[95] Inlet and outlet temperature ranges of the secondary fluid.

Due to the fact that two programs were used in design phase of the energy supply system for the passive house, this chapter will be divided in two separate sections.

6.3.1. Development of the heat pump model in Dymola

The newest data library for Dymola - TIL - allows for more efficient design of the heat pump, assuming similar accuracy of results. Based on this library, author created desired transcritical heat pump model with tripartite gas cooler (preheater and reheater for domestic hot water heating and one gas cooler segment for heating of the water circulating in the underfloor heating circuit), instead of intermediate water loop connecting gas cooler and ideal heat exchangers as Thoreby did in his work (fig. 6.3). Beside of this issue, there are some another relevant features which were changed comparing with the heat pump designed by Thoreby:

- an ambient air - heat source supplying the heat pump - was replaced with the ground. This change was caused by two following reasons:
  - it was desired to develop a transcritical heat pump which could function in monovalent mode of operation, without additional heating devices;
  - the author of this thesis wanted to design a heat pump model which would be relatively close to the reality - the assumption of constant annual ambient air temperature by Thoreby in his air - based heat pump project was not realistic at all. However, the assumption of constant temperature while ground source is considered is much more true.

  It was desired to create a brine loop burrowed in the ground, obtaining heat from the level below $10 \div 18$ m, where temperature is almost constant and equal to average annual temperature of the ambient air or even higher [81, 82, 86]. However, due to lack of brine in the media library of Dymola, water as a heat carrier was applied, with constant temperatures while flowing in and out of the evaporator (8$^{\mathrm{o}}$C and 1$^{\mathrm{o}}$C respectively). In the first approach (before optimization) water mass flow rate releasing the heat in evaporator was also assumed as constant (0,5 $\frac{\mathrm{kg}}{\mathrm{s}}$, i. e. 30 $\frac{\mathrm{l}}{\mathrm{min}}$). Water pumps currently available on the market requires $35 \div 200\,\mathrm{W}$ of electric power to maintain the waterflow on such level, depended of the head value[96] [97, 98];
- temperatures of technical water from underfloor heating circuit - Thoreby assumed constant supply and return temperatures of the water circulating in space heating distribution system (35$^{\mathrm{o}}$C and 30$^{\mathrm{o}}$C). The author of this master thesis assumed 34$^{\mathrm{o}}$C as desired supply temperature (lower temperature for better thermal comfort) and variable return temperature, varying in the range of $\sim 22\ \div \sim 29^{0}$C, dependent on the actual space heating demand;

[96] A height at which a pump can raise water up.

- temperatures of domestic hot water - Thoreby assumed 10°C as inlet temperature to the first heat exchanger (preheater) and 65°C as outlet temperature from second heat exchanger (reheater). The author of the thesis (due to willing of keeping DHW temperature in the upper part of storage tank always[97] above 65°C, in order to avoid Legionnaires disease - Legionella bacteries die within two minutes at 66°C [51]) decided to warm DHW up to 70°C after reheater, and assumed its constant inlet temperature of 9°C before preheater;
- the separator was applied after evaporator in order to separate gas and liquid phase of the refrigerant. Its usage protects the compressor against compression of the liquid, what may have destructive influence on its lifetime and proper work. The liquid part of refrigerant appearing in separator is actually a result of its non - complete evaporation in the evaporator, thus it is normally pumped back to the state before evaporator to be evaporated again. Separator utilized in TIL library works properly if the filling level is between 5% and 95%. If not, either some gaseous fraction is pumping before evaporator with some liquid part (when filling level is below 5%) or some liquid fraction is going through separator to be compressed with gaseous part (filling level above 95%);
- electric top up heaters presence for both space heating and domestic water heating was rejected (since the heat source changed from ambient air into the ground, author of this thesis assumed that heat pump can fulfill 100% heating demand of the passive house alone). The maximum heating power of the heat pump is therefore different than in Thoreby's project - it equals 16 kW instead of 9 kW.

It was desired to control the gas cooler pressure using equation 6.20 determined by Thoreby, however due to presence of two secondary fluids instead of one (caused by tripartite gas cooler construction) this correlation could not be used. A constant gas cooler pressure of 90 bar was therefore assumed as the optimum for the whole analysis. The outlook of the heat pump model developed in Dymola is shown on the figure 6.8. In turn, operation of the heat pump was continuous (not intermittent like in Thoreby's analysis) in wide range of required power (exluding only periods when space heating demand was below 85 W and DHW demand was 0 W). In practice, heat pump was working all the time with intervals existing only when both space heating and DHW heating was not needed (i. e. every day between 0 a.m. and 2 a.m during the summer months - when only DHW heating was required - and rarely or never during winter months).

[97] Even in periods of water consumption.

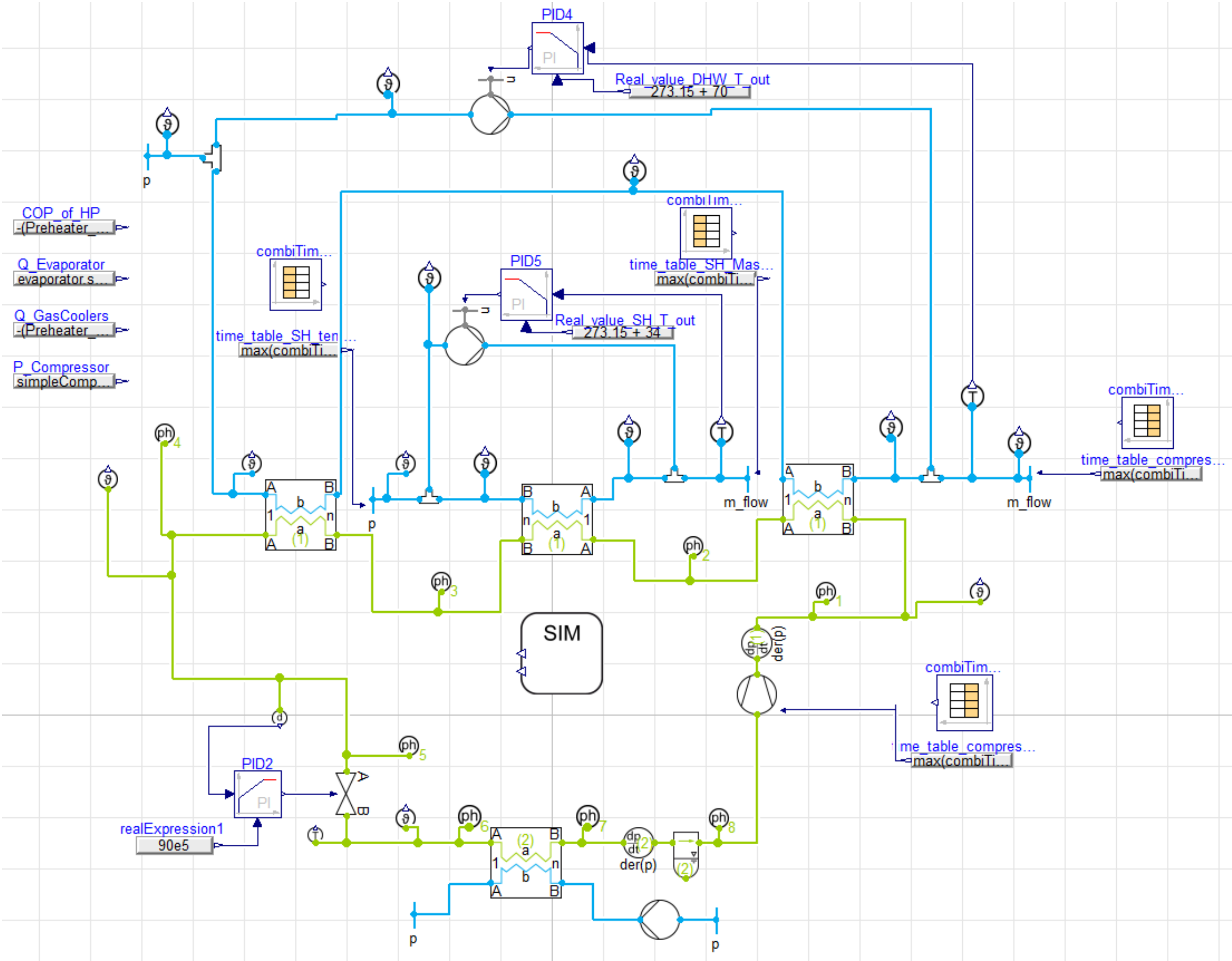


Figure 6.8. Transcritical heat pump model developed in Dymola by the author of this thesis

For the expansion valve a PI (Proportional - Integral) controller was applied, for maintaining the high pressure at desired level of 90 bar all the time, independently from variable mass flow rate flowing in the circuit (and changing heating needs of the building during the year). An orifice was assumed as the type of expansion valve, therefore equation 6.1 is surely utilized in calculations performed by the software.

In case of evaporator, calculations of the heat exchange made by the program are based on NTU method depicted in attachment 2. Many parameters were defined inside the evaporator model, among which the most important are presented in the table 6.2. Pressure drop in the evaporator was neglected on both CO_2 and water side.

For realization of the compression process it was desired to apply reciprocating compressor, with isentropic efficiency of 0,55 and volumetric efficiency of 0,7. It was also desired to control the compressor with PI controller, adjusting the current CO_2 mass flow rate to variable space heating and domestic hot water flow rates. Moreover, it was desired to define discharge state after compression - to do that it is required to provide an information about at least one more parameter than pressure on the heat pump demand side (which was set to 90 bar at it was mentioned recently). Inside TIL library of Dymola there are few types of compressor models, which can be defined by providing different input data (i. e. simple, reciprocating, efficient or scroll compressor). Unfortunately, there is no model which allows for both adjustment of refrigerant mass flow rate and definition of the discharge state. Furthermore, it is even not allowed to regulate two different mass flow rates by one compressor model (each compressor can regulate only one parameter). Thus it was decided to assume a simple compressor model, which allows for definition of discharge state after compression. In case of R744 flow its adjustment to variable demand was done by attaching its values in appropriate timetable and connecting the table with compressor. The values were roughly calculated in a function of the time based on total (space heating and DHW) heating power which had to be supplied to the building, with a timestep of 1 hour. Afterwards, these values were precisely customized manually. More details about implementation of the values placed in timetables into the program can be found in digital attachment 5.

Table 6.2. Parameters of the water - CO_2 evaporator defined in Dymola

Name of the parameter	**Value and unit**
Type of heat exchanger geometry	Tube and tube heat exchanger
Length and number of parallel tubes on the refrigerant side	0,5 m, 40
Length and number of parallel tubes on the secondary fluid side	0,5 m, 40
Inner diameter and cross section type of tubes on both sides	0,015 m, circular
Number of evaporator subsections (cells)	5
Heat transfer model of the evaporator and value of heat transfer factor	ConstantAlphaA, $2000\ \frac{W}{K}$
Thermal resistance factor of the wall separating fluids exchanging the heat	$6 \cdot 10^{-5}\ \frac{K}{W}$
Mass and material of the wall	6 kg, copper

Determination of all gas cooler parts in Dymola was identical and looked similar as in the case of evaporator. The only relevant difference was in definition of the heat transfer factor - $1300\ \frac{W}{K}$ was assumed instead of $2000\ \frac{W}{K}$. The number of subsections of each gas cooler part was one more time assumed as 5. It is relatively small amount compared with numbers of cells which are usually defined (e. g. Thoreby assumed 20 cells for both gas cooler and evaporator). However, assumption of lower number of cells is profitable from viewpoint of time - simulations in Dymola may last relatively long and many different errors may occure. When number of cells is limited, calculations performed by the program are clearly faster. Moreover, author of this thesis believes that 5 subsections allow for reaching the accuracy of calculations which is sufficient from viewpoint of this master thesis goals. There were also some differences between gas coolers in case of inlet and outlet specific enthalpies and mass flow rates of fluids exchanging heat in gas coolers, however due to the fact that these values were only needed for initialization and treated as start values, they were not mentioned in this section.

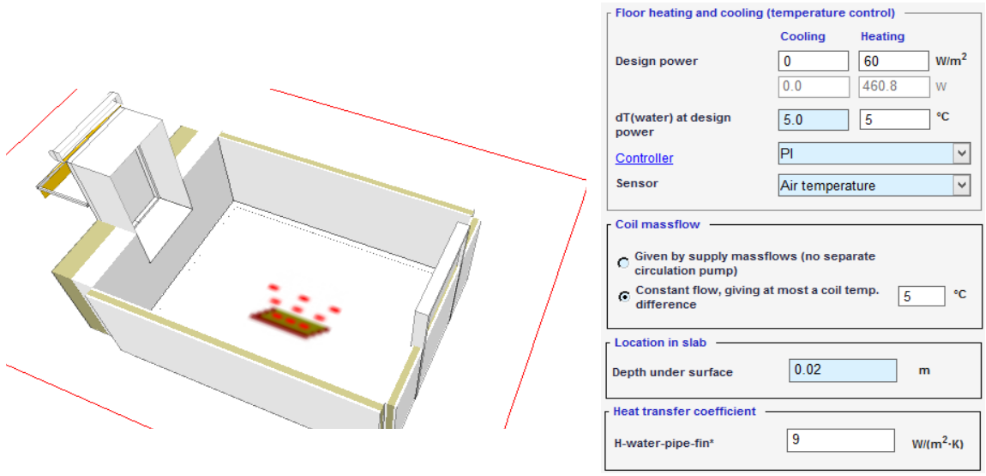
Due to the fact that both SH water and DH water flows were uptaking the heat from the same but variable refrigerant flow and their heating demands were varying throughout the time, their temperatures while flowing out of each gas cooler also varied, being often too high compared to desired values (i. e. 34°C and 70°C). In aim to equalize them for the whole simulation period (one year) additional water pumps with by-pass channels and PI regulators were added (they are visible in the upper part of fig. 6.8). These water pumps were working with speed dependent on current temperatures of water streams outflowing from gas cooler parts - if they were too high, pumps supplied appropriate quantity of colder water to three-way valves, where water streams were mixed and desired temperature values were reached.

Water pump adopted in aim to allow for circulation of technical water in the loop burrowed in the ground was assumed to have a constant nominal speed of 50 Hz, constant mass flow rate of $30\ \frac{\mathrm{l}}{\mathrm{min}}$, volume of liquid in the pump chamber equal to $0{,}4\ \mathrm{dm}^3$, nominal efficiency of 45% and nominal pressure head of 2,2 bar, what means that the water can be pumped upwards from the level placed even 22,4 m below the ground surface. As it turned out after simulation, electic power which have to be supplied to such pump was equal to ~52 W.

SIM (System Information Manager) visible in the middle part of fig. 6.8 is used for selection of fluids (media), which are considered in designed thermodynamic cycle. Hence in case of created heat pump $\mathrm{CO_2}$ was chosen as the gaseous medium and water as the liquid medium, with properties defined inside the TIL data library.

6.3.2. Heating distribution system, water storage tank and cooling system designed in IDA ICE

An underfloor water - based heating distribution system was created for the passive house in IDA ICE. Maximum capacity of designed installation is surely higher than heating demand of the building in each room throughout the simulated period. In all zones of the building maximum heating power of floor heating was defined as $60\ \frac{\mathrm{W}}{\mathrm{m}^2}$. Only in the magazines (due to technical equipment placed there, which generate the heat) a power of $30\ \frac{\mathrm{W}}{\mathrm{m}^2}$ was assumed. The outlook of waterborne floor heating panel model and its parameters which can be adjusted separately in each room are presented on the figure 6.9.a and 6.9.b respectively. As it can be noticed on fig. 6.9.b desired drop of space heating water temperature was set to 5°C (what results in return temperature of space heating water equal to 29°C), however due to variable heating demand during the time and different sizes of the zones the return temperature varied in practice between 22°C and 29°C.



a **b**

Figure 6.9. Design of the heating waterbased floor panel in IDA ICE; a - outlook of the water loop; b - parameters of the loop

In case of domestic hot water tank, due to inability of its design in Dymola it was decided to use IDA ICE. Thus a cylindrical thermally - stratified storage tank was modelled (instead of non - stratified tank, due to advantages of thermal stratification described in the chapter 3), with parameters similar to this developed by Thoreby - height of 2 m, volume of 500 dm^3 and relatively small ratio between diameter and height - 0,28, helping in maintaining of stable thermal stratification. The outlook of created tank can be noticed on the figure 6.10.

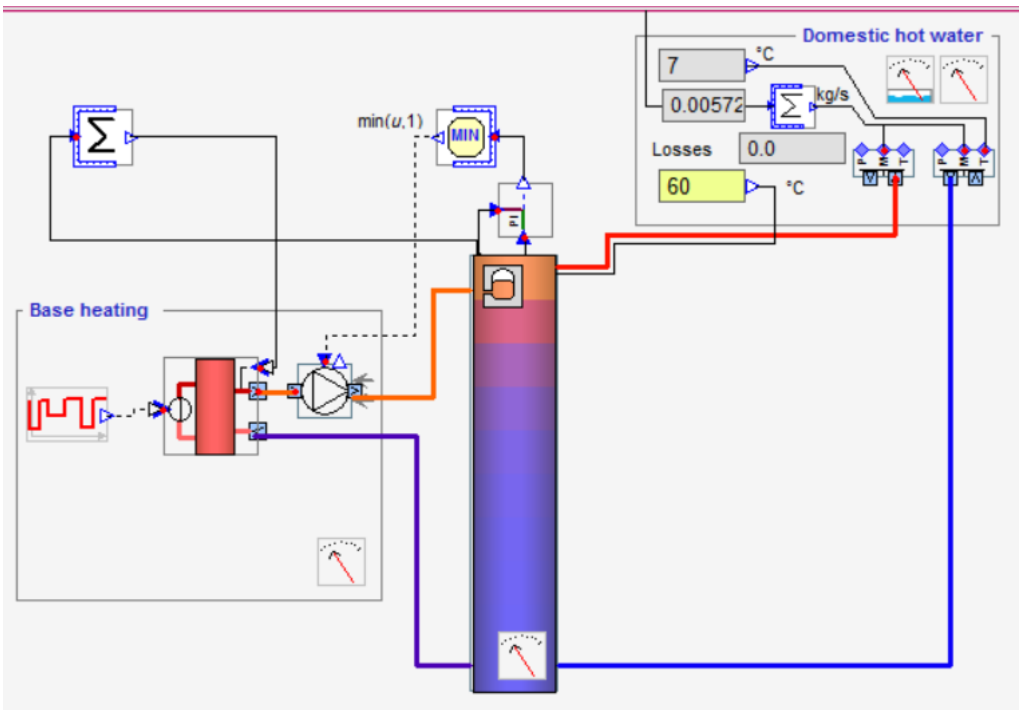


Figure 6.10. Outlook of the hot water storage tank after the whole day of operation, designed in IDA ICE

Ten layers of equal height were assumed for modeling of thermal stratification inside the tank. Although the mixing was decreased by relatively low ratio between diameter and height of the tank, in reality there always exists mixing to some extent. Inside IDA ICE many parameters of the tank can be defined, e. g. factor describing heat exchange between tank layers due to mixing processes (assumed as $50\ \frac{\mathrm{W}}{\mathrm{mK}}$) and factor describing the heat exchange due to stratification phenomenas (set as $800\ \frac{\mathrm{W}}{\mathrm{mK}}$). U-value for the whole tank envelope was set to $0{,}9\ \frac{\mathrm{W}}{\mathrm{m^2K}}$. As it was mentioned in the heat pump description, temperature of the water stored in upper part of the tank was maintained at the level of 65°C or above all the time, to maximally minimize a risk of legionella occurence. In turn, domestic hot water consumption was set as 33 litres of 60°C hot water per one occupant and day. Cold water from the main supplying the tank (in its lower part on the right side on fig. 6.10) was assumed to have a constant yearly temperature of 7°C at the inlet to the tank and 9°C while it starts to be heated in the gas cooler. As the temperature of ambient air surrounding the tank (designed to be placed inside the building) 21°C was assumed. In case of the IDA ICE model electric boiler with efficiency of 100% was adopted, playing a role of two gas coolers (preheater and reheater) designed in Dymola (fig. 6.8). In case of DHW tank utility, its daily schedule can be found in attachment 3. From viewpoint of the heat pump, it was set to heating up the water all the time excluding periods between 0 a.m. and 2 a.m.

Utility of hot water storage tank (compared to the situation in which electric boiler would be used for direct domestic water heating without tank) in the passive building resulted in the following issues:

- increase of total energy yearly spent on DHW preparation. Although the same amount of hot water is prepared (33 litres per person and day) with the same outlet temperature (60°C), water in storage tank have to be heated up to higher temperature (70°C), because water stored in tank have to have higher temperature. Therefore (even if water is started to be heated up from 7°C instead of 9°C and turning on / off cycles of the boiler appears clearly more often in case of direct heating without tank, what in overall increase the boiler energy consumption) the energy consumption in case of direct heating of the water without tank is lower. Annual energy usage would be equal to 10114 kWh without the tank and is equal to 11438 kWh with the tank, so the difference is 1054 kWh. Another factor which increase energy consumption when water tank is taken into account is the loss of heat to the ambient through the tank envelope (even if it is well - insulated);
- higher reliability of domestic hot water temperature while used. When water storage tank is applied, even if more people need hot water at the same time they can all use the water with the

same high and stable temperature of max. 60°C without its decrease, because the water is already heated. If DHW preparation system would not contain storage tank, dependent on maximum heating power of electric boiler after reaching certain level of water consumption in the building its occupants could feel a decrease of either the water temperature or its flow rate;

- immediate hot water appearance while used. When hot water tank is present in DHW system, the water is heated up to over 65°C before usage. Thus when the occupant turns on the tap, the hot water can be there instantly. Without tank utility the water is started to be heated when the user turns on the tap, hence it is necessary to wait some time (longer or shorter, depended of the maximum power of electric boiler) before the hot water will flow out from the tap;
- use of the heating device with lower maximum power is possible. If the tank is taken into account the water can be heated there for a long time (e. g. during all the day), getting prepared for its later usage. If DHW system is not equipped with storage tank, the water is heated up only while it is used, thus heating device has to work shorter and needs to have higher maximum power to meet the demand.

As it was mentioned in the first section of this chapter, Thoreby assumed no cooling system, due to supposedly short period during the year which would require cooling (10 ÷ 20 days [9]). Author of this thesis checked it, and he is not agree with Thoreby. First of all, in opinion of the author thermal comfort inside the building zones should exist all the year. Moreover (based on simulations performed in IDA ICE) there is clearly more days than 20 which require cooling during the year, especially if the house is built in passive standard (better overall tightness of the building results in clearly lower heating demand in the winter but also higher cooling demand during the summer, compared to the low energy building). Cooling needs of the passive building throughout the summer period can be seen on the figure 6.11.

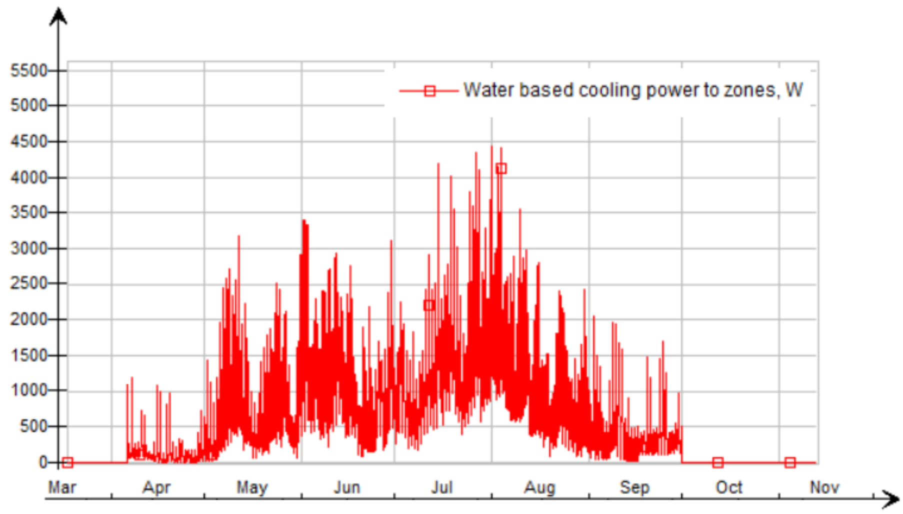


Figure 6.11. Required cooling power which have to be supplied to the passive house in order to keep indoor air temperature below 23°C

The cooling system consisted of water - based cooling beams and electric chiller with efficiency of 100% and maximum power of 5 kW was therefore applied. Similarily as in the case of floor heating panels, maximum power of each cooling beam was defined with a large reserve. More details can be found on the figure 6.12.

