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# Heating power at room and building levels in passive houses and low-energy buildings

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## Preface

The building standards for new buildings and comprehensive rehabilitative buildings have improved the last years. The goal is to reduce the heat demand in the building stock and in that way contribute to an environment friendly energy supply. The Norwegian building stock represent approximately 40% of the domestic energy consumption[2]. In the next technical regulations, *TEK15,* it is expected that the requirements of building standard will be tightened to the level we call passive house standard today. That will lead to a significant reduction in energy consumption and thus the power demand for heating purposes [3]. Hence, inaccurate calculation of power demand will have greater relative importance.

In this master thesis, the Norwegian and American standard for calculating power demand is compared, both on room level and on building level. The Swedish standard, *FEBY 12*, is also investigated regarding the choice of design outdoor temperature. The goal of this master thesis is an improvement of the Norwegian standard, *NS-EN 12831:2003- Varmesystemer i bygninger - Metode for beregning av dimensjonerende effektbehov*. Properly sized heating systems is of great importance for comfort, energy use, economy and environment and is further investigated in this master thesis.

There is little documentation and previous studies on power demand for room and building heating in passive houses and low energy buildings. In this master thesis, an office building is investigated and real power consumption is compared to theoretical calculated and simulated power demand. A lot information about heating systems is, among others, extracted from the report COWI prepared on behalf of Enova; ”*Hensiktsmessige varme og kjøleløsninger i bygninger mars 2013*”.

This master thesis is written in collaboration with «Erichsen & Horgen» and is linked to a research and development project by Enova. Thanks to my supervisor Laurent Georges and my co supervisor Ida Bryn from ”Erichsen & Horgen” who contributed to the task.

## Summary

This master thesis investigates the Norwegian standard for calculating necessary power demand, *NS-EN 12831:2003- Varmesystemer i bygninger - Metode for beregning av dimensjonerende effektbehov*. In that context, it is also investigated how power demand affects heating systems financially and environmentally, but also in terms of comfort and indoor air quality.

Improved energy efficiency are leading to a reduction in the power demand for heating purposes in buildings. However, there are few guidelines for how the power demand for heating should be calculated at room level or at building level and correspondingly little documentation on thermal comfort and power demand in low-energy and passive houses.

Dimensioning heating systems correct is very important for comfort, energy and finance. Net power demand over the year should determine the distribution between base load and peak load. It is crucial that the estimated power demand correspond to real power demand in order to find the optimal distribution. Gross power demand is the basis for net power demand. A more realistic net power duration curve leads to a more profitable distribution between base load and peak load. If a safety margin is desirable, one should install an additional peak load, which is a cheap investment. That will not affect the operating costs appreciable, as it hardly will be in use.

Buildings are complex and there could be many reasons why real power consumption do not match the calculated and intended power demand. Larger heat loss than calculated may be a result of an inaccurate calculation method, poor quality of the work in the construction phase and that the building is used in a different way than intended.

Choosing a reasonable ventilation airflow at design conditions is important. In the winter, there is rarely any cooling demand, and the necessary ventilation airflow only depends on the airflow needed to ensure good indoor air quality.

In passive houses and low-energy buildings it is cheaper to maintain good indoor air quality as the heat transfer coefficient is low and the heat recovery in the ventilation system has good efficiency. It would seldom make financial sense to reduce indoor air quality, neither thermal nor atmospheric. Especially in commercial buildings, where a small percentage drop in performance would constitute a major cost in terms of wages to non-productive time.

The main improvement potential to the Norwegian standard, *NS-EN 12831:2003*, found in this master thesis is:

- One of the most obvious potentials for improvement is to include internal loads as lighting, computers and people to reduce the power demand. Especially at building level were one use a central heating system, in which the need of installed capacity will be reduced. At room level, one should be more careful and only include internal load that most certainly occurs at design conditions. The sum of the power demand at room level might then be larger than power demand at the building level, but it includes the uncertainty of using different rooms at different times.

- The calculation method for infiltration, particularly mechanical infiltration, have a potential for improvement. The American standard, *ASHRAE 2013* takes into account that the airflow is not linear with the pressure difference across the building envelope. *NS-EN 12831:2003* does not, but adds mechanical ventilation airflow as an addition, distributed by permeability.

- Heat release from ventilation fans and heat loss due to transmission and leakage from ducts are included in *ASHRAE 2013,* and is an improvement potential of the Norwegian standard. Heat release from the ventilation system in the investigated building is estimated to about 4,8kW at design conditions, which is 6,7% of calculated design power demand in the ventilation system.

- Clever control of the installed power is also a possibility. It is possible to drop night setback during the coldest periods or exploit installed capacity of domestic hot water.

- The Swedish standard, *FEBY 12*, seems to have the most detailed approach of how to choose design outdoor temperature. One finds the number of days in the "n-days mean temperature" method by calculating the heat storage in the actual building. *FEBY 12* provides a method of grading the choice of design outdoor temperature based on heat storage in the building.

Several buildings should be examined in order to conclude that the recommendations given in this thesis are the best options. It will be much easier having a simpler control principle than in the investigated office in this master thesis. Being able to play with the control of the building will also make it a lot easier to eliminate sources of error. Nevertheless, the main principles found in this master thesis should be considered reliable.

## Sammendrag

Denne masteroppgaven tar for seg norsk standard for beregning av effektbehov, *NS-EN 12831:2003- Varmesystemer i bygninger - Metode for beregning av dimensjonerende effektbehov*. I den sammenheng blir det sett på hvordan effektbehov påvirker oppvarmingssystemer økonomisk, men også med tanke på komfort, inneklima og miljø.

Bedret energieffektivitet fører til kraftig reduksjon i effektbehov til oppvarming i bygninger. Det finnes imidlertid lite anvisninger for hvordan effekten for oppvarming sentralt og på romnivå bør beregnes og tilsvarende lite dokumentasjon av termisk komfort og effektbehov på vinterstid i lavenergi- og passivhus.

Riktig dimensjonerte varmeanlegg har stor betydning for komfort, energibruk og økonomi. Netto effektbehov over året bør avgjøre fordelingen mellom grunnlast og spisslast. Det er avgjørende at beregnet effektbehov samsvarer med virkelig effektbehov for å finne optimal fordeling. Brutto effektbehov legger grunnlaget for netto effektbehov. En mer realistisk netto effekt-varighetskurve gjør at det kan velges en mer lønnsom fordelingen mellom grunnlast og spisslast. Dersom det er ønskelig med en sikkerhetsmargin bør det installeres en ekstra spisslast, som er en billig innvestering. Denne vil ikke påvirke driftskostnadene nevneverdig da den nesten ikke vil bli brukt.

Bygninger er komplekse og det er derfor mange årsaker til at effektbehovet i virkeligheten ikke er likt det som ble beregnet. Større varmetap enn beregnet kan komme av unøyaktig beregningsmetode, kvaliteten på håndverket i byggefasen og at bygget brukes på en annen måte enn tiltenkt.

Å velge en fornuftig luftmengde for ventilasjon er viktig. Om vinteren er det sjelden behov for kjøling, men luftmengden må være stor nok til å sikre god innendørs luftkvalitet.

I passivhus og lavenergibygg er det billigere å opprettholde et godt inneklima, da varmetapskoeffisienten er lav og varmegjenvinneren i ventilasjonssystemet har god virkningsgrad. Økonomisk lønner det seg sjelden å redusere kvaliteten på inneklima, verken termisk eller atmosfærisk. Dette gjelder spesielt for yrkesbygg, der kun noen prosent fall i ytelse vil kunne utgjøre en stor kostnad i form av lønn til ikke-produktiv tid.

De viktigste forbedringspotensialene til den norske standarden, *NS-EN 12831:2003*, som ble funnet i denne masteroppgaven:

- En av de mest åpenbare forbedringspotensialene er å inkludere internlast, som lys, datamaskiner og personer for å redusere effektbehovet. Spesielt på bygningsnivå, som ved bruk av et sentralt varmesystem vil redusere behovet for installert effekt i varmeanlegget. På romnivå bør man være litt mer forsiktig og bare inkludere internlast man er sikker på at oppstår ved dimensjonerende forhold. Summen av effektbehov på romnivå kan da bli større enn effektbehovet på bygningsnivå, men det inkluderer usikkerheten ved bruk av forskjellige rom til forskjellige tider.

- Beregningsmetoden for infiltrasjon, og da spesielt mekanisk infiltrasjon, har et forbedringspotensial. Den amerikanske standarden, *ASHRAE 2013*, tar hensyn til at luftmengden ikke er lineær med trykkdifferansen over bygningsskallet. Det gjør ikke *NS-EN 12831:2003,* som legger til den mekaniske ventilasjonen som et tillegg fordelt etter permeabilitet.

- Oppvarmingseffekten fra ventilasjonsvifter, samt varmetap ved transmisjon og lekkasje fra ventilasjonskanaler tas med i *ASHRAE 2013* og er et forbedringspotensial til den norske standarden. I kontorbygget som ble studert nærmere utgjorde oppvarmingseffekten fra ventilasjon 4,8kW, hvilket er 6,7% av dimensjonerende effektbehov for varmebatteriet i ventilasjonsanlegget.

- Smart styring av den installerte effekten er det også en mulighet. Det er mulig å droppe nattsenking i de kaldeste periodene eller utnytte installert effekt til varmt tappevann.

- Den svenske standarden, *FEBY 12*, ser ut til å ha den mest detaljerte tilnærmingen til hvordan man bør velge dimensjonerende utetemperatur vinterstid. Man finner antall dager i "n-dagers middeltemperatur" metoden basert på varmelagringen i den aktuelle bygningen. Metoden gitt i *FEBY 12* gjør det altså mulig å differensiere valget av dimensjonerende utetemperatur på grunnlag av varmelagringen i den aktuelle bygningen.

For å kunne konkludere at forbedringspotensialene gitt i denne oppgaven er beste løsning, bør flere bygninger bli undersøkt. Det vil være mye lettere å konkludere dersom det er et enklere styringsprinsipp enn i kontorbygningen som ble undersøkt. Dersom man i tillegg kan leke litt med styringen vil det være enklere å utelukke feilkilder. Hovedprinsippene som ble funnet i denne masteroppgaven bør likevel bli regnet som pålitelige.

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# 1 Introduction

## 1.1 Background

This master thesis is written at the ”Department of Energy and Process Engineering» at NTNU. The goal is to find a better way to dimension heating systems and improve the Norwegian standard for calculating design power demand, *NS-EN 12831:2003,* especially for highly insulated buildings. The Swedish standard *FEBY 12* and the American standard *ASHRAE 2013* are compared to the Norwegian standard, and are used as inspiration to several of the improvements proposed. An office building located near Oslo is investigated as well.

Improved energy efficiency leads to a reduction in power demand for heating. Some buildings are even built without need for heating. Furthermore, there is little documentation about the thermal comfort and power demand for heating purposes in passive and low energy buildings. Correct dimensioning of heating installations is important for the thermal comfort, the energy consumption as well as the economic performance.

It is expected that the requirements in *TEK15* will be on the same level as the current passive house standard. That will result in significant reduction of energy consumption and power demand for heating buildings [3]. Hence, inaccurate calculation of power demand could have a greater relative importance. Therefore, it is desirable to improve the method within the next technical requirement is released.

## 1.2 Purpose

In order to dimension heating systems in an optimal way, it is crucial that the calculation of power demand is as realistic as possible. The aim of the master thesis is to propose better alternatives and methods for calculation of power demand on room and building level, especially in highly insulated buildings.

## 1.3 Limitation

This thesis examines the power demand for heating of the building, room heating and heating of ventilation air. Heating of domestic hot water is considered as constant over the year. This approach can be justified by using a large accumulation tank for domestic hot water.

An important test of several of the possible improvements is done by measuring the power consumption in an office building and compare with theoretical calculated values and SIMIEN simulations. Improvements to the Norwegian standard is presented on basis of theory and the comparison between real and calculated power demand in the office building. The improvements proposed should be tested on several buildings, as there are many factors that affects the power demand of a building. It will be an advantage if one tests buildings using different control principles, to exclude sources of error.

It would also be an advantage having colder outdoor temperature the days of measuring. The week measured was week 50 in 2014. The lowest outdoor temperature occurred Tuesday and varied between -2°C and -6°C. Results will be most lifelike if there is design temperature outside. Colder outdoor temperature will eliminate the possibility of cooling demand in most cases.

### 1.4 Readers Guide

The first part is about energy supply to buildings and requirements that must be followed. It will be looked upon base load and peak load and the power distribution between them. Control methods are discussed and put up against indoor climate and economy. Then the Norwegian and American standard for calculating power demand are compared. In addition, the method for calculating design outdoor temperature in the Swedish standard, *FEBY 12*, is investigated and compared to the Norwegian standard. Power consumption, user behavior, ventilation quantities among others is measured in an office building and real power demand is compared to theoretical calculation and SIMIEN simulations. Finally, it is proposed improvements to the Norwegian standard, as well as control and installation methods.

# 2 Method

This master thesis is answered by searching literature, making a survey and investigate an office building in use. Literature search was used to find relevant theory, while survey was useful for collecting empirical data and experience from the industry. Since passive houses and low energy buildings are among the newest building it was necessary to make direct contact with companies in the industry that can document how they work in practice.

## 2.1 Literature

The literature study includes identifying and reviewing relevant sources, materials and background information related to the project's theme. Literature of interest is how to calculate power demand. Studies of real and estimated power demand on both room and building level and correlations between measured and theoretically calculated power demand for heating is of particularly interest.

The Norwegian Building Regulations are used extensively in addition to Norwegian standards. Norwegian standard for calculating design power demand is compared to the American standard. Useful literature and theories found form the basis of the theory chapter and further analysis. It turned out to be difficult to find wide, nuanced and relevant theory. Therefore, it is performed a survey among relevant companies and an office building in use is examined.

## 2.2 Survey

The survey were sent out by mail to the most major and some minor district heating providers in Norway with questions including correspondence between ordered and delivered power.

Some of the companies contacted supplies electricity as well. The incoming information is therefore based on both the electricity supply and district heating delivery. Most companies do not distinguish whether delivered power are used for heating the building, heating of hot water or other power consumption. Still, some interesting information is found.

## 2.3 Investigation of an office building

An office building located around Oslo is investigated. Measured power consumption was provided over a long time span, but user behavior, ventilation quantities among others was not provided and was therefore measured in week 50. The measured real power consumption is compared to theoretical calculations and SIMIEN simulations in order to provide improvements to the Norwegian standard.

# 3 Energy and power in passive houses and low energy buildings

## 3.1 Energy supply

Building standards for new buildings and comprehensive rehabilitative buildings have been changed the last years. The goal is to reduce the heat demand in the building stock and in that way contribute to an environment friendly energy supply. The Norwegian building stock represent approximately 40% of the domestic energy consumption [2].

### 3.1.1 *TEK10* requirements, passive houses and low energy buildings

Current requirements for energy supply to heating purposes in Norway is found in the technical regulations; *TEK10* [4]:

**Table 1 *TEK10* – Energy supply**

| *TEK 10* § 14-7. | **Energy supply** |
|---|---|
| *1* | It is not allowed to install oil fired boiler for fossil fuel used for base load |
| *2* | Buildings over 500 m2 heated BRA[1] shall be designed and constructed so that a minimum of 60% of net heat demand can be covered with a different energy supply than direct-acting electricity or fossil fuels by the end user. |
| *3* | Buildings up to 500 m2 heated BRA shall be designed and constructed so that at least 40% of net heat demand can be covered with different energy supply than direct-acting electricity or fossil fuels by the end user. |
| *4* | The requirement for energy supply in the second and third paragraphs shall not apply if it is documented that natural conditions make it practically impossible to satisfy the requirement. For residential buildings, the requirement for energy supply neither apply if the net heating demand is estimated at less than 15,000 kWh / year or requirement leads to additional costs of residential building's life cycle. |
| *5* | Residential building that under the fourth paragraph is exempt from the requirement for energy supply should have a chimney and closed fireplace for the use of biofuels. This does not apply for units below 50 m2 heated BRA or a residential that meets the requirements of the passive house standard. |
| **§ 14-8. District heating** | If there is determined connection to the district heating system by the Planning and Building Act § 27-5 , new buildings shall be equipped with heating systems so that the district heating can be used for space heating, ventilation and hot water. |

[1] BRA is usable area for a building exclusive open covered area, according to *NS 3940 Areal- og volumberegninger av bygninger*. BRA for a building is the sum of the BRA for all measurable levels/floors, regardless if the use includes the buildings net area and the area of internal walls[2]

Passive and low energy residential buildings are often so energy efficient that they can use point 4. This means that there is no requirement for the amount of net heat demand that should be met with different energy supplies than direct-acting electricity or fossil fuels. However, there is a requirement in the passive house and low energy building standard, *NS3700 2013*, for residential buildings that:

*" Estimated amount of delivered electricity and fossil energy should be less than the total net energy minus 50% of net energy for hot water."*[5]

Unless natural conditions make it practically impossible to meet the requirement, passive and low-energy commercial buildings must follow point 2 and 3. If heated floor area is over 500m$^2$, 60% of net heating demand should have posibilities to be covered by other energy supply then direck-acting electrisity or fossil fuels. If the heated floor area is less than 500 m$^2$, 40% is the requirement.

#### 3.1.2 Energy and power

It is expected that the requirements for building standard in *TEK15* will be tightened and go towards passive house standard. Hence, power and energy requirements for heating in buildings will reduce [3].

The heat demand is divided into heating of the building and heating of domestic hot water. Heating of the building is then dividend into room heating and ventilation heating as shown in Figure 1.[2]

Heating of buildings is stretched depending of outdoor temperature and varies a lot through a year. The amount of energy and power used in passive and low energy buildings is substantial reduced compared to older houses.

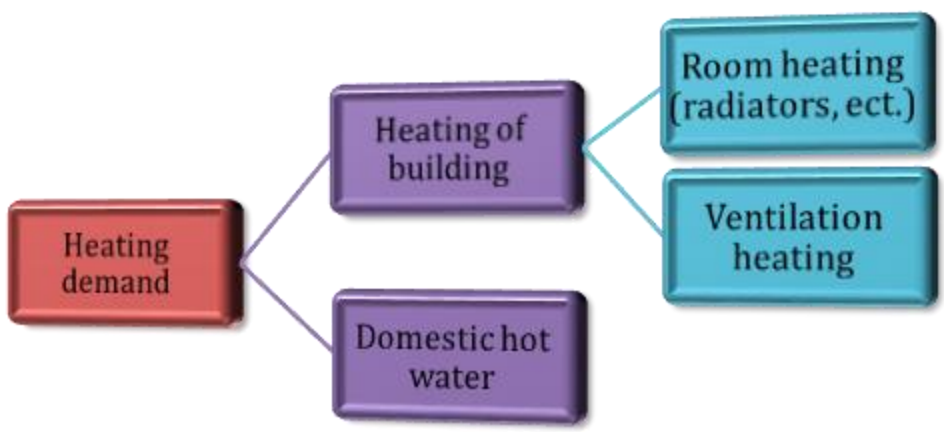


Figure 1: Heating demand

Figure 2 shows an example of the distribution of the energy needs in a family house with various building standards. Passive and low-energy buildings are tighter and better insulated than *TEK10*- and older standards and use efficient heat recovery systems as well. This implies that the share of temperature-dependent heat demand is stretched reduced[3]. Low Energy Buildings of Class 1 require that the heat exchanger should have an annual average temperature-efficiency of at least 70% and passive houses are required to have at least 80%.[5, 6] Approximately 50% of the total energy demand is heating of rooms or ventilation air with *TEK10* standard and about 25% with the passive house standard.

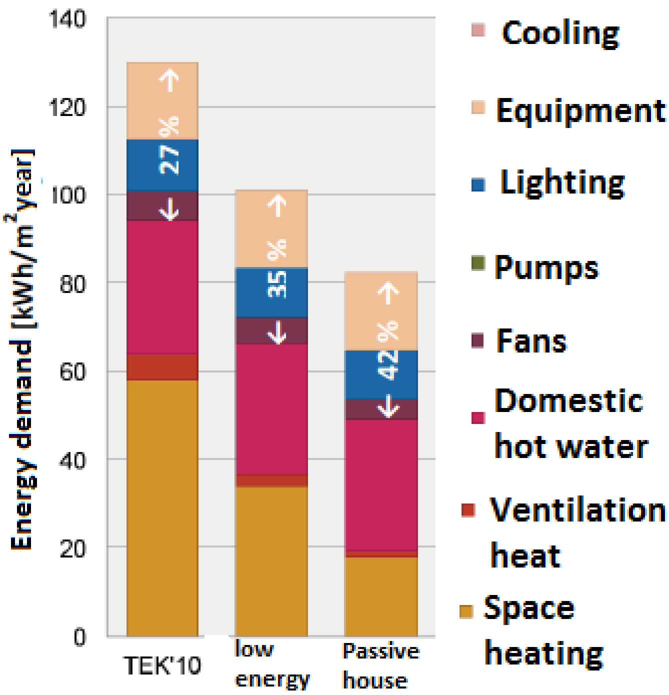


**Figure 2 Example of energy demand for a residential at different building standards. Electricity specific demand is given in % of total energy demand (the figure is translated from Norwegian)[3]**

For commercial buildings, the tendency is the same. However, there is a greater proportion electricity specific energy demand, which is due more internal load in terms of lighting and technical equipment.[3]

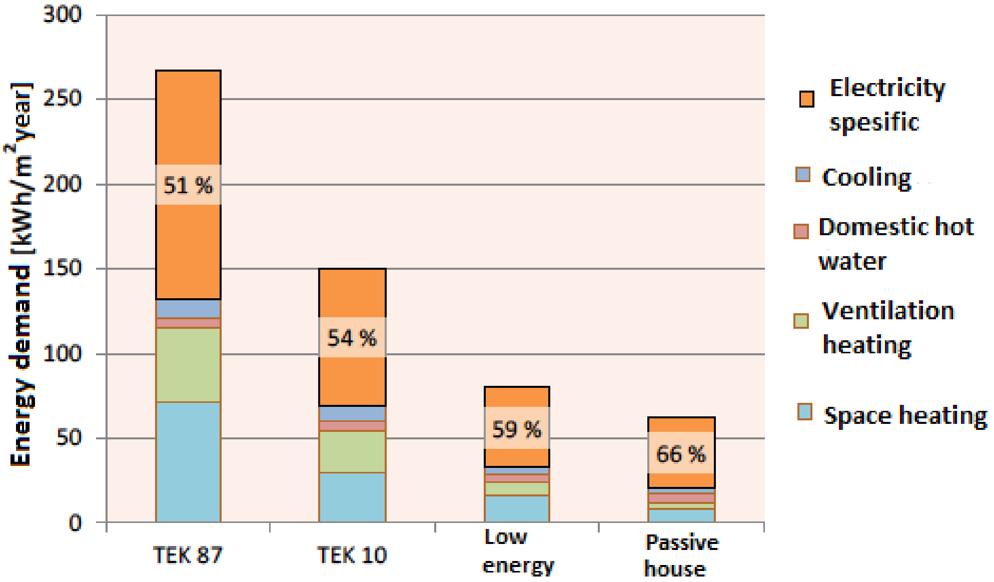


Figure 3 Example of energy demand in a office building at different building standards. Electricity specific demand is given in % of total energy demand (the figure is translated from Norwegian) [3]

Domestic hot water constitute only a small share of the total heat demand in office buildings.

Figure 4 shows a simulation of the absolute and relative need for heat in a residential in Oslo with different building standards. With improved building standards, the energy demand for heating of domestic hot water become greater relative to the total heat demand of the building. In residential buildings, apartment buildings and nursing homes with passive house standard, heating of the domestic hot water is typically 40-70% of the total energy demand for heating. Non-residential buildings use a smaller proportion of the energy for heating of domestic hot water. The three different heating purposes, room heating, ventilation heating and heating of domestic hot water represent different temperature requirements, which affects the heating facility framework[3].

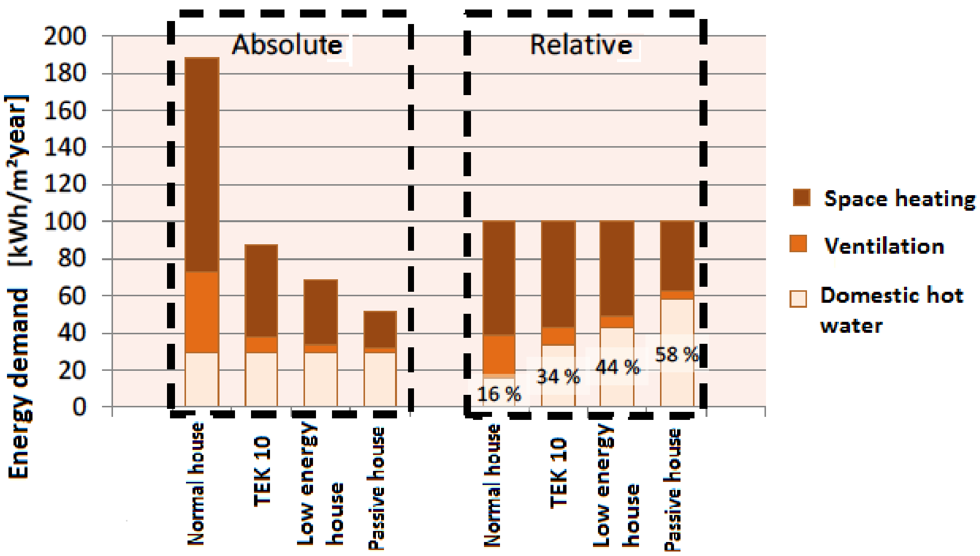


Figure 4 Example of simulated absolute and relative heating demand in a residential building in Oslo with different building standards (the figure is translated from Norwegian) [3]

### 3.1.3 Power duration curve

Power duration curves shows the power demand for heat over the year in descending order. The power profile in the diagrams varies a lot according to type of building, building standards and climate zones. It provides the basis for the choice of heating system[3]. Annual heat demand, often given in kWh, is the area under the power duration curve.

Power duration curves may be prepared with gross heating power[2] with or without regard to the power requirement for heating of domestic hot water. It is common to calculate the power demand of domestic hot water as constant throughout the year, which is quite realistic using an accumulation tank. Power duration curves may also be prepared with net heating power[3]. Internal heat loads from equipment and people, as well as solar radiation depends on the use and location of the building. Figure 5 shows examples of different power duration curves.

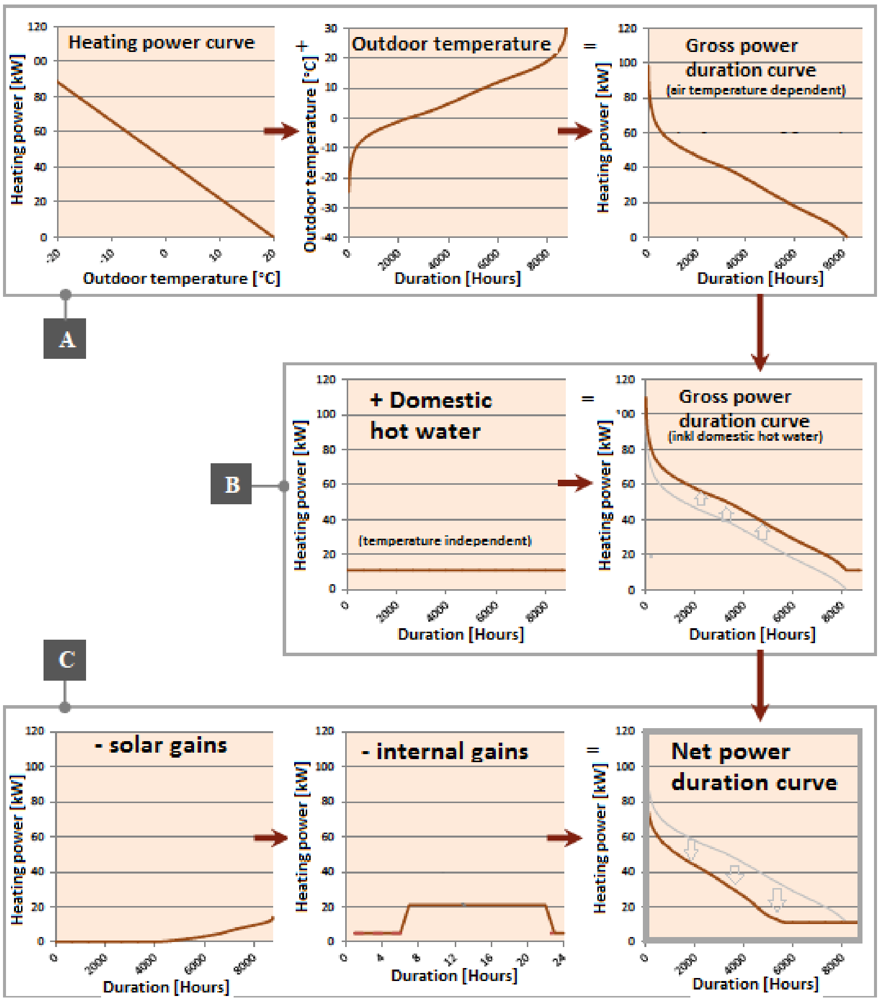


Figure 5 Power duration curve - structure of the diagram for gross and net power including heating of domestic hot water. (the figure is translated from Norwegian) [3]

[2] Gross heating power: heating power demand of the building without taking into account internal heat gains or heat gains from solar radiation
[3] Net heating power: gross heating demand minus internal heat gains and heat gains from solar radiation

Dependent on type of building, power and energy demand varies. In office buildings, the energy demand is reduced more than the power demand by improving building standard. For example *TEK10* to passive house standard. In residential buildings, the energy and power demand correlate largely. This is due to the heating demand composition. Residential buildings have a large energy demand related to heating of domestic hot water, where power demand is relatively small.[3]

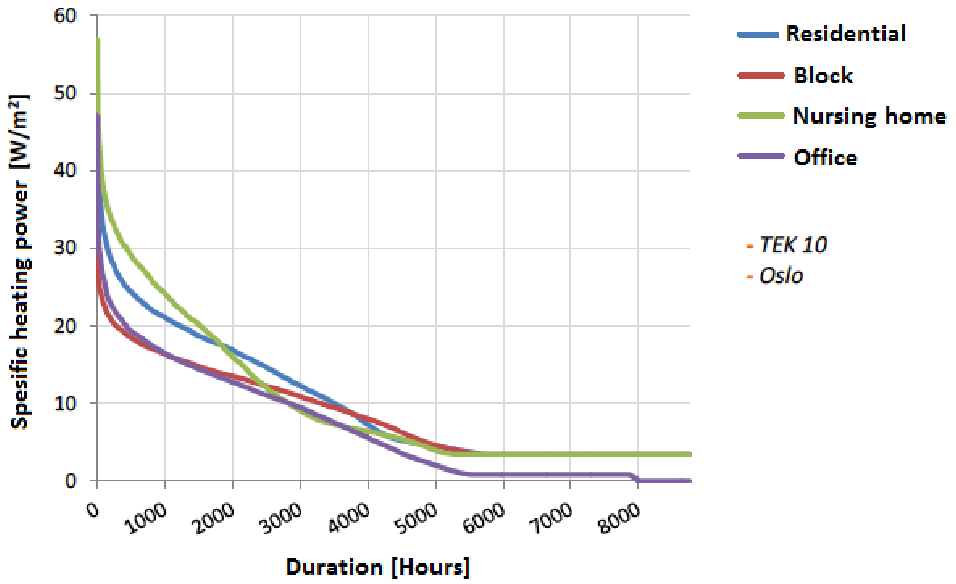


**Figure 6 Simulated net power duration curve for different types of buildings with *TEK 10* standard in Oslo. (the figure is translated from Norwegian) [3]**

The example in Figure 7 shows the simulated values of office buildings in Oslo climate. The heat demand is reduced by about 80% from normal house to passive house, while the design gross power demand is reduced by about 60%. A situation where the energy demand decreases more than the power demand reduces the useful life [4] of the heating system. It becomes an even bigger challenge to make heat distribution system based on renewable heat economically competitive compared to for example electrical heating[3].

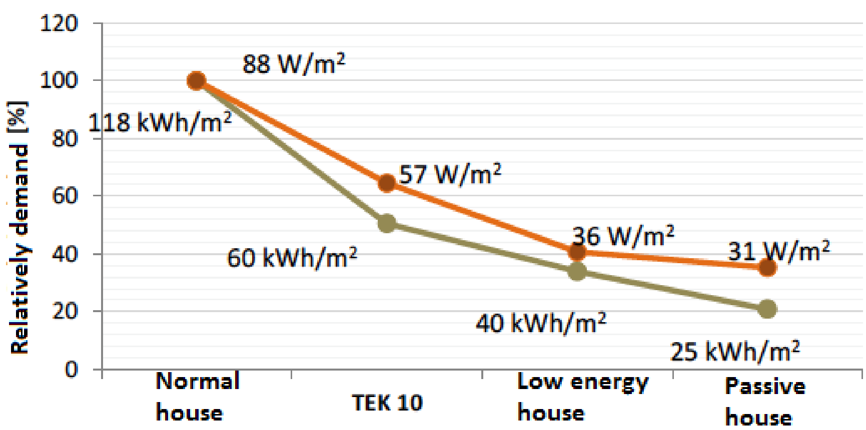


**Figure 7 Example of annual specific heating demand and design specific gross power demand for an office with varying building standards located in Oslo (the figure is translated from Norwegian) [3]**

[4] The useful life is the amount of heat divided by the thermal power. Also called equivalent operating time and is used as an indication of how well the heating system is utilized

### 3.1.4 Heating season

Based on the gross power demand, the heating season will occur in the same period of time for all kind of building standards and is only dependent on the temperature difference between inside and outside. By looking at net power demand, which is taking into account internal gains and heat gains from the sun, the heating season is reduced by improvements in the building standard.

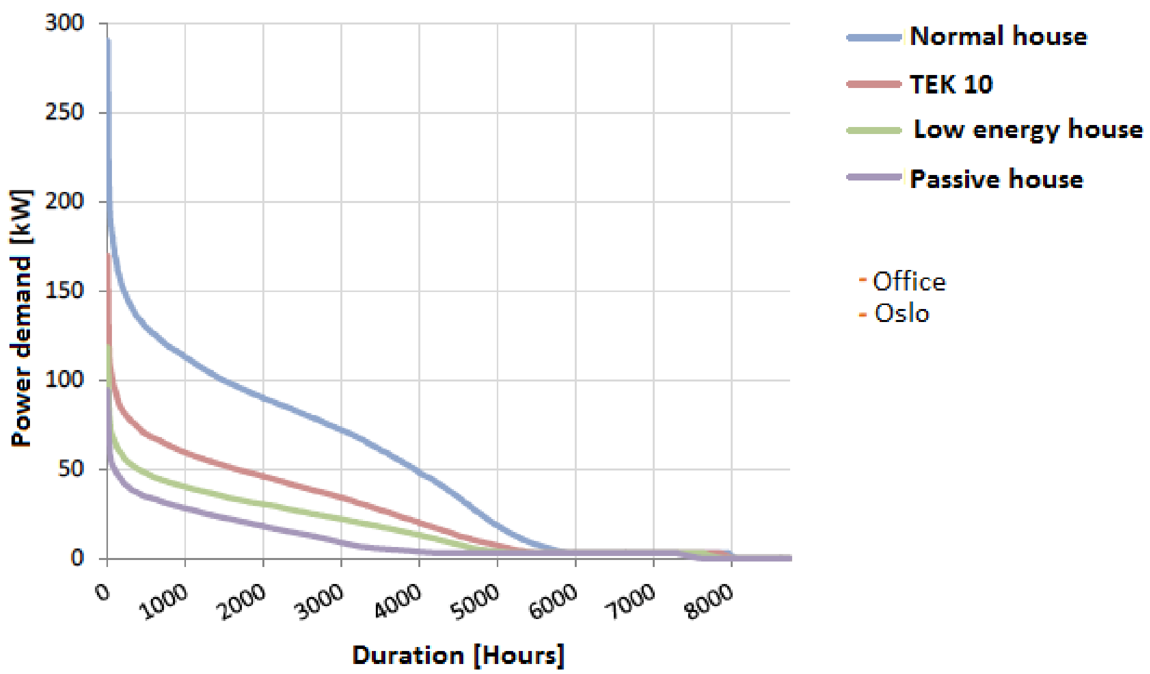


Figure 8 Example of reduced heating season - net power duration curve for an office in Oslo using different building standards (the figure is translated from Norwegian) [3]

### 3.1.5 Net and gross power demand for different building standards

The difference between net power demand and gross power demand depends on geographical location, building type and not least the building standard. There are also variations from building to building within the same building standard. Table 2 illustrates an example of the difference between net and gross power demand. Notice that the offices and nursing homes have much lower ratios than residential and blocks.

Table 2 Estimated net design power demand divided by gross design power demand (the table is translated from Norwegian) [3]

| | | Normal house | TEK 10 | Low energy house | Passive house |
|---|---|---|---|---|---|
| Residential | Oslo | 0,98 | 0,96 | 0,95 | 0,93 |
| | Bergen | 0,98 | 0,95 | 0,94 | 0,92 |
| | Trondheim | 0,98 | 0,96 | 0,95 | 0,93 |
| | Tromsø | 0,98 | 0,95 | 0,93 | 0,92 |
| | Røros | 0,99 | 0,97 | 0,96 | 0,95 |
| Block | Oslo | 0,97 | 0,95 | 0,94 | 0,92 |
| | Bergen | 0,96 | 0,94 | 0,92 | 0,90 |
| | Trondheim | 0,97 | 0,95 | 0,94 | 0,92 |
| | Tromsø | 0,96 | 0,94 | 0,93 | 0,90 |
| | Røros | 0,98 | 0,97 | 0,96 | 0,94 |
| Office | Oslo | 0,74 | 0,86 | 0,71 | 0,67 |
| | Bergen | 0,65 | 0,87 | 0,58 | 0,65 |
| | Trondheim | 0,73 | 0,86 | 0,62 | 0,64 |
| | Tromsø | 0,67 | 0,87 | 0,69 | 0,65 |
| | Røros | 0,82 | 0,72 | 0,73 | 0,72 |
| Nursing home | Oslo | 0,86 | 0,78 | 0,73 | 0,68 |
| | Bergen | 0,80 | 0,70 | 0,70 | 0,64 |
| | Trondheim | 0,85 | 0,77 | 0,74 | 0,69 |
| | Tromsø | 0,82 | 0,72 | 0,69 | 0,63 |
| | Røros | 0,90 | 0,85 | 0,82 | 0,79 |
| Color code | | 1 – 0,9 | 0,9 – 0,8 | 0,8 – 0,7 | 0,7 > |

### 3.1.6 Dimensioning of heating systems

Traditionally, thermal facilities in Norwegian buildings are designed by *gross* power demand at design outdoor temperature, DUT[5]. In other words, power demand when the building is not in use, but still should be heated to the selected indoor temperature [3]. Both for economic reasons and because of regulatory requirements or standards, both a base load and a peak load often cover the power and heat demand.

" Base load is the power up to a certain level that is needed to cover most of the annual heating demand in the most profitable way. The distribution between base load and peak load is in practice dependent on territorial climate, the building's power demand over the year and the heating system's properties. Typically, the base load constitute 70 - 90% of the building's heating demand over the year."[2]

70-90% of the energy demand corresponds to a far smaller proportion of the power demand due to a short annual period with high power demand. This phenomenon is easy to see in the power duration curve in Figure 9. For example, heat pumps have traditionally been sized to provide 40-60% of the design value of the net power demand after it has been performed a cost optimization of the heating plant [3]. The graph below shows the principle of oil boiler as peak load and air to air heat pump that base load. The graph is somewhat special as it used an air to air heat pump that gets poorer efficiency when the outdoor temperature drops. At the coldest period, for which most effect is needed, it would not be profitable to use the heat pump and is will be switched off. Other base loads would typically cover the entire area under "$P_{vp,dm}$".

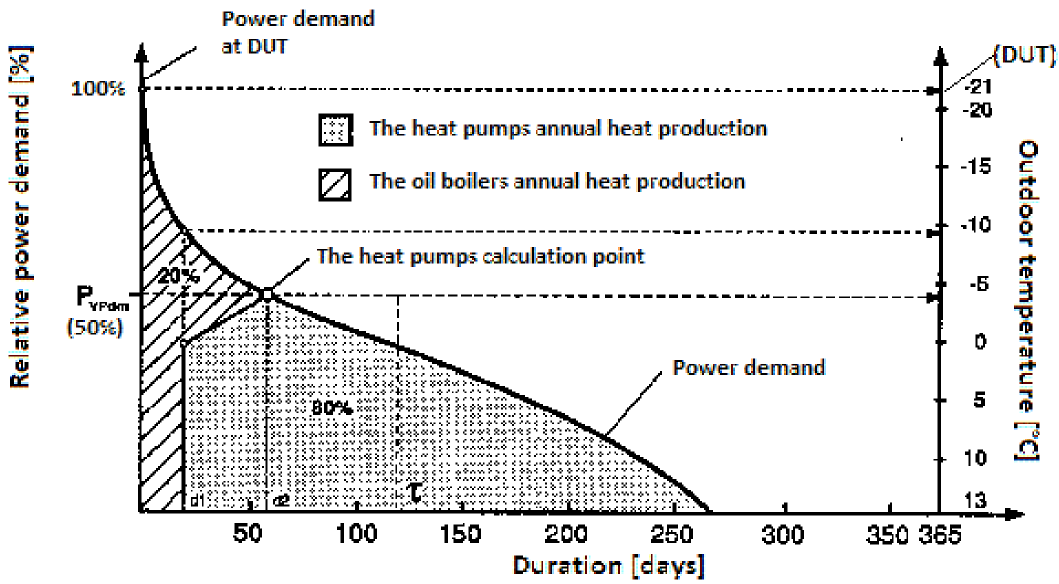


**Figure 9 Power duration curve - example of distribution between base load and peak load, air-air heat pump and an oil boiler (the figure is translated from Norwegian) [7]**

[5] DUT – Design outdoor temperature

The base load usually have high investment cost, but is inexpensive to operate. The important thing is to minimize the total costs. Cheap resources that makes the base load inexpensive to operate are often more environmental friendly then the peak load. The peak load is usually covered by a heating system that have low investment costs, but are more expensive to operate. The peak load covers only a small part of the energy demand, which usually is less environment friendly than the base load. The balance between base load and peak load is cost-optimized with respect to the current requirements for energy supplies.[8]

### 3.1.7 Part load operation

The efficiency at part load operation depends on what kind of heating system one are using and how it is adapted to part load operation. Many renewable systems, whether incinerators or heat pumps, have poorer efficiency at part load operation, depending on control possibilities and design.[9, 10]

### 3.1.8 Night setback of the indoor temperature

Night setback of the indoor temperature has been a central issue within energy conservation initiatives in existing buildings. Although it saves energy, it leads to increased power demand to reheat the building. Increased power demand leads too increased investment costs, while energy conservation reduces the operating costs. Night setback may be unfavorable using for example heat pump, which has a high specific investment cost (NOK/kW). Heat pumps often have poor efficiency while operating outside optimal operation point as well. [3]

### 3.1.9 Outdoor climate

The framework of conditions for thermal plants depends on climatic conditions, especially the heating season length and design outdoor temperature [3].The shape of the power duration curve provides a basis for designing the heating system.

Design gross power demand affects the investment cost of peak load, while the shape of the net power duration curve determines the distribution between base load and peak load. [3]

Figure 10 shows an example of a simulated net power-duration curve for different locations in Norway for a normal house. Normal house is a typical existing house with a relatively old building standard. The different curves show distinct climatic differences where both heating season and the design power demand varies.

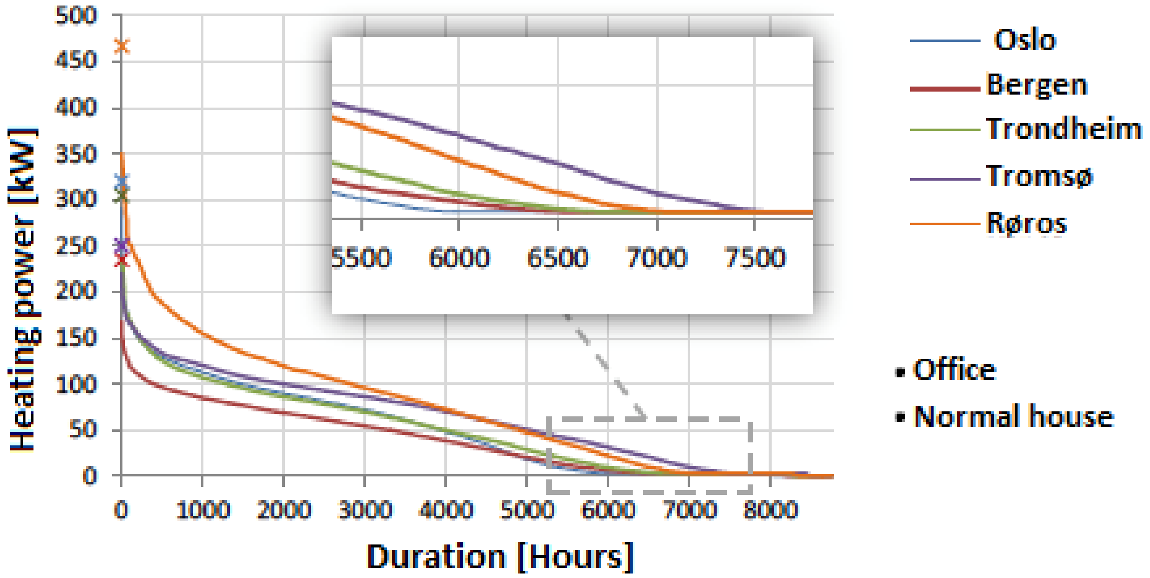


Figure 10 Net power duration curves for a normal house office at different climatic zones (the figure is translated from Norwegian) [3]

## 3.2 Indoor climate

### 3.2.1 Thermal indoor climate

People experience thermal comfort differently. The temperature of choice varies mainly with:

- the metabolism (heat output) of every single person
- age
- gender
- health condition
- medication
- acclimatization
- clothing
- activity level

[11]

Not everyone can be satisfied in the same thermal conditions. In the Norwegian standard for indoor climate, the optimal indoor temperature is based on the thermal comfort index PMV[6]-PPD[7] with presumed clothing and activity level.[12] Humidity is also an element affecting thermal comfort, but has the greatest impact at high temperature. High humidity leads to less evaporation of sweat, which would reduce the cooling effect evaporation of sweat provides.[13]

**Operative temperature**

The operative temperature, sensible temperature, is the average of the air temperature and the radiation temperature. [13]

$$T_{oprative} = \frac{T_{air} + T_{radiation}}{2}$$

**Local thermal discomfort**

Even if desired operating temperature is obtained on average, there may be local thermal discomfort. Local thermal discomfort can result from, among other things:

- Draft
- Asymmetric radiation temperature
- Vertical air temperature difference
- Too hot or cold floor temperature

[12]

**Optimal operative temperature – maximal performance**

It is conducted several studies examining the relationship between temperature and performance. The results vary, but there is an apparent trend [11]. Figure 11 shows a representative representation of the relationship between temperature and performance.[14]

[6] Predicted mean vote
[7] Predicted percentage dissatisfied

As the figure shows, the highest performance is obtained at 21,5°C ± 1°C. The performance is reduced for both higher and lower temperatures. The comfort interval is wider and extends from 20°C to 27°C.

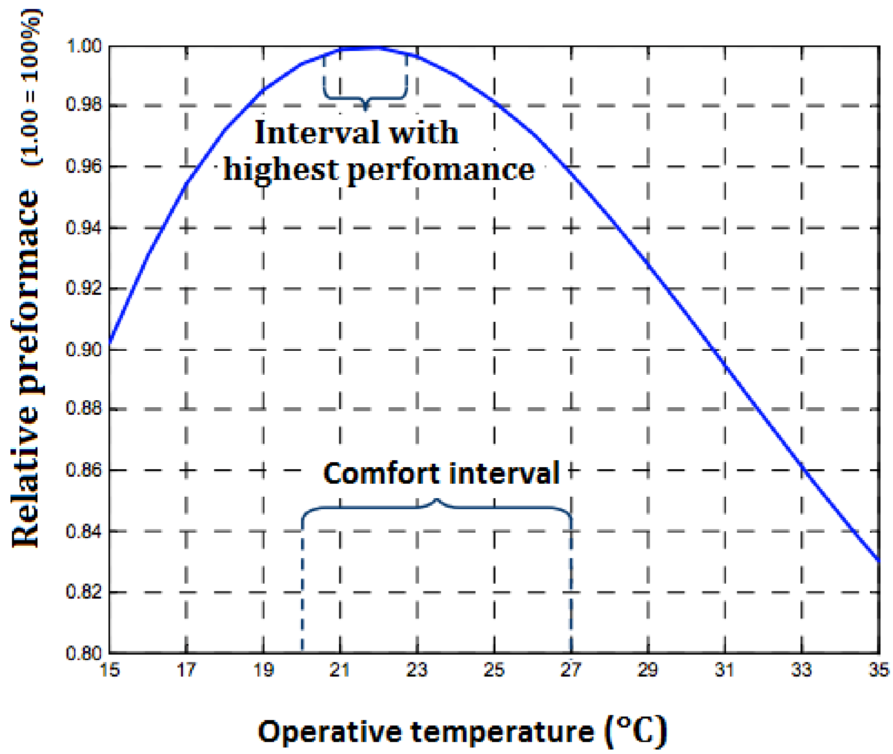


Figure 11 Relative performance and comfort as a function of operative temperature (the figure is translated from Norwegian) [14]

### 3.2.2 Air quality

Important health impacts of indoor pollution is:

- Allergies
- Irritations of the mucous membranes or skin
- Poisoning
- Infection and inflammation
- Cancer
- Affects human reproduction

[11]

Poor indoor air quality can lead to what is called "Sick building syndrome symptoms'. Typical symptoms include headache, difficulties breathing, feeling dizzy, irritation of eyes and mucous membranes. It is unclear what the cause is, but the magnitude has a correlation with:

- Humidity
- Ventilation airflow
- Organic material and particles in the air

[13]

Indoor air quality is influenced by emissions from humans and their activities, from building and furnishing and from the HVAC system [12]. Indoor air quality also varies with outdoor air quality. Air purification and location of air intakes and filters in the ventilation system influences indoor air quality. The same applies for the amount of ventilation air[13]

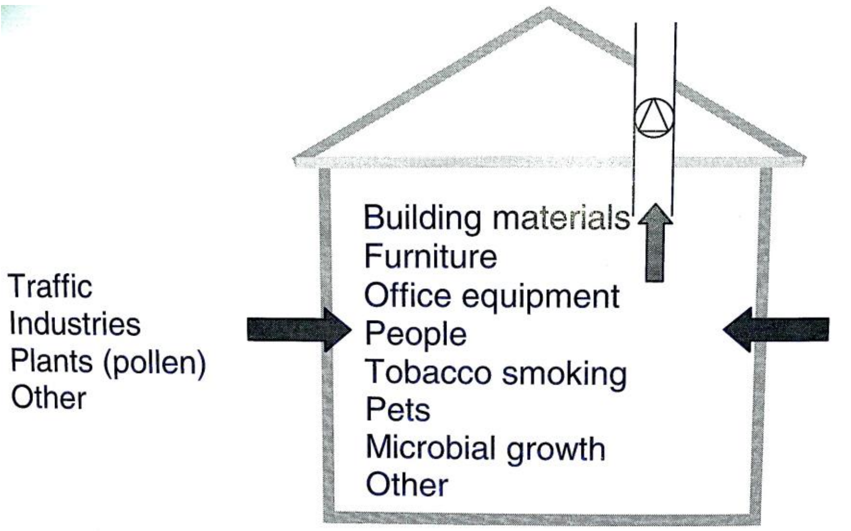


**Figure 12 Examples of indoor air pollution[13]**

*" Required ventilation airflow is based on health and comfort criteria. In most cases, the comfort criteria are sufficient enough to meet the requirements stated with regard to health "* [12]

Comfort is related to perceived air quality, i.e. odor and irritation.

Air quality has a correlation with productivity. Figure 13 is based on experiments and shows the relationship between perceived air quality or eventually PPD and performance at office work. In addition, the experiments have shown that a doubling of ventilation airflow results in 1.8% increase in performance. [13]

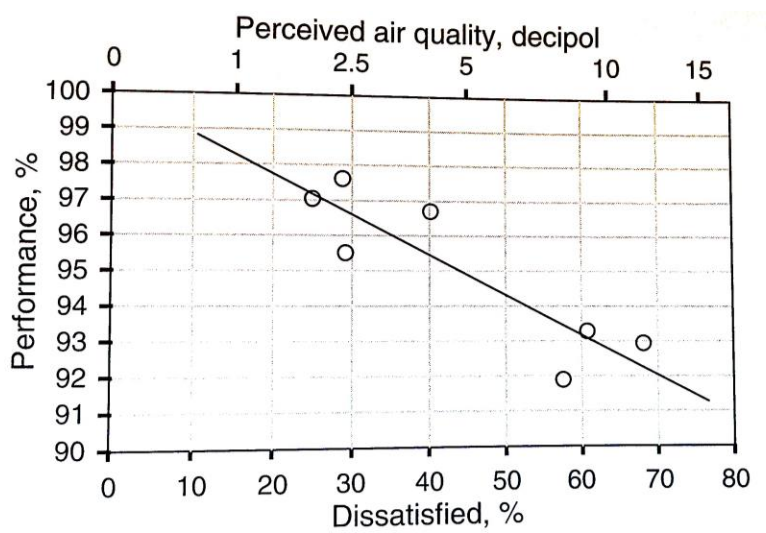


**Figure 13 Performance at office work as a function of perceived air quality [13]**

## 3.3 Norwegian and American standard for calculation of power demand, *NS-EN 12831:2003* and *ASHRAE 2013*

This chapter describes the methods for calculation of the power demand using the two standards, *NS-EN 12831:2003* and *ASHRAE 2013*. The tables are meant as a basis for comparison, and the methods are further described in the text.

According to "CEN/CENELEC International Regulations" the following countries are bound to implement the European standard, *NS-EN 12831:2003*: Belgium, Denmark, Finland, France, Greece, Iceland, Ireland, Italy, Luxembourg, Malta, Netherlands, Norway, Portugal, Slovakia, Spain, United Kingdom, Switzerland, Sweden, Czech Republic, Germany, Hungary and Austria. The standard specifies methods for the calculation of the design heat loss for basic cases under design conditions, which gives the design power demand.[15] *ASHRAE 2013* is an equivalent standard used in the United States.