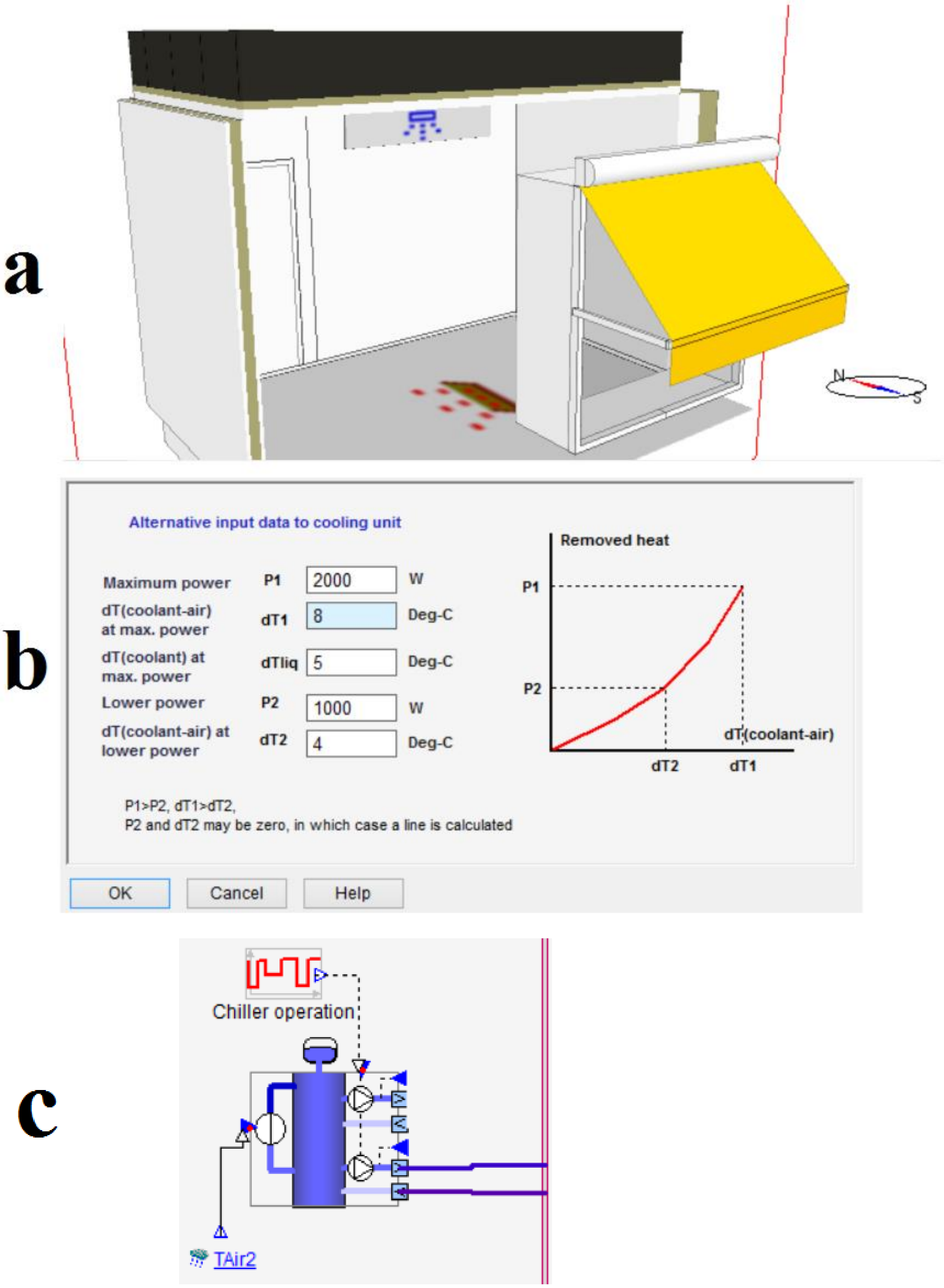


Figure 6.12. Design of the cooling system in IDA ICE for passive house; a - outlook of the cooling beam placed on the wall; b - parameters of the cooling unit; c - outlook of the electric cooler model

Presented cooling system allowed for maintaining of the indoor air temperature in the passive house on the desired level between 21ºC and 23ºC. In turn, air temperature in the low energy building without cooling system (as Thoreby proposed) is clearly higher during summer months. Comparison of achieved temperatures in both buildings, showing usefullness of the cooling system can be noticed on the figure 6.13. In addition, indoor air temperature curve for the passive house without installed cooling system was depicted, to underline higher necessity of installation of the cooling system in passive house than in low energy building.

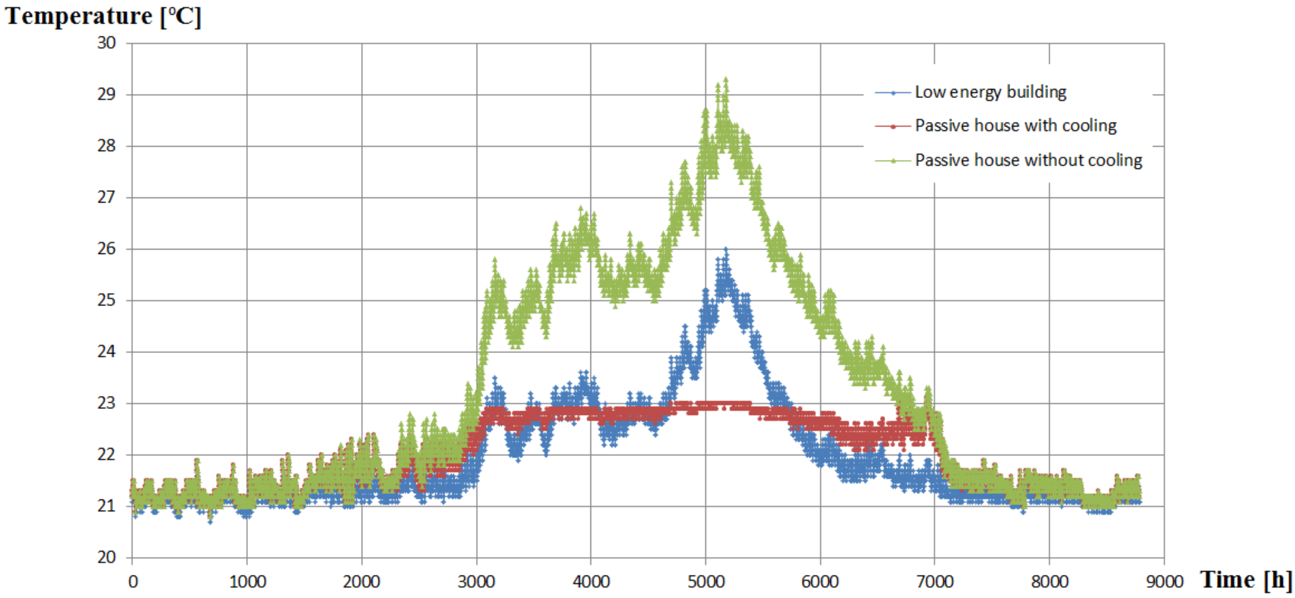


Figure 6.13. Mean indoor air temperature in considered building during the year, dependent on its type and presence of installed cooling system (all building types were simulated in IDA ICE)

6.4. Comparison of achievements between heat pump designed in EnergyPlus and Dymola

Thoreby in his thesis analyzed three different patterns of the heat pump utilization (base case with standard heat pump work as described in section 6.1, a case where DHW consumption was increased by 50% and a case where an indoor air temperature reduction was maintained during night hours [9]). For comparison of his achievements with the heat pump designed by the author of this master thesis only the base case considered by Thoreby will be taken into account.

As it was described in chapter 5, energy consumption (equal to electricity consumption) of the low energy building in Thoreby's considerations for only heating needs equalled 31031 kWh annually in the reference state (i. e. electric heating system with efficiency of 100%). In turn, when the heat pump was applied total electricity consumption decreased to 12363 kWh[98] per year [9], still covering 31031 kWh of required heating demand. Due to the fact that heat pump designed by Thoreby was air - based and co-operated with electric top - up heaters (for SH and DHW), the heat

[98] Including 11622 kWh supplied to the heat pump compressor and 741 kWh supplied to top - up electric heaters.

pump didn't cover 100% heating demand alone. However, as it turned out the heat pump was evaluated to have relatively high energy coverage factor - 96,2%, with a power coverage factor of 57% [9]. Comparison between electric reference case and heat pump base case from viewpoint of covering annual heating demand of the building can be seen on the figure 6.14.a. In turn, a heating load duration curve of low energy building when heat pump is utilized with top - up heaters can be noticed on the figure 6.14.b.

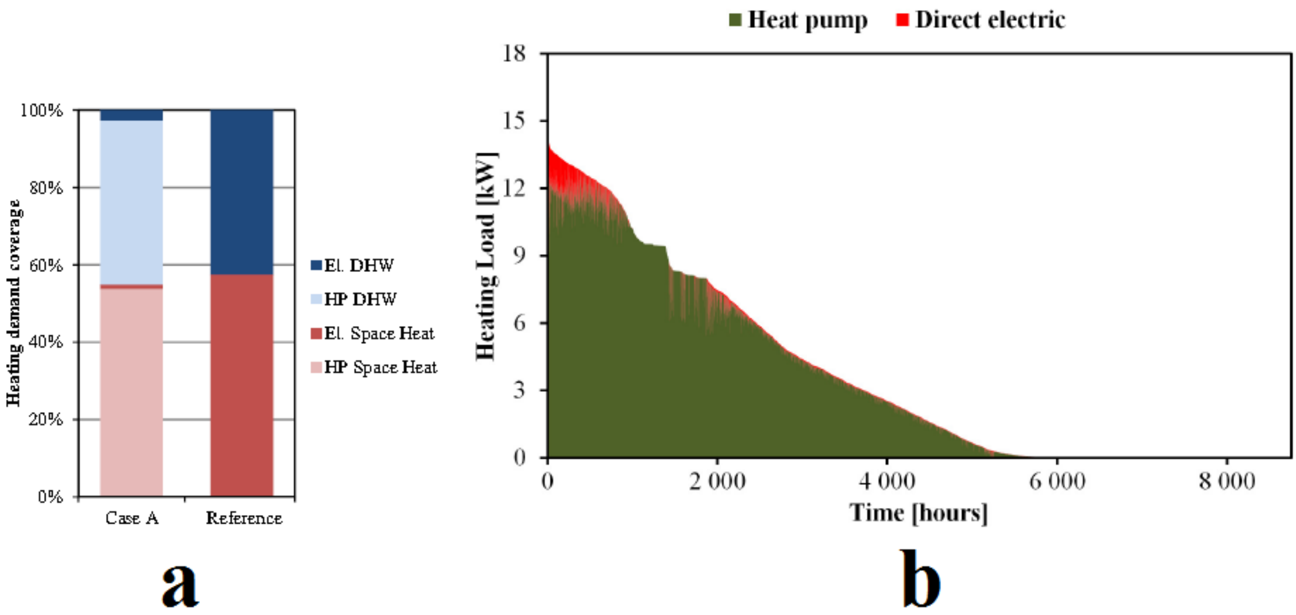


Figure 6.14. Results of the heat pump performance designed by Thoreby [9]; a - energy coverage in electric reference state and in heat pump base case; b - heat load duration curve with heat pump and top - up heaters in operation

Differences in shape between the curves depicted on fig. 6.14.b and on fig. 5.12 are the result of an intermittent operation of the heat pump developed by Thoreby, which prevented operation at low loads. The heat pump therefore operated less frequently but at higher capacity than it would with continuous operation. The highest heating loads occured when the heat pump supplied heat for both space heating and DHW simultaneously. Top - up heaters were thus almost exclusively necessary only in this operating mode. The duration curve on fig. 6.14.b also reveals that the heat pump often operated at higher capacity than its design capacity of 9 kW. This means that the operating conditions often were more favorable than design conditions. The equivalent operating time of the heat pump was equal to 3444 hours, what may indicate its relatively good utilization.

In case of COP factor achieved by air-based heat pump designed by Thoreby, it mostly varied between ~2,2 and 3,2 during the year [9]. Unfortunately, COP values changing during the simulation period cannot be shown, because Thoreby did not make such plot in his thesis. However, SPF factor of his heat pump for the analyzed year was equal to 2,67 and SPF for the whole system (heat pump and top - up heaters) equalled 2,51 [9]. However, Thoreby in his calculations of COP and SPF did not include a power of the fan working in the air - based heat exchanger.

In turn, ground - source heat pump designed by the author of this master thesis achieved SPF equal to 2,55 (excluding power of the water pump) and 2,40 (including this power). Its COP (assuming only compressor power) varied in the range of 1,09 ÷ 5,76 and in the range of 0,03 ÷ 4,95 (taking both water pump and compressor power into account). Changes of the COP throughout the simulated 2016 year are depicted on the figure 6.15.

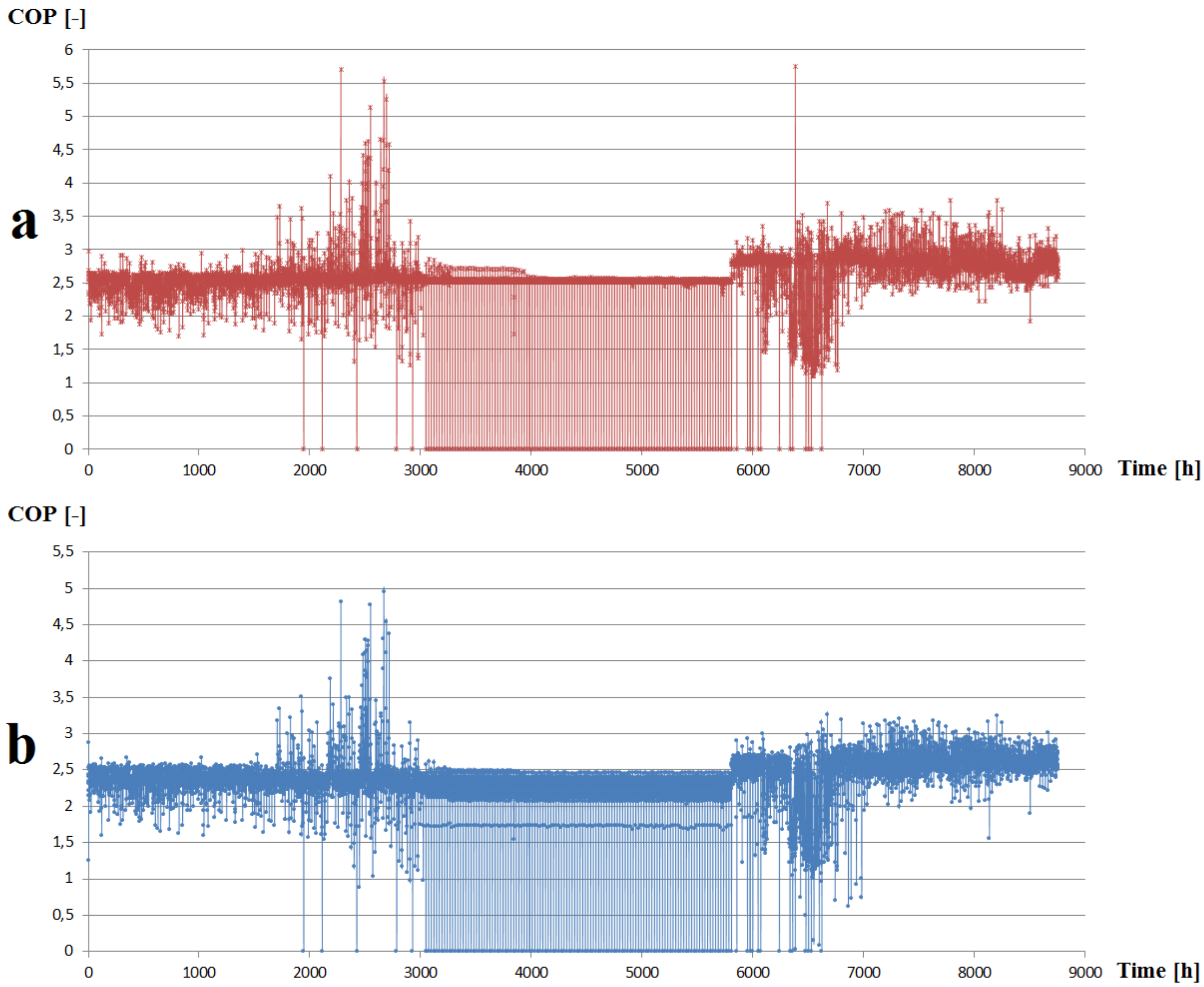


Figure 6.15. COP variations of the heat pump designed in Dymola during considered simulation period; a - excluding water pump power; b - including water pump power

COP of the heat pump had a value between 0 and 1 14 times during the year (i. e. during 14 hours), due to very low heating power supplied by heat pump, compared to which the water pump power of 52 W was a high value. However, heat pump worked properly all the time, as it can be seen looking at COP values excluding this factor (the minimum COP was equal to 1,09). Furthermore, even when the water pump power was included in calculations, during 92,5% of the year (i. e. during 8105 hours, whilst total duration of the year is 8760) COP of the heat pump was over 2. In turn, when water pump power was not taken into account, COP was over 2 during 95,5% of the year (8365 hours).

As it was described in chapter 5, total heating demand of the passive house was equal to 19368 kWh (based on simulations performed in IDA ICE with electric reference model, assuming the boiler efficiency of 100%). Hence the electricity yearly supplied to the building had the same value. The heat pump designed in Dymola met this demand in 100% - total energy annually supplied by all (three) gas cooler parts was equal to 19578 kWh, whilst only 8146 kWh (including water pump) or 7690 kWh (excluding pump power) had to be supplied to the compressor by means of electricity. Thus the water pump (having efficiency of 45%) consumed 456 kWh of electricity annually.

Among total heating demand of the building, 7930 kWh was spent on space heating and 11438 kWh was consumed for DHW preparation. In case of heat pump, middle part of gas cooler (responsible for space heating preparation) supplied 8256 kWh annually, whilst preheater and reheater (two other gas cooler parts) supplied 4737 kWh and 6585 kWh respectively (11322 kWh in total).

Heat load duration curve of the passive house (where heating needs were covered by the heat pump) is presented on the figure 6.16, in order to compare energy supplied by the heat pump during each hour of the year with heating demand of the house calculated taking 100% efficient electric boiler into account. As it was mentioned in chapter 5, maximum heating load of the passive building was equal to 15,44 kW, whilst maximum heating power provided by gas coolers equalled 15,97 kW.

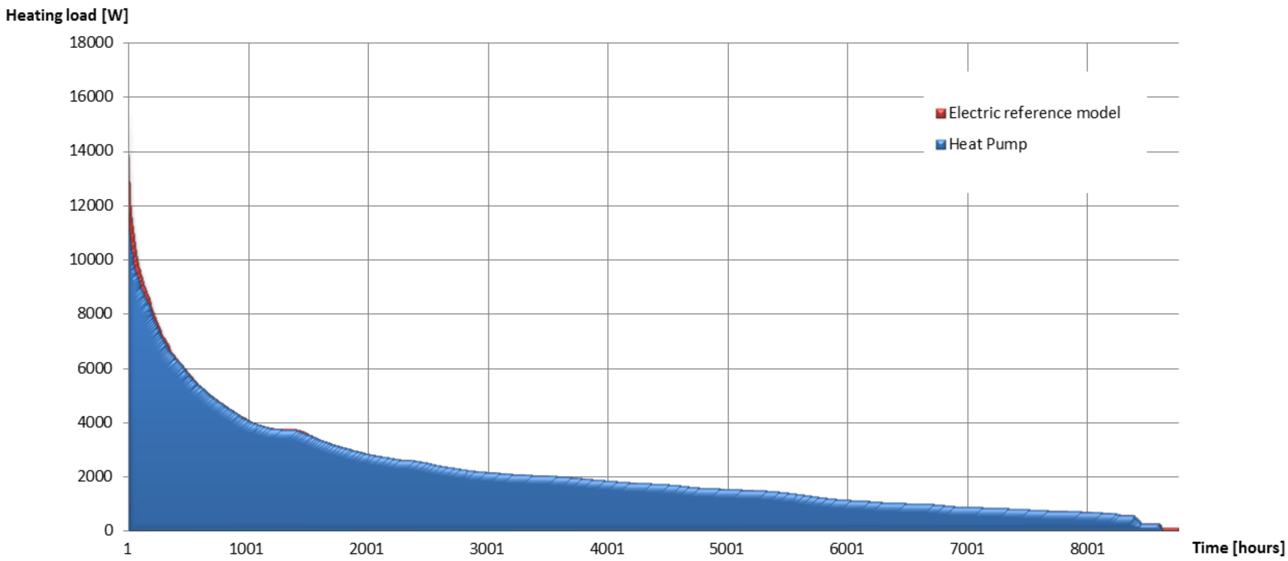


Figure 6.16. Heat load duration curve of the passive house with heat pump in operation and with utilization of electric boiler in the reference state

7. Optimization of heating and cooling system in the passive house

Optimization of the energy supply system in modern building is a really overall term, related with improving of the system performance, what can be done in many ways. It is not an easy task and type of its execution is depended of the level of complexity and design precision of the system before optimization.

In the first part of this chapter a theory containing main aims of the energy supply system optimalization and different optimization approaches will be described. As the aim of this master thesis is optimization of both heating and cooling system, in the second part of the chapter optimization of different parts of the energy supply system (transcritical heat pump, water storage tank and cooling system) will be characterized separately. Such layout of the chapter is appropriate, because specific optimization solutions are strictly dependent on type of the system, its components and aim of functioning. At the beginning of each section, possible optimization criteria will be therefore specified and afterwards the optimization of each part of the energy supply system will be performed, from viewpoint of chosen criteria. Subsequently, results of performed optimization will be described.

7.1. Basics of the energy supply system optimization

Among the main goals of the energy supply system (in terms of heating and cooling) optimalization, one can distinguish:

- improvement of the system efficiency - the most popular form of optimization, allowing for decrease of the system operation costs. It can be performed in two main ways: either by reduction of required energy supplied to the system (e. g. electricity) in order to cover energy demand of the building (in terms of heating and / or cooling) or by reduction of the energy demand;
- extension of the system lifetime - actions, which can be uptaken in order to allow for longer operation of the system, e. g. selection of more durable materials for some components of the system or improvement of conditions in which the system works, e. g. by decrease the number of turn on / off cycles or stabilization of its working conditions (e. g. avoiding too low or too high pressures inside the installation pipes);
- better control and regulation of the system operation - relevant from viewpoint of both efficient work of the system (allowing for further minimization of operation costs and even investment cost, by selection of a device with lower heating or cooling power) and quality of microclimate inside the building (increased thermal comfort experienced by occupants). It can be reached by i. a. more accurate steering of the system components, adjusting them to the actual demand varying

with the time and adopting modern regulators like PID (containing a D - derivative part, which is responsible for minimization of future errors (differences) between desired and actual value of measured parameter), application of the appropriate amount of various sensors in different thermal zones of the building (afterwards connecting them with the system) or using modern bluetooth applications, increasing awareness of occupants about the current energy usage what may help in its more efficient management by introducing individual control strategies into central control panel which steers both heating and cooling system in the building.

In case of each part of the energy supply system optimization can be conducted either:

- directly - focusing on the system only, changing its operating conditions or control strategy of its one or few elements. Direct optimization can be done in two ways:
  - introducing changes to only currently existing components (e. g. adjustment of currently existing single speed compressor, by checking which level of its speed is the most profitable to cover demand)
  - replacing some components with the new ones (e. g. exchange of old single speed compressor into the new one with variable speed, which after regulation will be more profitable in operation than the old one)
- indirectly - focusing on the other factors influencing the system operation, however not holistically related with the system (e. g. improvement of the cooling system achievements by modification of the ventilation system strategy). Similarily as in case of direct optimalization methods, indirect optimization can be also performed by modification of either existing elements or the new ones.

Although some of the people assume that optimization must be always direct and done via modification of only existing components, the author of this master thesis believes that the most important thing is to improve the energy supply system performance, whilst the way of doing it has less relevant meaning. However, it is always necessary to think about profitability of each improvement (sometimes costs of optimization may be higher than predicted savings).

7.2. Optimization of the heating system

In the passive house developed in IDA ICE by the author of this thesis, heating system consists of few main elements:

- transcritical ground - source heat pump (supplying the heat to SH and DHW subsystems),
- space heating distribution system,
- domestic hot water system.

In this work a scope of the heating system optimization was limited to the heat pump and domestic water storage tank, mostly due to limited time destined for preparation of the thesis.

7.2.1. Optimization of the heat pump

There are many optimization criteria, indicating how the performance of considered heat pump can be improved. Among these which are the most worth to mention one can distinguish:

- implementation of the suction line heat exchanger (also called as internal heat exchanger), transferring heat inside the transcritical cycle, uptaking the heat from CO_2 mass flow rate from the place located after it flows through all gas cooler parts and rejecting this heat to CO_2 flow in place before compressor. As the result, COP of the heat pump during the year and its overall SPF value increases;
- change of the orifice expansion valve into thermostatic expansion valve or pulse modulation valve, allowing for more accurate regulation of high pressure;
- find the most optimal pressure, which could be maintained in tri - partite gas cooler, what can be done by determination of correlation allowing for calculation of this pressure, mainly depended of desired secondary fluids inlet and outlet temperatures, similarily like Thoreby did for one secondary fluid by the method of least squares in his thesis [9] (it is wider described in section 6.2.2 of this work). Although the gas cooler pressure of 90 bar assumed in thesis does not give bad results, author of this thesis suppose that there surely exists more optimal pressure, which can result in higher COP and SPF of the heat pump. In overall, at each operating mode and temperature programme there exist optimum high pressure that leads to maximum COP [8];
- change of PI regulators (currently co-operating with expansion valve and water pumps on demand side of the heat pump) into PID, what could allow for even more stable steering of respective parameters;
- adjustment of volume flow rate of the water flowing through the evaporator to current heating demand of the passive house. It is highly probable that constant flow of the water ($30\ \frac{\mathrm{l}}{\mathrm{min}}$) is too high during periods when the building heating demand is low. In such periods, it decreases COP of the heat pump significantly. If to partly decrease nominal flow of the water pump during low demand periods, a drop of COP would be obviously smaller;
- increase of the water temperature while it flows into evaporator, rejecting the heat which is obtained from the ground source - by utilization of deeper boreholes. However, this modification should be particularly considered from viewpoint of profitability (increase of the investment cost related with drilling deeper boreholes may be too high compared to savings related with later

usage of the compressor; moreover, water pump which makes the water circulating in the ground loop would consume more energy, due to pumping the water from lower height level).

After deep analysis of each optimization criteria author of this thesis finally decided to consider two of them - internal heat exchanger and adjustment of the water flow in evaporator, via water pump regulation. At the first moment it was desired to consider also other optimization proposals, however due to limited time author had to reject them.

7.2.1.1. First modification - suction line heat exchanger (SLHX)

Internal heat exchanger added to the transcritical heat pump in Dymola was assumed to have construction of 4 parallel copper circular tubes on both sides (i. e. on CO_2 side releasing and uptaking the heat), with a length of 0,5 m each. Pressure drop in heat exchanger was neglected and assumed heat transfer model was the same as for evaporator and gas cooler, i. e. „ConstantAlphaA", with heat transfer factor of 350 $\frac{W}{K}$. Number of cells (subsections) was set to 4. Heat pump model designed in Dymola, including suction line heat exchanger can be noticed on the figure 7.1.

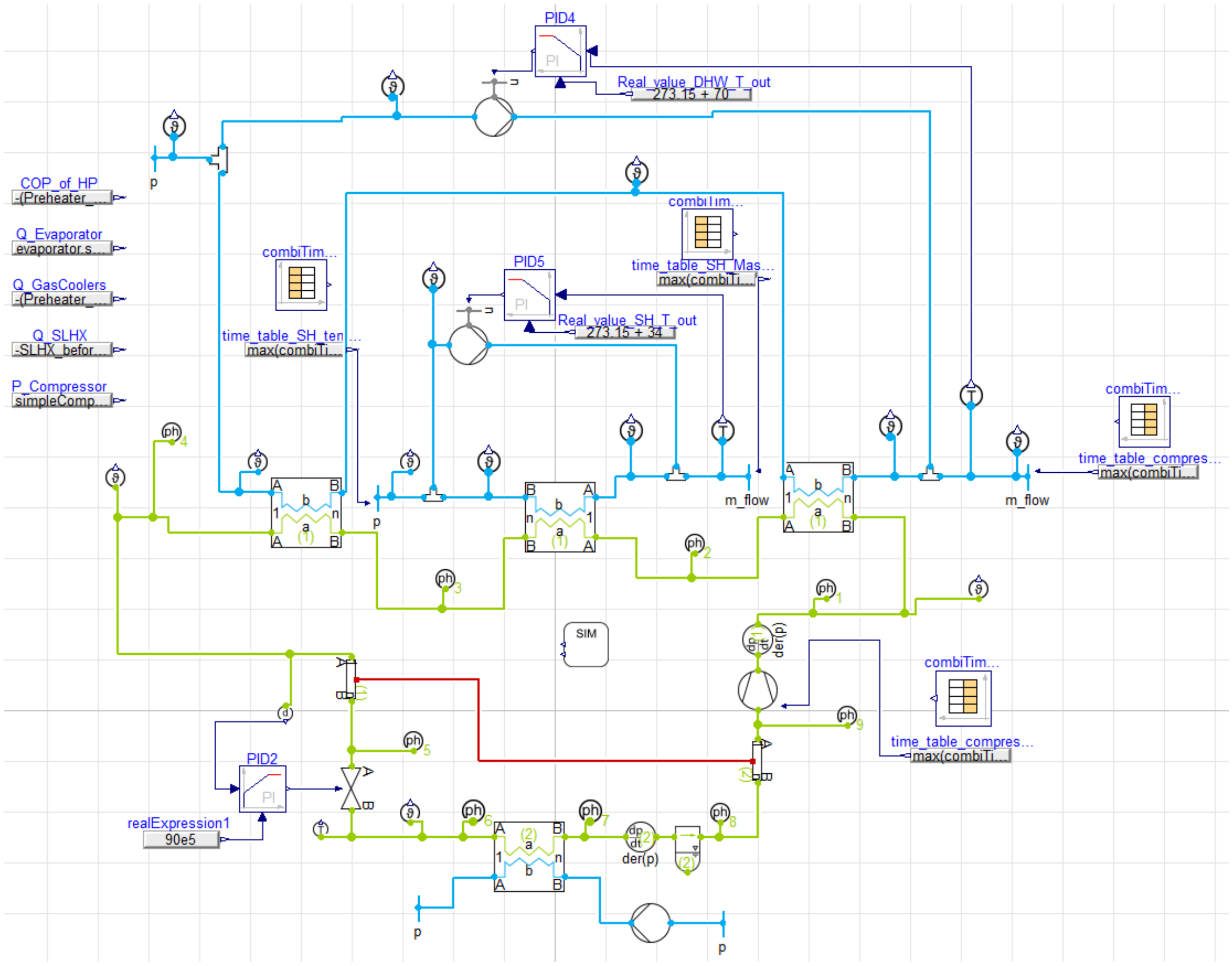


Figure 7.1. Heat pump model with SLHX developed in Dymola

After introduction of the internal heat exchanger, SPF of the heat pump raised up to 3,37 (excluding water pump (WP) energy consumption in calculations) and 3,12 (including this energy). In turn, its COP varied between 1,21 and 7,69 (excluding WP power) and in the range of 0,02 ÷ 4,75 (including the power). Total energy yearly supplied to the compressor was equal to 5817 kWh (decrease of ~24,5% compared to compressor energy consumption without SLHX, which equalled 7690 kWh), with the same overall heating energy provided by gas cooler as before optimization (i. e. 19578 kWh). COP changes of the heat pump with SLHX (including water pump power) are presented on the figure 7.2.a, while changes of heat transferred in SLHX inside the heat pump cycle are depicted on the figure 7.2.b. In turn, outlook of the heat pump cycle on p-h diagram (for the time step of 138 h[99]) can be noticed on the figure 7.3.

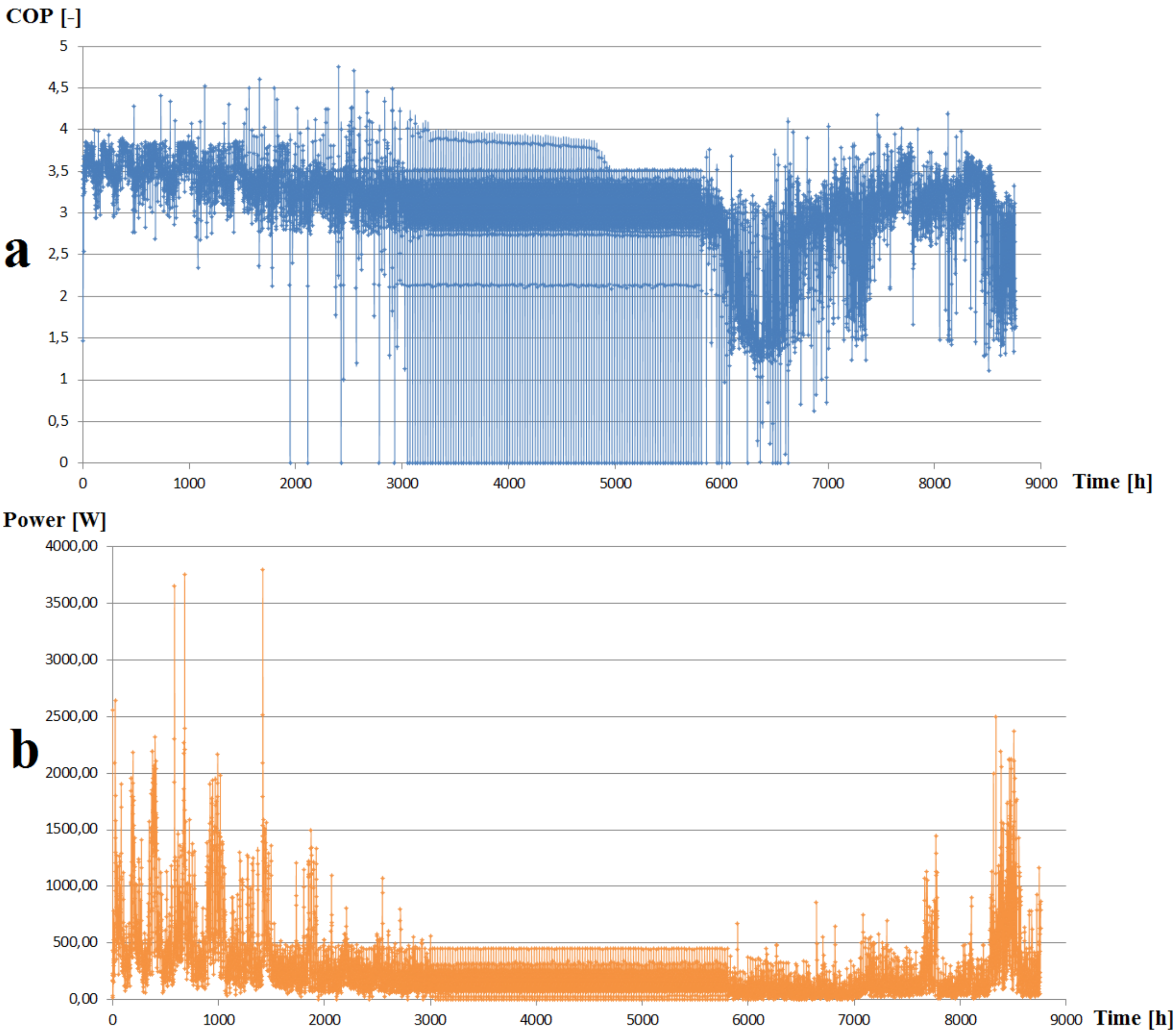


Figure 7.2. Performance of the heat pump equipped with SLHX; a - COP variations throughout the year; b - annual SLHX power changes

[99] The whole simulation period is one year, equal to 8760 hours.

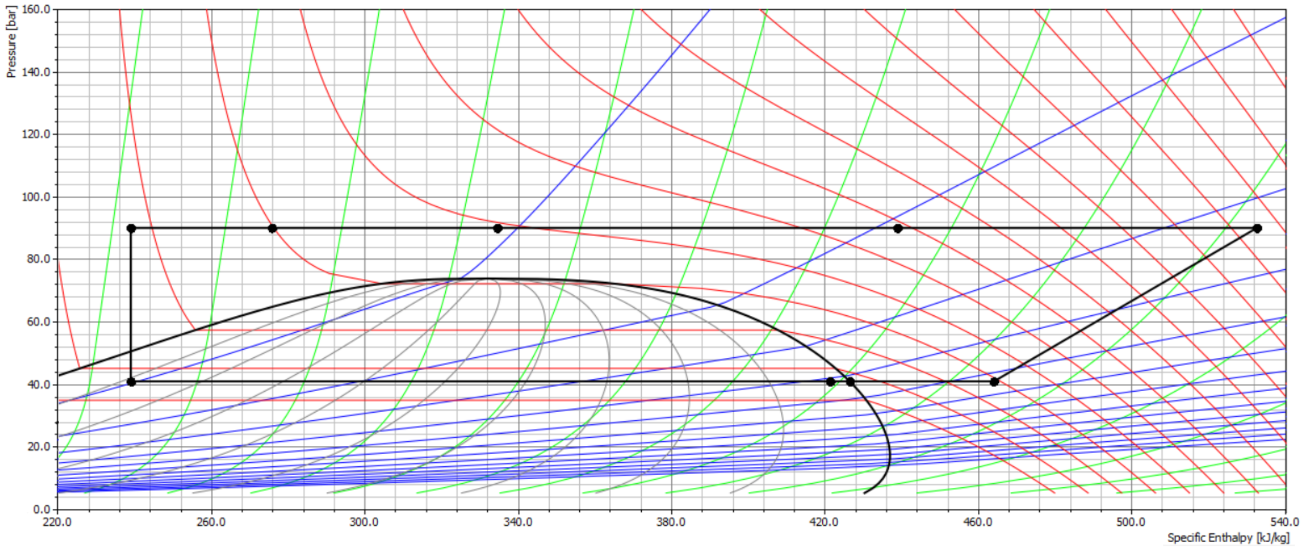


Figure 7.3. Refrigeration cycle of the transcritical heat pump equipped with SLHX depicted on p-h diagram for R744 in DaVE program

7.2.1.2. Second modification - water pump regulation

Although many heat pump sellers calculate COP of heat pumps which they are selling without including of energy consumed by water pumps (or fans, in case of air - based heat pumps), in reality every user of the heat pump will have to pay for that. Before regulation, nominal water flow of the water pump equalled 30 $\frac{l}{\min}$. Regulation of the water pump power in optimized heat pump was performed, in accordance with a pattern presented in the table 7.1. All other parameters of the pump were the same before and after optimization.

Table 7.1. Changes of the water pump nominal flow dependent on heating power of the heat pump

Total heating power of the heat pump [kW]	Nominal flow rate of the water pump $[\frac{l}{\min}]$	Electric power supplied to pump [W]
> 11	30	52,0
8 ÷ 11	25	43,3
6 ÷ 8	20	34,6
3 ÷ 6	15	26,0
< 3	10	17,3

As it was mentioned previously, before regulation water pump consumed 456 kWh of energy annually. After regulation, its energy usage dropped to 171 kWh (in both cases, i. e. either when SLHX was and was not installed). In case of SPF, water pump optimization resulted in increase SPF from 2,4 up to 2,49 (heat pump without SLHX) and from 3,12 up to 3,27 (heat pump with SLHX). It is therefore clearly seen that this optimization was worth to be conducted.

7.2.1.3. Results of the heat pump optimization

In this section a summary of optimization results is presented in the table 7.2, comparing the heat pump state before execution of any optimization with its state with SLHX only, water pump regulation only and with both optimization executed.

Table 7.2. Results of the heat pump (HP) optimization designed for the passive house

State of the heat pump	SPF [-]	Compressor [kWh]	Evaporator [kWh]	Gas cooler [kWh]	Heating demand of the building [kWh]	SLHX [kWh]	Water pump [kWh]
Basic HP	2,40	7690	11888	19578	19368	0	456
HP + SLHX	3,12	5817	13761	19578	19368	1873	456
HP + WP	2,49	7690	11888	19578	19368	0	171
HP + SLHX + WP	3,27	5817	13761	19578	19368	1873	171

7.2.2. Optimization of the water storage tank

Water storage tank utilized in considered passive house constitutes an integrated part of the whole DHW system and was wider described in the chapter 6. Its optimization can be done by:

- modification of its insulation coat. Due to the fact that analyzed tank is stratified, water is colder in its lower part and warmer its upper part. Insulation layer could be therefore increased in its upper part, to decrease heat losses from the tank (main part of heat losses is generated above the water layer with the same temperature as the air surrounding the tank placed in the house). In turn, below certain height of the tank, insulation thickness could be decreased, to allow for preheating of the cold water by surrounding air (with higher temperature than water in the lower part of the tank). Overall heat losses could be therefore minimized and energy required for DHW preparation would be lower. Furthermore, if water in the bottom part of the tank would be slightly preheated by indoor air of the house, temperature of water flowing into the heat pump gas cooler would be higher (e. g. 10°C or 11°C instead of 9°C) what would lead to further drop of required heating energy;
- change of the heating strategy. Water in the storage tank is currently heated up to 70°C, in order to keep water temperature in the tank always above 65°C (even during periods of the water consumption), to <u>definitely</u> avoid legionella (as it was mentioned previously, at 66°C they die within 2 minutes). However, maintaining of water temperature in upper part of the storage tank above the level of 60°C is usually enough - at 60°C legionella die within 32 minutes [51]. To keep the water temperature above such value all the time, it seems that heating the water up to 65°C instead of 70°C would be enough, thereby allowing for energy savings. However, keeping the temperature above 60°C in the tank all the time would enforce lower temperature of the water in faucets, probably with a value around 55°C. Continuing, if

to assume that occupants use the water with temperature varying mostly between 30ᵒC and 45ᵒC (reached by mixing tank water with cold water from the mains in taps or showers) that would result in a decrease of daily water volume allowed for usage per each occupant;

- improvement of thermal stratification inside the tank, what can result in further decrease of the energy required for DHW preparation. Thermal stratification can be partially corrected by upgrade of the insulation coat but also by elimination of thermal bridges in the top part of the tank, decrease of the mixing level inside the tank by adding modern diffusors with plate partitions instead of normal inlets and outlets and optimization of its height to diameter ($\frac{\mathrm{H}}{\mathrm{D}}$) ratio. As it was mentioned in the previous chapter, a relatively high value of this ratio was assumed (i. e. 3,57), however in opinion of the author of this thesis there surely exists more suitable ratio which can result in lower DHW heating demand. Most of researchers agree that optimal $\frac{\mathrm{H}}{\mathrm{D}}$ value is in the range of 3 ÷ 4 [74], although it is worth to remember that together with increase of $\frac{\mathrm{H}}{\mathrm{D}}$ total area of the tank is also getting higher. That may result in increase of the heat losses to the ambient [71, 74], what is unwanted;
- optimization of the tank volume - volume of currently designed thermally stratified tank was assumed in the same way as for standard, non - stratified tanks. However, when the tank is stratified, a smaller tank can be usually applied, offering the same comfort (what was wider described in chapter 3). Optimization (decrease) of the tank volume would result in its lower investment cost (smaller tank costs less).

After review of possible optimization criteria, due to limited time and limited possibilities of the program (IDA ICE) the author finally decided to only upgrade insulation coat of the tank. In IDA ICE U-value of the insulation coat can be defined separately for lateral surface of the tank, its top and its bottom, however not separately for its higher and lower part. Previously assumed value (i. e. 0,9 $\frac{\mathrm{W}}{\mathrm{m^2K}}$ for all tank parts) was thus replaced with 0,7 $\frac{\mathrm{W}}{\mathrm{m^2K}}$ for the tank side surface, 0,6 $\frac{\mathrm{W}}{\mathrm{m^2K}}$ for its top and 0,9 $\frac{\mathrm{W}}{\mathrm{m^2K}}$ for its bottom area.

Annual heating demand of the passive house for DHW preparation before modification of the tank insulation coat equalled 11438 kWh. After performed optimization, it dropped to 11355 kWh, what means a decrease of 83 kWh (0,73%). In turn, maximum heating power required from viewpoint of DHW needs was almost the same before and after optimization - 3,83 kW. One can thus conclude that in considered case improvement of the tank insulation coat is only slightly profitable.

7.3. Optimization of the cooling system

Cooling system initially designed for the passive house by the author of this thesis consists of the following main parts:

- electric chiller (with overall efficiency assumed as 100%), cooling the water which is utilized as the coolant;
- space cooling distribution system, containing cooling beams placed in rooms, filled with the cold water.

There are many ways for improvement of the cooling system performance (similarily as for the heating system), mainly divided into direct methods (more accurate steering of the system, adoption of modern PID regulators) and indirect solutions (allowing for change of the building cooling demand), applying without modification of integer parts of the system.

Cooling system currently assumed in the building is relatively inefficient, because it needs to provide the same amount of electricity as the cooling demand is. To improve its efficiency without changing of the cooling device (electric chiller) cooling demand should be either covered by providing less amount of electricity or should be decreased. Thus the author of this master thesis finally decided to execute two modifications of the cooling supply system:

- replacement of the actual space cooling distribution system (cooling beams) with the simple cooling coil placed in the ventilation duct. This alteration is implemented together with a change of the ventilation system strategy and desired supply air temperature setpoint during summer months of the year. After modification, during winter part of the year (i. e. in periods of 01.01.2016[100] ÷ 05.04.2016 and 18.10.2016 ÷ 31.12.2016) a balanced CAV system was assumed to work in the passive house (with supply air temperature of 21°C and air flow rate of 1,2 $\frac{\mathrm{m^3}}{\mathrm{m^2h}}$, i. e. similarily like before optimization), however during summer part of the year (05.04.2016 ÷ 18.10.2016) a VAV system was adopted. Desired air flow rate was set vary in the range of 1,2 $\frac{\mathrm{m^3}}{\mathrm{m^2h}}$ ÷ 3,6 $\frac{\mathrm{m^3}}{\mathrm{m^2h}}$ (or 0,3333 ÷ 0,9999 $\frac{l}{\mathrm{m^2s}}$ respectively), same as the supply air temperature setpoint, whose changes are depicted on the figure 7.4. The whole modification will decrease an investment cost of the cooling system and allow for covering of the cooling demand with lower amount of delivered electricity, due to more intensive utilization of the ambient air, which often has lower temperature than 23°C[101] during period when cooling is required - passive house with

[100] As the time period for all simulations performed in this master thesis a year of 2016 was assumed.
[101] Maximum temperature desired in the passive house.