### 3.3.1 General method of calculation

The six steps given below shows the general procedure for calculating the design power demand at room or building level, depending on the placement of the system boundary. When calculating the power demand at room level, heat exchange with adjacent rooms must be taken into account as well as heat loss to the outside. At building-level heat exchange between adjacent heated room is not take into account.[15-17]

1. Select the outdoor design conditions:
   - Temperature, humidity, wind direction and wind speed.
2. Select the indoor design conditions:
   - Select the lowest temperature within the decided temperature range
3. Estimate the temperature in unheated rooms
4. Select the transmission coefficients and find the heat loss through walls, ceilings, floors, windows, doors and ground.
5. Calculate the heat loss due. infiltration and ventilation
6. Add up losses from transmission, infiltration and ventilation.

*The principle of the calculation method that applies to both NS-EN 12831:2003 and ASHRAE 2013:*

Both calculate the maximum heat loss from each room to determine how much heating power is necessary to install in each room. The maximum simultaneous heat loss in the building are calculated and used to determine the power required by the building's heat exchanger or other heat production.

The first step is to calculate the design heat losses. The results are then used to determine the design power demand. When calculating the design power demand it is assumed steady state, in which temperatures and other properties are constant.

Power calculations are conservative methods which:

- Ignores heat gains from the sun
- Ignores internal heat gains
- Ignores heat storage in the building
  (steady state, no inertia in the system)

The result of these assumptions gives, in most cases, a built-in safety margin since not all these assumptions usually will occur simultaneously.

It have to be calculated heat loss due to:

- Heat conduction through the surrounding surfaces
- Infiltration
- Ventilation

In addition, one must take into account the heating-up capacity, an additional power demand, which is necessary when using periodic heating. This is due to the necessary raise in temperature within a reasonable time. Table 3 shows reheat factors given in *NS-EN 12831:2003.* They depends on temperature rise, building mass and the time of reheating. [15-17] Heating up capacity is found by multiplying the reheat factor with floor area of the investigated building or zone.

**Table 3 Reheat factor, $f_{RH}$, for non-residential buildings, night setback for maximum 12 hours[15]**

| Reheat time hours | $f_{RH}$ W/m² | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | Assumed internal temperature drop during setback [a] | | | | | | | | |
| | 2 K | | | 3 K | | | 4 K | | |
| | building mass | | | building mass | | | building mass | | |
| | low | medium | high | low | medium | high | low | medium | high |
| 1 | 18 | 23 | 25 | 27 | 30 | 27 | 36 | 27 | 31 |
| 2 | 9 | 16 | 22 | 18 | 20 | 23 | 22 | 24 | 25 |
| 3 | 6 | 13 | 18 | 11 | 16 | 18 | 18 | 18 | 18 |
| 4 | 4 | 11 | 16 | 6 | 13 | 16 | 11 | 16 | 16 |

[a] In well insulated and airtight buildings, an assumed internal temperature drop during set back of more than 2 to 3 K is not very likely. It will depend on the climate conditions and the thermal mass of the building.

### 3.3.2 Design temperature

Design outdoor temperature is used in all heat loss calculations and is therefore very interesting to examine. In Norwegian standard, *NS-EN ISO 15927-5,* there are two ways of calculating the design outdoor temperature. One of these is similar to the method in *ASHRAE 2013*, which is the 99% temperature value method. This method implies that there is warmer than the design outdoor temperature 99% of the time in the coldest month of the year. Thus, only colder than the design outdoor temperature 1% of the time in the coldest month. *ASHRAE 2013* state that 99.6% method also is an opportunity if it is desirable to be even more certain that the power demand will be large enough. The Norwegian standard states: "The coldest month is the month with the lowest 20-years average monthly mean temperature".[17-19]

The second method described in the Norwegian standard uses n-day mean air temperature. Number of days can be one, two, three or four, and shall occur on average every year (i.e. 20 times in 20 years).[18]

**Table 4 Design temperature**

| | ***NS-EN 12831:2003* [15]** | ***ASHRAE 2013*** |
|---|---|---|
| Outdoor temperature | Air temperature and operative temperature is equal (applies for well insulated houses) | Air temperature and operative temperature is equal [16, 17] |
| | Several methods. One of them is 99% temperature value method. Another is n-day mean air temperature method[18] | Uses 99% (99.6%) temperature values method [19] |
| Indoor temperature | The indoor temperature is selected in the lowest level within the temperature boundaries for good indoor air quality | The indoor temperature is selected in the lowest level within the temperature boundaries for good indoor air quality [16, 17] |

### 3.3.3 Heat loss due to transmission

*The main principles*

The main principles for calculation of heat loss due to transmission is equal in both standards, although the formulas are written differently. Both standards are based on the product of U-values, area and temperature difference across the actual building component (Table 5)

*Heat loss directly to the outside*

Heat loss is measured in the same way for the two standards, although the formulas in *NS-EN 12831:2003* looks more advanced.

The U-value which is used in *ASHRAE 2013* includes the effect of the thermal bridge[20]. $e_k$ and $e_l$, which is used in *NS-EN 12831:2003* is correction factors that take into account climate changes that affect and weaken the U-value (assuming it is not already taken into account in the determination of U-values). Climate Influences can be moisture absorption in building components, wind speeds and temperature effects [15].

*Heat loss through unheated spaces*

Heat loss through unheated areas are calculated in the same way as the heat loss to the outside, but the temperature of the unheated room is to be used instead of the ambient temperature (the same will be the case for partial heated room).

*ASHRAE 2013:*

If the temperature in the unheated room, $t_b$, is not given it can be determined in following ways using *ASHRAE 2013*[16]:

1. $t_b$ can be assumed equal to the outdoor temperature
    i. It may be a good assumption if the unheated room is heavily ventilated
2. $t_b$ can be assumed to be the intermediate temperature between the unheated and heated rooms
    i. It could be a good assumption if the unheated room is isolated
3. $t_b$ can be calculated in detail:

$$t_b = \frac{C_s * Q * t_0 + \Sigma A_x * U_x * t_x + q}{C_s * Q + \Sigma A_x * U_x}$$

$t_b$ = Unheated buffer room temperature [°C]

$C_s$= The sensible heat coefficient $\left[\frac{W}{l*s*K}\right]$ (1,23 at sea level)

Q = Unheated buffer room infiltration/ventilation [l/s]

$t_0$ = Outdoor temperature [°C]

$A_x$= Area of surface x in the unheated room

$U_x$= U-factor of surface x in the unheated room

$t_x$ = temperature outside surface x, typical outdoor temperature at the outdoor wall (eventually the ground temperature)

q= additional heat contribution to the room (for example, from the distribution system)

*NS-EN 12831:2003:*
The temperature reduction factor, $b_u$ takes into consideration the temperature inside the unheated room [15].

1. If the temperature in the unheated room, $\theta u$, is given or can be calculated:
   - $b_u = \frac{\theta i - \theta u}{\theta i - \theta e}$

   This means that the formula become equivalent to the one used in *ASHRAE 2013*.

2. If $\theta u$ is not known one use that:
   - $b_u = \frac{H_{ue}}{H_{iu} - H_{ue}}$

   $H_{iu}$ and $H_{ue}$ are respectively the heat transfer coefficient from the heated room (i) to the unheated room (u) and the heat transfer coefficient from the unheated room (u) to the outside (e). The heat transfer coefficients are measured in [W/K] and take in to account:
   1. Heat transmission
   2. Heat loss cause of ventilation (air streams)
   3. One can use default values based on experience

*Heat loss to the ground*
*NS-EN 12831:2003* refers to *NS-EN ISO 13370* where there is a detailed method and a simplified method. The simplified method is given in Table 5 and does not include thermal bridges. $U_{equiv,k}$ is the equivalent U-value for a building part, and depends on the U-value of the building component, the depth below ground level and thermal properties of the ground. It is given tables in *NS-EN 12831:2003* where one can find these $U_{equiv,k}$ values. When calculating heat loss due to floors at or below ground level, the area and the exposed circumference(perimeter) is taken into account by $B' = \frac{A_g}{0,5 * P}$.[15]

*ASHRAE 2013* calculates the factor, HF, for surfaces below ground level as follows: $HF = U_{avg}(t_{in} - t_{gr})$. It is taken into account that the temperature of the ground is not the same as the outdoor temperature. For walls below ground level, $U_{avg}$ take into account the thermal properties of the ground, the wall's original resistance and depth below ground level. For basement floor, it is taken into account the shortest width of the basement, similar to the effect of factor B in *NS-EN 12831:2003*. For floors that stand on the ground there is given a simplified method, which takes into account the perimeter. The building component then has a coefficient per meter perimeter listed in a table in the standard. [17]

There are many ways of calculating heat loss through the ground. In *NS-EN ISO 13370,* it is given several detailed methods, while *ASHRAE 2013* refers to more detailed methods in "Bahnfleth and Pedersen" (1990), which shows an *area to circumference effect* or "Beausoleil-Morrison and Mitalas"(1997)," CAN/CSA Standard F280"," HRAI" (1996) and "Krarti and Choi" (1996).

Table 5 Heat loss

| HEAT LOSS | *NS-EN 12831:2003* [15] | *ASHRAE 2013* |
|---|---|---|
| Main principle | $\Phi = H * \Delta\ \theta$ [W]<br>• $\Delta\theta = \theta_{\dim in} - \theta_{\dim out}$[K] (difference between indoor and outdoor temperature)<br>• H = Heat loss coefficient $\left[\frac{W}{K}\right]$ | q=A*HF [W]<br>• HF=UΔt $\left[\frac{W}{m^2}\right]$<br>• U=U-verdi $\left[\frac{W}{m^2*K}\right]$<br>• A=Area [$m^2$]<br>• Δt= difference between indoor and outdoor temperature<br>[16, 17] |
| Directly to the outside | H=$\sum_k(A_k * U_k * e_k)$+$\sum_l(\psi_l * l_l * e_l)$ $\left[\frac{W}{K}\right]$<br>• $A_k$=Area of wall k<br>• $U_k$= U-value of wall k<br>• Thermal bridges is considered by $\psi_l * l_l \left[\frac{W}{m^2*K}\right]$, linear heat transfer coefficient multiplied by length<br>• $e_k$ and $e_l$ → Correction factors because of climate impacts | U-value $\left[\frac{W}{m^2*K}\right]$ includes the effect of thermal bridges.[20] |
| Through unheated rooms | H=$\sum_k(A_k * U_k * b_u)$+$\sum_l(\psi_l * l_l * b_u)$ $\left[\frac{W}{K}\right]$<br>• $b_u$ = temperature-reduction factor taking into account the difference between the temperature in the unheated room and the design outdoor temperature<br>→ Can be determined in 3 ways | HF=U($t_i$-$t_b$)<br>• $t_i$=indoor temperature[K]<br>• $t_b$ = temperature in the unheated room [K]<br>→ can be determined in 3 ways<br><br>[16, 17] |
| Through the ground (simplified methods) | H=$f_{g1} * f_{g2} * \sum_k(A_k * U_{equiv,k}) * G_w$ $\left[\frac{W}{K}\right]$<br>• $f_{g1}$= correction factor which takes into account variations in the outside temperature, are determined nationally.<br>• $f_{g2}$ = temperature reduction factor taking into account the difference between the annual average and the design outdoor temperature<br>  ○ $f_{g2} = \frac{\theta i - \theta m,e}{\theta i - \theta e}$<br>• $U_{equiv,k}$= equivalent heat transfer coefficient (U-value)<br>• $A_k$=Area<br>• $G_w$= Correction factor that takes into consideration the effect from the groundwater (must be used if the distance is less than 1 m) | Below ground level[17]<br>HF=$U_{avg}$($t_{in}$-$t_{gr}$)<br>• $t_{gr} = t_{m,gr} - A$<br>• $t_{m,gr}$ =average ground temperature<br>• A= the temperature amplitude on the surface of the ground.<br>• Using average U-values:<br>  ○ for the wall: $U_{avg,bw}$<br>  ○ for the floor: $U_{avg,bf}$<br><br>On ground level[17]<br>q=p*HF [W]<br>• where HF=$F_p\Delta t$ $\left[\frac{W}{m}\right]$<br>• p = Perimeter<br>• $F_p$= Heat transfer at the perimeter$\left[\frac{W}{m*K}\right]$ |

#### 3.3.4 Heat loss because of infiltration

Infiltration is driven by pressure differences across the building elements and occurs because a building is not completely sealed. The pressure differences is a result of the chimney effect, wind pressure and unbalanced ventilation.[21]

The chimney effect is an effect that creates a pressure difference due to differences in temperature inside and outside and the fact that hot air rises. High and open buildings thereby gets huge chimney effect. Lower temperatures outside than inside will give under pressure in the lower part of a building or a room and provides over pressure in the upper part. This is the principle of natural ventilation.[21]

Mechanical exhaust from the kitchen fan or unbalanced ventilation, will give under pressure in the building. Although balanced ventilation is used, there may be pressure differences in some rooms because of the location of the intake and exhaust. Leakage from channels will also affect the pressure differences.[21]

Residential and commercial buildings have a common way of calculate infiltration according to *NS-EN 12831:2003*. *ASHRAE 2013* calculates the volume flow due to infiltration for residential somewhat different from for commercial buildings. [15, 22]

The principle of the methods in both standards is to multiply the volume flow by the density, specific heat and temperature difference to find the sensible heat loss. *ASHRAE 2013* also shows an equation for latent heat, which is used if it is necessary to add moisture to keep the relative humidity (RH) over a certain level.[15-17] It requires a lot of energy to evaporate water. It is recommended to keep the relative humidity relatively low to reduce building-related illness, also called "Sick building syndrome", which among others may come from condensation, providing fungus and mold damages. Studies show that negative effects of dry air, such as irritation of the eyes and skin, does not occur until the relative humidity is below 15%.[13]

*How to calculate the airflows*

In *NS-EN 12831:2003* the volume flow cause of infiltration, $\mathbf{V_{inf,i}}$, is calculated based on the room volume multiplied by the air change (leakage number) at a pressure difference of 50 Pa between inside and outside. The leakage number, $n_{50}$, varies according to the tightness of the building, sealing of windows and doors is especially important.[15] The requirement for passive houses is that $n_{50}$ must be less than 0,6$h^{-1}$. For commercial buildings of low energy standard the requirement is 1,5$h^{-1}$.

Residential low energy buildings have two categories. Class 1 requires a leakage number lower than1 $h^{-1}$, while for class 2 the requirement is 3 $h^{-1}$.[5, 6] The leakage number and correction factors for shielding and height is then multiplied (Table 6). Wind speed increases with height.

At room level, it is taken into account that all infiltration can occur on one side of the building and nothing on the other. Designing power demand, one must consider the worst possible situation. That is why it is multiplied by a factor of 2 at room level, but not at building level. [15]

Mechanical ventilation is a result of unbalanced ventilation and/or exhaust air directly. *NS-EN 12831:2003* calculate mechanical infiltration, $\mathbf{V_{mech,\,inf}}$, independent of other infiltration. By removing air mechanically, there will be an additional negative pressure sucking in outside air. The volume flow is distributed by the permeability of each room compared to the building's permeability and not multiplied by any factor at room level.[15]

The volume flow in *ASHRAE 2013* is given by the equation; $Q_i = A_L * IDF \ \left[\frac{l}{s}\right]$.
$A_L$ is the effective leakage area of the building at reference pressure difference of 4Pa calculated theoretically, $C_d$=1.

$$A_L = 10\,000 * Q_r * \frac{\sqrt{\rho / 2\Delta P_r}}{C_D}$$

$\Delta P_r$=reference pressure difference
$Q_r$= assumed airflow at $\Delta P_r$, often given by a pressure test
$C_D$= emission coefficient (theoretical value: $C_D$=1)

[21]

The effective leakage area of the building, $A_L$, can also be found in a simplified way:

$$A_L = A_{es} * A_{ul}$$

$A_{es}$ =surface area of the building above ground level
$A_{ul}$ =leakage area [cm$^2$/m$^2$]

[16]

The effective leakage area, $A_L$, is often converted from the estimated air flow, $Q_r$, which can be found by pressure measurement with a given reference pressure difference, $\Delta P_r$[21]. The airflow due to infiltration is found by multiplying $A_L$ with the driving forces, IDF. IDF can be calculated using several methods and usually account for differences in pressure because of temperature difference, chimney effect and wind. As well as the phenomena that half the building could experience double infiltration, while the other half have almost no infiltration. Considering calculations on room level, the worst possible situation is accounted for.[16]

Infiltration from doors towards the outdoor, indoor elevators and stairs is accounted for in the calculation of infiltration in commercial buildings. The reason is that commercial buildings often is well sealed so this type of infiltration constitute a significant part. Otherwise, infiltration is calculated in a similar way as for residential buildings. *ASHRAE 2013* notes that the method for commercial buildings can also be used for residential buildings [17]:

$$Q=c(\Delta p)^n \left[\frac{m^3}{s}\right].$$

c is the buildings flow coefficient and is connected to $A_L$ as follows:

$$c = \frac{C_D A_L}{10\ 000} * \sqrt{\frac{2}{\rho}} * (\Delta P_r)^{0.5-n}$$

[21]

This flow coefficient is multiplied with the pressure difference across the building body, which have the exponent, n. Typical value for n is 0,65 [21]. The pressure difference primarily varies with the temperature difference in passive houses and thus chimney effect, but wind and unbalanced ventilation is important factors as well.

$$\Delta P = s^2 C_p P_U + H P_T + \Delta P_I$$

s = shielding factor
$C_p$ = wind pressure coefficient (Depends on location of the building and wind direction. Varies with the height of the building)
$P_U = \rho_0 * \frac{U_H^2}{2}$
$\rho_0$ =density of outdoor air
$U_H$=effective wind speed

H = Height
$P_T = g\rho_0(T_i - T_o)/T_i$
g = gravitation forces, 9,81 m/s$^2$
$T_i$= indoor temperature
$T_o$= outdoor temperature

$\Delta P_I$ is the pressure arising because of unbalanced ventilation, mechanical exhaust air included.[21]

### 3.3.5 Heat loss because of ventilation

The ventilation air flow are dependent on the chosen ventilation principle, displacement or mixing ventilation and variable, demand controlled or constant air flow.[23]
To fulfill requirements of fresh air, the ventilation system must cover the remaining quantity of air that is not covered by infiltration. When calculating power demand at room level using *NS-EN 12831:2003,* one must take into account if the supply air is pre-heated at a central place, coming from an adjacent room or have received heat from a heat exchanger. This is done by multiplying by a temperature reduction factor, $f_{v,i}$.

$$f_{v,i} = \frac{\theta \mathrm{i} - \theta \mathrm{su,i}}{\theta \mathrm{i} - \theta \mathrm{e}}$$

$\theta$su, i = Supply air temperature.
$\theta$i = Indoor temperature
$\theta$e = Outdoor temperature

At building level, one must include the power consumption of the central heating system for supply air. [15]

Power demand to heat the supplied ventilation air are not included at room level according to *ASHRAE 2013.* In the United States it is common that the air is heated to indoor temperature in a central facility [17].

At building level there is, in the chapter on housing, described how one can take in to account the distribution losses in the ventilation system [16]:

- $q_{\boldsymbol{d}} = F_{dl} * q_{bl}$
    - $q_{\boldsymbol{d}}$= distribution losses [W]
    - $F_{dl}$=Channel loss/ supplementation factor
    - $q_{bl}$ = total building load [W]

Distribution losses occur due to leakage and heat losses from the channels and is especially applicable when the channels runs through unheated spaces such as basements and attics. Distribution losses have less importance in heated and ventilated rooms. To meet the need for fresh air in every room, in addition to the leakage in unwanted rooms, the total airflow must be increased [24].

It is taken into account similarly considerations for commercial buildings. It is referred to more advanced methods to calculate heat loss and leakage.

The supply air fan provides heat, which is considered as well. This heat release in the ventilation system varies with location and efficiency of the motor and fan, as well as pressure losses in the system [17]. This is not taken into account in *NS-EN 12831:2003*[15].

The standard for passive and low-energy buildings set requirements for power consumption per airflow supplied. It is expressed by the SFP factor "specific fan power" and is measured in $\frac{kW}{m^3/_s}$.
The requirement is:

- Passive houses: SFP≤1,5 $\frac{kW}{m^3/_s}$
- Low energy buildings: SFP≤2$\frac{kW}{m^3/_s}$

[5, 6]

Table 6 Infiltration and ventilation

| Infiltration &Ventilation | *NS-EN 12831:2003* [15] | *ASHRAE 2013* |
|---|---|---|
| Heat loss | $\Phi = H * \Delta\ \theta$ [W]<br>• $\Delta\theta = \theta_{\dim in} - \theta_{\dim out}$[K]<br>• H = Heat loss coefficient$\left[\frac{W}{K}\right]$<br><br>H=$\boldsymbol{V_i} * \rho * C_p \left[\frac{W}{K}\right]$<br>• $\boldsymbol{V_i}$=Volume flow $\left[\frac{m^3}{s}\right]$<br>• $\rho$=Density of air at indoor temperature$\left[\frac{kg}{m^3}\right]$<br>• $C_p$=Specific heat capacity of air at indoor temperature$\left[\frac{kJ}{kg*K}\right]$ | Sensible heat[16, 17]<br>$q_s = \frac{Q}{\nu} * c_p (t_{in} - t_o)$ [W]<br>• $t_{in}$, $t_o$=Indoor temperature. Outdoor temperature<br>• Q =Volume flow $\left[\frac{m^3}{s}\right]$<br>• $c_p$ = Specific heat capacity of air$\left[\frac{kJ}{kg*K}\right]$<br>• ν = Specific volume$\left[\frac{m^3}{kg}\right]$<br>→ $q_s$ (kW)=1,23*Q*($t_{in}$-$t_o$), assumed 15°C at sea level $\left(\frac{Cp}{\nu} = 1{,}23\right)$<br>Latent heat[16, 17]<br>$q_l$ (W)=(Q/ν)($W_{in}$-$W_o$)$D_h$<br>• $W_{in}$ = Humidity rate of indoor air $\left[\frac{kg_w}{kg_a}\right]$<br>• $W_o$= Humidity rate of outdoor air $\left[\frac{kg_w}{kg_a}\right]$<br>• $D_h$ = Enthalpy change of a phase change of water between the gas and liquid $\left[\frac{kJ}{kg}\right]$<br>→ $q_l$ (W)=3010*Q*($W_{in}$-$W_o$), (assumed 15°C at sea level) |
| Volume flow Infiltration | $\mathbf{V_{inf,i}}$ =2 $* V_i * n_{50} * e_i * \epsilon_i \left[\frac{m^3}{s}\right]$<br>• The factor 2 is valid at room level and consider that all infiltration could occur at one part of the building. This factor is excluded when calculating at building level → $\mathbf{V_{inf}}$=0,5* Σ$\mathbf{V_{inf,i}}$<br>• $V_i$=Volume of the room [$m^3$]<br>• $e_i$ = Shielding coefficient<br>• $\epsilon_i$ = Correction factor for height, which consider that the wind speed increase with height above ground.<br>• $n_{50}$=air change [$h^{-1}$] at a pressure difference of 50Pa<br>Excess exhaust airflow:<br>• $\mathbf{V_{mech,\ inf}} \left[\frac{m^3}{s}\right]$: Excess exhaust is replaced by outdoor air coming through the construction. → Distributed according to the permeability in each room compared to the permeability of the building. | Residential[16]<br>$Q_i = A_L * IDF\ \left[\frac{m^3}{s}\right]$<br>• $A_L$[$m^2$]=The effective leakage area of the building (incl. chimneys) at reference pressure difference=4Pa<br>• IDF= driving forces of infiltration [L/(s*$cm^2$)], which depends on temperature difference, the chimney effect and wind, as well as the possibility that half the building gets all the infiltration in some periods (advanced calculation method)<br><br>$Q_{vi}$=max($Q_{unbal}$, $Q_i$+0,5$Q_{unbal}$)<br>• $Q_{vi}$= Combined airflow$\left[\frac{m^3}{s}\right]$<br>• $Q_{unbal}$= Mechanical or unbalanced ventilation $\left[\frac{m^3}{s}\right]$<br>• $Q_i$= Airflow of infiltration assuming no mechanical pressurization $\left[\frac{m^3}{s}\right]$<br>Commercial buildings[21]<br>Q=c(Δp)$^n$ $\left[\frac{m^3}{s}\right]$ → <u>Could also be used for residential buildings</u><br>• c=Flow coefficient $\left[\frac{m^3}{s*Pa^n}\right]$ varies dependent on how sealed the building is.<br>• n=Pressure exponent [–], typical value of n=0,65<br>• Δp=Pressure difference between outdoor and indoor → $\Delta P = s^2 C_p P_U + HP_T + \Delta P_I$ (are explained in the text)<br><br>Calculate the leakage number $\left[\frac{l}{s}\right]$ for doors facing outside as well as indoor elevators and stairs etc. |
| Volume flow Ventilation | • Total airflow, $\mathbf{V_i}$, must be greater or equal the minimum requirement to achieve good indoor environment[4]<br>• The ventilation must cover the airflow which is not covered by infiltration | • Total airflow, Q, must be greater or equal the minimum requirement to achieve good indoor environment.<br>• The ventilation must cover the airflow which is not covered by infiltration (Must be assumed to be comparable) |

## 3.4 Method of finding design temperature in the Swedish standard for passive houses, *FEBY 12*

The Swedish passive house standard, *FEBY 12*, differs from the Norwegian by taking into account the heat storage at the election of the design winter outdoor temperature. This is done by calculating the time constant for the building by the following formula [25]:

$$\boldsymbol{\tau_b} = \sum(\boldsymbol{m_i} \cdot \boldsymbol{c_i})/\boldsymbol{H_T} \qquad [\boldsymbol{s}]$$

The time constant, $\tau_b$, is a measurement of the time a building use to respond to a rapid change in temperature or interrupted heat supply.

- $\sum(\boldsymbol{m_i} \cdot \boldsymbol{c_i})$ is the heat capacity in the building components, which is all layers located within the insulation layer, including interior walls and beams up to 10cm. $\left[{}^{J}/_{K}\right]$
- $H_T$ is the heat loss coefficient of the building and is calculated on the basis of transmission losses, infiltration losses and ventilation losses. $\left[{}^{W}/_{K}\right]$

Based on climate data the "n-day mean air temperature" can be found. The time constant determines n and then the design temperature is the average temperature over the n days.