CAV ventilation (during all the year) and without cooling system is getting overheated even when the ambient air has a temperature of ~15^{o}C;

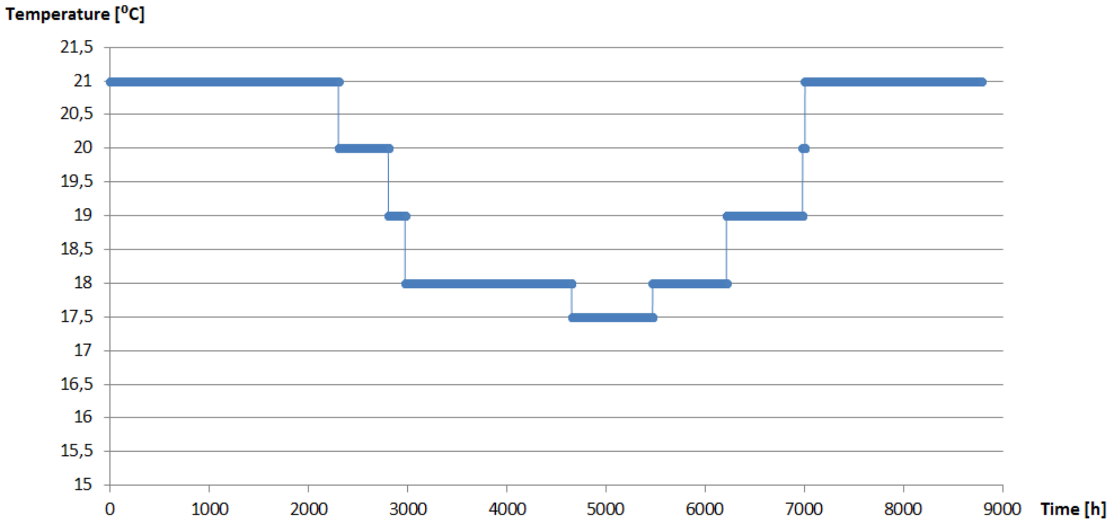
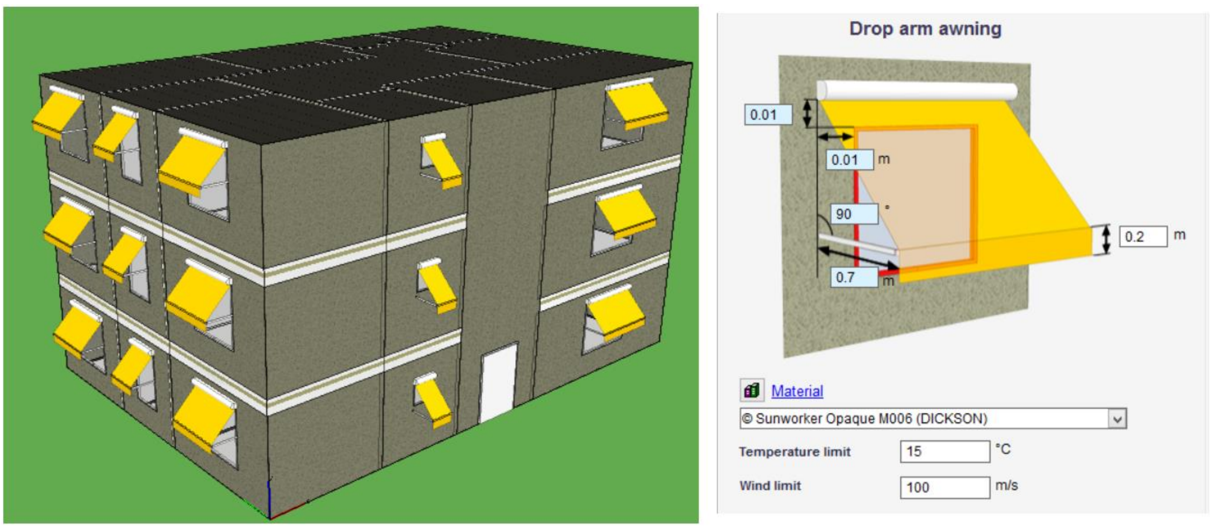


Figure 7.4. Changes of the ventilation supply air temperature setpoint during the year; where temperature is set on 21^{o}C, ventilation of the passive house operates in CAV mode

- addition of drop arm awnings, with geometry depicted on the figure 7.5. They were adopted to all windows of the building and set to be controlled by both time schedule (awnings were set to operate in the period between 15 april and 15 september), intensity of solar radiation (when solar radiation intensity is below 100 $\frac{W}{m^2}$, awnings are not working), desired indoor air temperature (when the outdoor temperature is below 15^{o}C, awnings are turned off) and even wind speed - they are closed after the wind speed reaches 100 $\frac{m}{s}$ (such high wind speed limit was set intentionally, to let the awnings working independently of the wind speed). Utility of awnings will result in decrease of the annual cooling demand of the passive huse.



a **b**

Figure 7.5. Presentation of drop arm awnings; a - outlook of the awnings while mounted above the windows; b - parameters of the awning (not all)

After implementation of two modifications influencing performance of the cooling system into IDA ICA, comparison of the results is depicted in the table 7.3. In overall, four states of the cooling system were compared (basic cooling system before optimization - state 1, cooling system with cooling beams replaced by cooling coil and with changed ventilation strategy - state 2, basic cooling system with awnings - state 3, and cooling system after execution of both modifications - state 4.

Table 7.3. Results of the cooling system optimization in the passive house

State of the cooling system	Maximum cooling power [kW]	Annual cooling energy demand [kWh]
State 1	4,41	3668
State 2	3,96	820 + 367[102] = 1187
State 3	3,01	2511
State 4	3,25	680 + 122 = 802

Based on performed optimization, one can notice that installation of drop arm awnings was not a bad choice. In case of basic cooling system, thanks to their utility cooling demand of the passive house decreased for about 31,5%. In case of the new cooling system (i. e. cooling coil and variable ventilation strategy) percentage benefit from the awnings usage was even higher - 32,4%, although the difference in kWh was smaller than for the basic system. In case of the systems in state 2 and 4 one can notice that low amount of energy was consumed by electric chiller but some extra energy was consumed by fans. It is caused by increased air flows in ventilation ducts, which cool the indoor space of the house (particularly in case of the supply airflow) - comparison between ventilation supply and return flow rates for the cooling system in state 1, 2 and 4 can be found in the appendix 3. Besides, optimization of the cooling system by means of cooling coil and VAV is even more profitable than appliance of the awnings - a decrease of 67,7% in energy consumption between state 1 and 2 and drop of 78,1%, comparing state 1 and 4. Although the system efficiency is still 100%, less amount of electricity have to be supplied in order to reach similar indoor climate inside the house, due to engaging of the ventilation system.

[102] Extra electricity consumption of fans.

8. Conclusions, recommendations and suggestions for further work

Optimization of energy supply system in considered passive house resulted in decrease of total energy (electricity) which have to be supplied to the building in order to fulfill its heating and cooling needs. Comparison of energy amounts supplied to the building before and after performed optimization is presented in the table 8.1. In addition, passive house in the reference state[103] and low energy state of the building designed by Thoreby in his master thesis [9] were collated.

Table 8.1. Overall results of conducted optimization of the energy supply system designed for passive house, collated with previous (low energy) state of analyzed building. All numbers in the table have a unit of kWh.

Name of compared parameter	Passive building in the reference state	Passive building before optimization	Passive building after optimization[104]	Low energy building developed by Thoreby
Total heating demand	19368	19368	19285	31031
Total cooling demand	3668	3668	802	0[105]
Overall heating and cooling demand	23036	23036	20087	31031
Electricity supplied to cover the heating demand	19368	8146	5988	12363
Electricity supplied to cover both heating and cooling demand	23036	11814	6790	12363
Total electricity supplied to the building (incl. auxiliary equipment[106])	28177 (23036 + 5141)	16955 (11814 + 5141)	11931 (6790 + 5141)	22311 (12363 + 9948)
Total energy demand	28177	28177	25228	40979

Based on analysis done in this master thesis the following **main conclusions** can be formed:

- retrofit of desired building from its low energy state considered by Thoreby [9] to the passive standard resulted in 31% drop of its total energy demand (from 40979 kWh down to 28177 kWh);
- heat pump developed in Dymola achieved SPF of 2,40 (without optimization, including water energy consumption), what resulted in 40% decrease of total electricity supplied to the building (from 28177 kWh to 16955 kWh);
- thanks to performed optimization of the energy supply system (transcritical heat pump, DHW thermally - stratified storage tank and cooling system), further drop of total electricity supplied to the building was possible. A drop of 30% (from 16955 kWh to 11931 kWh) was achieved;

[103] All heating and cooling needs of the building were covered by the electric energy supply system with efficiency of 100%.
[104] In terms of all performed modifications, i. e. two for heat pump, one for storage tank and two for cooling system.
[105] There was no cooling system in the building developed by Thoreby and hence there is a lack of data about its cooling demand.
[106] In terms of lighting, fans and pumps.

- optimization of transcritical heat pump utilizing Dymola software with TIL library and DaVE support resulted in increase of its SPF. Thus this factor rised from 2,40 up to 3,27. For comparison, overall SPF achieved by heating system developed by Thoreby equalled 2,51, therefore slightly higher than SPF value reached in this thesis before optimization but significantly lower than after its conductance. However, Thoreby did not include power of fans supplying the airflow into air-based evaporator of the heat pump which he designed.

Summing up the whole analysis, it is always worth to remember that not only a standard in which the residential building is built (e. g. low energy, passive, zero energy) determines its operation costs. Of course, it may decrease energy consumption of the building, however it will never improve the ratio between its energy demand and energy which have to be supplied to the building to cover that demand. To achieve this (what is maybe even more important) properly designed and energy efficient energy supply system have to be implemented. Moreover, one should know that in practice even the best energy supply system can always be optimized, what may result in further drop of operation costs of the system (and hereby the building).

Based on comprehensive work done in this master thesis, the following **broad recommendations** for design of energy supply systems equipped with heat pump and DHW storage tank (desired to be utilized in residential buildings) are listed below:

- dependent on energy standard in which the building is built (and especially type of the heat distribution system), another refrigerant is recommended to be used in the heat pump. In case of low energy and passive houses where waterborne underfloor heating is installed an utility of transcritical heat pumps working in combined mode (i. e. simultaneous SH and DHW heating) with CO_2 as refrigerant is definitely recommended, due to their higher efficiency in terms of SPF (and COP). However, in case of less - insulated buildings or buildings equipped with heat distribution system which is based on convectors or radiators then convenctional (subcritical) heat pump will be a better choice, with utility of one of popular refrigerants (e. g. R-134a, R-410a, R-600, R-407c). Furthermore, independent of the type of used heat pump it is always worth to equip the heat pump in three important components: internal suction line heat exchanger, water pump (or fan) operating in the heat source loop with variable speed and separator, placed between compressor and evaporator. These elements will always influence the heat pump work and its operation costs in positive manner;
- irrespective of the type of utilized heat pump, application of variable - speed scroll compressor[107] and modern (thermostatic) expansion valve with regulation of effective flow area available in

[107] Scroll compressors have higher volumetric efficiency than other types.

wide range is recommended, to achieve smoother operation of the heat pump and hereby minimize its operating costs;

- utility of DHW tank, especially these which are thermally - stratified is highly recommended. It is always profitable to use domestic hot water storage tank in heating system of the building. Although the operation costs of the building will be higher, the difference can be minimized via proper design of the tank and maintenance of its thermal stratification. If needed, the user can also think about utility of second tank for space heating loop (buffer) but not stratified (utilization of thermally - stratified tanks in space heating loop does not make sense);
- usage of PCM - based products is desired. Their utilization always positively influence a sense of thermal comfort inside the building, allowing for decrease of operation costs related with its heating and cooling system. The only problem may be their relatively high investment cost;
- ventilation system should be designed and adjusted with particular care. For residential buildings a VAV system is recommended. Its proper operation allow for both significant decrease of the building cooling demand during summer and minimization of its heating demand during winter;
- cooling distribution system carried out by waterborne cooling coil placed in the ventilation duct is recommended. In climate which occurs in central and northern Europe (e. g. in Norway, Poland and Germany) usage of separated cooling systems (e. g. in form of cooling beams), allowing for coverage of relatively high cooling demands of the buildings is not profitable. When installed ventilation system is VAV, it is normally enough to use cooling coil and cool the building inner space by chilled air. Such solution is clearly cheaper in case of both the investment cost and operation costs. In overall, it is also possible to adopt more energy efficient solutions, e. g. heat pump working in cooling mode, located and operated in the ventilation system (between supply and return ventilation ducts). However, such system would be also more expensive, therefore it should be first considered if it is really profitable.

In opinion of the author of this master thesis energy supply system which he developed for the passive building can be characterized by relatively good performance. Optimization of this system conducted afterwards also resulted in significant improvement of the heating and cooling system operation. However, many issues and cases which were desired to be considered as integral parts of this master thesis had to be neglected, mainly due to time restrictions. As the passive house achievements are very good and the building was re - designed to be more realistic, these issues were mostly related with either more accurate and realistic design of the heat pump, implementation and adjustment of other thermal energy storage methods utilized in residential buildings or with further optimization of considered energy supply system, from viewpoint of other criteria depicted in the chapter 7. Moreover, some other technical solutions which could support the energy supply system in further decrease of annual net energy supplied to the building were thought.

Among all ideas which were thought and analyzed by the author of this thesis the most interesting and noteworthy **concepts** are proposed to be considered in **further work** with this topic:

- further optimization of the energy supply system:
  - ground - source transcritical heat pump - especially from viewpoint of finding the optimal pressure which maintenance in gas coolers would result in better performance of the heat pump (increase of SPF, COP). It is highly probable that to achieve the best performance this pressure should vary during the year, dependent on current heating demand of the building [8];
  - hot water storage tank - particularly by investigation of its optimal heating strategy and optimal volume, which could be surely lower than currently existing (500 dm^3);
- implementation and optimization of PCM - based products into analyzed passive house - in order to do that, it is necessary to find and select the appropriate program allowing for PCM modeling in modern buildings. Afterwards (dependent on capabilities of the program) phase change materials can be adopted to the passive house indoor environment which is currently defined (data from IDA ICE should be sufficient);
- combination of solar collectors with hot water storage tank - their usage can be very profitable if properly maintained, especially while combining solar panels with thermally - stratified tank (as this one existing in analyzed energy supply system);
- upgrade of the heat pump by adding photovoltaic cells transferring gained electricity to the compressor of the heat pump, hereby decreasing electricity amount which have to be supplied from the grid and for which users of the building have to pay;
- modification and optimization of the connection between cooling coil and electric chiller, by adding new by - pass channel allowing the water absorbing the heat in ventilation duct to release this heat in the bottom part of hot water storage tank via additional internal copper coil placed inside the tank, where (due to its thermal stratification) temperature is often relatively low (around $9 \div 10^{o}C$). Naturally, additional sensors and automatics should be applied to avoid heat exchange in periods when temperature of water in the bottom part of tank would be higher than temperature of the water inflowing from the cooling coil. Such solution would surely support the electric chiller and would help in decrease of total amount of electricity supplied from the grid to cover the building cooling demand.

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10. List of figures

11. List of tables

12. List of digital attachments

The following digital appendices are attached to this master thesis:

1. IDA ICE file, containing low energy building without cooling system
2. IDA ICE file, containing passive house with CAV ventilation and awnings
3. IDA ICE files, containing passive house with VAV ventilation and awnings
4. IDA ICE file, containing water storage tank
5. Dymola file, containing heat pump without SLHX
6. Dymola file, containing heat pump with SLHX

Appendix 1. Draft proposal for a scientific paper

Optimization of the heating and cooling system for a residential passive house equipped with heat pump and energy storage

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Norwegian University of Science and Technology (NTNU), june 2016

Abstract

Modern heating system consisting of air - based transcritical heat pump and domestic hot water storage tank is considered in the low energy building designed to be located in Oslo. However, the system optimization has to be performed for the passive house, thus the house has to be retrofitted. Furthermore, due to the lack of files containing the building project and analyzed heating system designed previously in EnergyPlus, they have to be first designed again and only afterwards heating and cooling system (created separately from the beginning, because there was no cooling system designed for low energy building) can be optimized. After evaluation of various simulation tools, IDA ICE is assumed as the program for creation of desired passive building and hot storage tank[108], while Dymola with TIL library package and DaVE extension is utilized for the heat pump design. Heat source of the heat pump (working with CO_2 as refrigerant) is finally modified from air into ground and optimization of the heat pump is performed using Dymola, while the cooling system and DHW[109] storage tank are optimized in IDA ICE.

Keywords: passive house, low energy building, energy supply system, transcritical heat pump, carbon dioxide, heat storage, domestic hot water tank, suction line heat exchanger, water pump, ground source

1. Introduction - theoretical background

From above the last two decades a design of residential buildings in Europe (especially in well - developed countries) is getting more and more ecological meaning. That resulted in the necessity of extreme reduction in overall energy consumption of the buildings, especially from viewpoint of the building heating needs[110], in order to limit emission of CO_2 and other green house gases to the atmosphere. Hence, highly insulated buildings - low energy, passive and even zero energy buildings[111] are becoming more and more popular, not only in commercial appliances (as it was at the beginning of their development) but also in residential sector. Today many building companies offer their customers complete projects of residential buildings which are well - insulated and on the market exist many technical solutions describing how to retrofit the older buildings to current requirements. One could therefore say, that potential of further reduction related with the building energy consumption is already significantly lower. It is true - at least from viewpoint of the building construction. Now, when modern residential buildings are characterized by clearly lower net energy consumption, people are trying to develop new ideas and solutions increasing energy efficiency of the buildings energy supply systems or comfort of their occupants. Examples of such modern technologies are transcritical heat pumps, DHW thermally - stratified tanks or PCMs (phase change materials).

2. Utilized simulation software

In aim to select the most appropriate software, evaluation of different simulation tools[112] was performed, based on the following criteria:

- complexity - the variety of options which a program offers, related with the passive house design,

[108] At the beginning it was desired to model also PCM (phase change materials) in analyzed building, however due to lack of programs allowing for relatively precise PCM modeling the energy storage system was finally limited to DHW storage tank.
[109] Domestic hot water.
[110] Heating needs of the buildings built in accordance with so-called old building requirements (in Norway old building code concern all building law regulations released before TEK10) constituted the main part of their overall energy consumption, therefore there was the highest potential in reduction of energy consumed in the building sector.
[111] In terms of annual net energy consumption of the building covered by external energy sources, e. g. electricity from the grid.
[112] Programs which had been taken into account are: CASAnova, SIMIEN, EnergyPlus, OpenStudio, Beopt, IDA ICE, TRNSYS, Dymola and Audytor OZC.

- overall precision - the accuracy level of calculations performed by the software,
- heat pump model accuracy - the way of the heat pump system design in the program. The accuracy is high when a flexible, random construction of the heat pump coupled with water storage tank on a component level is available. Possibility of regulation of the heat pump components work also increases a precision of the model,
- optimization options - the variety of possible ways for energy optimization of designed heat pump and water storage tank,
- independency - if the program can reach its desired goals (conduct the whole building and energy supply system design process, simulation and optimization) alone, without installing other familiar programs (or subprograms),
- interface and 3D options - the intuity level and simplicity of the program interface. Visualization options of the program related with the building design and ability of performing various animations, either of the building and its energy supply system work.
- accessibility - if the program is licence - free or the user has to pay for its utility. If the software requires licence, the price is also taken into account (it is better if the price is low). Moreover, available language of the menu is considered (multi - language menu / English menu / menu in other language),
- reliability - if the program works fluently or it crashes, contains bugs, reports errors, etc.

Based on evaluation performed in the master thesis [2] **IDA ICE** seems to be the best choice among the compared programs and therefore this program was selected for further analysis work. However, the author felt too small possibilities of IDA ICE in case of desired heat pump model (with tri-partite gas cooler). Furthermore, IDA has not great possibilities from viewpoint of the heat pump work regulation and steering. Hence (although the stratified hot water storage tank can be modelled in IDA with relatively good accuracy) the author decided to use also second program **- Dymola**, for more accurate development of the heat pump system.

3. Energy supply system in previous and current state

The energy supply system for low energy state of considered building was designed by A. O. Thoreby in his master thesis [3]. In case of the heating system, air to water heat pump was applied for both domestic hot water and space heating. The topic of Thoreby's master thesis was related with analysis of CO_2 heat pump in low energy residential building, thus the heat pump which he designed works in transcritical conditions, utilizing R744 as refrigerant. In order to determine input parameters of the heat pump, an assessment of the peak heating load of the building had to be made. A simulation of the reference model of considered building (equipped with direct electric heating) was therefore performed in EnergyPlus[113]. Based on its results, Thoreby decided to implement heat pump with a heating capacity of 9 kW. As heat pump capacity drops with decreasing outdoor temperature, it was decided that full capacity of 9 kW should be able to be delivered at outdoor temperature of -7°C (based on technical report of J. Stene and T. Skiple [6]). After evaluation of different gas cooler configurations conducted by Thoreby in his master thesis, it was finally desired to model a system consisting of tripartite gas cooler, presented on the figure 1.a. Such solution is very profitable in co-operation with low - temperature underfloor water heating distribution system. First, superheated CO_2 releases heat in the reheat DHW gas cooler to domestic hot water, which has higher temperature than in case of other compared systems (various single and bipartite gas cooler configurations) [2], because it was earlier preheated in the preheat DHW gas cooler. Tripartite system is thus taking full advantage of the R744 temperature glide when cooling the CO_2.

The original EnergyPlus module for a conventional heat pump model existing in the program was made by Murugappan in 2002 [5] and it provided basics for the transcritical heat pump designed by Thoreby. Unfortunately, in EnergyPlus the heat pump could only be connected to one single loop on the gas cooler side. The system was therefore modeled with an intermediate water loop containing three ideal heat exchangers, what can be seen on the figure 1.b.

[113] Thoreby used EnergyPlus software for all needs of his master thesis.

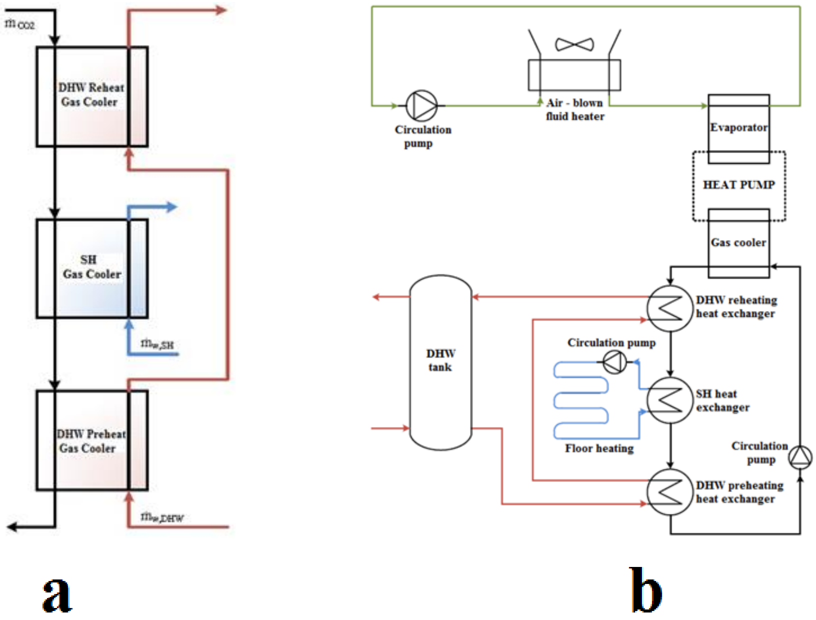

Figure 1. Heat pump model eveloped by Thoreby in Energy Plus [3]; a - desired gas cooler configuration; b - final outlook of the heat pump model

Due to the fact that desired building location is Oslo, where climate is rather cold (based on investigation performed by Thoreby, only 10 ÷ 20 days per year would require cooling) Thoreby assumed no cooling system [3].

In case of heat storage, cylindrical DHW storage tank was modeled, with height of 2 m and volume of 500 dm^3, what resulted in relatively small ratio between its diameter and height (~ 0,28). For the thermal water stratification modeling, the tank was divided into ten parts (nodes) of equal height. In addition, 7 kW electric heater was assumed for supplementary DHW heating, in order to support the heat pump in peak heating demand periods. More details about energy supply system developed by Thoreby can be found in his master thesis [3].