In project planning and as a guide one can use the following [25]:

**Table 7 Guidance values for the time constant for different types of buildings [25]**

| | | |
|---|---|---|
| Light building: | 3 days | Lightweight construction and crawl space |
| Half-light building: | 6 days | Lightweight construction, concrete slabs on the ground |
| Half heavy building: | 12 days | Heavy construction, beams of concrete, lightweight facades |
| Heavy building: | 12 days | Max 12 days (considering the heat loss number) |

## 3.5 Ventilation systems - mode of action

Balanced ventilation is most common in new buildings. The ventilation systems are typically a HVAC aggregate and ducts distributing the air throughout the building. [26]

The HVAC aggregate usually have filters, fans, and a heat recovery unit. A common composition of the HVAC aggregate is shown in Figure 14. [27]

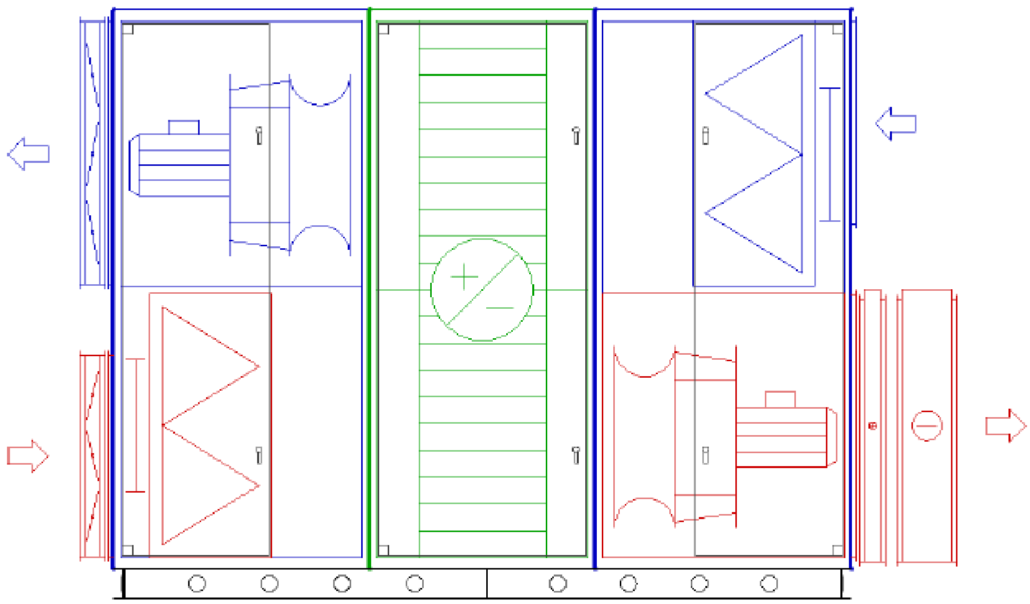

Figure 14 Typical HVAC aggregate used in Norway[27]

To bring enough air to every room, the fan have to overcome the pressure loss in the ducts and in the HVAC aggregate, as well as the pressure loss due to the fact that air is distributed with a certain speed. The pressure loss in the ducts and aggregate represents the static pressure and the pressure in the moving air is the dynamic pressure. A rule of thumbs says that the static pressure is equally distributed between ducts and aggregate, but this strongly depends on the design of the ventilation system[28].

When the air meets friction on its way through the ventilation system, a pressure loss occurs and produce heat. Heat will also arise when the air, which comes out of the ventilation system with a certain speed, becomes stagnant. In addition, the fan and motor also produce heat dependent on the efficiency of these components. Finally, there is some heat release due to losses in the frequency converter.

Dependent on the design of the system and the placement of the fan, motor and frequency converter one can calculate the heat supplied to the ventilation air. Heat released to the supply air follows the air stream into the building. Heat released to the exhaust air could be heat exchanged if the heat release happens upstream the heat exchanger. Heat release in the exhaust air downstream the heat exchanger is lost.

## Pressure loss at different airflows

To calculate the heat release in the different parts of the ventilation system at different operational modes, one need to find the pressure loss using different airflows.

One could assume incompressible flow if the Mac number, which is speed of the airflow divided on speed of sound, becomes lower than 0,3. Then the density effects are negligible. The speed is quite low in a ventilation system and the speed of sound for air is in the range of 300 to 350 m/s, depending on the temperature. It is safe to assume incompressible airflow and use Bernoulli's equation.[29]

$$Ma = \frac{V}{a} \ll 0{,}3 : \quad Incompressible\ flow$$

[29]

***Bernoulli's incompressible frictionless flow equation:***

$$0{,}5\rho_1 v_1^2 + \rho_1 g z_1 + P_1 = 0{,}5\rho_2 v_2^2 + \rho_2 g z_2 + P_2$$

$\rho_1$= density of the air at a chosen starting point, point 1.
$\rho_2$= density of the air at a chosen end point, point 2
$v_1$= speed of the air at point 1
$v_2$= speed of the air at point 2
$z_1$, $z_2$= is the elevation of the point above a reference plane, with the positive z-direction pointing upward.
$P_1$, $P_2$= The pressure at the chosen point
g = acceleration due to gravity
[30]

The expression of ΔP:

$$\Delta P = 0{,}5\rho_1 v_1^2 - 0{,}5\rho_2 v_2^2 + \rho_1 g z_1 - \rho_2 g z_2$$

To simplify one can assume that:

$$\rho_1 = \rho_2$$
$$z_1 = z_2$$
$$\Rightarrow \quad \Delta P = 0{,}5\rho(v_1^2 - v_2^2)$$

If one choose point 2 to be in the room, where the air speed, $v_2$, is zero one gets that:

$$\Delta P = 0{,}5\rho v_1^2$$

**Friction in a pipe:**
There is friction in a ventilation system, which cause extra pressure loss.
Friction in a pipe (given as head loss):

$$\Delta h_{tot} = h_f + \sum h_m = \frac{V^2}{2g} \sum \left( \frac{fL}{d} + \sum K \right)$$

$\Delta h_{tot}$=Total head loss
$h_f$= Pipe head loss due to friction
$h_m$= pipe loss due to bends, pipe entrance/exit, valves etc.
V= speed of the air
f = Darcy's friction factor
L = Length of the pipe
d= hydraulic diameter of the pipe
K = pipe loss factor due to bends
[31]

The cross section area in ventilation systems differs along the path. Then the speed also will vary, therefore one have to divide into parts with same conditions and find the $\Delta h_{tot}$ for each part. Then one can summarize all actual parts.

The friction factor f are dependent on the Reynolds number, while K usually is a constant given by producers of components. Reynolds number for air in ventilation systems is usually turbulent because of the low viscosity (kinematic viscosity (at 0°C) = 1,33 * $10^{-5}$). [31] High Reynolds number implies that friction factors does not varies much. This can be shown using Blasius approximation to calculate the friction factor and insert the actual Reynolds numbers:

$$f = 0{,}316 * {Re_d}^{-0{,}25} \qquad Re_d \in \langle 4000, 10^5 \rangle$$

[31]

Eventually one can use the moody diagram, which shows that the friction factor does not vary that much for turbulent flow with high Reynolds number.[31]

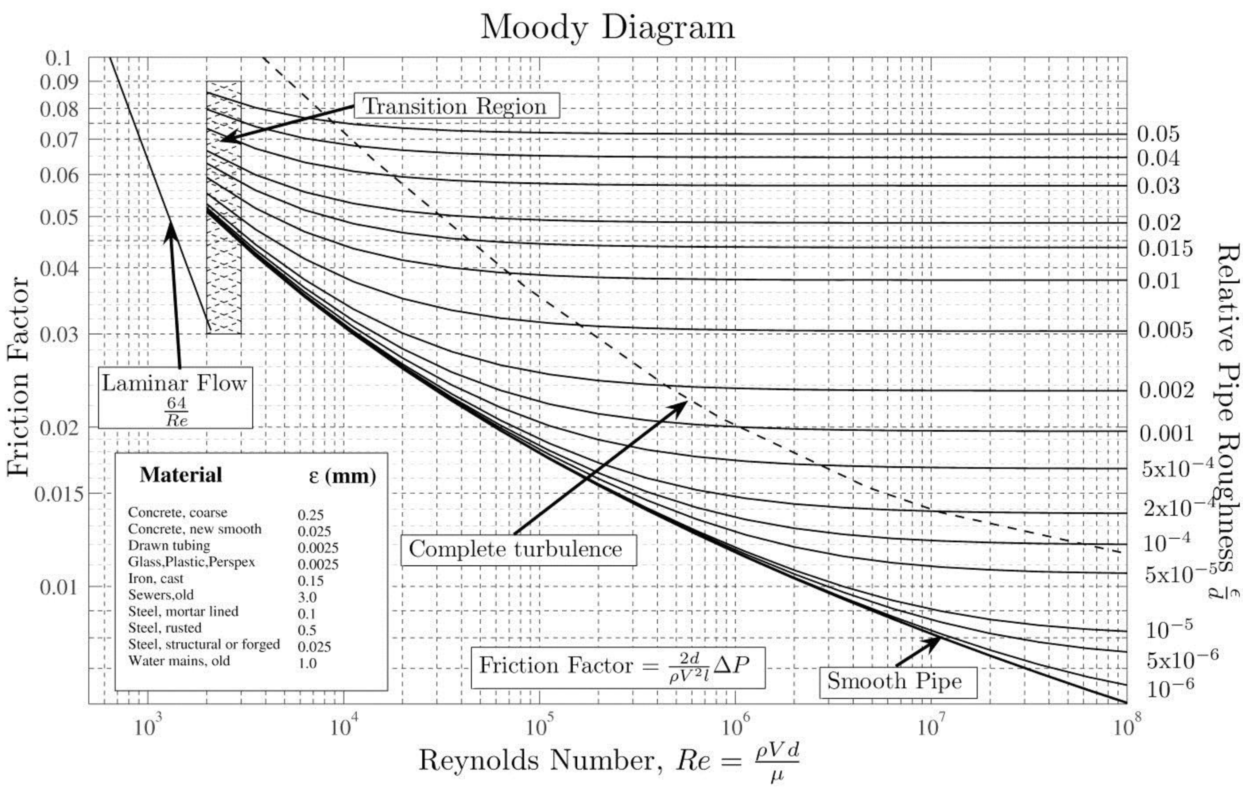


Figure 15 Moody diagram [32]

Taking friction into account the total pressure loss, ΔP is:

$$\boldsymbol{\Delta P = 0{,}5\rho\left(v_1^2 - v_2^2\right) + \rho g \Delta h_{tot}}$$

Using the simplification that the friction factors is constant, one gets that the pressure loss is only dependent on the speed squared. Lower airflow is equivalent with lower speed as the cross section area of the ducts is always constant after installation.

$$v_{dim} = \frac{Q_{dim}}{A}, \quad v_{low\ flow} = \frac{Q_{low\ flow}}{A}$$

$$v_{low\ flow} = v_{dim}\frac{Q_{low\ flow}}{Q_{dim}} \qquad (A = A_{dim} = A_{low\ flow})$$

[30]

The pressure loss in a ventilation system at lower airflow can be calculated using following formula:

$$\Delta P_{low\ flow} = \Delta P_{dim}\left(\frac{v_{low\ flow}}{v_{dim}}\right)^2 = \Delta P_{dim}\left(\frac{Q_{low\ flow}}{Q_{dim}}\right)^2$$

***Energy calculations and SFP:***

To calculate heat loss because of pressure loss in the system, one can use following formula:

$$Heat\ loss = \Delta P * Q = \Delta PAV$$

To calculate energy in moving air, either in the airflow entering the rooms or the outside, one only consider the dynamic pressure in the air leaving the duct, P=$0{,}5\rho V^2$, times the airflow, Q=AV:

$$Dynamic\ energy = 0{,}5\rho AV^3$$

[30]

Heat released from the fan, motor and frequency converter is dependent on the efficiency of the components. Since energy does not disappear, losses in the system becomes heat. The efficiencies depends on the design of the ventilation system. To minimize the total energy use or the average SFP, the optimal design is to have best efficiency at the airflows used most during operation. Components tends to get lower efficiencies at small airflows. Figure 16 are showing different motors from ABB at varying load. The largest motors is those with best efficiency (eg.90-95% at 100% nominal motor power), smaller motors is not that efficient. The efficiency drops rapidly when the airflow gets low(below 30% of nominal motor power) [24]

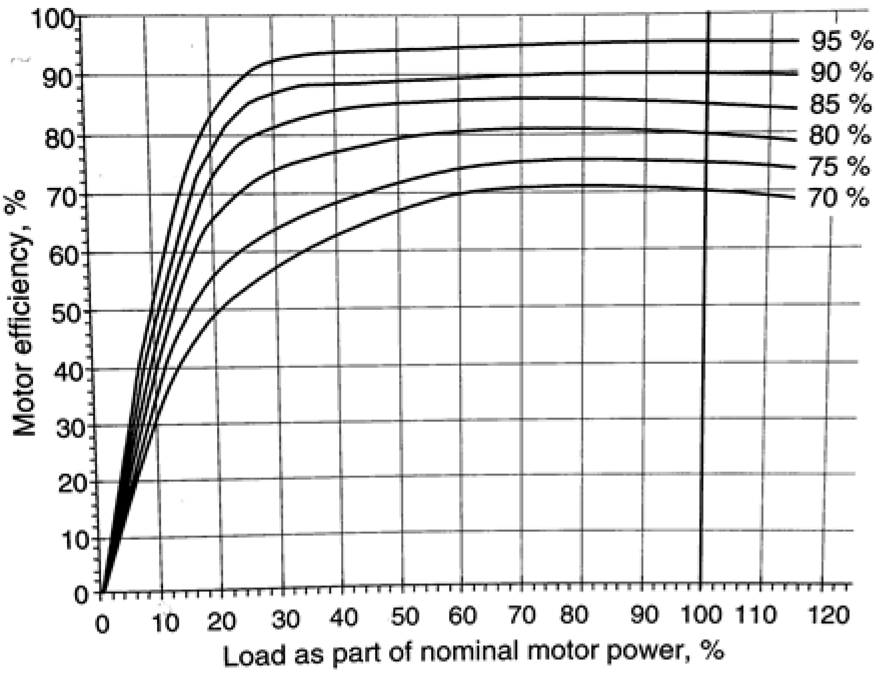


Figure 16 Example of part load efficiency for 2 and 4 pole AC motors. Data from ABB Motors(1999) [24]

The fan and the frequency converter also tends to get lower efficiencies when operating on part load[27].

«Specific fan power», SFP, is the energy use divided by the airflow rate. The energy use because of pressure loss in the ventilation system is decreasing proportional to speed in the power of three. The airflow rate is only decreasing proportional to the speed. This implies that SFP can decrease rapidly with lower airflow[24]. Figure 17 are showing how the SFP could vary in a ventilation system with variable airflow dependent on control and design. One can see that SFP have the possibility to get significantly lower at lower airflows. Lower efficiencies in motor, fan and frequency converter causes the curve to point up or flatten out at lower airflows[33].

NS3031:2007 calculate the SFP factor by the formula:

$$SFP_{red} = SFP_{on} * (1{,}6 * r^2 - r + 0{,}6)$$

$SFP_{red} = SFP\ at\ reduced\ airflow$
$SFP_{on} = SFP\ at\ average\ airflow$
$r = \ Fraction\ of\ maximum\ flow\ rate\ = \frac{\dot{V}_{red}}{\dot{V}_{on}} = \frac{reduced\ airflow}{average\ airflow}$
[34]

This formula gives a result following the curve for «Normal» control of the ventilation system for airflows between 60 and 100% of maximal. For airflows between 20 and 60% of maximal airflow the curve will be between «Normal» and «Good» control.[33]

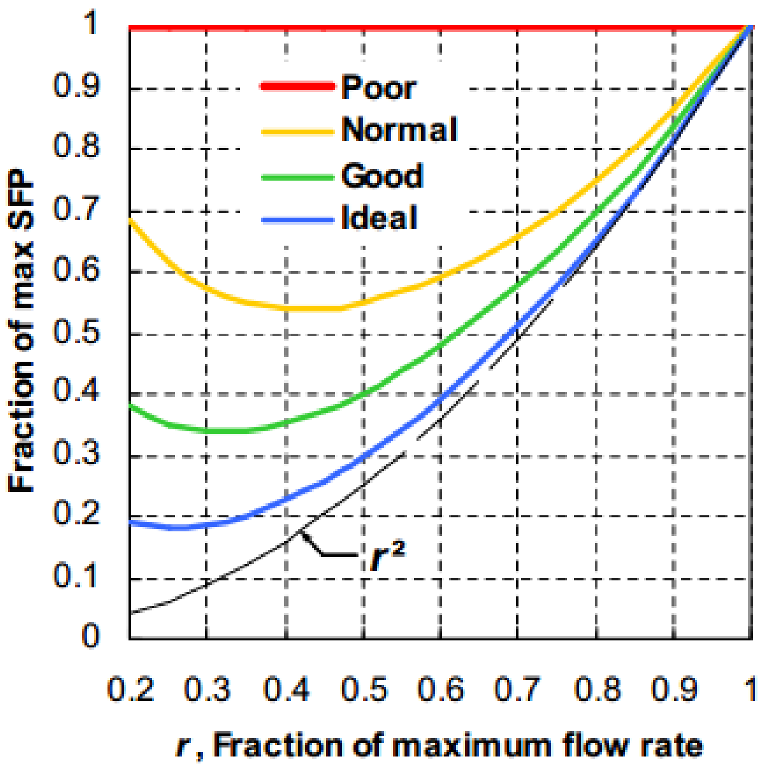


Figure 17 Relation between SFP and airflow in DCV-systems from poor to ideal control[33]

# 4 Electricity and district heating suppliers experiences

District Heating Suppliers in Norway were questioned about compliance between ordered and supplied power. The results of the survey are based on both electricity supply and district heating supply. Most companies did not distinguish whether delivered power were used to heat the building, heat domestic hot water or used to other purposes.

## 4.1 BKK

*BKK* supplies electricity to Løvåshagen cooperatives in Bergen. It consists of 80 units of which 28 are passive houses and 52 are low energy buildings. It is installed solar collectors on the roof of the passive houses, which produce $17\frac{kWh}{m^2*year}$.

It was estimated that:
Energy consumption for heating rooms and ventilation are:

- $15\frac{kWh}{m^2*year}$ for the passive houses
- $25\frac{kWh}{m^2*year}$ for the low energy buildings

Total estimated energy use is:

- $91\frac{kWh}{m^2*year}$ for the passive houses
- $101\frac{kWh}{m^2*year}$ for the low energy buildings

The low energy buildings has electric water heaters, underfloor heating in bathrooms and one electrical heater in the living room of approximately 1000 W. The apartments are 50 $m^2$ - 95 $m^2$ and the average apartment size is 80 $m^2$ for both the low-energy houses and passive houses.
(Appendix II – Løvåsen cooperatives)

Unfortunately, BKK are only measuring the total power and energy consumption for the entire housing cooperative. They does not differentiate between distribution to passive houses or low energy buildings. There is 80 units in total, with a total residential area of 6353m. The maximum power demand registered is 229kW and the total annual energy output is approximately 900 000kWh.

This gives the following:

- Electricity consumption pr. year: $141\frac{kWh}{m^2*year}$
- Power demand: $36\frac{W}{m^2*year}$
    - Power pr residential unit become about 2,9kW.

The total power use became larger than estimated, but BKK do not know what are using the extra power. The circuit breaker was designed on the basis of low power requirement, but has been modified since it was triggered multiple times.[35]

## 4.2 Hafslund nett

*Hafslund nett* are experiencing that oversizing of installed power in general applies to all buildings, and writes by email:

” It seems that installers, our customer contact, want to ensure enough power. Secure that the customers overload protection will never break down due to overload”[36]

## 4.3 Statkraft varme

*Statkraft varme* delivers district heating to a heat exchanger at the building site, hence they cannot differentiate between what goes to domestic hot water, space heating or ventilation air heating. Still, they have made an analysis of some passive houses in Trondheim and concluded that:

- Maximum power demand for heating is in the range 12-15 $\frac{W}{m^2}$ for each unit.
    - For example, a unit of 150m$^2$ will have a maximum power demand of approximately 2kW

*Statkraft varme* states that the domestic hot water determine the power demand in their systems:
”A single residential passive house may for example order 34 kW (4 kW to room heating + 30 kW to domestic hot water). Meaning that even if ordered power demand to heat the building become twice the necessary, it would not affect the sizing of our plant since domestic hot water dominates. Dimensioning the district heating pipe in a building for 32 or 34 kW does not matter.”[37]

## 4.4 Agder energi varme

*Agder energy varme* have little experience with passive and low energy buildings, but writes by email:

” Usually the power demand is oversized a lot. We cannot remember having had to increase the capacity”[38]

## 4.5 Norsk fjernvarme

*Norsk fjernvarme* has received feedback that the consumption of district heating usually is higher than intended, but have little experience of passive and low energy buildings.

” Some builders and owners of passive and low energy buildings do not want to make public that energy consumption is higher than expected, especially if they have expressed in the press about little energy consumption in advance. Hence, that information is not easy to obtain.”[39]

# 5 Discussion/Results

## 5.1 Design outdoor temperature using the Swedish standard *FEBY 12*

To find the time constant, $\tau_b$, for the whole building, results from SIMIEN fit directly in the formula:

$$\boldsymbol{\tau_b = \sum (m_i \cdot c_i)/H_T} \qquad [s]$$

Inserting all information about the building, SIMIEN gives a normalized heat capacity of $106 \frac{Wh}{m^2 K}$ and a total heat loss number of $0{,}58 \frac{W}{m^2 K}$. The time constant then becomes:

$$\tau_b = 657936 \text{ s} = 182{,}76 \text{ h} = 7{,}61 \text{ days}$$

According to *FEBY 12* one can use a 7,61 days mean temperature as the design winter outdoor temperature.

The n-day mean temperatures shown in Table 8 are determined according to "NS-EN ISO 15927-5"[18]. Meteorological data from the years 1970-2000 is obtained from "Meteriologisk institutt"[40]. The nearest place to the analyzed building, which also fits with the altitude of the actual building, are "Fornebu" weather station.

**Table 8: n-day mean outdoor temperature, temperatures in °C**

| Place | 1-day | 2-day | 3-days | 4-days | 5-days | 6-days | 7-days | 8-days | 9-days | 10-days | 11-days | 12-days |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Fornebu | -22 | -20,7 | -20,1 | -19,5 | -19,3 | -18,4 | -18,2 | -18,2 | -17,8 | -17,3 | -16,3 | -15,5 |

Taking into account the effect of thermal mass, the proper design outdoor temperature is -18,2°C according to *FEBY 12*. In the project plan of the building, it is assumed -20°C, which may lead to oversizing of the heating system.

## 5.2 Useful heat release in ventilation systems

Useful heat release from a ventilation system is dependent on the airflow and the placement and efficiency of the motor, frequency converter, fan and heat exchanger. Calculations made for the different aggregates in different operation modes shows that approximately 50-65% of the power used can be considered as internal load (useful heat release).”Appendix I – Calculation of heat release from the ventilation system”, shows the excel sheet used to calculate Table 9 and explain the “Method of calculations”.

**Table 9 Useful heat release to the building through the ventilation system**

| DVN 100 | | | DVN 40 | | |
|---|---|---|---|---|---|
| **Airflow rate [m³/h]** | **Useful heat release to the building** | | **Airflow rate [m³/h]** | **Useful heat release to the building** | |
| | Given in: kW | Given in: % of power use for ventilation | | Given in: kW | Given in: % of power use for ventilation |
| **26 000** | 9,92 | 64% | **3000** | 0,44 | 57 % |
| **24 264** | 8,12 | 64 % | **250** | 0,03 | 50 % |
| **20 000** | 4,8 | 63% | | | |
| **4 320** | 0,57 | 52 % | | | |
| **3 000** | *0,56 | *51,2 % | | | |

*DVN 100 with airflow of 3000m³/h use as input that SFP (total power use) follows Figure 17 with “good control” and that the “% of power usage for ventilation” is linear with the airflow

## 5.3 Simulations and measurements in an office building

In this chapter, it is first looked upon SIMIEN simulations of the preliminary project. The building is divided in two in the preliminary project, the main part of the building and one smaller part. One predict ventilation airflows, lights and internal load based on the passive house standard and assumptions of user behavior. The building body is updated to "as built" condition. Looking at this preliminary project gives a rough overview and indicates which factors that is interesting to investigate more closely.

The goal of this master thesis is to improve the Norwegian standard for design power demand. In order to achieve this goal, a new SIMIEN model that matches the measurements from the actual building are made, so that one are able to compare simulated power demand to the measured. The building is divided into four parts, 2-5 floor northern zone, 2-5 floor southern zone, first floor and basement. Obtaining good measurements from the first floor and basement were hard, as it was not allowed to observe user behavior in these floors cause of confidentiality. The investigated zones are therefore those in the 2-5 floor. Outdoor temperature and solar radiation are implemented from climate stations nearby.

The original measurements were missing some important values to make a good model, such as ventilation flow rates, supply air temperature and indoor temperature as well as user behavior and number of users. Therefore, these factors are measured in week 50, 8-12 December. This week is the main basis for the whole simulation. Ideally, one should have measure more than one week and include the coldest days of the year. Looking upon several buildings would eliminate individual inaccuracies as well. Regardless uncertainties cause of limited data and measurements, many interesting results were found.

To improve the standard, comparison of SIMIEN simulations, theoretical calculated values and real measurements is necessary. It is important to have in mind that many parameters could influence the need of heat and power.

### 5.3.1 The preliminary project

Figure 18 shows the yearly energy budget for the main part of the building. It is easy to see that energy is used for many purposes and not only for heating. Fans, lighting and technical equipment together use 57,6% of the total yearly energy use. [41]

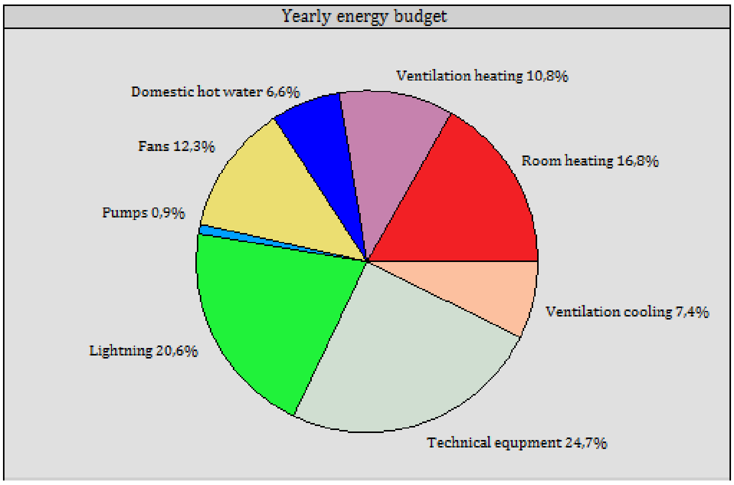


Figure 18 Yearly energy budget for the main part of the building, simulation of the preliminary project[41]

In the winter time, heating of the building is of course a larger part of the energy use as shown in Figure 19. Nevertheless, fans, lighting and technical equipment is a significant part of the energy use. Then, how much of the energy use contribute with heat as internal load? Are internal load from these factors contributing with heat when the need of power is largest? Is it reasonable to install less power because of this contribution?

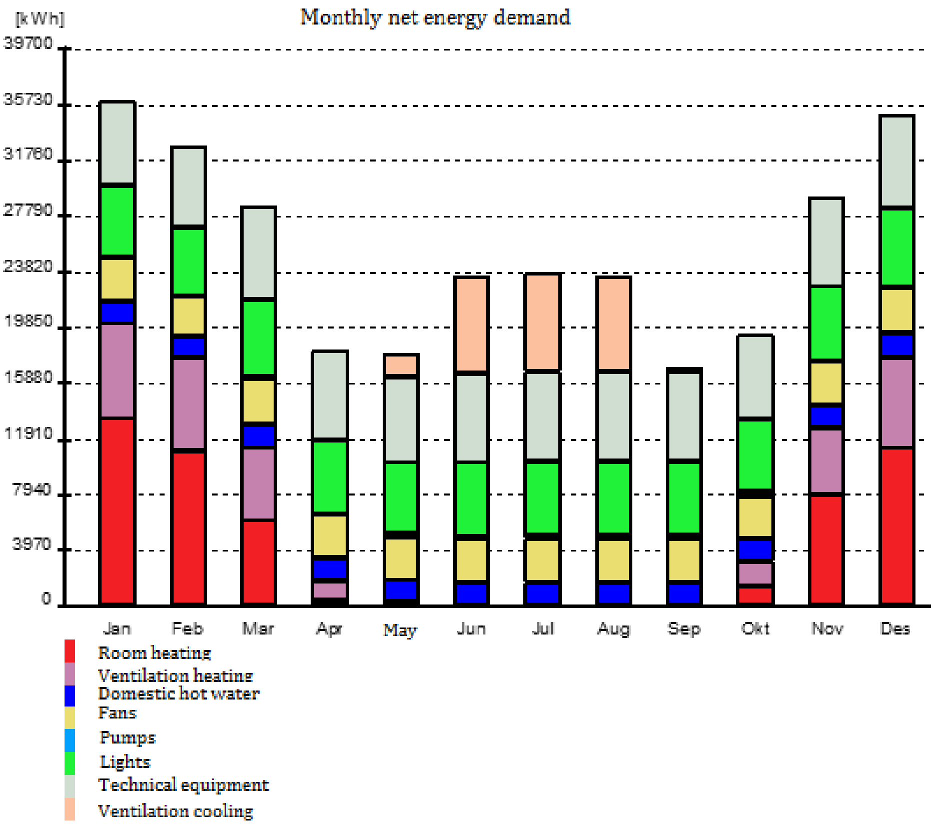


Figure 19 Monthly net energy demand for the main part of the building, preliminary project [41]

## Ventilation

It is interesting to look further into the heat release in the ventilation system. Some of this heat is lost in the exhaust or in the machine room, but quite a lot of the heat release follows the supply air as well.

***Ventilation principle****:*
There is 3 ventilation aggregates in daily use in the building. The largest aggregate, DVN 100, serves the office area in 2 to 5 floor. The smaller ventilation aggregates, DVN 40, serves lecture rooms and halls in the ground floor and basement. Exhaust air from these two aggregates enters the open parking space in the basement. [42]

The lecture rooms and halls have a very variable need for ventilation. In use, there is many people in the rooms and need for fresh air is huge. Although the rooms is probably not in use all the working hours, the estimated average ventilation is 12 $\frac{m^3}{h \cdot m^2}$. Normal office areas do not need so much ventilation, it is actually estimated an average ventilation airflow as low as $6\frac{m^3}{h \cdot m^2}$. Outside operating hours, the ventilation airflow is decided to be $1\frac{m^3}{h \cdot m^2}$ in projecting calculations.[41]

## Internal load

### *Lighting*

The power consumption of lighting is planned to be 5 $\frac{W}{m^2}$ during operation and 0 $\frac{W}{m^2}$ outside operation. 5 $\frac{W}{m^2}$ during operation is the passive house demand for lighting in *NS 3701:2012* and all energy used becomes heat. Figure 18 shows that heat released to the rooms represents 20,6% of the energy demand in the main part of the building. In the coldest period of the year, the percentage part is not that high, but looking at the coldest month in Figure 19, typically January; it is still almost 20% of the energy use.

### *Technical Equipment*

The power consumption of technical equipment is planned according to the passive house standard *NS 3701:2012*, 6 $\frac{W}{m^2}$ during operation and 0 $\frac{W}{m^2}$ outside operation. All energy used becomes heat.

### *People*

Internal load cause of presence of people is planned according to the passive house standard *NS 3701:2012* as well. 4 $\frac{W}{m^2}$ during operation and 0 $\frac{W}{m^2}$ outside operation.

## Design power demand

***Planned internal load according to NS 3701:2012 and night setback to 19°C from 21°C during operation, looking at the main part of the building.***

SIMIEN are including heat gains from the sun and internal heat gains from people, technical equipment and lighting. Heat because of the pressure loss in the ventilation system is NOT included in SIMIEN. Looking at the winter simulation, there is a large power demand for heating at the start of the operating hours because of the night setback. Assuming deign outdoor temperature of -20°C, SIMIEN calculates that the design power demand is:

Table 10 Power demand at design conditions

| *Power consumers* | *Power demand [kW]* |
|---|---|
| Room heating | 104 |
| Ventilation heating coil | 52,8 |
| ***Net power demand*** | **156,8** |
| Internal load: | 60,7 |
| ***Gross power demand*** | **217,5** |

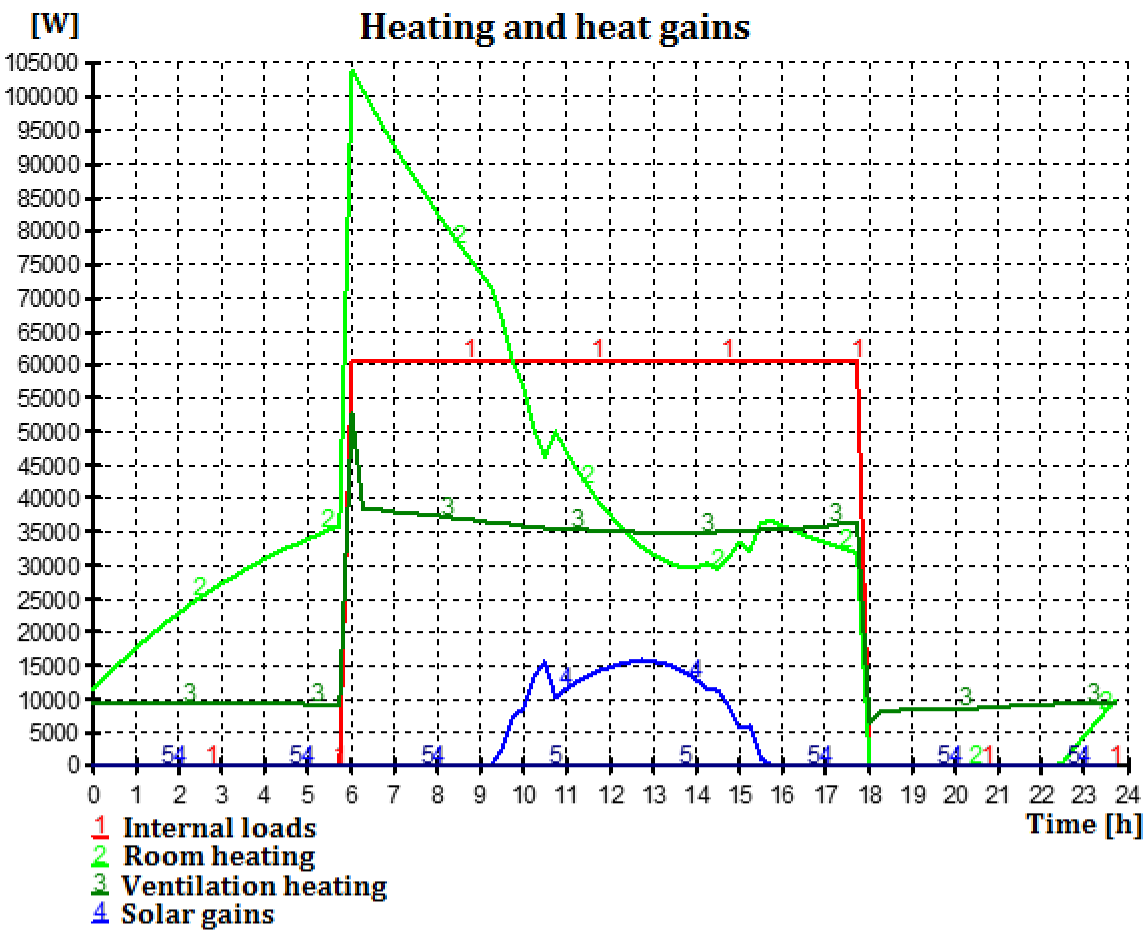


Figure 20 Heating gains in winter simulation in SIMIEN

Internal load covers 60,7 kW, which is 27,9% of the total need at the time where one needs most power. Taking into account the usable heat from the ventilation system, 8,12 kW ($\approx 2\frac{W}{m^2}$). Internal heat gains covers 31,6% of the actual need of power.

Notice that heat gains from the sun only occur in the middle of the day and would not contribute at design conditions.

This simulation is at building level. When picking out radiators one cannot assume equally distributed internal loads, one have to look at each room separately. When choosing the design power of the central heating plant this could come in hand. An oversized heat pump for example will, as mentioned earlier, get lower efficiency.

## Heating up capacity

Another uncertainty is how long time one accept for the reheating after night setback. Starting the reheating for example at 5 o'clock, the power demand for the heating-up capacity will decrease. According to Figure 21, SIMIEN increase the temperature within approximately 15 minutes. There is no way to adjust the time of reheating in SIMIEN.

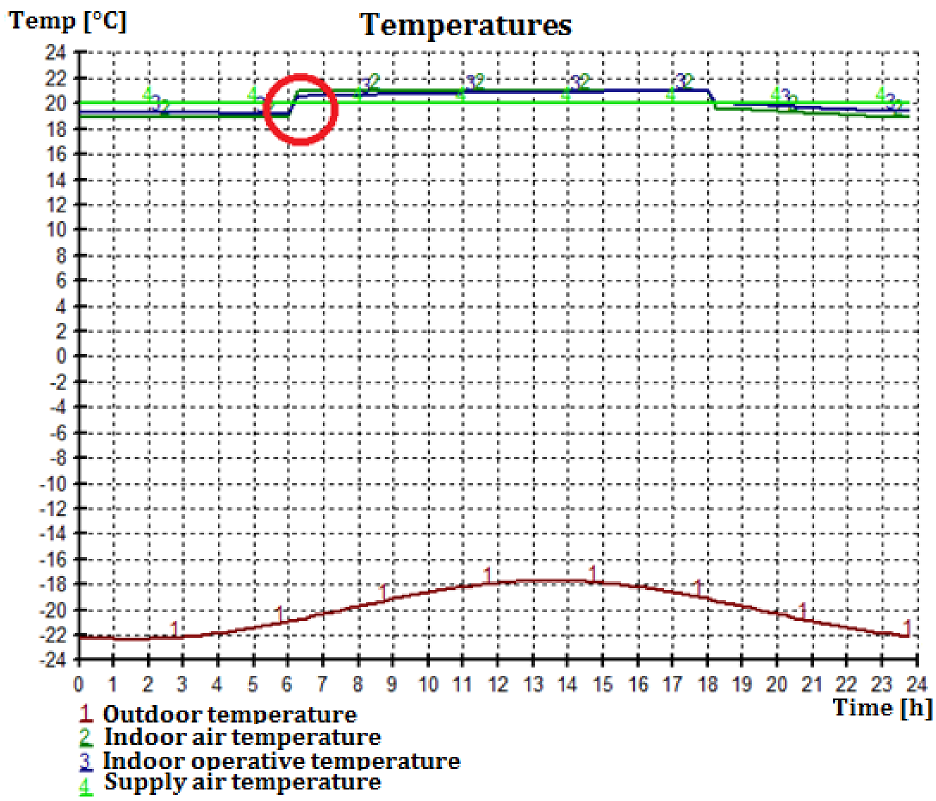


Figure 21 Temperatures from the winter simulation in SIMIEN

The heating-up capacity, which varies with temperature raise, building mass and time of reheating is calculated in Table 11, using the Norwegian standard, *NS-EN 12831:2003*, given in Table 3. The classification of the actual building is a “heavy building” and there is planned a temperature raise of 2K. The area of the simulated part of the building, the main part, is 4044 $m^2$.

Table 11 Heating-up capacity according to *NS-EN 12831:2003* using different “time of reheating”

| Time of reheating [h] | Heating-up capacity [kW] |
|---|---|
| **1** | 101,1 |
| **2** | 89 |
| **3** | 72,8 |
| **4** | 64,7 |

Table 11show that the heating-up capacity decrease with longer time of reheating. Nevertheless, Table 3 clearly shows that the heating-up capacity would decrease even more in buildings with lower thermal mass.
As there is no way of choosing time of reheating in SIMIEN, one could look upon the power demand without night setback and then manually choose a longer time of reheating if that is desirable. The heating-up capacity used in SIMIEN is found by comparing the gross power demand from Table 10 and Table 12(simulations with and without night setback). This implies that SIMIEN in this case use a heating up capacity of 97 kW, which again more or less correspond to one hour reheat time using the Norwegian standard given in Table 11.

***Planned internal load according to NS 3701:2012, 21°C constant (not night setback), looking at the main part of the building***

Figure 22 shows the heat gains in a winter simulation in SIMIEN. Gross power demand is largest at the start of the operational hours. Nevertheless, the net power demand is largest at about 3 o’clock in the night (extracting internal load).

Table 12 Power demand at design conditions

| *Power consumers* | Start of operational hours: *Power demand [kW]* | 3 o’clock at night: *Power demand [kW]* |
|---|---|---|
| Room heating | 21 | 60,7 |
| Heating coil | 38,8 | 7 |
| ***Net power demand*** | **59,8** | **67,7** |
| Internal load | 60,7 | 0 |
| ***Gross power demand*** | **120,5** | **67,7** |

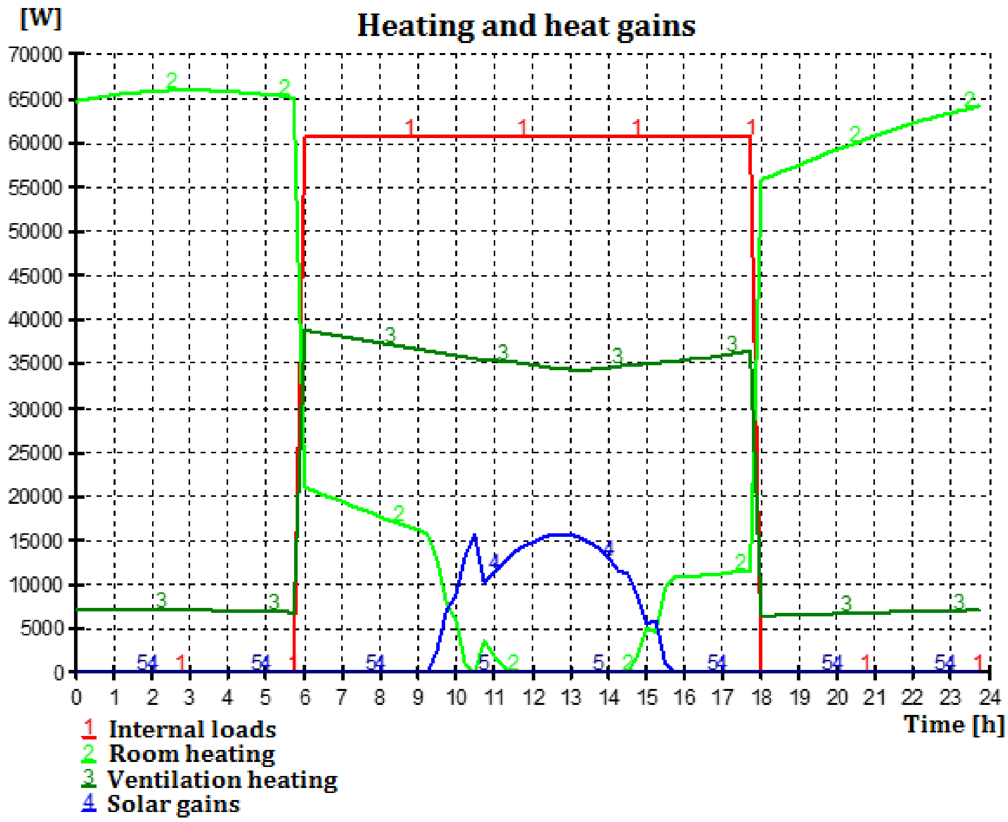


Figure 22 Heat and heating gains in the winter simulation in SIMIEN

The Norwegian standard assume no internal load, i.e. gross power demand. In that case, "Room heating" must cover the heat given by internal load in the simulation. The need of heat is largest at the beginning of operational hours, 120,5kW for this part of the building.

Using the same conditions and airflows as simulated, theorehical power demand calculated according to *NS-EN 12831:2003* is given in Table 13. Theoretical vaules is very similar to the simulated values, which makes sence as the simulation are based on norwegian standards.

Comparing Table 13 to the SIMIEN simulation with night setback (Table 10), one must add heating up capacity from Table 11. Using 1 hour reheat time will give amost equal design power demand.

Table 13 Theoretical Power demand [kW] (Start of operational hours)

| *Power consumers* | *Theoretical Power demand [kW] (Start of operational hours)* |
|---|---|
| Room heating (no internal load) | 82 |
| Ventilation heating coil | 42,8 |
| ***Gross power demand*** | **124,8** |

As seen in Figure 22, internal load covers about half of the gross power demand at building level in the beginning of operational time, leading to a low net power demand.

Taking heat release from ventilation (8,12 kW ) into account as well, internal load will cover about 60% of the gross power demand simulated. The net power demand is largest outside operation hours when there is no night setback. Heat contribution from the ventilation system outside operational hours is very small (0,57kW) and would not make any difference. The net power demand outside operating hours is 67,7 kW.

Assuming there is no night setback, the question would be if there would be reasonable to use the peak of gross power demand (120,5kW) or the peak of net power demand (67,7kW) as design power demand when dimensioning the heating system, or maybe something in between.

The difference between gross and net power demand is large during operation hours and an accurate way of calculate the power demand is essential to design the heating system in a cost effective way.

#### 5.3.2 Real measurements at building site

In this section, measurements from the actual building are implemented in a SIMIEN model. Input variables in the SIMIEN model must equal real conditions to compare simulated and measured power demand and evaluate the Norwegian standard, *NS-EN12831:2003*. A theoretical calculated design power demand is compared to the measured and simulated values as well.

Measured power demand is available for the building, divided in three part, and does not include the basement. The parts is 2-5 floor northern zone, 2-5 floor southern zone and the ground floor. Measured power demand is actually hourly measured heat demand. This may lower the peaks of power demand somehow, but are quite good measurements for power demand.

Because of confidential activities in the ground floor, user behavior were not possible to measure. Since the ground floor mainly consist of halls and lecture rooms where user behavior is important in order to get reasonable results, only 2-5 floor were analyzed in this master thesis. 2-5 floor is divided in two parts of 1424m$^2$ each, northern and southern zone, and consists of offices and one computer room. The computer room was not in use during the period of measuring, week 50 in 2014.

## Inaccuracies in the SIMIEN simulation

There are programs more accurate than SIMIEN that allow more input parameters etc. One is IDA ICE, but cause of limited time and limited input variables, SIMIEN is used in this master thesis.

Ideally, one should have divided the zones in rooms or groups of rooms. Doing this in SIMIEN seemed unnecessary, as the real measurements were divided in huge zones.

Using another more advanced program makes it more important to divide into smaller zones. For every zones, one shall in SIMEN, choose between one or more than one facade exposed to wind. Using only two zones, both have more than one facade exposed to wind. Dividing into many zones, only the corner rooms are exposed to wind at more than one facade. This makes a difference and will be discussed and adjusted by theoretical calculations.

## Climate

SIMEN uses several climate variables such as outdoor temperature, radiation, wind, wind direction, humidity etc. Outdoor temperature have from experience the largest impact on the power demand in a building. Hourly temperature values from a climate station nearby and are implemented in SIMIEN. Radiation and wind is also quite important factors. The influence because of wind is strongly dependent on the shielding level. Wind speeds varies a lot between the nearby climate stations as well. Therefore, it is used values for a normal year for wind. Measurements of radiation on the other hand, is implemented in SIMIEN. Radiation varies a lot by location and time. There might be sun at the nearby climate station, while it is cloudy at the building site. Nevertheless, it is reasonable to believe that the average radiation is quite similar at the building and at the climate station.

## Ventilation

The ventilation airflow and temperatures are measured in the ventilation aggregate, which is described in chapter 0. There is a joint ventilation aggregate in 2-5 floor. According to the operation manager, the distribution between the northern and southern zone is more or less equal as long as there is no special need for cooling. Almost every rooms have VAV-boxes and some very few have constant airflow. Temperature sensors and CO2 sensors controls the VAV-boxes individually. The ventilation system is in operational mode from 06:30 to 17:00.[43] In operational mode the minimum supply airflow is about 17 800$\frac{m^3}{h}$ (6,3$\frac{m^3}{m^2 h}$). When the VAV-boxes ask for it, it is used higher airflows. Outside operational hours the ventilation airflow is 3000$\frac{m^3}{h}$ (1,05$\frac{m^3}{m^2 h}$), constant.

## Supply airflow

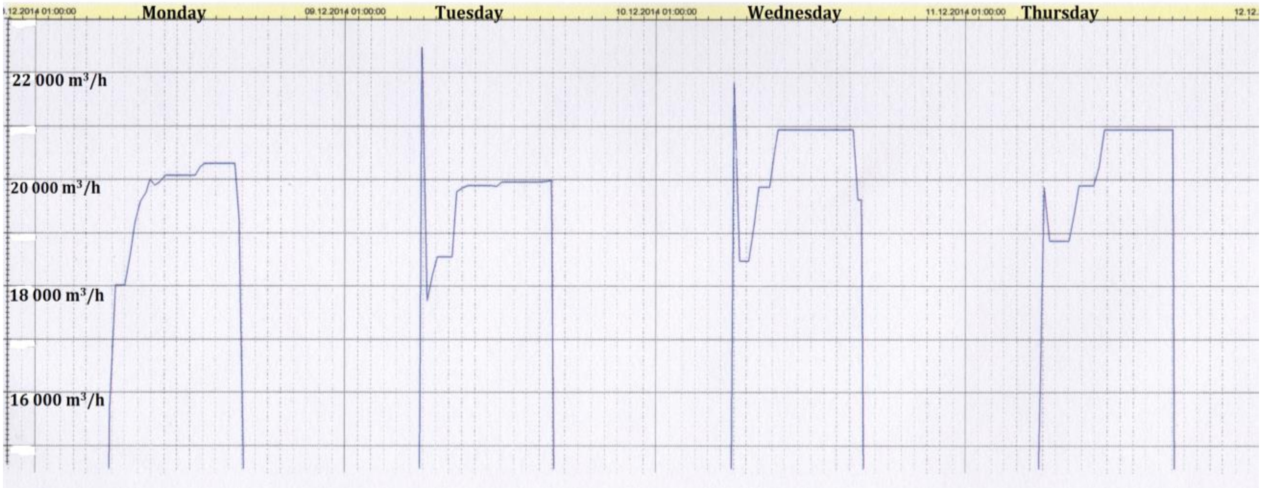


Figure 23 Supply airflow in week 50[44]

## Exhaust airflow

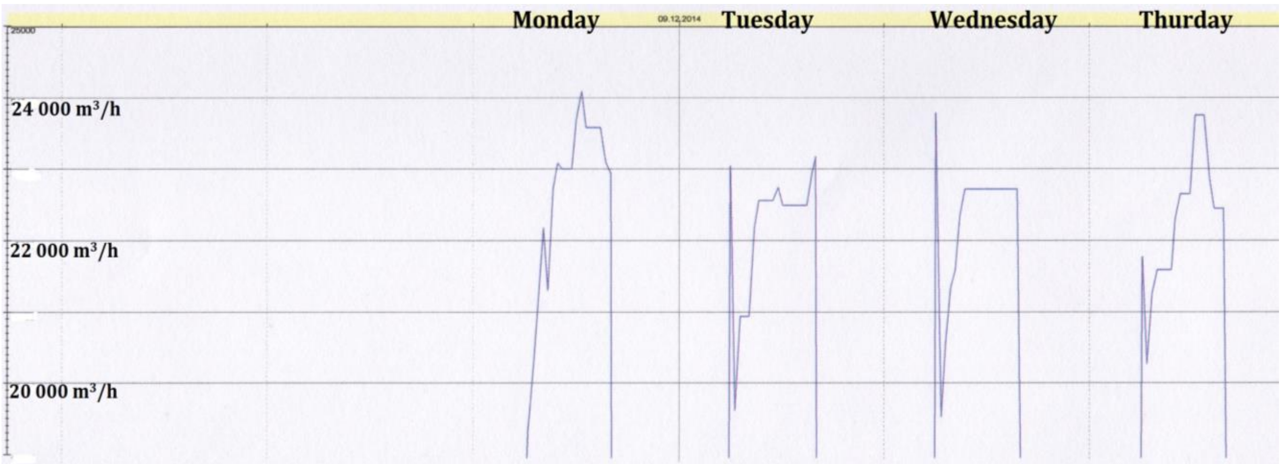


Figure 24 Exhaust airflow in week 50[44]

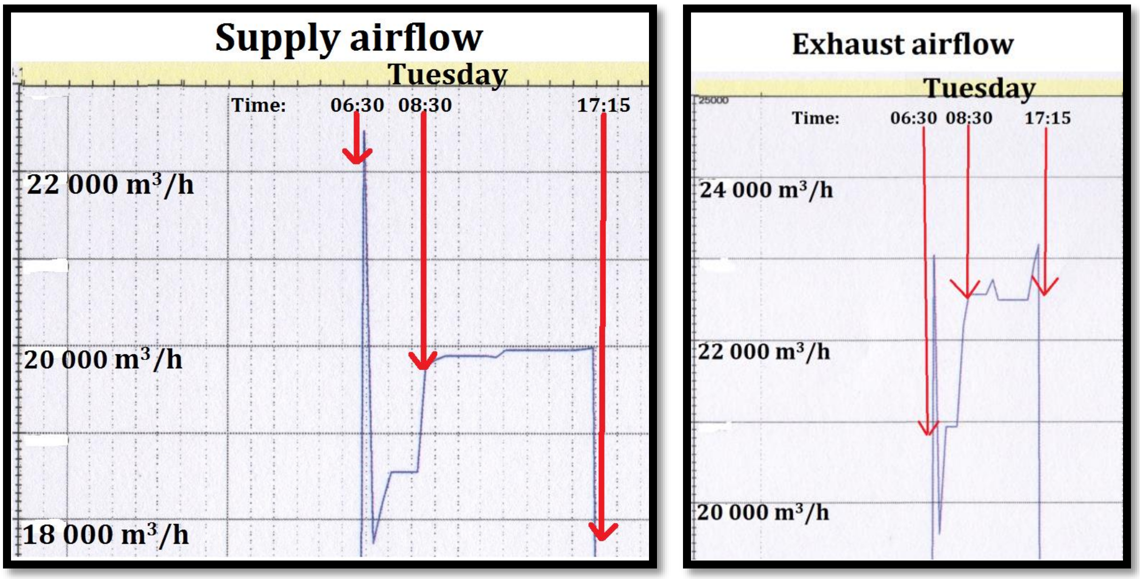


Figure 25 Ventilation airflows a typical day, Tuesday in week 50, 2014[44]

Although it turns out that there most likely is balanced ventilation (discussed in chapter 5.3.2), Figure 23, Figure 24 and Figure 25 show unbalanced ventilation. Unbalanced ventilation must be taken into consideration as the difference in airflow have to leak in trough doors and other unsealed spots in the wall etc. According to the operation manager, it should not be under-pressure ideally. Nevertheless, the benefit of having under-pressure in a building in cold climate is decreasing possibility of condensation cause of chilled indoor air going through poorly sealed spots in the wall. [43]

Table 14 shows airflows used in a SIMIEN simulation (Later the SIMIEN simulation is changing, using balanced ventilation based on supply airflow). Tuesday is used as a basis because it is the coldest day measured in week 50.

Table 14 airflows used in SIMIEN, measured week 50

| Timespan | Supply airflow [m$^3$/h] | Exhaust airflow [m$^3$/h] | Difference (in % of supply airflow) |
|---|---|---|---|
| 17:15 – 06:30 | 3 000 | 3 100 | 3,3 |
| 06:30 - 08:30 | 18 500 | 21 000 | 13,5 |
| 08:30 – 17:15 | 20 000 | 22 600 | 13 |

The supply air temperature was 18,5°C during the time of measuring in week 50. It varies with the outdoor temperature as shown in Figure 26. The system will give higher supply air temperature in the winter when it is cold outside and the maximum power demand occur.