The newest data library for Dymola - TIL[114] - allows for more efficient design of the heat pump, assuming similar accuracy of results. Based on this library, author of this article created desired transcritical heat pump model with tripartite gas cooler, instead of intermediate water loop connecting gas cooler and ideal heat exchangers as Thoreby did in his work (fig. 1.b). Beside of this issue, there are some another relevant features which were changed comparing with the heat pump designed by Thoreby:

- an ambient air - heat source supplying the heat pump - was replaced with the ground;
- temperatures of technical water from underfloor heating circuit - Thoreby assumed constant supply and return temperatures of the water circulating in space heating distribution system (35°C and 30°C). The author of this master thesis assumed 34°C as desired supply temperature (lower temperature for better thermal comfort) and variable return temperature, varying in the range of ~22 ÷ ~29°C, dependent on the actual space heating demand;
- temperatures of domestic hot water - Thoreby assumed 10°C as inlet temperature to the first heat exchanger (preheater) and 65°C as outlet temperature from second heat exchanger (reheater). The author of the thesis (due to willing of keeping DHW temperature in the upper part of storage tank always[115] above 65°C, in order to avoid Legionnaires disease - Legionella bacteries die within two minutes at 66°C [8]) decided to warm DHW up to 70°C after reheater, and assumed its constant inlet temperature of 9°C before preheater;
- the separator was applied after evaporator in order to separate gas and liquid phase of the refrigerant. Its usage protects the compressor against compression of the liquid, what may have destructive influence on its lifetime and proper work;

[114] Developed by TLK-Thermo GmbH in Germany [9].
[115] Even in periods of water consumption.

- electric top up heaters presence for both space heating and domestic water heating was rejected.

Finally developed heat pump model in Dymola is shown on the figure 2. More details about this heat pump can be found in the master thesis [2].

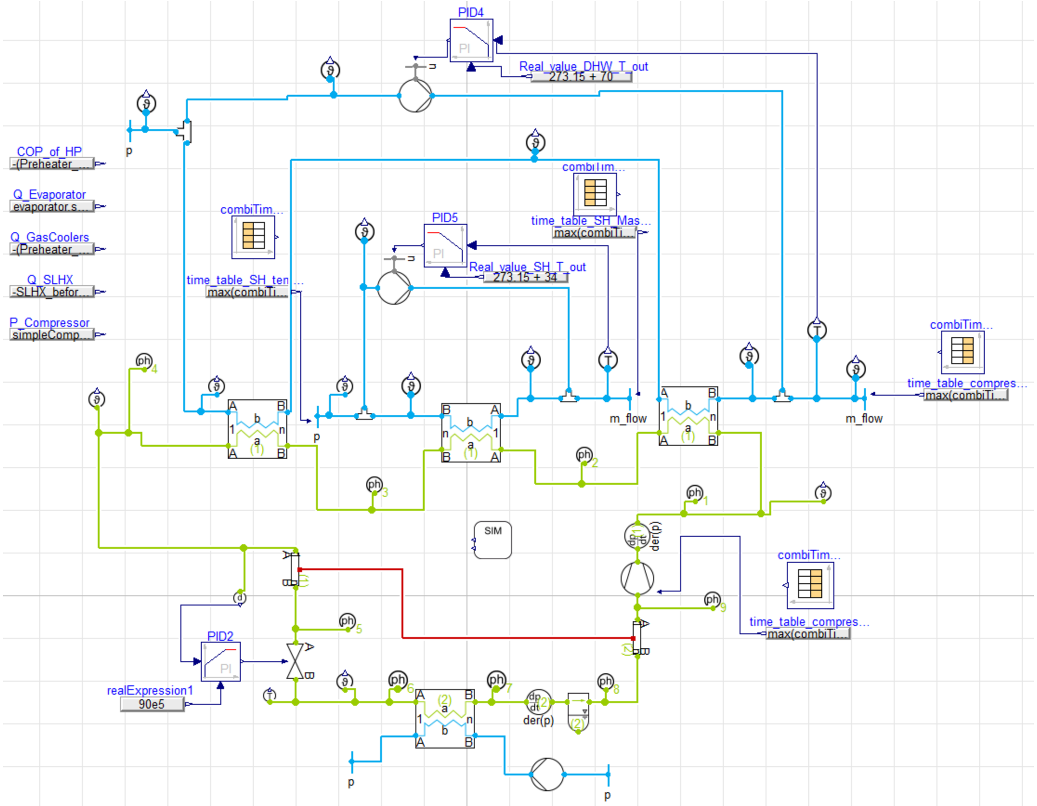

Figure 2. Heat pump developed in Dymola for the passive house by the author of this article [2]. As the red line and two tubes suction line heat exchanger is visible, which was used for optimization of the heat pump.

Heat storage tank was modelled in IDA ICE [2] with most of the parameters similar to these contained in model developed by Thoreby [3]. Factor describing heat exchange between tank layers due to mixing processes (assumed as 50 $\frac{\mathrm{W}}{\mathrm{mK}}$) and factor describing the heat exchange due to stratification phenomenas (set as 800 $\frac{\mathrm{W}}{\mathrm{mK}}$). U-value for the whole tank envelope was set to 0,9 $\frac{\mathrm{W}}{\mathrm{m^2K}}$. DHW consumption was set as 33 litres of 60°C hot water per one occupant and day. Cold water from the mains supplying the tank in its lower part was assumed to have a constant yearly temperature of 7°C at the inlet to the tank and 9°C while it starts to be heated in the gas cooler. As the temperature of ambient air surrounding the tank (designed to be placed inside the building) 21°C was assumed.

Author of this article assumed the cooling system consisted of water - based cooling beams and electric chiller with efficiency of 100% and maximum power of 5 kW. Although in Thoreby's considerations cooling system was not needed in the low energy building, in the passive house cooling is much more required, what was proved in the master thesis [2].

Comparison of achievements between heat pump designed in EnergyPlus by Thoreby for low energy building and in Dymola by the author of this article for the passive house is included in the table 1.

Table 1. Comparison of two transcritical heat pumps developed for low energy building (HP1) and passive house (HP2) [2, 3]

Compared parameter	HP1	HP2
Total heating demand of the building, covered by the heat pump	31031 kWh	19578 kWh[116]

[116] 7930 kWh for space heating needs and 11438 kWh for DHW preparation.

Table 1. Continuation [2, 3]

Compared parameter	HP1	HP2
Maximum peak heating load of the building	15,7 kW	15,8 kW
Maximum designed heating power of the heat pump	9 kW (25 kW[117])	16 kW
Energy supplied to compressor of the heat pump	11622 kWh (12363 kWh[118])	7690 kWh (8146 kWh[119])
Achieved SPF of the heat pump	2,67 (2,51)	2,55 (2,40)

4. Optimization of the energy supply system for the passive house

In case of transcritical heat pump developed in Dymola, there are many ways to improve its performance in aim to reach the optimization goals listed above, i. a.:

- implementation of the suction line heat exchanger (also called as internal heat exchanger). As the result, COP of the heat pump during the year and its overall SPF value increases;
- change of the orifice expansion valve into thermostatic expansion valve or pulse modulation valve, allowing for more accurate regulation of high pressure;
- find the most optimal pressure, which could be maintained in tri - partite gas cooler, what can be done by determination of correlation allowing for calculation of this pressure, mainly depended of desired secondary fluids inlet and outlet temperatures. Although the gas cooler pressure of 90 bar assumed for the heat pump in Dymola does not give bad results, author of this article suppose that there surely exists more optimal pressure, which can result in higher COP and SPF of the heat pump;
- change of PI regulators (currently co-operating with expansion valve and water pumps on demand side of the heat pump) into PID, what could allow for even more stable steering of respective parameters;
- adjustment of volume flow rate of the water flowing through the evaporator to current heating demand of the passive house. It is highly probable that constant flow of the water ($30\ \frac{\mathrm{l}}{\mathrm{min}}$) is too high during periods when the building heating demand is low. In such periods, it decreases COP of the heat pump significantly. If to partly decrease nominal flow of the water pump during low demand periods, a drop of COP would be obviously smaller;
- increase of the water temperature while it flows into evaporator, rejecting the heat which is obtained from the ground source - by utilization of deeper boreholes. However, this modification should be particularly considered from viewpoint of profitability (increase of the investment cost related with drilling deeper boreholes may be too high compared to savings related with later usage of the compressor; moreover, water pump which makes the water circulating in the ground loop would consume more energy, due to pumping the water from lower height level).

After deep analysis of each optimization criteria author of this article finally decided to consider two of them - internal heat exchanger and adjustment of the water flow in evaporator, via water pump regulation. At the first moment it was desired to consider also other optimization proposals, however due to limited time author had to reject them.

In turn, thermally - stratified DHW storage tank developed in IDA ICE can be optimized in many ways:

- modification of its insulation coat. Due to the fact that analyzed tank is stratified, water is colder in its lower part and warmer its upper part. Insulation layer could be therefore increased in its upper part, to decrease heat losses from the tank (main part of heat losses is generated above the water layer with the same temperature as the air surrounding the tank placed in the house). In turn, below certain height of

[117] In addition, 16 kW was added in form of top - up electric heaters (9 kW for space heating and 7 kW for DHW).
[118] Including electricity supplied to top - up heaters.
[119] With electricity supplied to water pump, operating in the ground source water loop.

the tank, insulation thickness could be decreased, to allow for preheating of the cold water by surrounding air (with higher temperature than water in the lower part of the tank). Overall heat losses could be therefore minimized and energy required for DHW preparation would be lower. Furthermore, if water in the bottom part of the tank would be slightly preheated by indoor air of the house, temperature of water flowing into the heat pump gas cooler would be higher (e. g. 10⁰C or 11⁰C instead of 9⁰C) what would lead to further drop of required heating energy;

- change of the heating strategy. Water in the storage tank is currently heated up to 70⁰C, in order to keep water temperature in the tank always above 65⁰C (even during periods of the water consumption), to <u>definitely</u> avoid legionella (as it was mentioned previously, at 66⁰C they die within 2 minutes). However, maintaining of water temperature in upper part of the storage tank above the level of 60⁰C is usually enough - at 60⁰C legionella die within 32 minutes [8]. To keep the water temperature above such value all the time, it seems that heating the water up to 65⁰C instead of 70⁰C would be enough, thereby allowing for energy savings. However, keeping the temperature above 60⁰C in the tank all the time would enforce lower temperature of the water in faucets, probably with a value around 55⁰C. Continuing, if to assume that occupants use the water with temperature varying mostly between 30⁰C and 45⁰C (reached by mixing tank water with cold water from the mains in taps or showers) that would result in a decrease of daily water volume allowed for usage per each occupant;
- improvement of thermal stratification inside the tank, what can result in further decrease of the energy required for DHW preparation. Thermal stratification can be partially corrected by upgrade of the insulation coat but also by elimination of thermal bridges in the top part of the tank, decrease of the mixing level inside the tank by adding modern diffusors with plate partitions instead of normal inlets and outlets and optimization of its height to diameter ($\frac{\mathrm{H}}{\mathrm{D}}$) ratio. As it was mentioned in the previous chapter, a relatively high value of this ratio was assumed (i. e. 3,57), however in opinion of the author of this thesis there surely exists more suitable ratio which can result in lower DHW heating demand. Most of researchers agree that optimal $\frac{\mathrm{H}}{\mathrm{D}}$ value is in the range of 3 ÷ 4 [7], although it is worth to remember that together with increase of $\frac{\mathrm{H}}{\mathrm{D}}$ total area of the tank is also getting higher. That may result in increase of the heat losses to the ambient [7, 4], what is unwanted;
- optimization of the tank volume - volume of currently designed thermally stratified tank was assumed in the same way as for standard, non - stratified tanks. However, when the tank is stratified, a smaller tank can be usually applied, offering the same comfort (what was wider described in chapter X). Optimization (decrease) of the tank volume would result in its lower investment cost (smaller tank costs less).

After review of possible optimization criteria, due to limited time and limited possibilities of the program (IDA ICE) the author finally decided to only upgrade insulation coat of the tank. In IDA ICE U-value of the insulation coat can be defined separately for lateral surface of the tank, its top and its bottom, however not separately for its higher and lower part. Previously assumed value (i. e. $0{,}9\ \frac{\mathrm{W}}{\mathrm{m^2K}}$ for all tank parts) was thus replaced with $0{,}7\ \frac{\mathrm{W}}{\mathrm{m^2K}}$ for the tank side surface, $0{,}6\ \frac{\mathrm{W}}{\mathrm{m^2K}}$ for its top and $0{,}9\ \frac{\mathrm{W}}{\mathrm{m^2K}}$ for its bottom area.

Cooling system currently assumed in the building is relatively inefficient, because it needs to provide the same amount of electricity as the cooling demand is. To improve its efficiency without changing of the cooling device (electric chiller) cooling demand should be either covered by providing less amount of electricity or should be decreased. Thus the author of this master thesis finally decided to execute two modifications of the cooling supply system:

- replacement of the actual space cooling distribution system (cooling beams) with the simple cooling coil placed in the ventilation duct. This alteration is implemented together with a change of the ventilation system strategy (from CAV into VAV) and desired supply air temperature setpoint during summer months of the year. The whole modification will decrease an investment cost of the cooling system and allow for covering of the cooling demand with lower amount of delivered electricity, due

to more intensive utilization of the ambient air, which often has lower temperature than 23°C[120] during period when cooling is required - passive house with CAV ventilation (during all the year) and without cooling system is getting overheated even when the ambient air has a temperature of ~15°C;
- addition of drop arm awnings, adopted to all windows of the building. Utility of awnings will result in decrease of the annual cooling demand of the passive house.

5. Results and conclusions

After optimization of the heat pump, water storage tank and cooling system from viewpoint of selected criteria depicted in the previous point, the following results were obtained:

- for the heat pump:
  - after introduction of the internal heat exchanger, SPF of the heat pump raised up to 3,37 (excluding water pump (WP) energy consumption in calculations) and 3,12 (including this energy). In turn, its COP varied between 1,21 and 7,69 (excluding WP power) and in the range of 0,02 ÷ 4,75 (including the power). Total energy yearly supplied to the compressor was equal to 5817 kWh (decrease of ~24,5% compared to compressor energy consumption without SLHX, which equalled 7690 kWh), with the same overall heating energy provided by gas cooler as before optimization (i. e. 19578 kWh);
  - in case of the water pump operation optimization, before regulation water pump consumed 456 kWh of energy annually. After regulation, its energy usage dropped to 171 kWh (in both cases, i. e. either when SLHX was and was not installed). In case of SPF, water pump optimization resulted in increase SPF from 2,4 up to 2,49 (heat pump without SLHX) and from 3,12 up to 3,27 (heat pump with SLHX). It is therefore clearly seen that this optimization was worth to be conducted;
  - the summary of optimization results is depicted in the table 2, comparing the heat pump state before execution of any optimization with its state with SLHX only, water pump regulation only and with both optimization executed;

Table 2. Results of the heat pump (HP) optimization designed for the passive house [2]

State of the heat pump	SPF [-]	Compressor [kWh]	Evaporator [kWh]	Gas cooler [kWh]	Heating demand of the building [kWh]	SLHX [kWh]	Water pump [kWh]
Basic HP	2,40	7690	11888	19578	19368	0	456
HP + SLHX	3,12	5817	13761	19578	19368	1873	456
HP + WP	2,49	7690	11888	19578	19368	0	171
HP + SLHX + WP	3,27	5817	13761	19578	19368	1873	171

- for hot water storage tank:
  - annual heating demand of the passive house for DHW preparation before modification of the tank insulation coat equalled 11438 kWh. After performed optimization, it dropped to 11355 kWh, what means a decrease of 83 kWh (0,73%). In turn, maximum heating power required from viewpoint of DHW needs was almost the same before and after optimization - 3,83 kW. One can thus conclude that in considered case improvement of the tank insulation coat is only slightly profitable;
- for cooling system:
  - after implementation of two modifications influencing performance of the cooling system into IDA ICA, comparison of the results is depicted in the table 3. In overall, four states of the cooling system were compared (basic cooling system before optimization - state 1, cooling system with cooling beams replaced by cooling coil and with changed ventilation strategy - state 2, basic cooling system with awnings - state 3, and cooling system after execution of both modifications - state 4;
  - based on performed optimization, one can notice that installation of drop arm awnings was not a bad choice. In case of basic cooling system, thanks to their utility cooling demand of the passive house decreased for about 31,5%. In case of the new cooling system (i. e. cooling coil and variable ventilation strategy) percentage benefit from the awnings usage was even higher - 32,4%, although

[120] Maximum temperature desired in the passive house.

the difference in kWh was smaller than for the basic system. In case of the systems in state 2 and 4 one can notice that low amount of energy was consumed by electric chiller but some extra energy was consumed by fans. Besides, optimization of the cooling system by means of cooling coil and VAV is even more profitable than appliance of the awnings - a decrease of 67,7% in energy consumption between state 1 and 2 and drop of 78,1%, comparing state 1 and 4. Although the system efficiency is still 100%, less amount of electricity have to be supplied to reach similar indoor climate inside the house.

Table 3. Results of the cooling system optimization in the passive house [2]

State of the cooling system	Maximum cooling power [kW]	Annual cooling energy demand [kWh]
State 1	4,41	3668
State 2	3,96	820 + 367[121] = 1187
State 3	3,01	2511
State 4	3,25	680 + 122 = 802

To sum up, optimization of the energy supply system in considered passive house resulted in decrease of total energy (electricity) which have to be supplied to the building in order to fulfill its heating and cooling needs. Comparison of energy amounts supplied to the building before and after performed optimization is presented in the table 4. In addition, passive house in the reference state[122] and low energy state of the building designed by Thoreby in his master thesis [2] were collated.

Table 4. Results of conducted optimization of the energy supply system designed for passive house, collated with previous (low energy) state of analyzed building [2]. All numbers in the table have a unit of kWh.

Name of compared parameter	Passive building in the reference state	Passive building before optimization	Passive building after optimization[123]	Low energy building developed by Thoreby
Total heating demand	19368	19368	19285	31031
Total cooling demand	3668	3668	802	0[124]
Overall heating and cooling demand	23036	23036	20087	31031
Electricity supplied to cover the heating demand	19368	8146	5988	12363
Electricity supplied to cover both heating and cooling demand	23036	11814	6790	12363
Total electricity supplied to the building (incl. auxiliary equipment[125])	28177 (23036 + 5141)	16955 (11814 + 5141)	11931 (6790 + 5141)	22311 (12363 + 9948)
Total energy demand	28177	28177	25228	40979

Based on investigation performed in this master thesis the following **main conclusions** can be formed:

- retrofit of desired building from its low energy state considered by Thoreby [2] to the passive standard resulted in 31% drop of its total energy demand (from 40979 kWh down to 28177 kWh);
- heat pump developed in Dymola achieved SPF of 2,40 (without optimization, including water pump power), what resulted in 40% decrease of total electricity supplied to the building (from 28177 kWh to 16955 kWh);

[121] Extra electricity consumption of fans.
[122] All heating and cooling needs of the building were covered by the electric energy supply system with efficiency of 100%.
[123] In terms of all performed modifications, i. e. two for heat pump, one for storage tank and two for cooling system.
[124] There was no cooling system in the building developed by Thoreby and hence there is a lack of data about its cooling demand.
[125] In terms of lighting, fans and pumps.

- thanks to performed optimization of the energy supply system consisting of transcritical heat pump, DHW thermally - stratified storage tank and cooling system, further decrease of total electricity supplied to the building was possible. A drop of 30% (from 16955 kWh to 11931 kWh) was achieved;
- optimization of transcritical heat pump utilizing Dymola software with TIL library and DaVE support resulted in increase of its SPF. Thus this factor rised from 2,40 up to 3,27. For comparison, overall SPF achieved by heating system developed by Thoreby equalled 2,51, therefore slightly higher than SPF value reached in this thesis before optimization but significantly lower than after its conductance. However, Thoreby did not include power of fans supplying the airflow into air-based evaporator of the heat pump which he designed.

Summing up the whole analysis, it is always worth to remember that not only a standard in which the residential building is built (e. g. low energy, passive, zero energy) determines its operation costs. Of course, it may decrease energy consumption of the building, however it will never improve the ratio between its energy demand and energy which have to be supplied to the building to cover that demand. To achieve this (what is maybe even more important) properly designed and energy efficient energy supply system have to be implemented. Moreover, one should know that in practice even the best energy supply system can always be optimized, what may result in further drop of operation costs of the system (and hereby the building).

6. Acknowledgements

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Appendix 2. NTU and LMTD methods

1. NTU approach - evaporator and gas cooler

For calculations of heat transfer rates in heat exchangers (evaporator and gas cooler) the **Number of Transfer Units** (**NTU**) method [5, 69] was applied, because there are insufficient information to calculate them using LMTD (Logarithmic Mean Temperature Difference) approach. It is due to the fact that to calculate LMTD either inlet and outlet temperature of both fluids exchanging the heat must be specified or determined by simple energy balance. However in the analyzed case only inlet temperatures of CO_2 and H_2O (gas cooler) or air (evaporator) are known. NTU method is especially recommended for the analysis of counter-flow heat exchangers, as these in considered energy supply system.

In overall, the energy effectiveness of a heat exchanger (ε) can be defined as:

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}}, \tag{1}$$

where:

$\dot{Q}$ - actual heat transfer rate in the heat exchanger [W],

$\dot{Q}_{max}$ - maximum possible heat transfer rate in heat exchanger that can be hypothetically achieved [W].

In NTU approach the main goal is to find $\dot{Q}$ of the heat exchanger (and outlet temperatures of both fluids). In order to do that, $\dot{Q}_{max}$ and ε have to be first defined. The method proceeds by calculating **heat capacity rates** of both fluids exchanging the heat in the following way:

$$C = c \cdot \dot{m}, \tag{2}$$

where:

C - heat capacity rate [$\frac{J}{K \cdot s}$],

c - specific heat capacity (specific heat) [$\frac{J}{kg \cdot K}$],

$\dot{m}$ - mass flow rate of considered fluid [$\frac{kg}{s}$].

A heat transfer rate in NTU method is defined as the multiplication of a heat capacity rate and a temperature difference. Fluids which exchange the heat in heat exchanger are normally different and hence they have different specific heats. Maximum possible temperaure difference which a fluid can experience in heat exchanger is equal to:

$$\Delta T_{max} = T_{h,i} - T_{c,i}, \tag{3}$$

where:

$T_{h,i}$ - inlet temperature of the hot stream [°C],

$T_{c,i}$ - inlet temperature of the cold stream [°C].

Big specific heat of a fluid means that a lot of heat needs to be supplied to the fluid to change its temperature per kg of mass. This fluid will therefore experience smaller temperature difference than the fluid with lower specific heat (if their mass flow rates are similar). Thus, $\dot{Q}_{max}$ can be calculated from the following equation:

$$\dot{Q}_{max} = C_{min} \cdot \Delta T_{max} = c_{min} \cdot \dot{m}_x \cdot \left(T_{h,i} - T_{c,i}\right), \qquad (4)$$

where:

C_{min} - minimum heat capacity rate[126] $[\frac{J}{K \cdot s}]$,

$\dot{m}_x$ - mass flow rate of the fluid with smaller heat capacity rate $[\frac{kg}{s}]$.

For any heat exchanger it can be shown that:

$$\varepsilon = f\,(NTU, C_r), \qquad (5)$$

where:

NTU - number of transfer units [-],

C_{ra} - heat capacity ratio [-].

For a given geometry of heat exchanger *NTU* can be defined as:

$$NTU = \frac{U \cdot A}{C_{min}}, \qquad (6)$$

where:

U - overall U-value (heat transfer coefficient) of the heat exchanger $[\frac{W}{m^2 \cdot K}]$,

A - heat transfer area $[m^2]$.

In turn, heat capacity ratio is equal to:

$$C_{ra} = \frac{C_{min}}{C_{max}}, \qquad (7)$$

where C_{max} means higher heat capacity rate among two *C* calculated for fluids exchanging the heat in heat exchanger $[\frac{J}{K \cdot s}]$.

There can be formed many formulas to calculate ε from equation 5 for different heat exchangers, dependent on their geometry (number of passes - in shell and tube exchangers), type of the flow (counter-flow, co-current or cross-flow), the occurence of phase change (if phase change of at least one fluid in heat exchanger occurs) and whether a flow stream is mixed or unmixed. In the case analyzed in this thesis energy effectiveness of gas cooler is calculated as below (for counter-flow type and $C_{ra} < 1$):

[126] After counting of heat capacity rates of both fluids the smaller C can be denoted as C_{min}.

$$\varepsilon_{GC} = \frac{1 - e^{-NTU_{GC}(1-C_{ra,GC})}}{1 - C_{ra,GC} \cdot e^{-NTU_{GC}(1-C_{ra,GC})}} \tag{8}$$

Due to the fact, that in evaporator the phase change of CO_2 occurs, effectiveness must be calculated in another way. In this special case $C_{ra} = 0$ and the heat exchanger behaviour is independent of the flow arrangement. The following formula is therefore applied:

$$\varepsilon_{EV} = 1 - e^{-NTU_{EV}} \tag{9}$$

For the case when both fluids in heat exchanger are identical (i. e. $C_{ra} = 1$) effectiveness can be calculated as follows:

$$\varepsilon = \frac{NTU}{1 + NTU} \tag{10}$$

If $\dot{Q}_{max}$ and ε are defined one can use transformed equation 1 to calculate actual heat transfer rate in heat exchanger:

$$\dot{Q} = \varepsilon \cdot \dot{Q}_{max} \tag{11}$$

It is obvious that the heat transferred from one fluid to another equals to the heat received by this second fluid from the first one (if heat losses are neglected, as in the analyzed case). Hence the actual heat transfer rate in heat exchanger can be also defined as:

$$\dot{Q} = C_h \cdot \left(T_{h,i} - T_{h,o}\right) = C_c \cdot \left(T_{c,o} - T_{c,i}\right), \tag{12}$$

where:

C_h - heat capacity rate of the hot fluid [$\frac{J}{K \cdot s}$],

C_c - heat capacity rate of the cold fluid [$\frac{J}{K \cdot s}$],

$T_{h,o}$ - outlet temperature of the hot stream [^oC],

$T_{c,o}$ - outlet temperature of the cold stream [^oC].

From the equation 12 outlet temperatures of both fluids can be also counted.

To sum up, NTU and actual heat transfer rate in **evaporator** are calculated as below:

$$NTU_{EV} = \frac{U_{EV} \cdot A_{EV}}{C_{sf,EV}}, \tag{13}$$

$$\dot{Q}_{EV} = \varepsilon_{EV} \cdot C_{sf,EV} \cdot \left(T_{r,EV} - T_{sf,EV,i}\right), \tag{14}$$

where:

$C_{sf,EV}$ - heat capacity rate of the secondary fluid (ambient air) in evaporator [$\frac{J}{K \cdot s}$],

$T_{r,EV}$ - constant temperature of the refrigerant (CO_2) in evaporator [^oC],

$T_{sf,EV,i}$ - inlet temperature of the secondary fluid in evaporator [^oC].

In turn, for the **gas cooler** these values can be counted as:

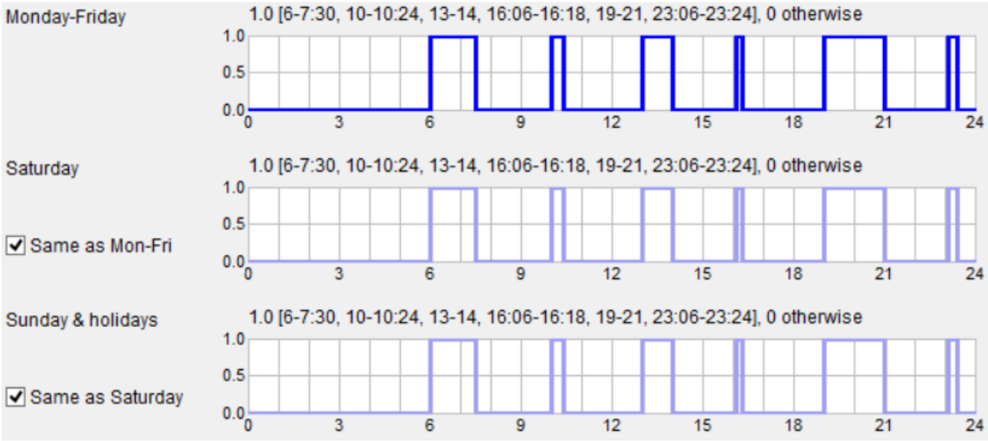
$$NTU_{GC} = \frac{U_{GC} \cdot A_{GC}}{C_{sf,GC}} \tag{15}$$

$$\dot{Q}_{GC} = \varepsilon_{GC} \cdot C_{sf,GC} \cdot \left(T_{r,GC,i} - T_{sf,GC,i}\right) \tag{16}$$

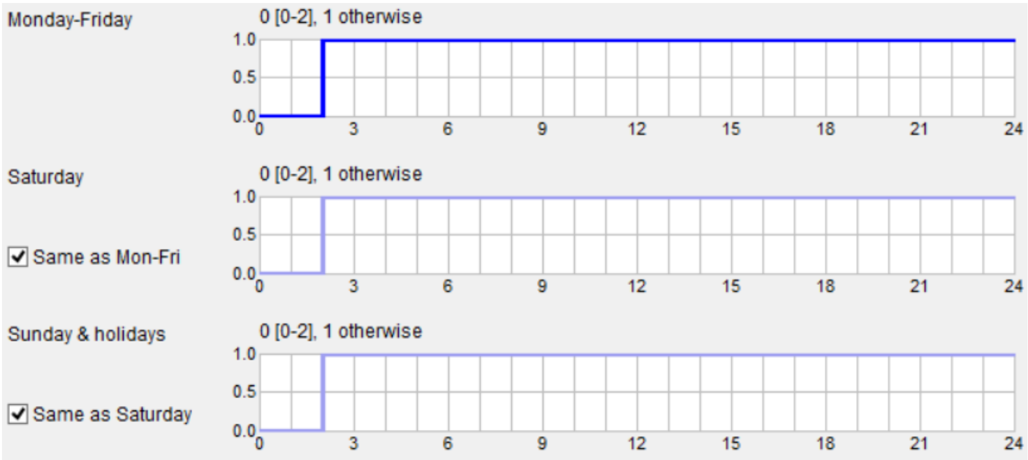
Appendix 3. Other passive building results

1. Schedules

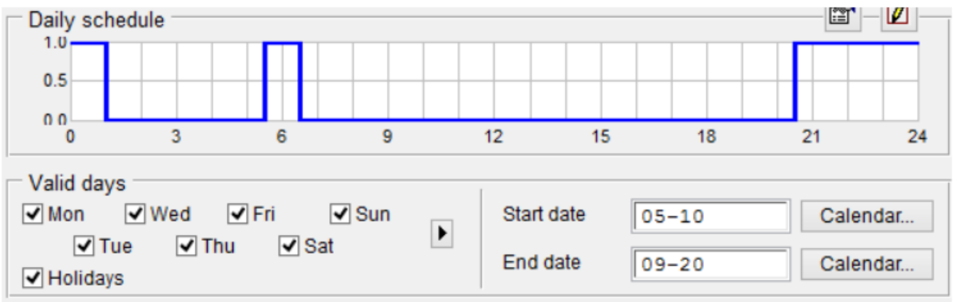
- **DHW storage tank - water consumption inside the building**

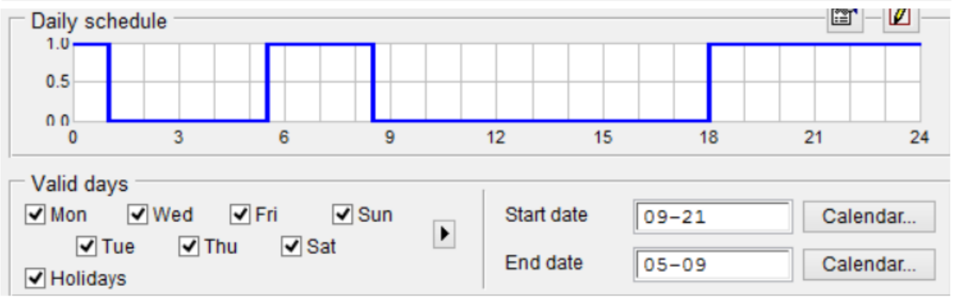


- **DHW storage tank - heat pump operation schedule**



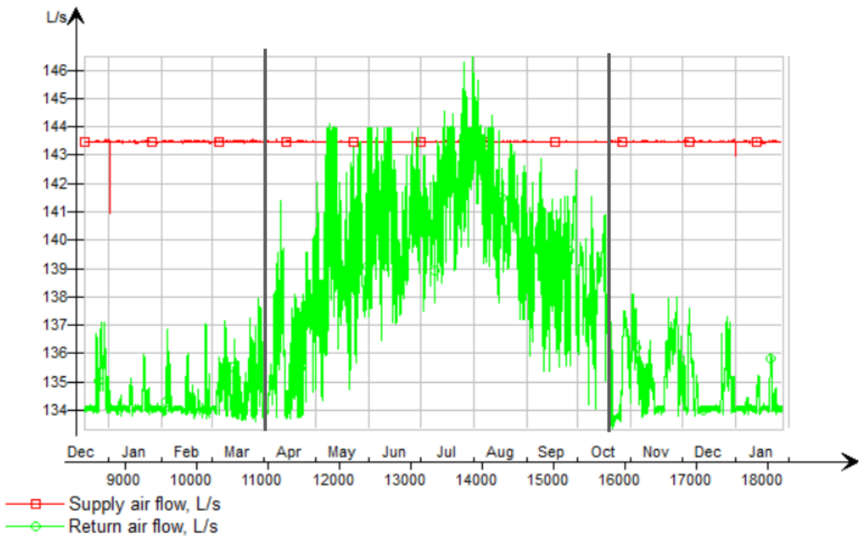
- **Lighting**



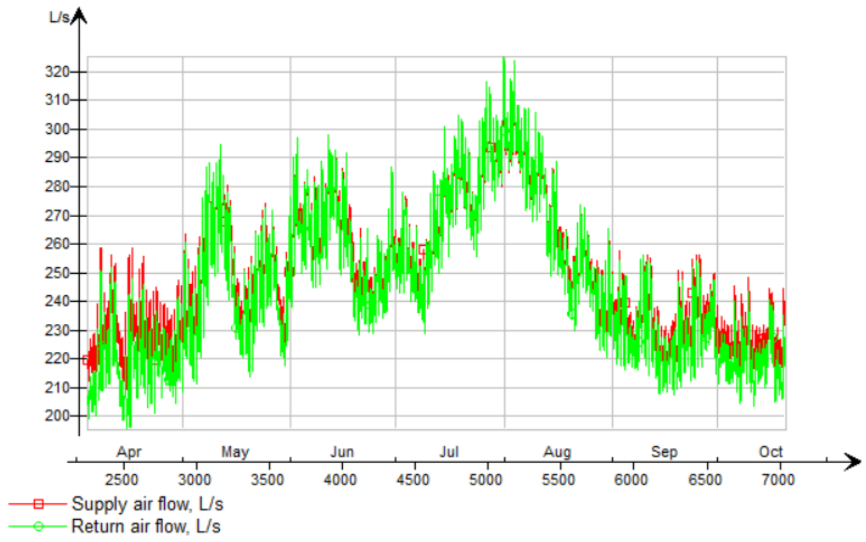


2. Comparison of supply and return ventilation airflows for different cooling system states

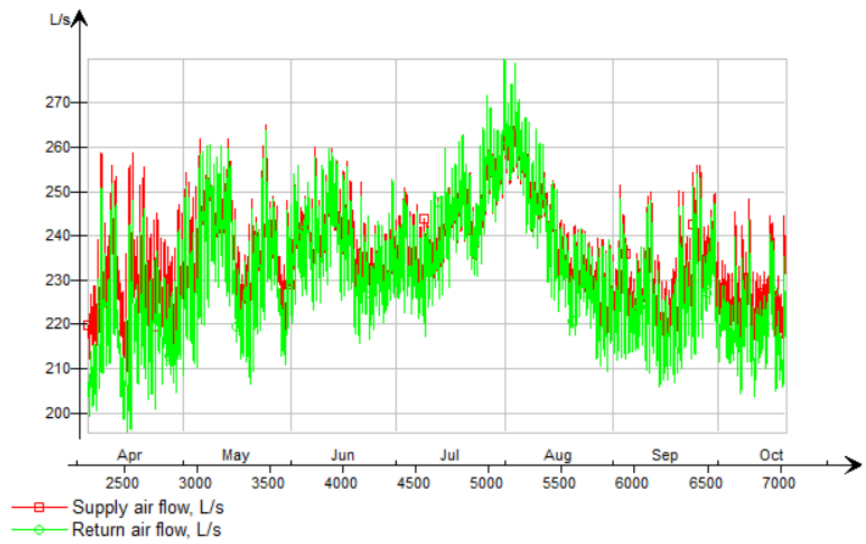
Basic cooling system before optimization (state 1 described in section 7.3)



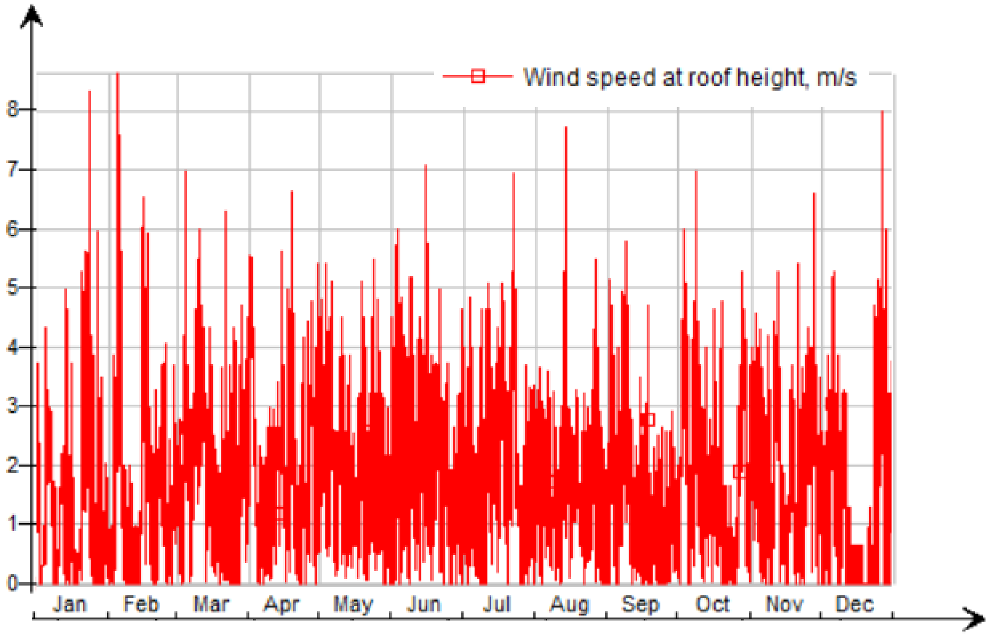
Cooling system after one optimization modification (state 2 described in section 7.3)



Cooling system after two optimization modifications (state 4 described in section 7.3)

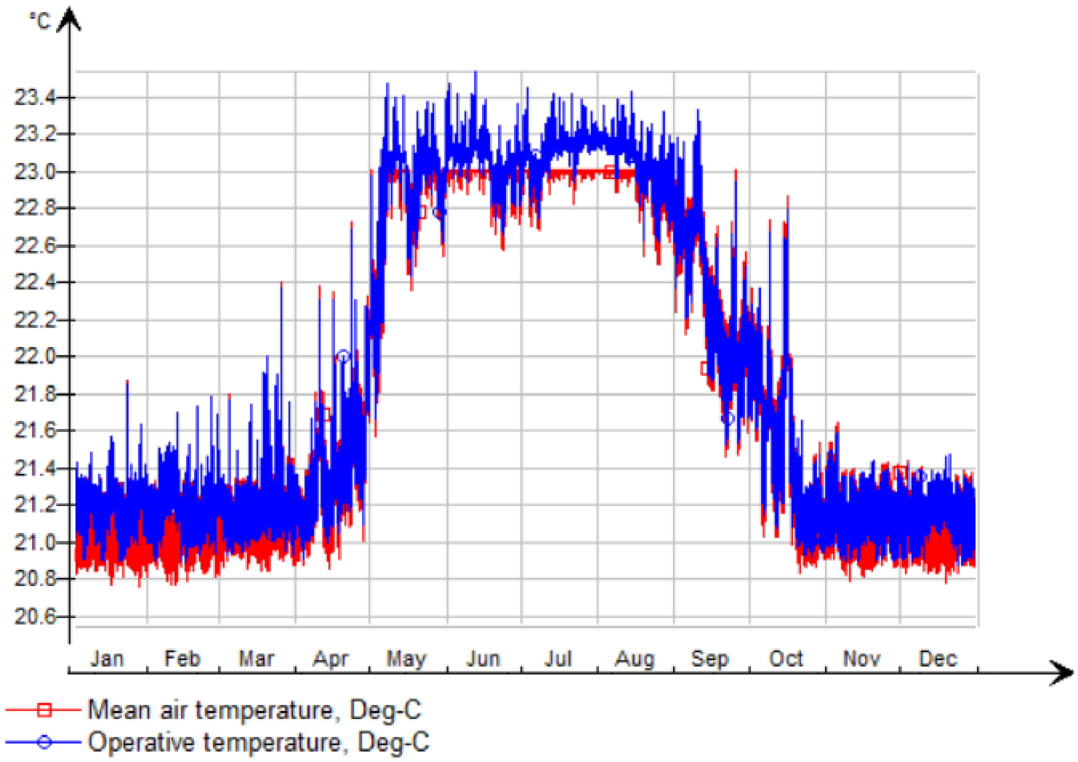


3. Wind speed at the roof height of the building

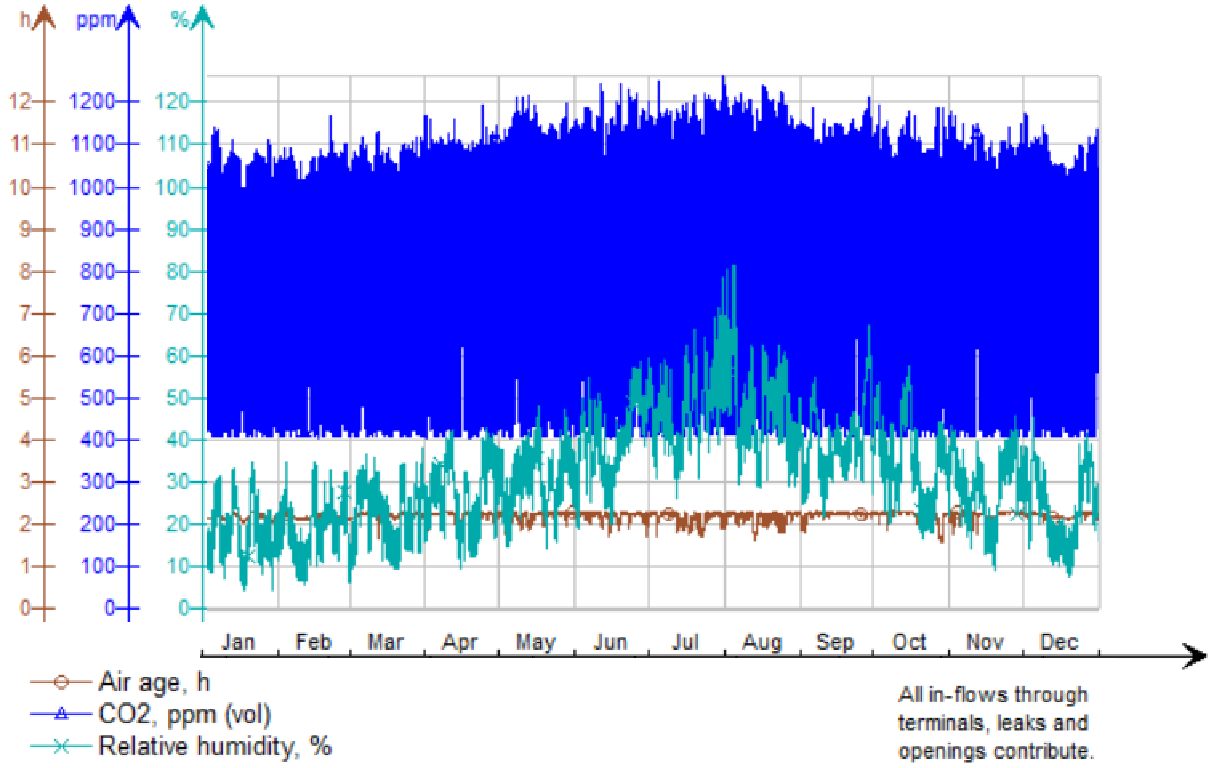


4. Air temperature, age, humidity and CO_2 level in the example room - children room 2, F3 (passive house with CAV ventilation all the year, awnings included - state 3 of the cooling system described in the section 7.3)

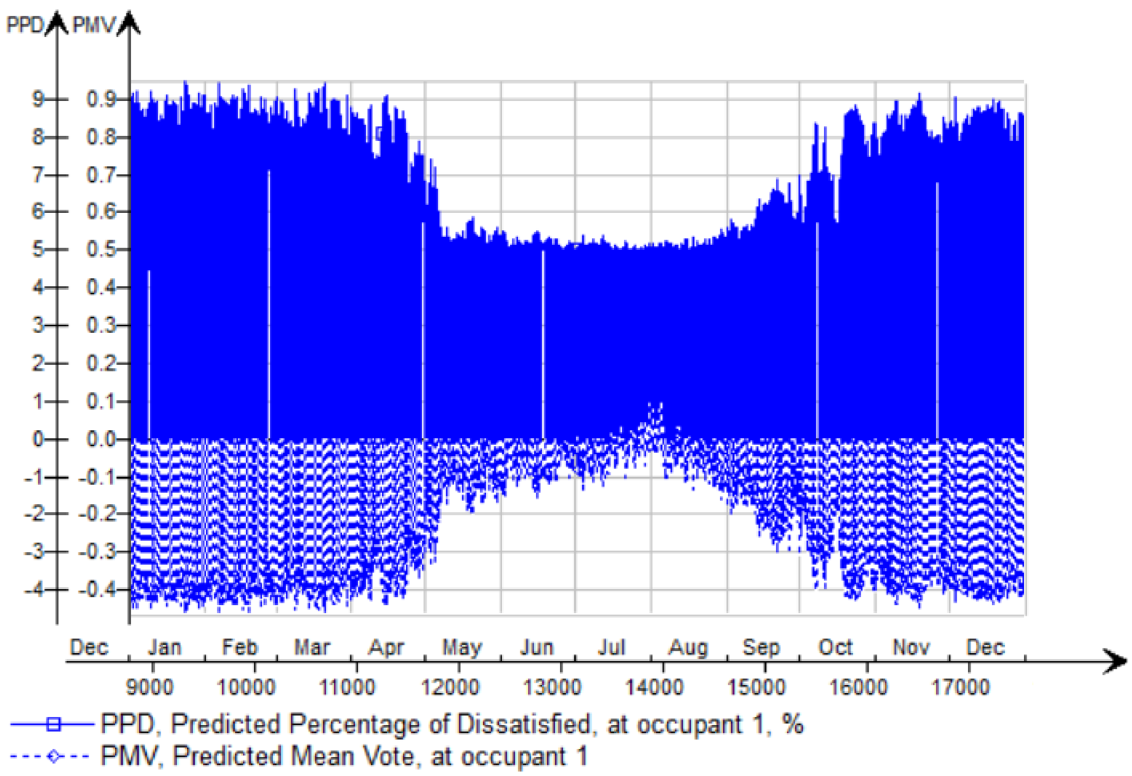
- **Dry bulb and operative air temperatures in the zone during reference 2016 year**