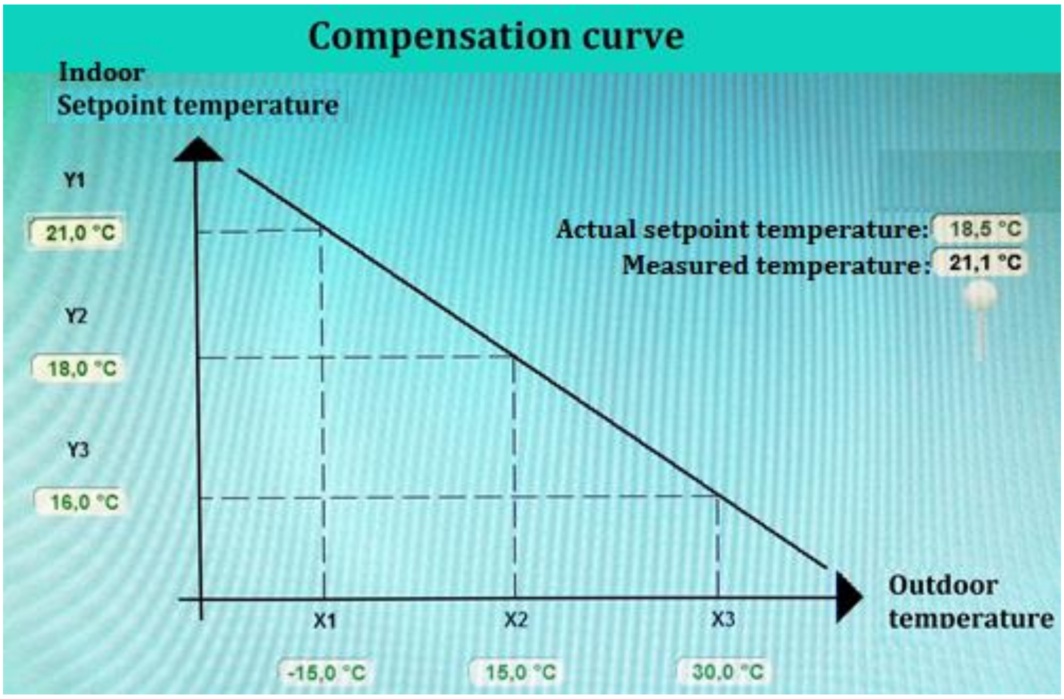

Figure 26 Compensation curve, supply air temperature depends on the outdoor temperature[44]

**Heat recovery**
The outdoor temperature differed from -5°C to 5°C in week 50. Because of these relatively high outdoor temperatures, the heat recovery unit did not always operate at maximum. Nevertheless, there is no doubt that the heat recovery system have to operate at maximum the coldest day, Tuesday. The heating coil in the ventilation system operates the whole day.

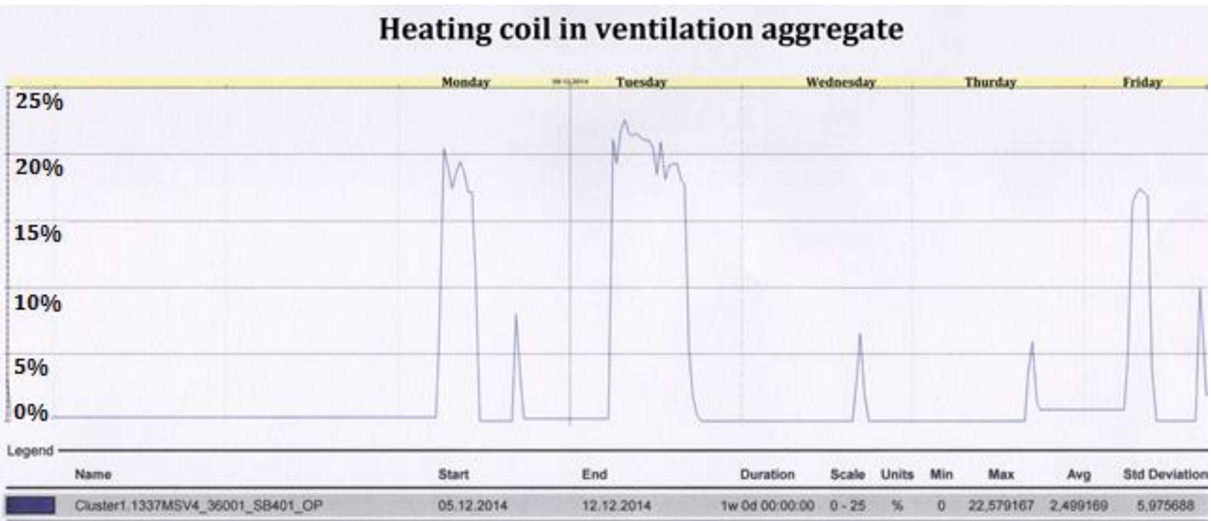

Figure 27 The heating coil in the ventilation aggregate, power use given in % of maximum. (More than 0% means that the heat recovery system will also work at maximum and try to recover as much as possible)[44]

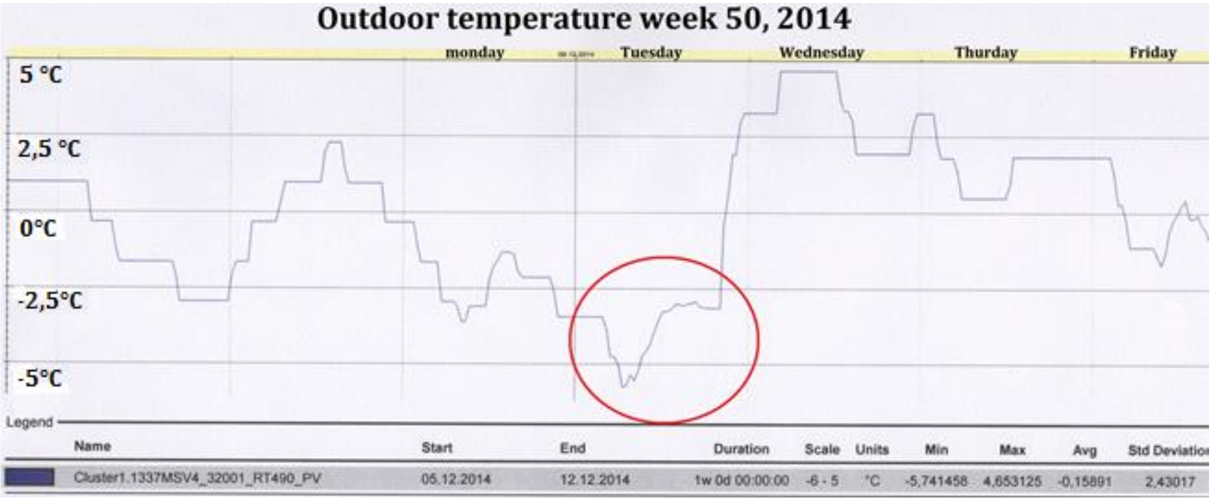

Figure 28 Outdoor temperature at building site week 50, 2014[44]

Another issue is that the measured temperature-efficiency of the heat recovery unit is not 85% on average as stated in the preliminary project. From Figure 29 we can see that the efficiency is about 72% during operating hours, when the heat recovery unit try to recover as much energy as possible, on Tuesday.

Figure 29 show an efficiency of 0% outside operating hours. As shown in Figure 25, the airflow is only 3000 $m^3/h$ outside operating hours and the efficiency of the heat recovery unit should become even higher than 72%, not 0%. Bad control of the system could have been the reason, by using ventilation air to cool the building because of cooling demand occurring with night setback of the indoor temperature. The compensation curve in Figure 26 is a screenshot outside operational hours and show that the supply air temperature is high. Measurements in of power use by the ventilation system, shown in Figure 40 and Figure 41, show very small power demand outside operational hours. All this implies that the efficiency of the heat recovery unit not 0% as in the figure! Based on simulations in SystemairCAD, there is reason to believe that the efficiency is supposed to be 77% outside operational hours, which also is the value used in the simulations.

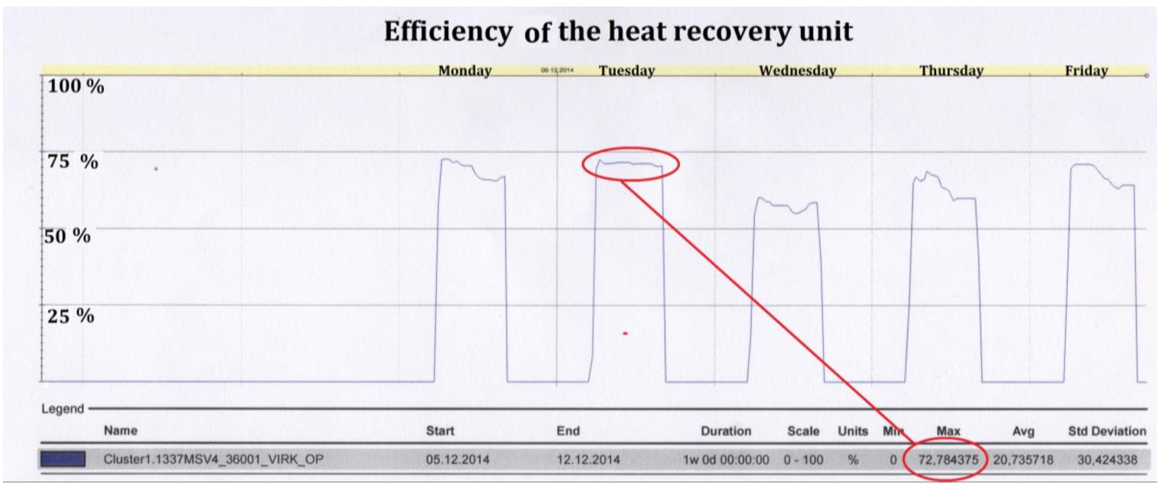

Figure 29 Efficiency of the heat recovery unit, week 50, 2014. [44]

## User behavior and indoor temperature

It was necessary to measure the user behavior in the building as it turned out that the control of the building is different from assumed in the preliminary project. There is presence sensors in every room. If there is people in the room the setpoint temperature is 23°C in floor 2, 3 and 5 and 22°C in floor 4. With no people in a room, the set point temperature is 21°C and 20°C respectively.

Both number of people in each zone and the proportion of area with people present were manually accounted in week 50. Results found this week is considered as a reference of user behavior every week through the year.

Figure 30 are showing the average setpoint temperature in the two zones analyzed based on percentage of area with people present. These values are implemented in SIMIEN. In the night, with no presence of people, the average setpoint temperature is 20,33°C.

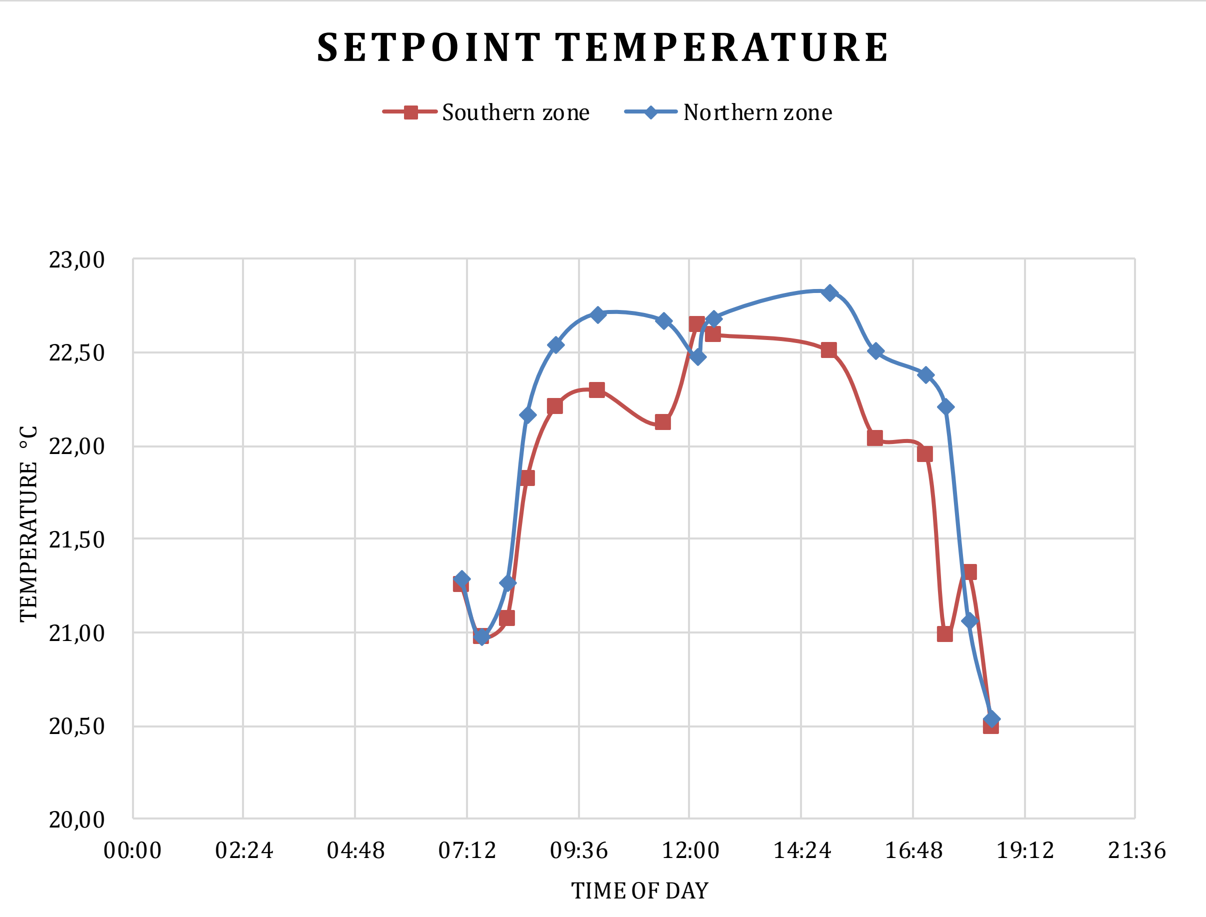


**Figure 30 Setpoint temperature a weekday based on counting of people and presence in each room**, "**Error! eference source not found.**"

Figure 31 are showing the heat release from people in the two zones analyzed. It is used 123,9W/person, assuming that people have sedentary activity. The background for this number is an activity level of 1,2 met, which correspond to 70W/m$^2$ human surface area. A Scandinavian person have approximately 1,77 m$^2$surface area.[13]

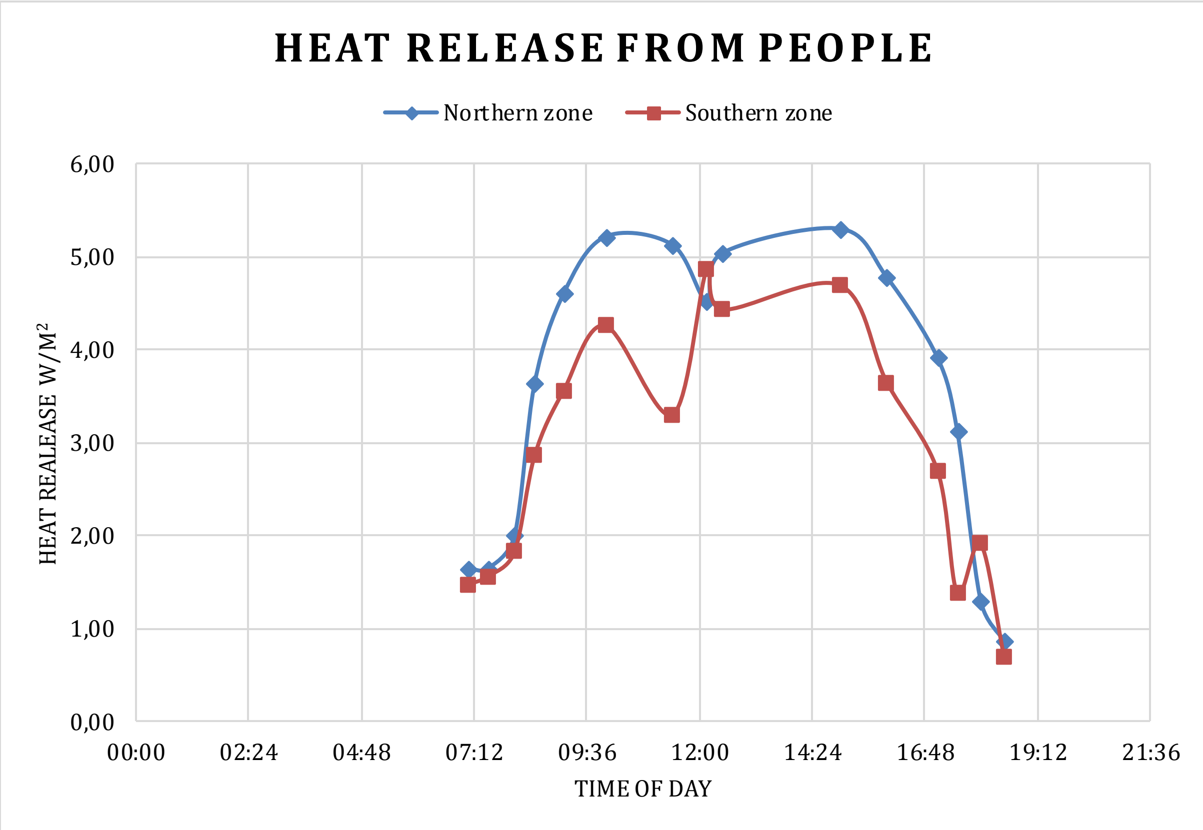


**Figure 31 Heat release from people based on counting of people,** "Error! Reference source not found."

Heat release from people and the indoor setpoint temperature obviously follows the same pattern, which makes sense, as the area is mostly offices. Lunch break for 2nd and half 3rd floor outside the building zone causes the lowering just after 11 o'clock, companies in other floors have own lunch spaces within the zone.

**Lights and technical equipment**
There is assumed that all energy from lights and technical equipment become heat. Technical equipment could be an element of uncertainty; one lose some energy though the dishwasher for example. Anyway, compared to the total energy use the amount is extremely low, almost all energy use by technical equipment in an office building comes from computers etc.

Figure 32 shows that the energy use by lights and technical equipment is quite similar every day. For this reason, Figure 33, Figure 34, Figure 35, Figure 36 and Figure 37 take a closer look at two days, Tuesday and Wednesday.

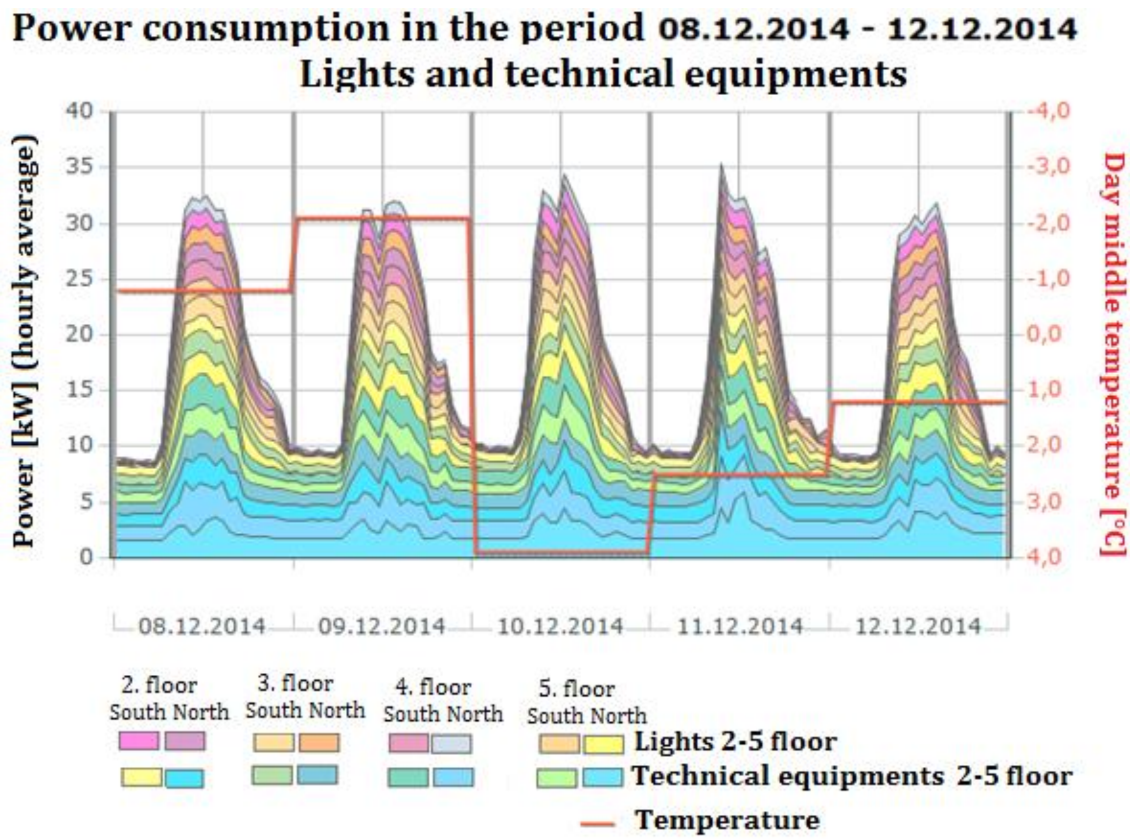


**Figure 32 Power use by lights and technical equipment week 50 in 2014, northern and southern zones [45]**

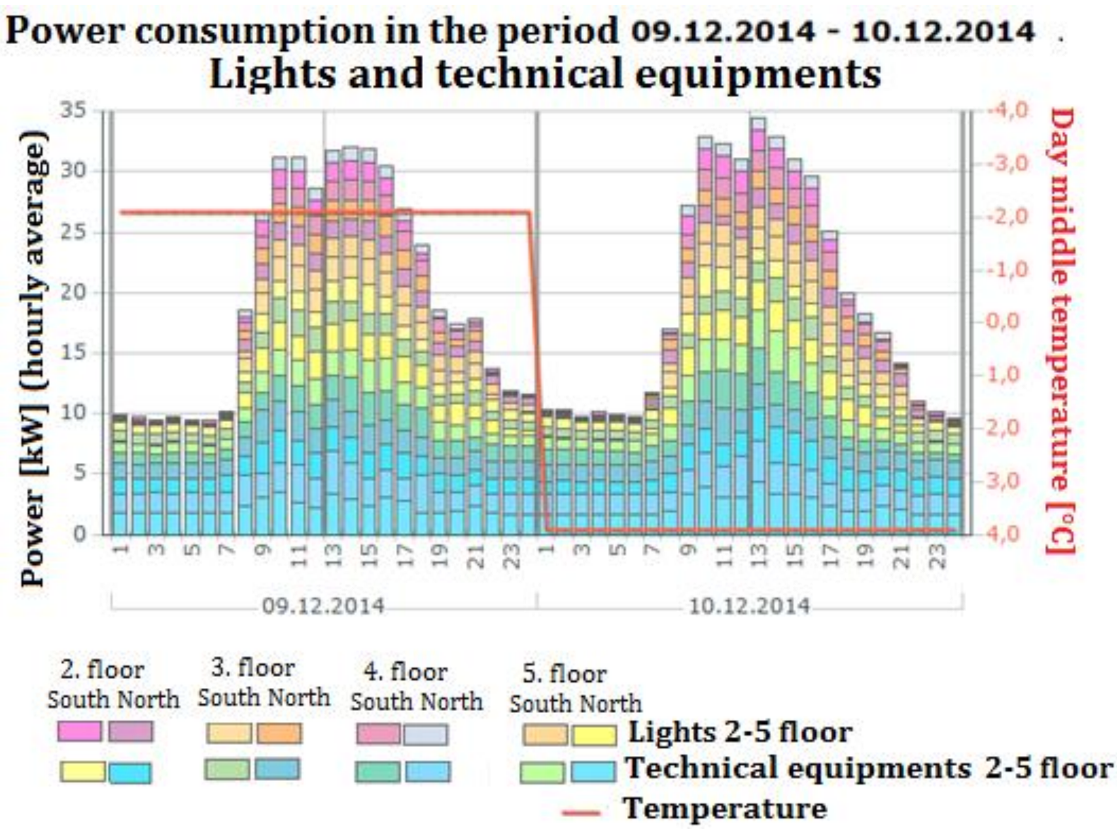


**Figure 33 Power use by lights and technical equipment Tuesday and Wednesday week 50, 2014, northern and southern zones [45]**

***LIGHTS***

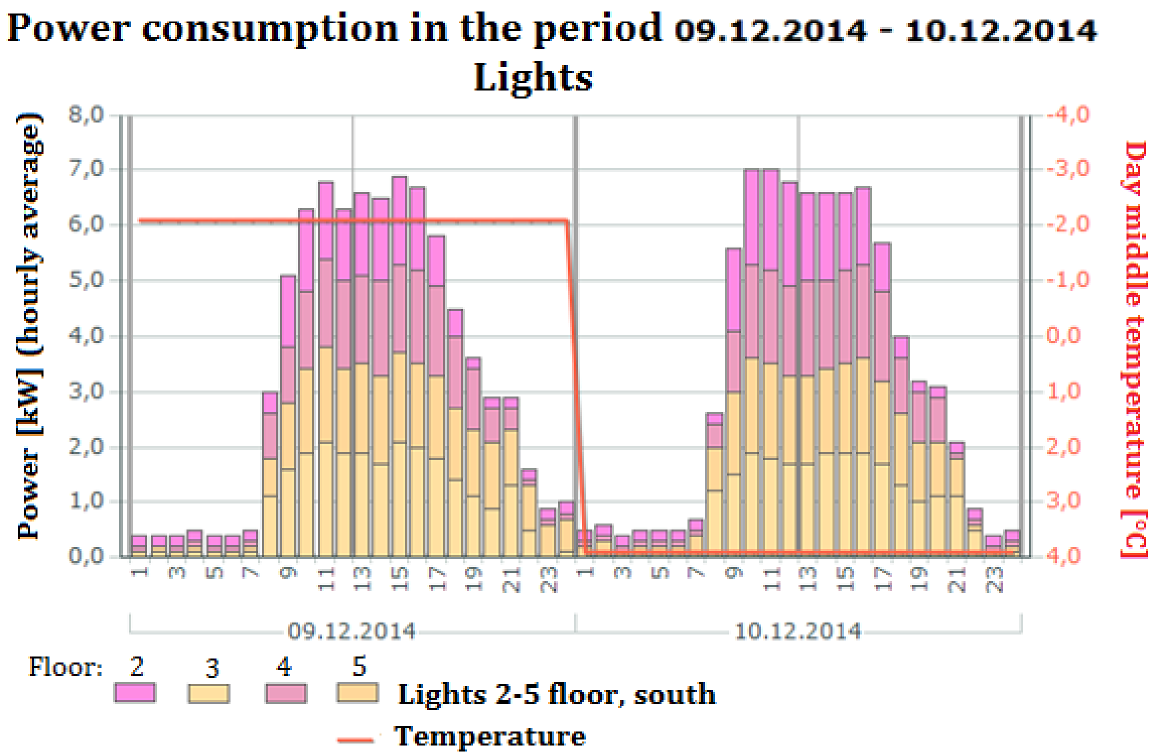


Figure 34 Power use of lights Tuesday and Wednesday week 50, 2014. Southern zone[45]

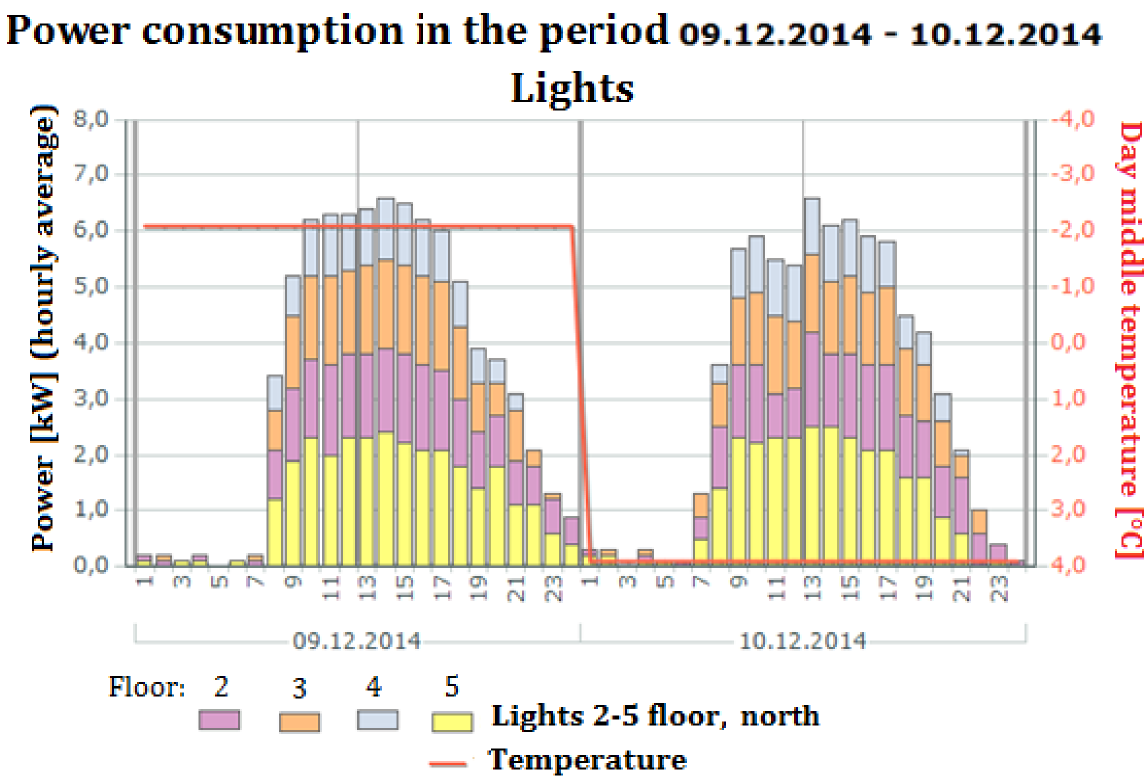


Figure 35 Power use of lights Tuesday and Wednesday week 50, 2014. Northern zone[45]

Figure 34 and Figure 35 shows the actual power use by lightning. The shape of the curves are quite similar. Table 15 shows the input values due to power use by lightning used in SIMIEN.