- **Age, CO_2 level and relative humidity of the air in the zone during reference 2016 year**

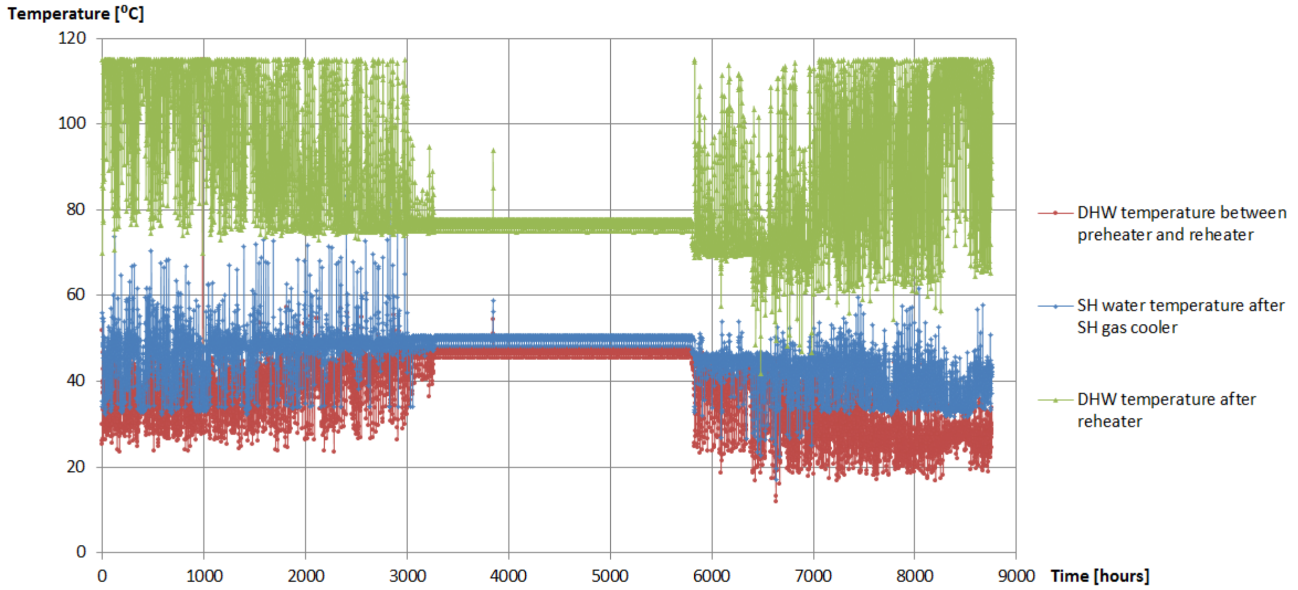


- **PPD and PMV from viewpoint of the occupant living in the zone, during reference 2016 year**

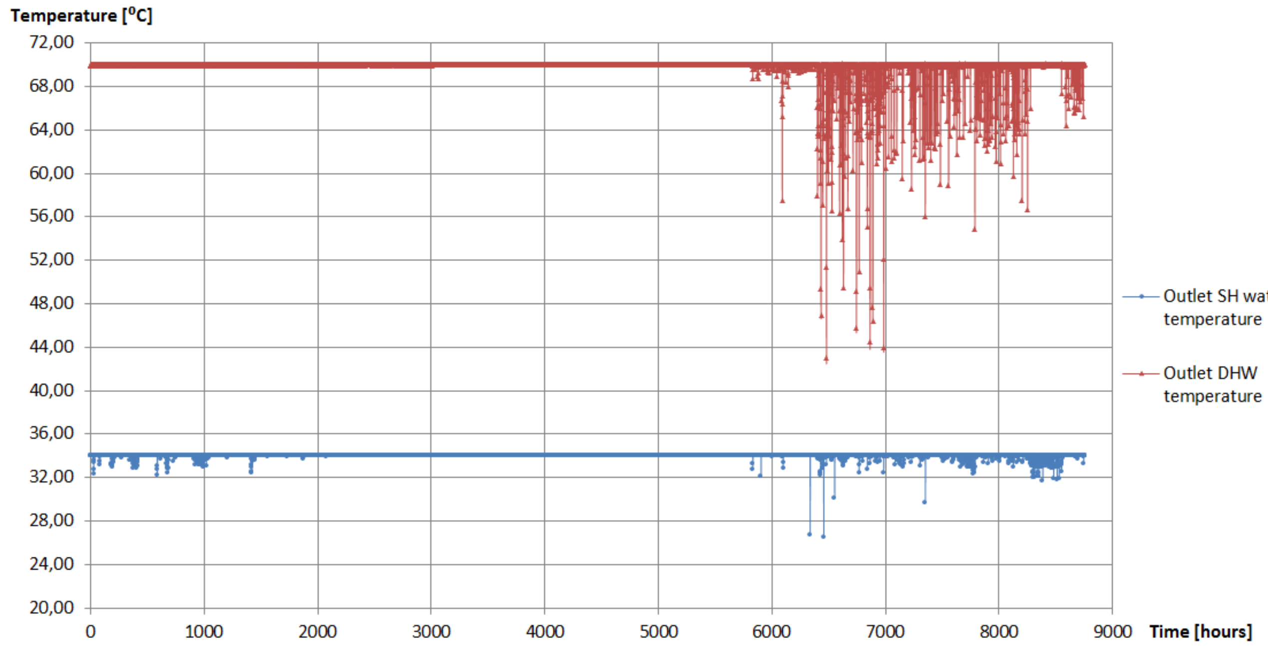


Appendix 4. Other heat pump results (after its optimization)

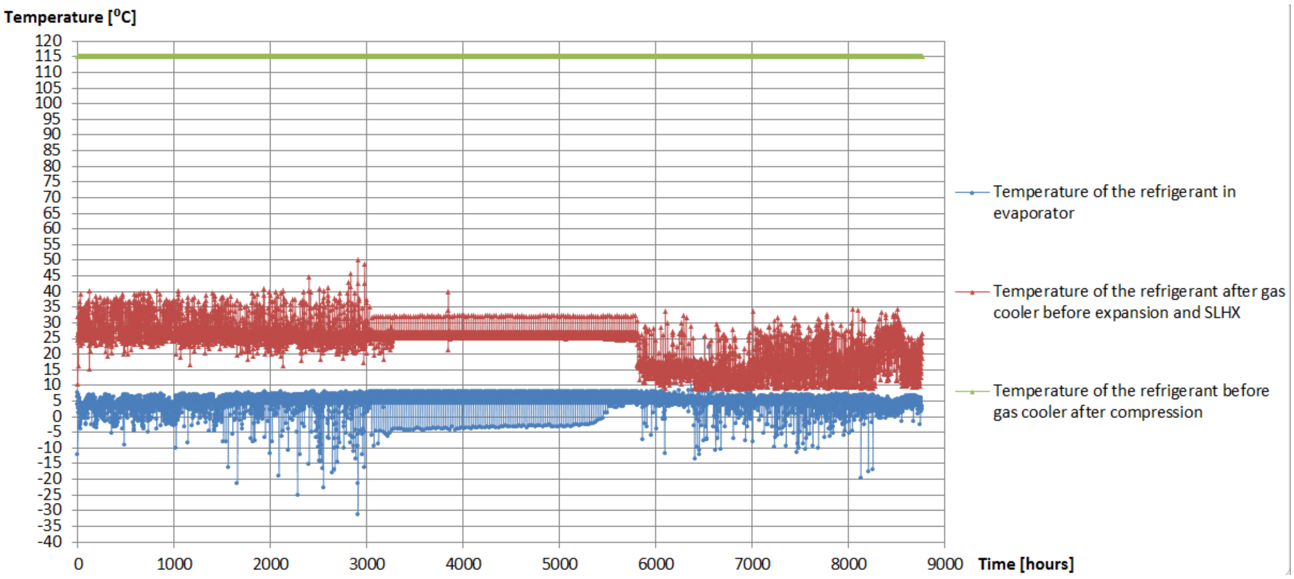
1. Installation water temperatures - after each gas cooler part (without regulation performing by by - passes and water pumps)



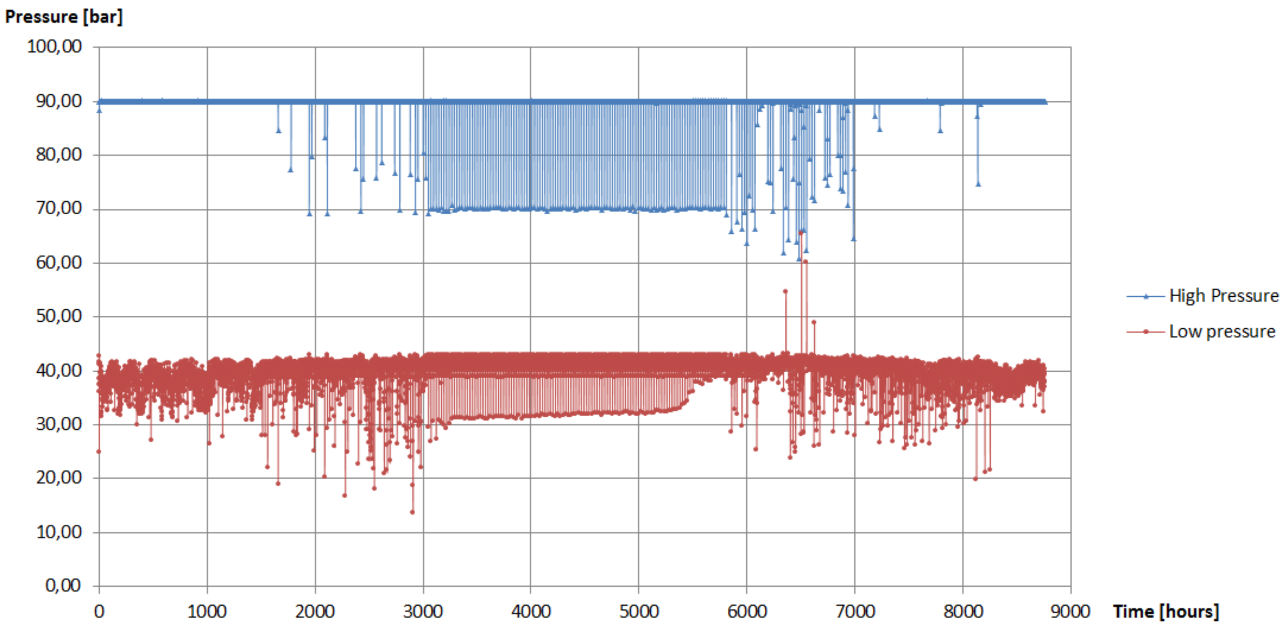
2. Installation water temperatures - final outlet temperatures (with regulation performing by by - passes and water pumps)



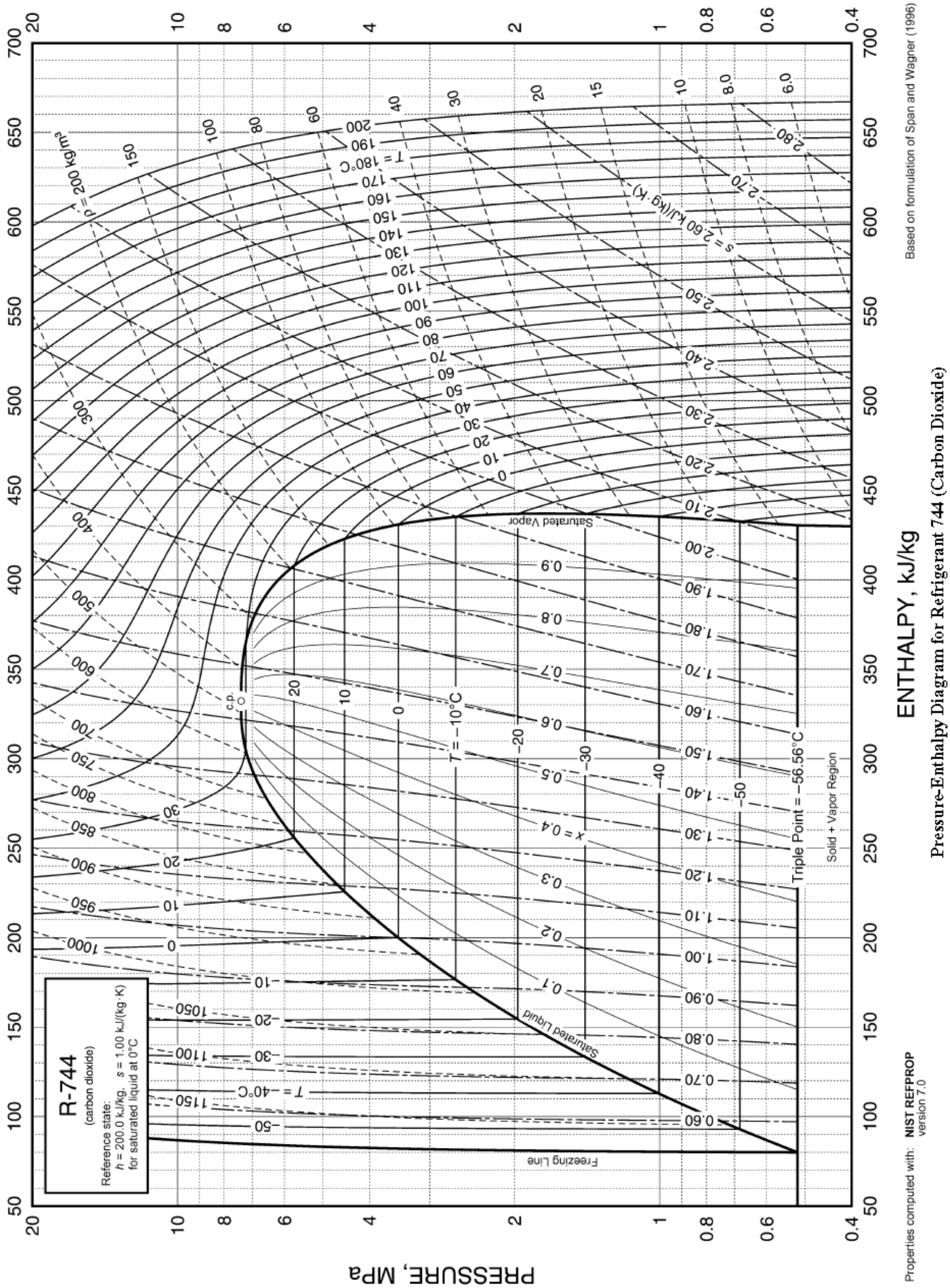
3. Temperatures of the refrigerant (CO_2) in main heat pump cycle points



4. High and low pressure - duration during simulated period (1 year)



Appendix 5. Pressure - enthalpy diagram for R744 (CO2) and its properties



Refrigerant 744 (Carbon Dioxide) Properties of Saturated Liquid and Saturated Vapor

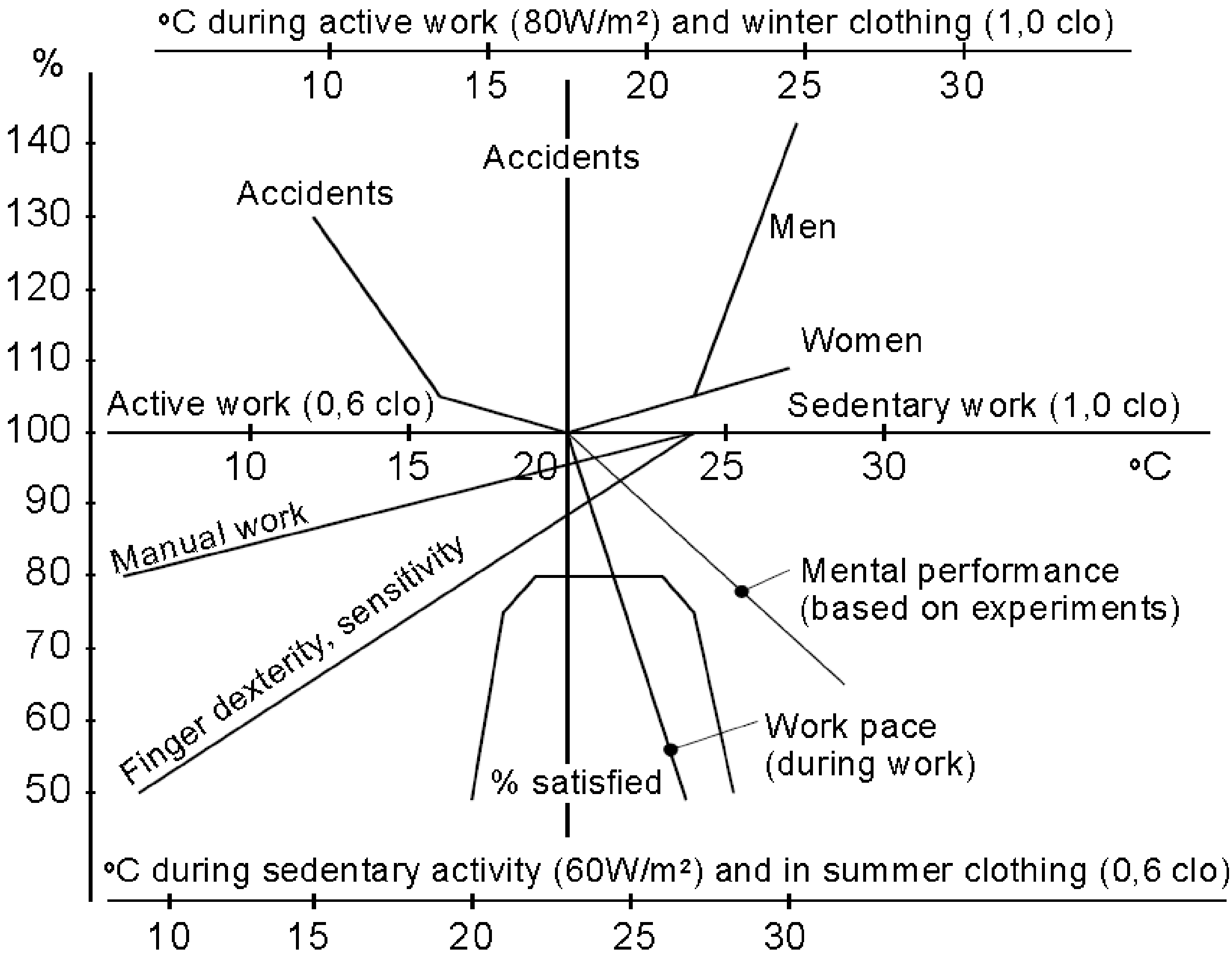
Temp.,* °C	Pressure, MPa	Density, kg/m³ Liquid	Volume, m³/kg Vapor	Enthalpy, kJ/kg Liquid	Enthalpy, kJ/kg Vapor	Entropy, kJ/(kg·K) Liquid	Entropy, kJ/(kg·K) Vapor	Specific Heat c_p, kJ/(kg·K) Liquid	Specific Heat c_p, kJ/(kg·K) Vapor	c_p/c_v Vapor	Velocity of Sound, m/s Liquid	Velocity of Sound, m/s Vapor	Viscosity, µPa·s Liquid	Viscosity, µPa·s Vapor	Thermal Cond., mW/(m·K) Liquid	Thermal Cond., mW/(m·K) Vapor	Surface Tension, mN/m	Temp.,* °C
−56.56[a]	0.51796	1178.5	0.07267	80.04	430.42	0.5213	2.1390	1.953	0.909	1.444	976	222.8	256.7	10.95	180.6	11.01	17.16	−56.56
−50	0.68234	1154.6	0.05579	92.94	432.68	0.5794	2.1018	1.971	0.952	1.468	928	223.4	229.3	11.31	172.1	11.58	15.53	−50
−48	0.73949	1147.1	0.05162	96.90	433.29	0.5968	2.0909	1.978	0.967	1.477	914	223.5	221.6	11.42	169.5	11.76	15.04	−48
−46	0.80015	1139.6	0.04782	100.88	433.86	0.6142	2.0801	1.985	0.982	1.486	900	223.6	214.3	11.53	166.9	11.95	14.56	−46
−44	0.86445	1132.0	0.04435	104.87	434.39	0.6314	2.0694	1.993	0.998	1.496	885	223.6	207.2	11.64	164.4	12.14	14.07	−44
−42	0.93252	1124.2	0.04118	108.88	434.88	0.6486	2.0589	2.002	1.015	1.507	871	223.6	200.3	11.75	161.8	12.34	13.60	−42
−40	1.00450	1116.4	0.03828	112.90	435.32	0.6656	2.0485	2.012	1.033	1.518	856	223.5	193.8	11.87	159.3	12.54	13.12	−40
−38	1.08050	1108.5	0.03562	116.95	435.72	0.6826	2.0382	2.022	1.052	1.530	842	223.4	187.4	11.98	156.8	12.75	12.65	−38
−36	1.16070	1100.5	0.03318	121.01	436.07	0.6995	2.0281	2.033	1.072	1.544	827	223.2	181.3	12.10	154.3	12.97	12.18	−36
−34	1.24520	1092.4	0.03093	125.10	436.37	0.7163	2.0180	2.045	1.094	1.558	813	223.1	175.4	12.22	151.8	13.20	11.72	−34
−32	1.33420	1084.1	0.02886	129.20	436.62	0.7331	2.0079	2.059	1.116	1.573	798	222.8	169.7	12.34	149.3	13.43	11.26	−32
−30	1.42780	1075.7	0.02696	133.34	436.82	0.7498	1.9980	2.073	1.141	1.590	783	222.5	164.2	12.46	146.9	13.68	10.80	−30
−28	1.52610	1067.2	0.02519	137.50	436.96	0.7665	1.9880	2.089	1.166	1.608	768	222.2	158.9	12.59	144.4	13.94	10.35	−28
−26	1.62930	1058.6	0.02356	141.69	437.04	0.7831	1.9781	2.105	1.194	1.627	753	221.8	153.8	12.72	141.9	14.20	9.90	−26
−24	1.73750	1049.8	0.02205	145.91	437.06	0.7997	1.9683	2.124	1.223	1.648	738	221.4	148.8	12.85	139.5	14.49	9.46	−24
−22	1.85090	1040.8	0.02065	150.16	437.01	0.8163	1.9584	2.144	1.255	1.671	723	220.9	144.0	12.98	137.1	14.78	9.02	−22
−20	1.96960	1031.7	0.01934	154.45	436.89	0.8328	1.9485	2.165	1.289	1.696	708	220.4	139.3	13.12	134.6	15.09	8.59	−20
−19	2.03100	1027.0	0.01873	156.61	436.81	0.8411	1.9436	2.177	1.307	1.709	700	220.1	137.1	13.18	133.4	15.25	8.37	−19
−18	2.09380	1022.3	0.01813	158.77	436.70	0.8494	1.9386	2.189	1.326	1.723	692	219.8	134.8	13.26	132.2	15.42	8.16	−18
−17	2.15810	1017.6	0.01756	160.95	436.58	0.8576	1.9337	2.201	1.346	1.738	684	219.5	132.6	13.33	131.0	15.59	7.95	−17
−16	2.22370	1012.8	0.01700	163.14	436.44	0.8659	1.9287	2.215	1.366	1.753	676	219.2	130.4	13.40	129.8	15.77	7.74	−16
−15	2.29080	1008.0	0.01647	165.34	436.27	0.8742	1.9237	2.228	1.388	1.768	668	218.8	128.3	13.47	128.6	15.95	7.53	−15
−14	2.35930	1003.1	0.01595	167.55	436.09	0.8825	1.9187	2.243	1.410	1.785	660	218.5	126.2	13.55	127.4	16.14	7.32	−14
−13	2.42940	998.1	0.01545	169.78	435.89	0.8908	1.9137	2.258	1.433	1.802	651	218.1	124.1	13.63	126.2	16.34	7.11	−13
−12	2.50100	993.1	0.01497	172.01	435.66	0.8991	1.9086	2.273	1.457	1.821	643	217.7	122.0	13.70	125.0	16.54	6.90	−12
−11	2.57400	988.1	0.01450	174.26	435.41	0.9074	1.9036	2.290	1.483	1.840	635	217.4	120.0	13.78	123.8	16.74	6.70	−11
−10	2.64870	982.9	0.01405	176.52	435.14	0.9157	1.8985	2.307	1.509	1.860	626	216.9	118.0	13.86	122.5	16.96	6.50	−10
−9	2.72490	977.7	0.01361	178.80	434.84	0.9240	1.8934	2.325	1.537	1.881	617	216.5	116.1	13.95	121.3	17.18	6.29	−9
−8	2.80270	972.5	0.01319	181.09	434.51	0.9324	1.8882	2.345	1.566	1.904	609	216.1	114.1	14.03	120.1	17.42	6.09	−8
−7	2.88210	967.1	0.01278	183.39	434.17	0.9408	1.8830	2.365	1.597	1.927	600	215.6	112.2	14.12	118.9	17.66	5.89	−7
−6	2.96320	961.7	0.01238	185.71	433.79	0.9491	1.8778	2.386	1.629	1.952	591	215.2	110.3	14.20	117.7	17.91	5.70	−6
−5	3.04590	956.2	0.01200	188.05	433.38	0.9576	1.8725	2.408	1.663	1.979	582	214.7	108.4	14.30	116.5	18.17	5.50	−5
−4	3.13030	950.6	0.01162	190.40	432.95	0.9660	1.8672	2.432	1.699	2.007	573	214.2	106.6	14.39	115.3	18.44	5.30	−4
−3	3.21640	945.0	0.01126	192.77	432.48	0.9744	1.8618	2.457	1.737	2.037	564	213.7	104.8	14.48	114.1	18.73	5.11	−3
−2	3.30420	939.2	0.01091	195.16	431.99	0.9829	1.8563	2.484	1.777	2.068	555	213.1	102.9	14.58	112.9	19.03	4.92	−2
−1	3.39380	933.4	0.01057	197.57	431.46	0.9914	1.8509	2.512	1.819	2.102	546	212.6	101.2	14.68	111.6	19.34	4.73	−1
0	3.48510	927.4	0.01024	200.00	430.89	1.0000	1.8453	2.542	1.865	2.138	536	212.0	99.4	14.79	110.4	19.67	4.54	0
1	3.57830	921.4	0.00992	202.45	430.29	1.0086	1.8397	2.574	1.913	2.176	527	211.5	97.6	14.89	109.2	20.02	4.35	1
2	3.67330	915.2	0.00961	204.93	429.65	1.0172	1.8340	2.609	1.965	2.218	518	210.9	95.9	15.00	108.0	20.38	4.17	2
3	3.77010	909.0	0.00931	207.43	428.97	1.0259	1.8282	2.645	2.020	2.262	508	210.3	94.2	15.12	106.8	20.76	3.99	3
4	3.86880	902.6	0.00901	209.95	428.25	1.0346	1.8223	2.685	2.080	2.309	499	209.6	92.5	15.24	105.5	21.17	3.80	4
5	3.96950	896.0	0.00872	212.50	427.48	1.0434	1.8163	2.727	2.144	2.360	489	209.0	90.8	15.36	104.3	21.60	3.62	5
6	4.07200	889.4	0.00845	215.08	426.67	1.0523	1.8102	2.772	2.213	2.416	480	208.3	89.1	15.49	103.1	22.06	3.45	6
7	4.17650	882.6	0.00817	217.69	425.81	1.0612	1.8041	2.822	2.289	2.476	470	207.6	87.5	15.62	101.8	22.54	3.27	7
8	4.28310	875.6	0.00791	220.34	424.89	1.0702	1.7977	2.875	2.370	2.541	460	206.9	85.8	15.76	100.6	23.06	3.10	8
9	4.39160	868.4	0.00765	223.01	423.92	1.0792	1.7913	2.934	2.460	2.612	451	206.2	84.2	15.91	99.4	23.61	2.93	9
10	4.50220	861.1	0.00740	225.73	422.88	1.0884	1.7847	2.998	2.558	2.690	441	205.4	82.6	16.06	98.1	24.21	2.76	10
11	4.61490	853.6	0.00715	228.49	421.79	1.0976	1.7779	3.068	2.666	2.776	431	204.6	80.9	16.22	96.9	24.84	2.59	11
12	4.72970	845.9	0.00691	231.29	420.62	1.1070	1.7710	3.145	2.786	2.871	421	203.8	79.3	16.39	95.6	25.53	2.42	12
13	4.84660	837.9	0.00668	234.13	419.37	1.1165	1.7638	3.232	2.919	2.977	411	203.0	77.7	16.56	94.4	26.27	2.26	13
14	4.96580	829.7	0.00645	237.03	418.05	1.1261	1.7565	3.328	3.068	3.095	401	202.1	76.1	16.75	93.1	27.08	2.10	14
15	5.08701	821.2	0.00622	239.99	416.64	1.1359	1.7489	3.436	3.237	3.228	391	201.2	74.4	16.95	91.9	27.96	1.95	15
16	5.21080	812.4	0.00600	243.01	415.12	1.1458	1.7411	3.558	3.429	3.378	381	200.3	72.8	17.16	90.6	28.93	1.79	16
17	5.33680	803.3	0.00578	246.10	413.50	1.1559	1.7329	3.698	3.649	3.550	370	199.3	71.2	17.39	89.4	29.99	1.64	17
18	5.46510	793.8	0.00557	249.26	411.76	1.1663	1.7244	3.858	3.905	3.748	360	198.3	69.5	17.64	88.1	31.16	1.49	18
19	5.59580	783.8	0.00536	252.52	409.89	1.1769	1.7155	4.044	4.204	3.979	349	197.2	67.8	17.90	86.9	32.47	1.35	19
20	5.72910	773.4	0.00515	255.87	407.87	1.1877	1.7062	4.264	4.560	4.252	338	196.1	66.1	18.19	85.7	33.94	1.20	20
21	5.86480	762.4	0.00494	259.33	405.67	1.1989	1.6964	4.526	4.990	4.578	326	194.9	64.4	18.50	84.5	35.61	1.06	21
22	6.00310	750.8	0.00474	262.93	403.26	1.2105	1.6860	4.846	5.519	4.976	314	193.6	62.7	18.85	83.4	37.52	0.93	22
23	6.14400	738.4	0.00453	266.68	400.63	1.2225	1.6749	5.248	6.185	5.472	302	192.3	60.9	19.23	82.4	39.74	0.80	23
24	6.28770	725.0	0.00433	270.61	397.70	1.2352	1.6629	5.767	7.049	6.107	288	190.8	59.0	19.66	81.5	42.35	0.67	24
25	6.43420	710.5	0.00412	274.78	394.43	1.2485	1.6498	6.467	8.212	6.949	274	189.1	57.0	20.16	80.8	45.51	0.55	25
26	6.58370	694.5	0.00391	279.26	390.71	1.2627	1.6353	7.460	9.862	8.121	259	187.2	55.0	20.73	80.5	49.44	0.44	26
27	6.73610	676.4	0.00369	284.14	386.39	1.2783	1.6189	8.970	12.38	9.870	243	185.0	52.8	21.42	80.7	54.56	0.33	27
28	6.89180	655.3	0.00346	289.62	381.20	1.2958	1.5999	11.550	16.69	12.780	225	182.1	50.3	22.27	81.9	61.73	0.23	28
29	7.05090	629.4	0.00320	296.07	374.61	1.3163	1.5763	16.950	25.74	18.630	205	178.2	47.5	23.41	85.2	73.19	0.13	29
30	7.21370	593.3	0.00290	304.55	365.13	1.3435	1.5433	35.340	55.82	36.660	177	171.3	43.8	25.17	95.4	98.02	0.05	30
30.98[c]	7.37730	467.6	0.00214	332.25	332.25	1.4336	1.4336	∞	∞	∞	0	0.0	—	—	∞	∞	0.00	30.98