Table 15 Power use by lightning at different timespans, used in SIMIEN simulations

| Timespan | Power use northern zone | | Power use southern zone | |
|---|---|---|---|---|
| 07:00-08:30 | 4 kW | 2,8 W/m² | 4 kW | 2,8 W/m² |
| 08:30-17:00 | 6 kW | 4,2 W/m² | 6 kW | 4,2 W/m² |
| 17:00-21:00 | 3,5 kW | 2,5 W/m² | 3 kW | 2,1 W/m² |
| 21:00-07:00 | 0,2 kW | 0,14 W/m² | 0,5 kW | 0,35 W/m² |

## *Technical equipment*

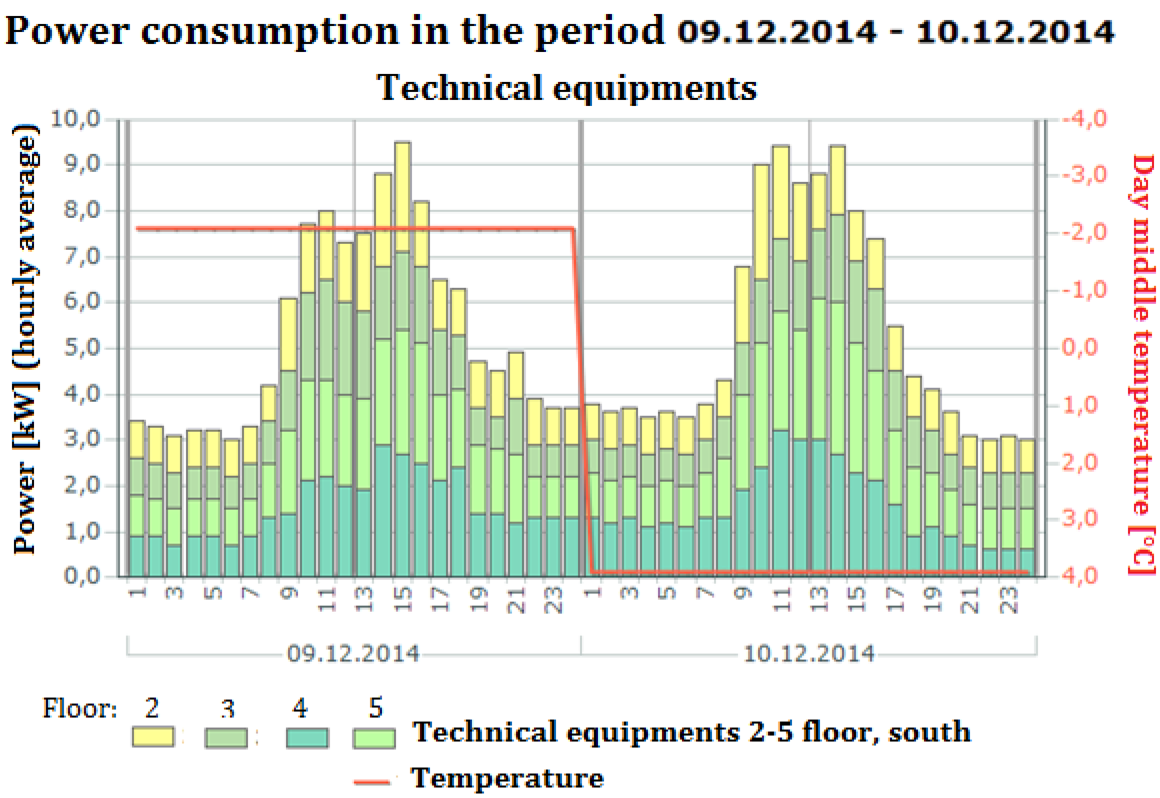


Figure 36 Power use of technical equipment Tuesday and Wednesday week 50, 2014. Southern zone [45]

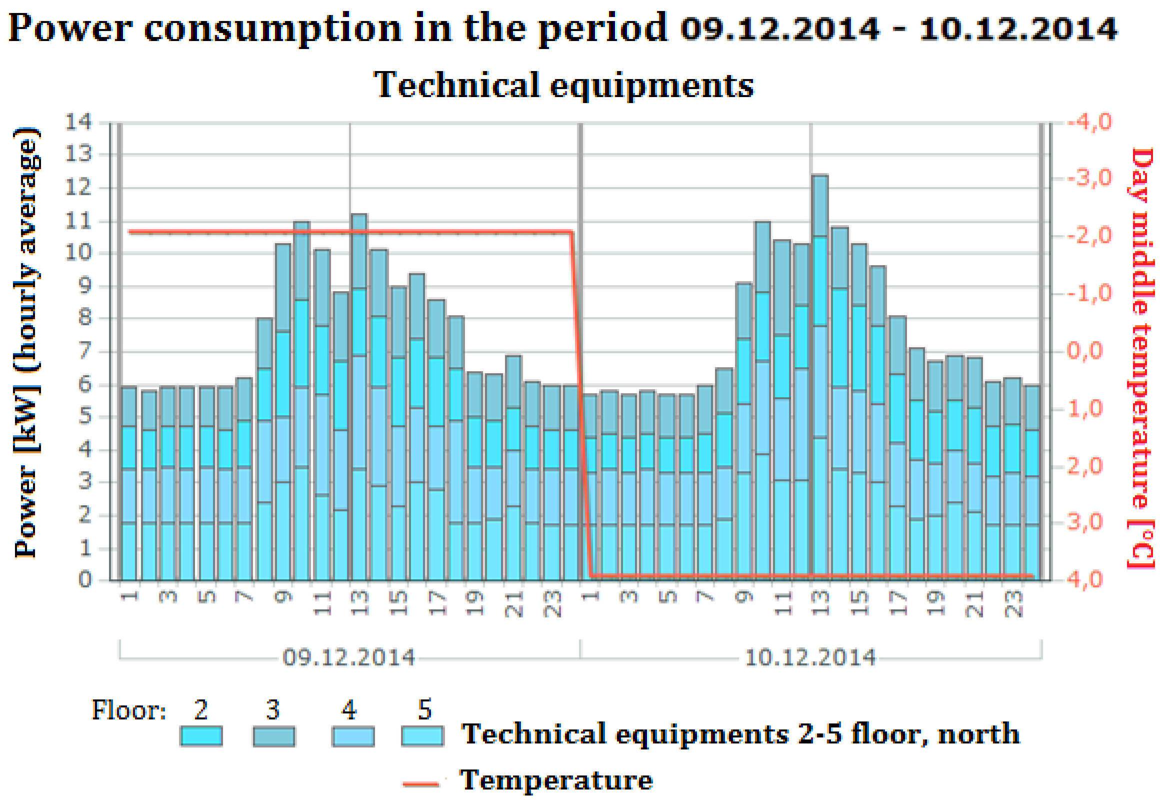


Figure 37 Power use of technical equipment Tuesday and Wednesday week 50, 2014. Northern zone [45]

Figure 36 and Figure 37 shows the actual power use by technical equipment. One can notice that the power use during the night is almost double at the northern zone than on the opposite side of the building. Most likely more computers are still on during the night. Table 16 shows the input values due to power use by technical equipment used in SIMIEN simulations.

Table 16 Power use of technical equipment at different timespans, used in SIMIEN simulations

| Timespan | Power use northern zone | | Power use southern zone | |
|---|---|---|---|---|
| 07:00-09:00 | 8,5 kW | 6 W/m$^2$ | 5,5 kW | 3,9 W/m$^2$ |
| 09:00-16:00 | 10 kW | 7 W/m$^2$ | 8 kW | 5,6 W/m$^2$ |
| 16:00-18:00 | 8 kW | 5,6 W/m$^2$ | 5,5 kW | 3,9 W/m$^2$ |
| 18:00-07:00 | 6,3 kW | 4,4 W/m$^2$ | 3,5 kW | 2,5 W/m$^2$ |

## ***Domestic hot water***

Large modifications were needed to be able to measure domestic hot water; this was not economical to do. It theory one could use the total energy consumption and subtract all other energy use. The problem in this case is that the hot water central delivers to both this building and another building. Therefore, it is assumed domestic hot water use according to the passive house standard instead.

### 5.3.3 Comparison of simulated, measured and theoretical calculated power demand

The comparison of simulated and measured power demand reveals weaknesses in the simulation, which need to be changed or adjusted manually. This is discussed further in this chapter. Results for the coldest day in the measured week, Tuesday, is most relevant. Using “winter simulation” in SIMIEN it is not possible to use exact temperature values for outdoor temperature. One define the middle temperature for the simulated day and the amplitude of the temperature difference during the day, with a specified time for maximum temperature.

**Simulated power use Tuesday week 50:**

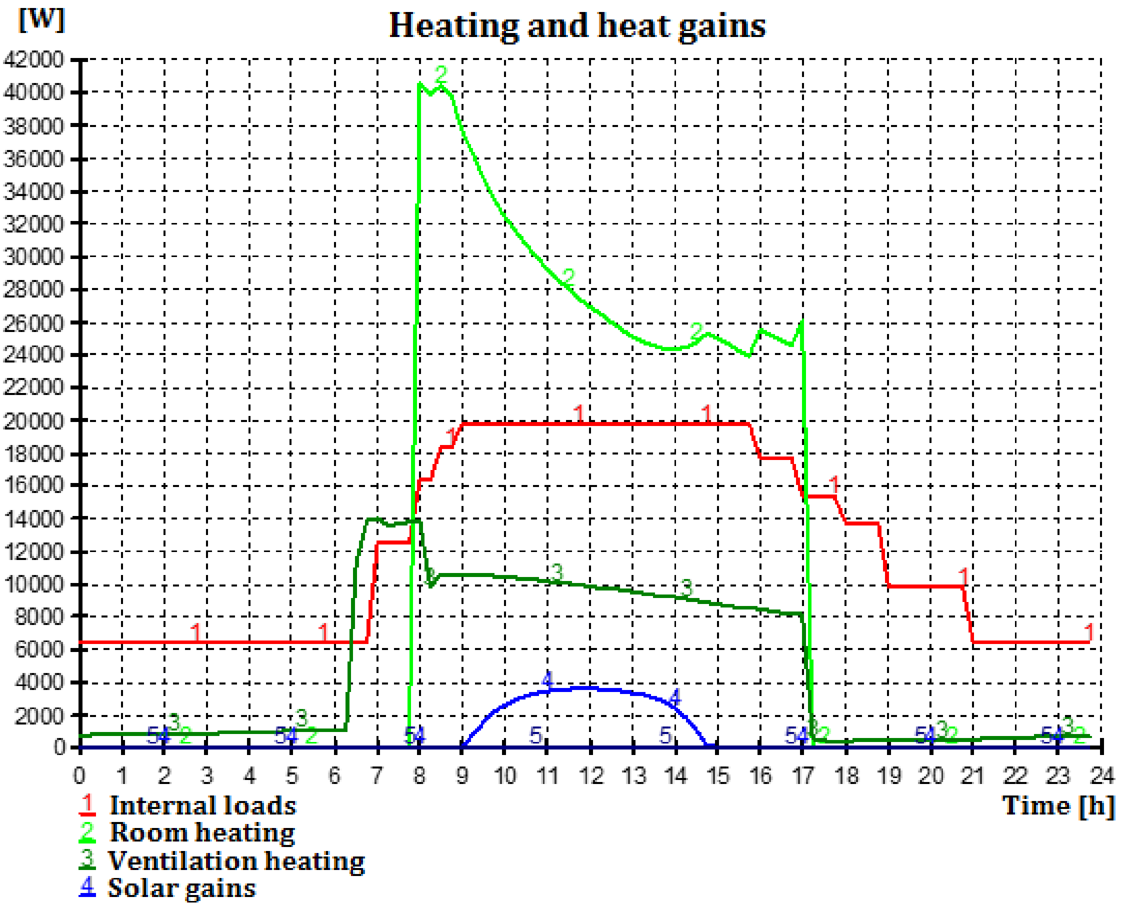


Figure 38 Simulated power use Tuesday week 50, northern zone, unbalanced ventilation

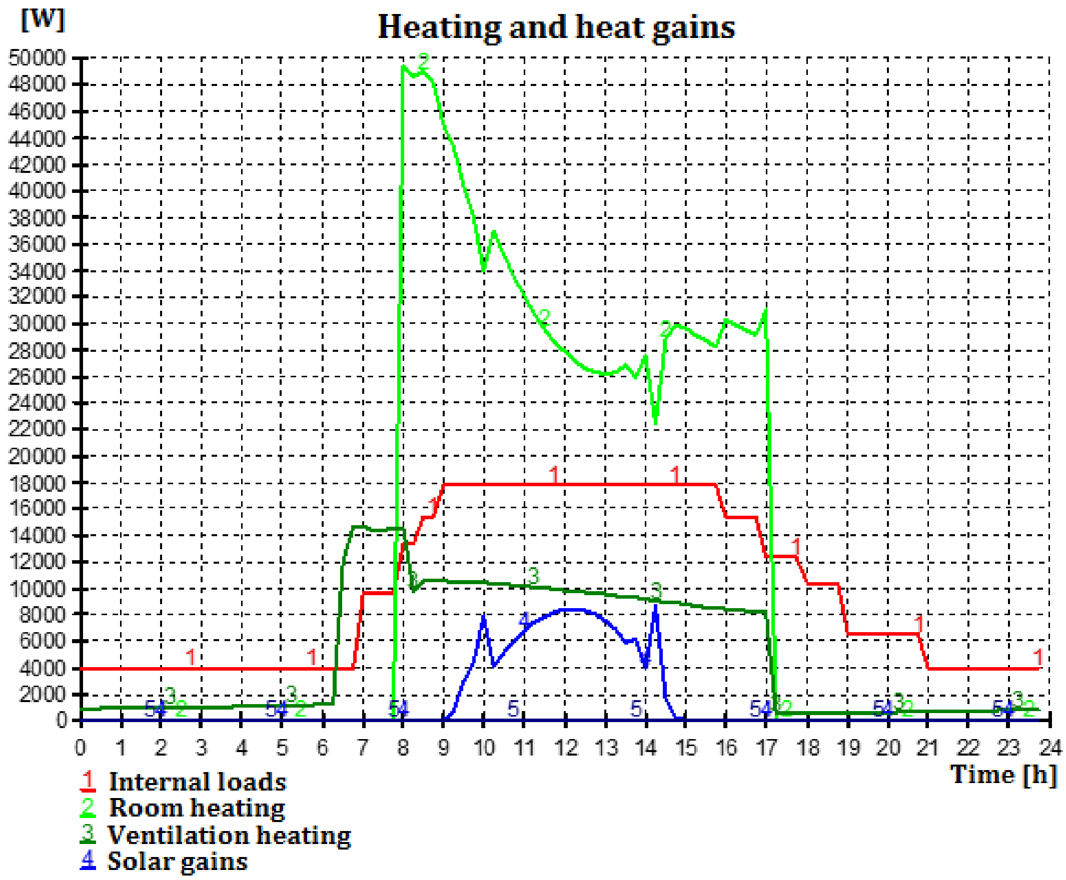


Figure 39 Simulated power use Tuesday week 50, southern zone, unbalanced ventilation

## The ventilation heating coil

The heating coil in the ventilation system serves both zones analyzed. Therefore, one must add the simulated power use from both zones to compare. Figure 40 shows measured and simulated power demand for ventilation heating Tuesday in week 50.

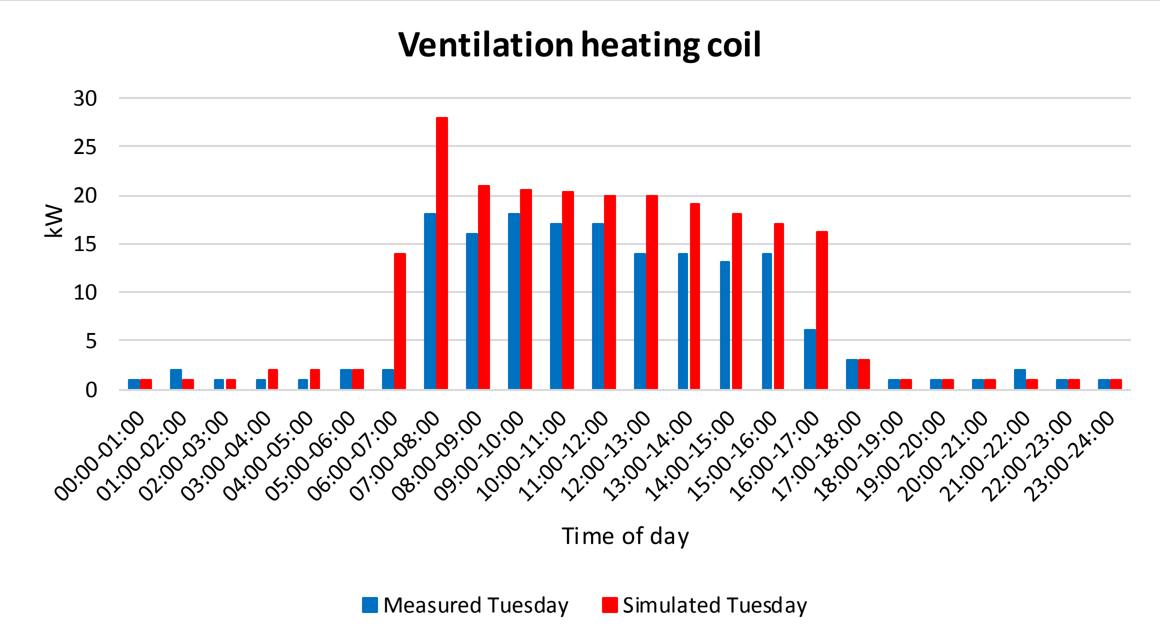


Figure 40 Measured and simulated power use in the ventilation heating coil Tuesday week 50

The measured power use from the heating coil in the ventilation system have quite similar shape as the simulated, except from 06:00-07:00 and from 16:00-17:00, where simulated values are way higher. Neither of these two timespans gives maximum power demand and would therefore not matter looking at maximum power demand. Maximum measured power demand in the ventilation coil occur between 07:00 and 12:00, and is relatively constant. One can notice that the simulated value is not as constant, with a top from 07:00-08:00. This is because SIMIEN do not allow gradually change of the indoor setpoint temperature. One can see how this affect the simulation in Figure 38 and Figure 39. The ventilation power demand suddenly falls at the same moment as the indoor setpoint temperature goes from 20,3°C to 22,5°C, at 8 o'clock.

In theory, one can explain that the power demand in the ventilation drops with higher indoor temperature by looking at the supply air temperature after it has passed through the heat exchanger. The air temperature after the heat exchanger depends on the indoor temperature, outdoor temperature, airflow and the efficiency of the heat exchanger. The heating coil must heat the supply air the remaining degrees up to the supply air setpoint temperature, in this case 18,5°C. Between 07:00-08:30 measurements shows that outdoor temperature and efficiency of the heat exchanger where more or less constant. The indoor setpiont temperature gradually increased from 20,3°C to 22,5°C and the airflow gradually increase from 18 500m$^3$/h to 19 250m$^3$/h. . According to *NS-EN 12831:2003* one can find the theoretical power demand by the formula:

$$Power = C_p * \rho * \dot{Q} * \Delta T$$

In this case, specific heat capacity ($C_p$) and density (ρ) are constant because of a small ΔT from the outlet of the heat exchanger to obtain the setpoint temperature of supply air. Using heat exchanger efficiency of 72% ($n_{hx}$) and an outdoor temperature of -3,5°C, theoretical ΔT is found as follows:

$$\Delta T = T_{supply} - (T_{indoor} - T_{outdoor}) * n_{hx} - T_{outdoor}$$

**Table 17 Theoretical calculated power use and ΔT of the ventilation heating coil**

| Ventilation condition | ΔT | Power |
|---|---|---|
| As simulated from 07:00-08:00:<br>$T_{indoor}$=20,3°C ($T_{outdoor}$= -3,5°C $T_{supply}$=18,5°C)<br>$\dot{Q}$=18 500 $\frac{m^3}{h}$ ($n_{hx}$=72% $C_p$=1,01 $\frac{kJ}{kg*K}$ ρ=1,22 $\frac{kg}{m^3}$) | 4,9 K | 30,8 kW |
| As simulated from 08:00-09:00:<br>$T_{indoor}$=22,5°C ($T_{outdoor}$= -3,5°C $T_{supply}$=18,5°C)<br>$\dot{Q}$=19 250 $\frac{m^3}{h}$ ($n_{hx}$=72% $C_p$=1,01 $\frac{kJ}{kg*K}$ ρ=1,22 $\frac{kg}{m^3}$) | 3,3 K | 21,6 kW |

Theoretical calculated values is a little higher than the simulated values, but quite similar.

The simulated and theoretical values are generally higher than measured. This could indicate that it would be appropriate to implement some heat release from the ventilation system. Theoretically it could be calculated that the heat/power release to the building from the ventilation system with an airflow of 20 000 $m^3/h$ is 4,8 kW, 63% of the calculated power demand to run the ventilation fans (Appendix I – Calculation of heat release from the ventilation system). This correspond very well with the average difference between measured and simulated (or calculated) values. The theoretical calculation of heat release cause of friction in the ventilation system seems to be a good estimate. Figure 41 shows graphically the relation between measured and simulated power demand adjusted cause of heat release from the ventilation system.

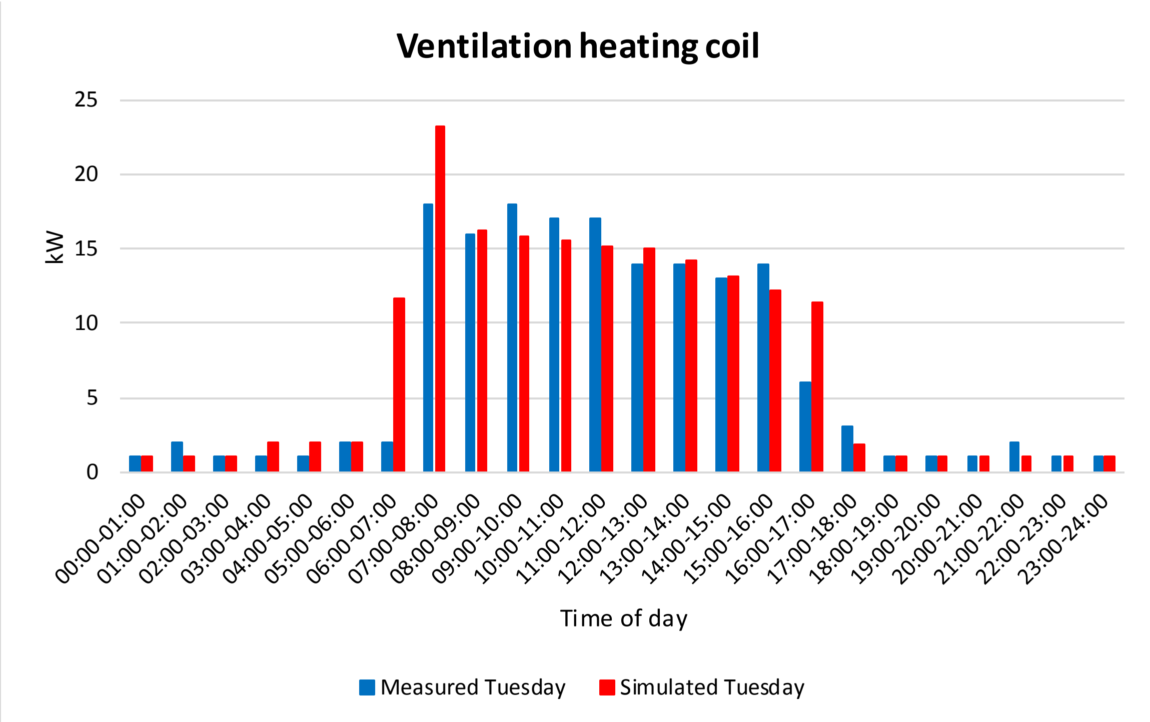


**Figure 41 Measured and simulated power use in the ventilation heating coil Tuesday week 50, adjusted for heat release**

To support this statement, Figure 42 shows the electrical power use of the ventilation system. Some of this power goes to other purposes, as pumping water in the heating coil, but most of it goes to drive the ventilation system. Using a heat (power) release of 63% to the building, as theoretically calculated, the heat release could actually be even higher depending on how much of the power going to other purposes. Anyway, the estimate of 4,8 kW usable power release seems like a cautious assumption compared to the actual power use by the ventilation system.