*Temperatures on ITS-90 scale [a]Triple point [c]Critical point

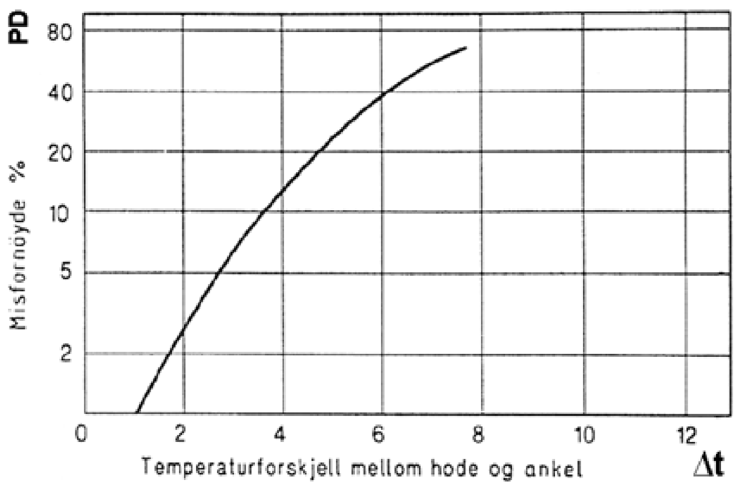
Both pressure - specific enthalpy diagram and properties of CO_2 as the refrigerant were taken from chapter 30 „Thermophysical properties of refrigerants" of publication released by ASHRAE Inc. in 2009.

Appendix 6. Interesting graphs

1. Thermal room climate vs. efficiency and productivity



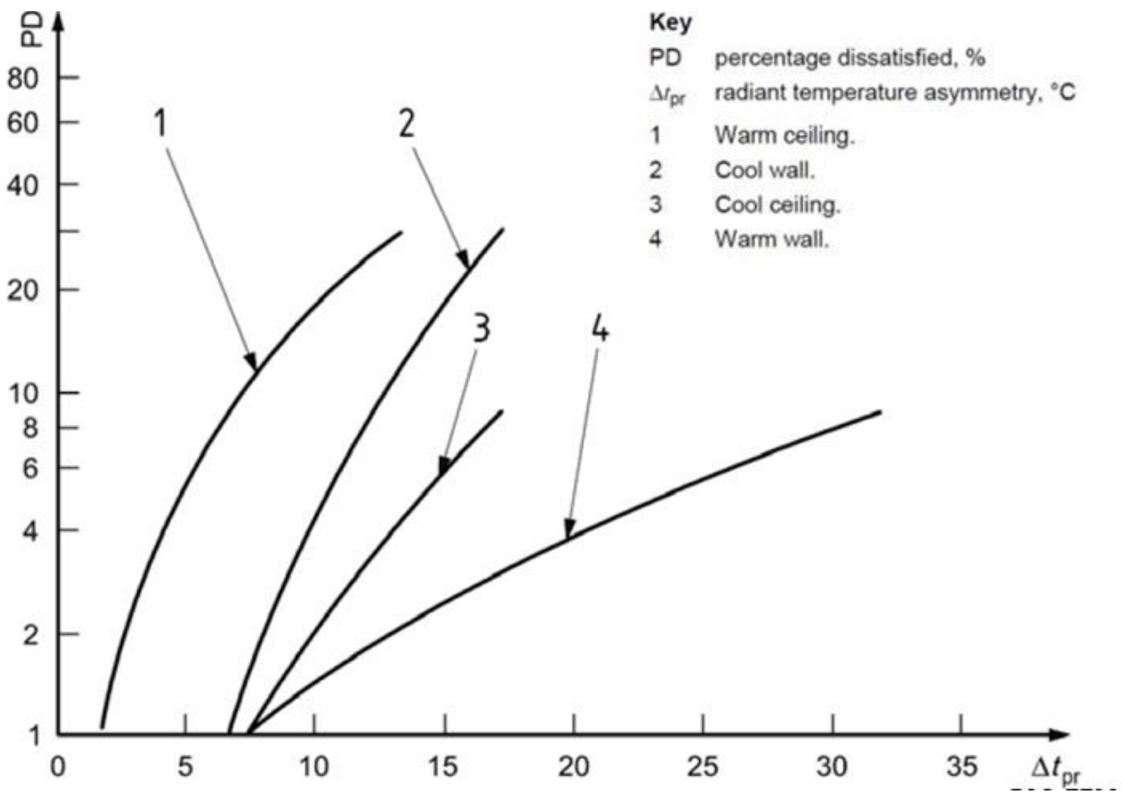
2. Local discomfort caused by vertical air temperature difference



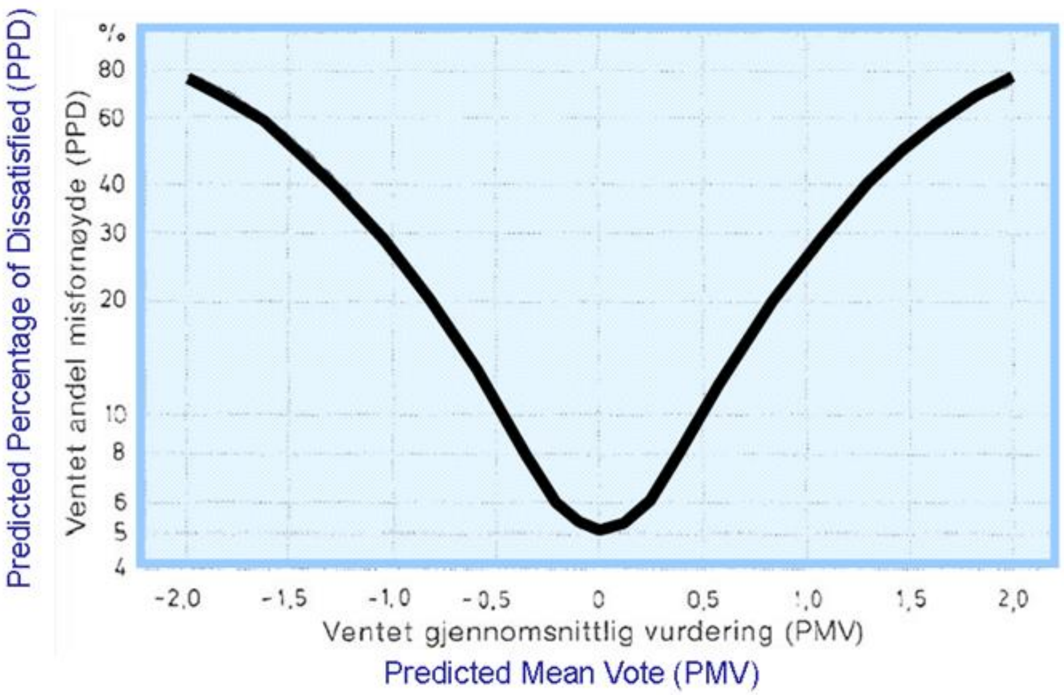
3. Examples of recommended categories for design of mechanical heated and cooled buildings (based on NS-EN 15251)

Category	Thermal state of the body as a whole		Local thermal discomfort			
	PPD %	Predicted Mean Vote	PPD Draught %	PPD vertical temp. diff. %	PPD warm or cold floor %	PPD temp. asymmetry %
I	< 6	-0.2 < PMV < + 0.2	< 15	< 3	< 10	< 5
II	**< 10**	**-0.5 < PMV < + 0.5**	< 20	< 5	< 10	< 5
III	< 15	-0.7 < PMV < + 0.7	< 25	< 10	< 15	< 10
IV	> 15	PMV<-0.7; or +0,7<PMV	> 25	> 10	> 15	> 10

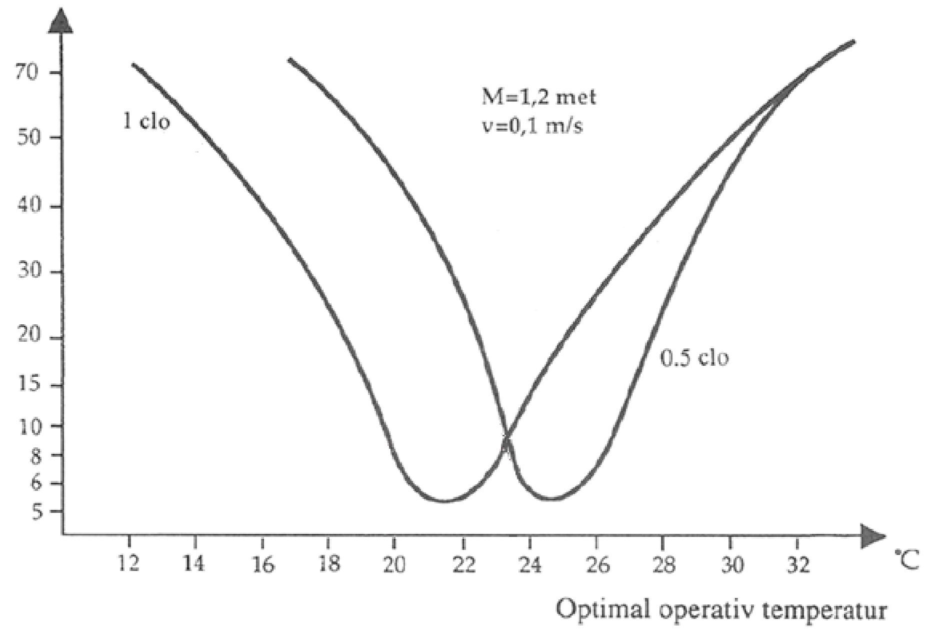
4. Local thermal discomfort caused by radiant temperature assymetry (based on ISO 7730)



5. PPD (Predicted Percentage of Dissadisfied) in a funtion of PMV (Predicted Mean Vote)

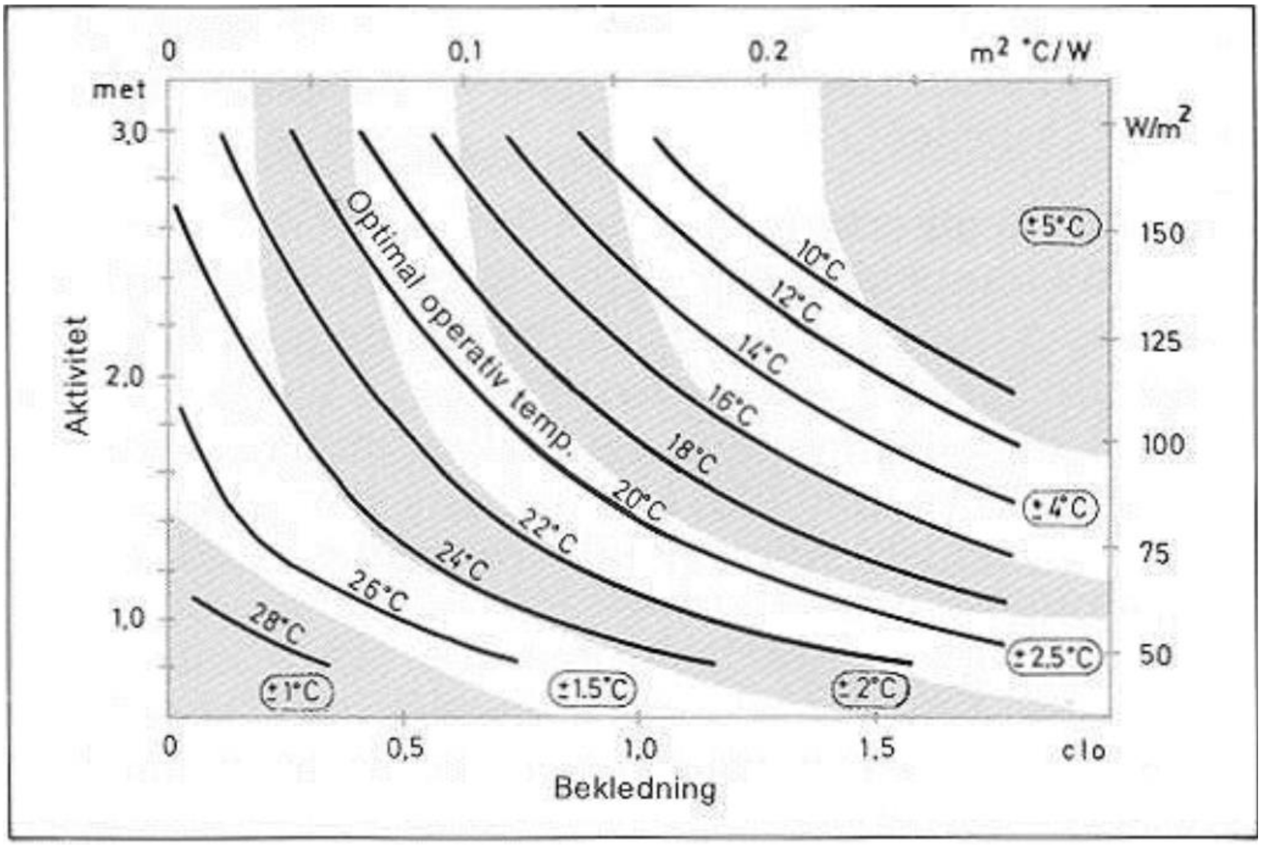


6. Relationship between optimal operative temperature and expected PPD for winter and summer clothing

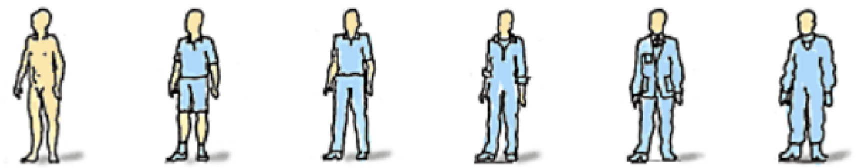


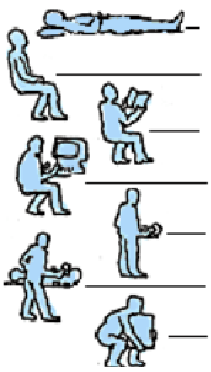
7. Optimal operative temperature in a function of clothing and activity level

- **Plot form**



- **Tabular form with schematic drawings**





°C	0 Clo	0,3	0,5	0,8	1,0	1,5
0,8 MET	29 ± 1	28,5 ± 1	27 ± 1	26 ± 1,5	24,5 ± 1,5	22 ± 2
1,0	28,5 ± 1	27 ± 1	26 ± 1	24,5 ± 1,5	23 ± 2	20 ± 2
1,2	28 ± 1	26 ± 1	24,5 ±1,5	22,5 ± 2	21 ± 2,5	18 ± 3
1,4	27 ± 1	25 ± 1,5	23,5 ± 2	21,5 ± 2	20 ± 2,5	16 ± 3
1,6	26,5 ± 1,5	24 ± 1,5	23 ± 2	20,5 ± 2,5	19 ± 3	15 ± 4
2,0	26 ± 1,5	23 ± 2	21 ± 2,5	18, 5 ± 3	16 ± 3	(12 ± 3)
2,4	25,5 ± 1,5	21,5 ± 2	20 ± 2,5	(16 ± 3)	(14 ± 3)	(9 ± 5)

8. Total and sensible heat emitted from human body, dependent on the activity type

Activity	Total heat Met [1)]	Total heat $W.person^{-1}$ [2)]	Sensible heat $W.person^{-1}$
Reclining	0.8	80	55
Seated, relaxed	1.0	100	70
Sedentary activity (office, school, laboratory)	1.2	125	75
Standing, light activity (shopping, laboratory, light industry)	1.6	170	85
Standing, medium activity (shop assistant, machine work)	2.0	210	105
Walking on the level:			
2 km/h	1.9	200	100
3 km/h	2.4	250	105
4 km/h	2.8	300	110
5 km/h	3.4	360	120

1) 1 met = 58 $W.m^{-2}$

2) rounded values for a human body with a surface of 1.8 $m^2.person^{-1}$

Most of graphs and tables presented in this attachment are taken from academic lectures from TEP4235 course („Energy Management in Buildings"), performed at NTNU during fall semester in 2015. The rest of them is gained from Google Graphics website.