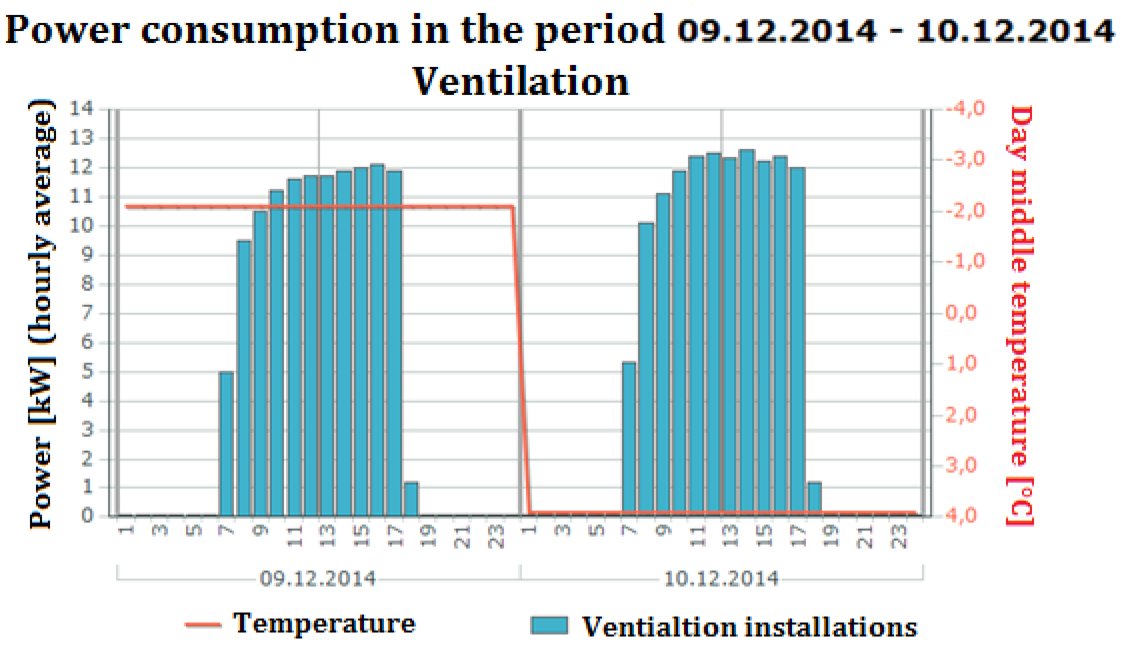


Figure 42 Power (electricity) consumption by the ventilation installations for the zones investigated[45]

## The room heating system

Figure 43 are showing power use for room heating in both zones in week 50. Obviously, power use and outdoor temperature are very dependent on each other. The shape of the graphs are quite equal day by day. Hence, in further investigation it is looked upon the coldest day, Tuesday.

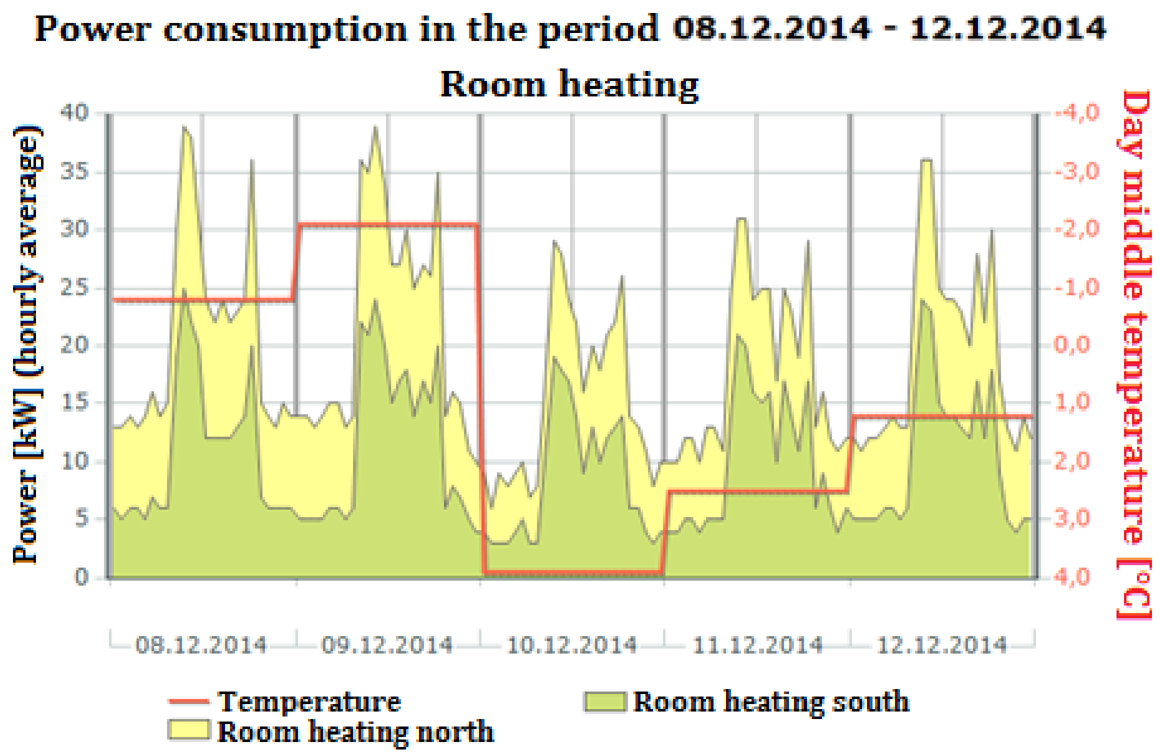


Figure 43 Room heating in both zones during week 50 and daily average temperature[45]

The SIMIEN simulation shown in Figure 38 and Figure 39 shows no need for heat during the night. Room heating is needed the moment the indoor setpoint temperature is adjusted from 20,3°C to 22,5°C (Colder outdoor temperatures will give some need from the moment the ventilation switch to operational airflows). The heating up capacity is huge the first hour in the simulation. In reality, the indoor setpoint temperature gradually changes as people come to work, so the peak should not be quite as high as the simulation indicate. To make the simulation more real, power (heat) demand is distributed gradually from about 07:00-08:30. It is not possible in SIMIEN, but an estimation is illustrated in Figure 48 and is discussed more in the following sections. The timespan to cover the heating up capacity influences the peak in power demand a lot.

Real measurements shown in Figure 44 and Figure 45 have quite similar shape as the simulated graphs (Figure 38 and Figure 39), although there are some differences between the zones. The northern zone have smaller differences in power demand over the day. The zones are connected and mainly analysis are based on the sum of both zones. The values of the measured power demand for room heating on the other hand, is quite different from the simulated!

- From 08:00 – 17:15, the simulated power use is about double of the measured.
- There is no use of power from 17:15-08:00 in the simulation, but about 14 kW measured

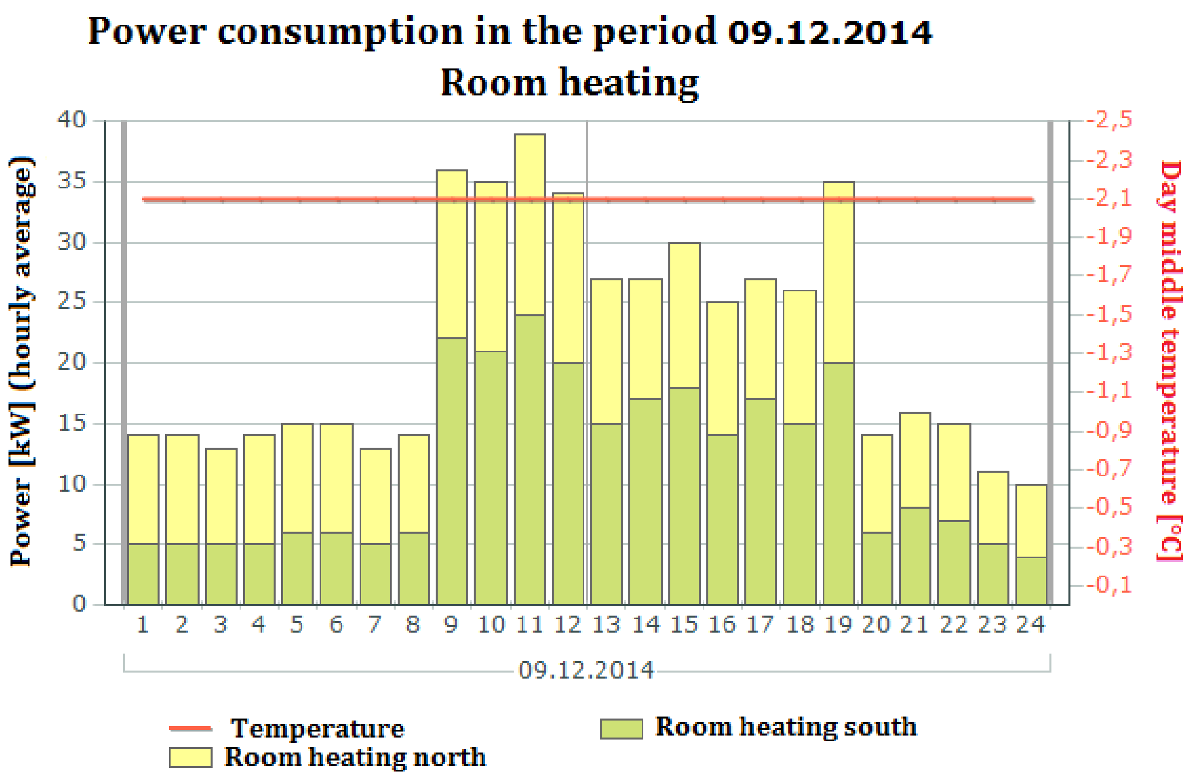


Figure 44 Power use for room heating both zones Tuesday week 50[45]

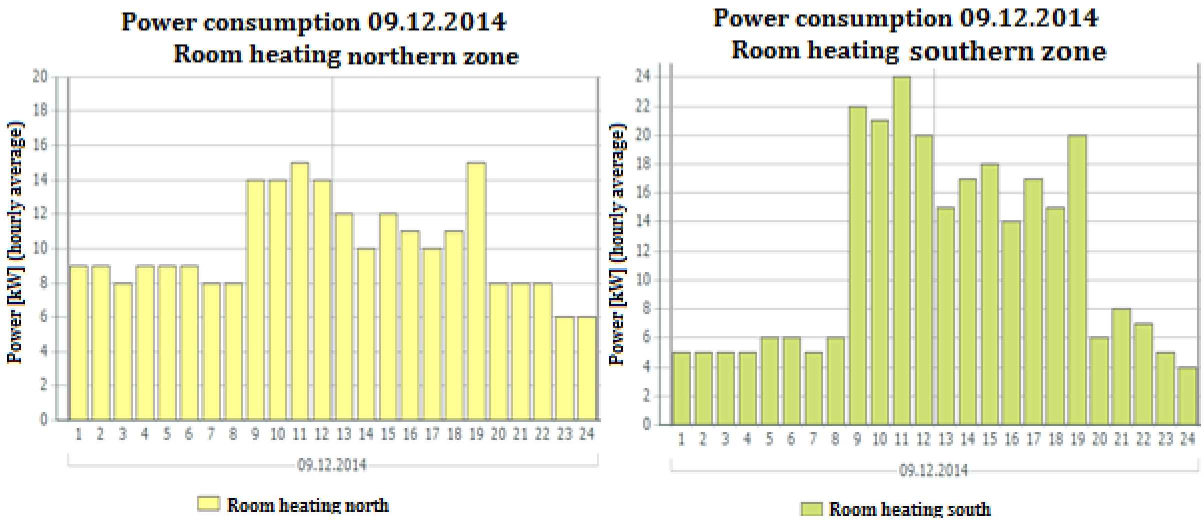


Figure 45 Power use for room heating for each zone Tuesday week 50[45]

Some input variables must have a fatal error and there are many sources of error. The unsure input values could be the grade of insulation, natural infiltration and infiltration caused by unbalanced ventilation. The error between measured and simulated power demand is quite huge and it is unlikely caused by insulation alone. Use of materials with a lot better U-values than projected is very unlikely. Heat loss cause of natural infiltration is supposed to be low, and even dropping natural infiltration would not reduce the simulated power demand enough. Infiltration because of unbalanced ventilation on the other hand causes a huge amount of extra infiltration and is interesting to investigate.

### Changing to balanced ventilation

Unbalanced ventilation were not a part of the plan. In principle, the exhaust airflow is pressure controlled, and should follow the supply airflow. Simulated results become more reasonable by implementing balanced ventilation, shown in Figure 46 & Figure 47.

Uncertainty in the measuring device could be the source of error in this case. It is well known that measuring devices for airflow could have uncertainties quite big, depending on the measuring method. 13% more exhaust airflow than supply, which were measured in our case during operational hours could all come from uncertainty in the measuring devices. In further analysis, it is assumed balanced ventilation.

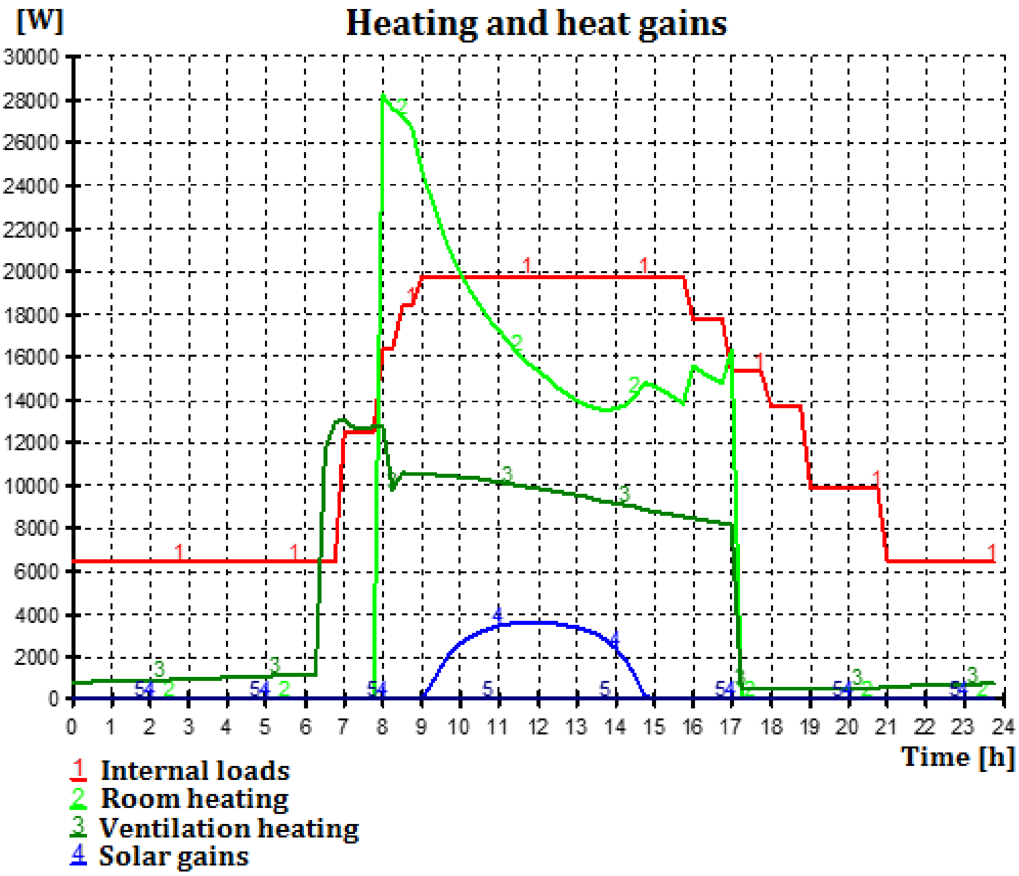

Figure 46 Tuesday week 50 northern zone, assuming balanced ventilation

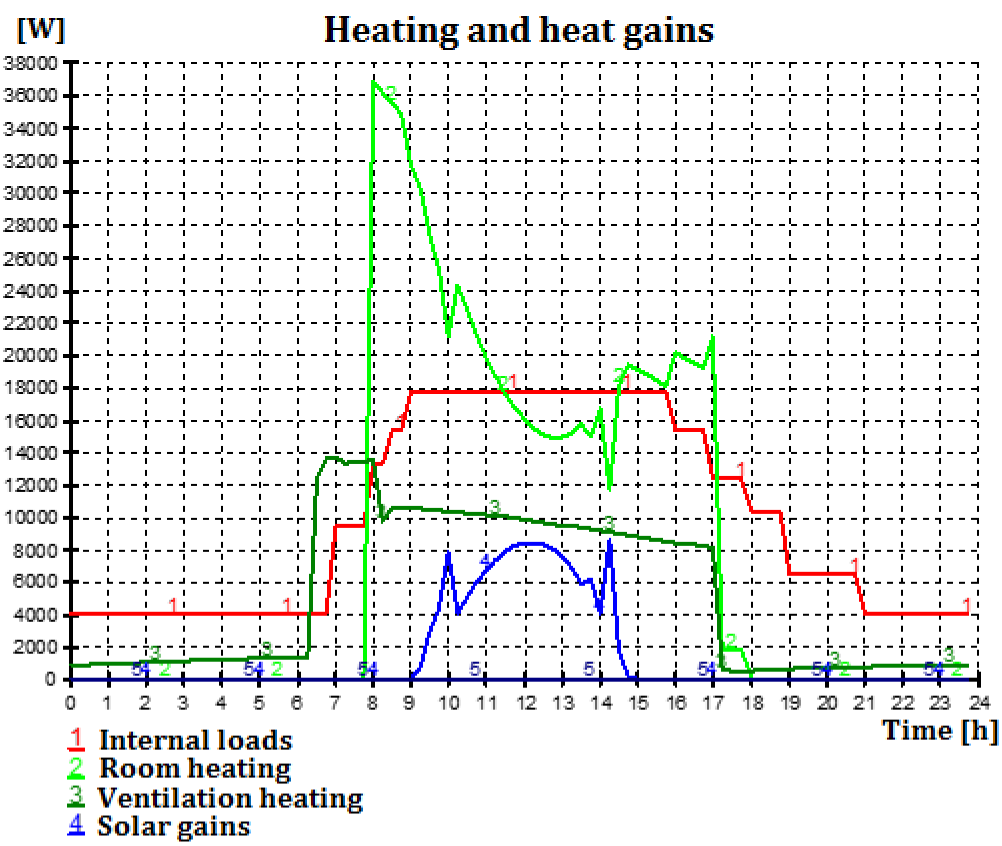

Figure 47 Tuesday week 50 southern zone, assuming balanced ventilation

Figure 48 shows measured and simulated power use for room heating as a sum of both zones investigated. There is no simple explanation of why the graphs does not match completely. The green graph is modified to be more like reality, where people arrive gradually and thereby the average indoor setpoint temperature gradually increase s.

The measured power demand increase at 08:00-09:00. It is a little strange that it do not start to increase at least one hour before. In reality, ventilation airflow starts at 06:30 and the indoor setpoint temperature increase as people arrive gradually from about 07:00 to 08:30. This pattern is no coincidence, but applies every day in week 50. It seems like the measurements is misplaced by one hour.

Even if the measured, blue graph is moved by one hour to the left, there would be high measured power demand from 17:00-18:00 (18:00-19:00 on Figure 48), where simulated power demand very low. This is harder to explain. As Figure 30 shows, the indoor setpoint temperature is high until 18:00, but the ventilation turns to a minimum from 17:15. The simulation results strongly depends on the amount of fresh air that must be heated from 18,5°C to 22,5°C.

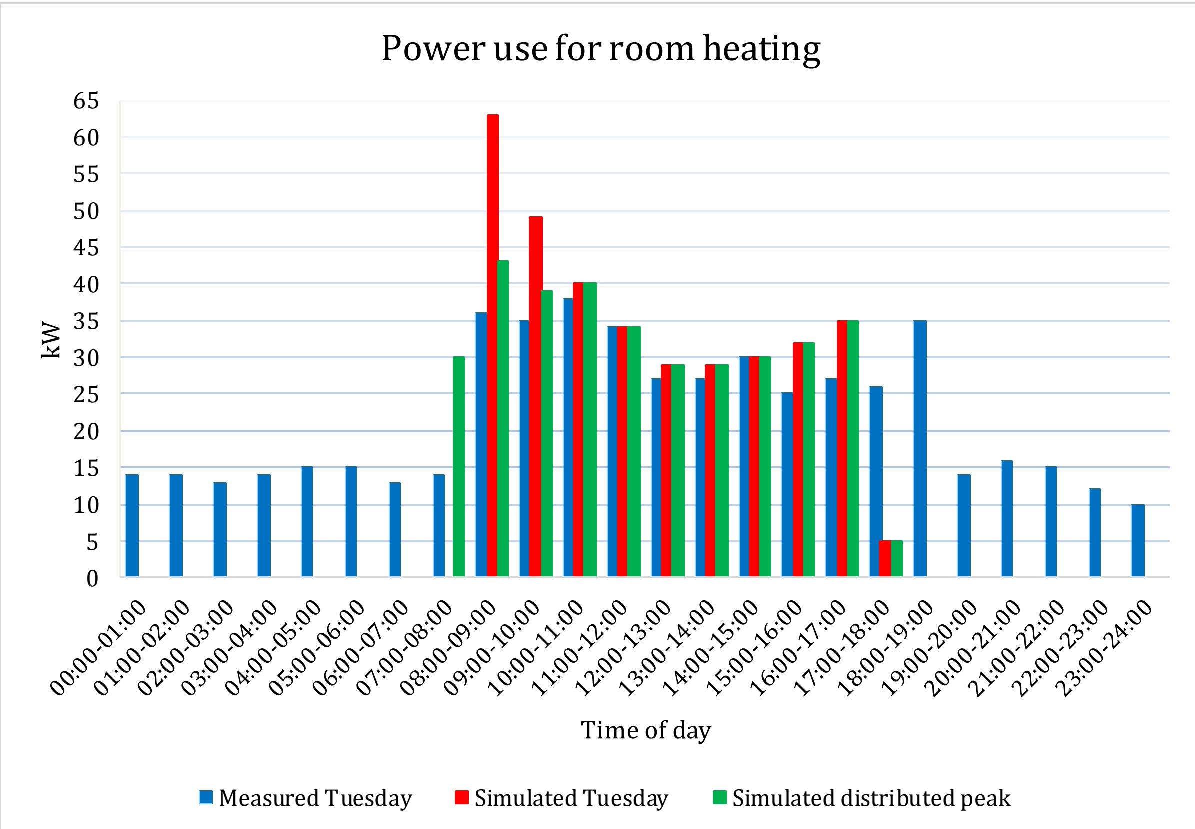


Figure 48 Power use for room heating as a sum of both zones investigated

As seen in Figure 48, maximum simulated power demand is higher than measured, also when using the more realistic green graph with distributed peak load. A reason is that one in SIMEN must choose, for every zone, if there is one or more than one facade exposed to wind. Only using two zones, both have more than one facade exposed to wind. Dividing into many zones, only the corner rooms have more than one façade exposed to wind. This influence the infiltration rate and give higher power demand than in the simulation. Table 18 shows theoretical calculated power demand based on *NS-EN 12831:2003*, using both that the zones have two facades exposed to wind and a division in smaller zones, where only 26% is corner room with two facades exposed to wind. 26% is found from drawings of the building. One can't compare this table directly to the graph in Figure 48 as this do not take into account the heating up capacity and are not extracting internal load. The difference because of extra infiltration by choosing only two zones instead of dividing into many zones on the other hand, will be the equal. It turns out that one theoretically should extract 7,1 kW from the simulated power demand. Then the simulated graph with distributed peak gets a little lower than the measured graph, shown in Figure 49. Many factors could cause these small differences, the insulation may be poorer or the building may not be sealed as well as projected. Adjusted simulated maximum power demand is almost equal to the measured though.

**Table 18 Theoretical calculated power demand for room heating, Tuesday week 50**

| Power demand room heating, Tuesday week 50 | | | |
|---|---|---|---|
| Part of the building | North [kW] | South [kW] | Total [kW] |
| Rooms with one façade exposed to wind (74%) | 18,2 | 18,3 | 36,5 |
| Rooms with two facades exposed to wind (26%) | 7,6 | 7,7 | 15,3 |
| Sum of the rooms | 25,9 | 25,9 | 51,8 |
| Zones with two facades exposed to wind | 29,4 | 29,5 | 58,9 |
| Difference | 3,6 | 3,6 | 7,1 |

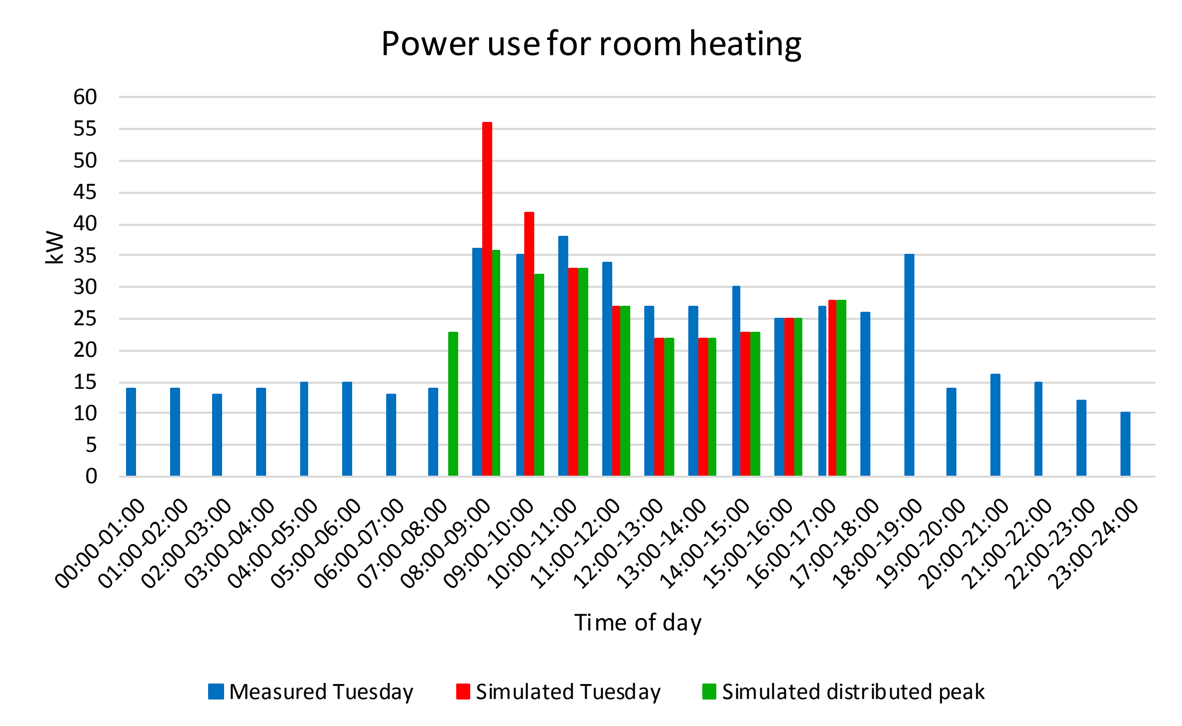


**Figure 49 Power use for room heating Tuesday week 50, simulations adjusted according to Table 18**

Why it is measured power demand outside operational hours (during night) is hard to explain. The SIMIEN simulation gives no power demand outside operation hours even if the outdoor temperature is -20°C as shown in Figure 53 and Figure 54. One possible answer is that there is some circulation of hot water in the room heating system, working as a frost protection of the system.

The assumption of balanced ventilation even though the measurements shows unbalanced, reduces the power demand both during operation and outside operational hours. One could think that this was the reason, but Figure 38 and Figure 39, which include unbalanced ventilation, shows that the SIMIEN simulation still do not need room heating outside operating hours, but during operation, power need will as discussed earlier almost double. As Table 14 shows, extra infiltration cause of unbalanced ventilation during operational hours is much bigger than outside operational hours. Balanced ventilation still seems like a correct assumption.

An important reason why the SIMIEN simulation shows no power demand outside operating hours is the heating capacity of the building during the setback of indoor temperature. As shown in Figure 50, the temperature do not reach 20,3°C on average at any time outside operating hours.

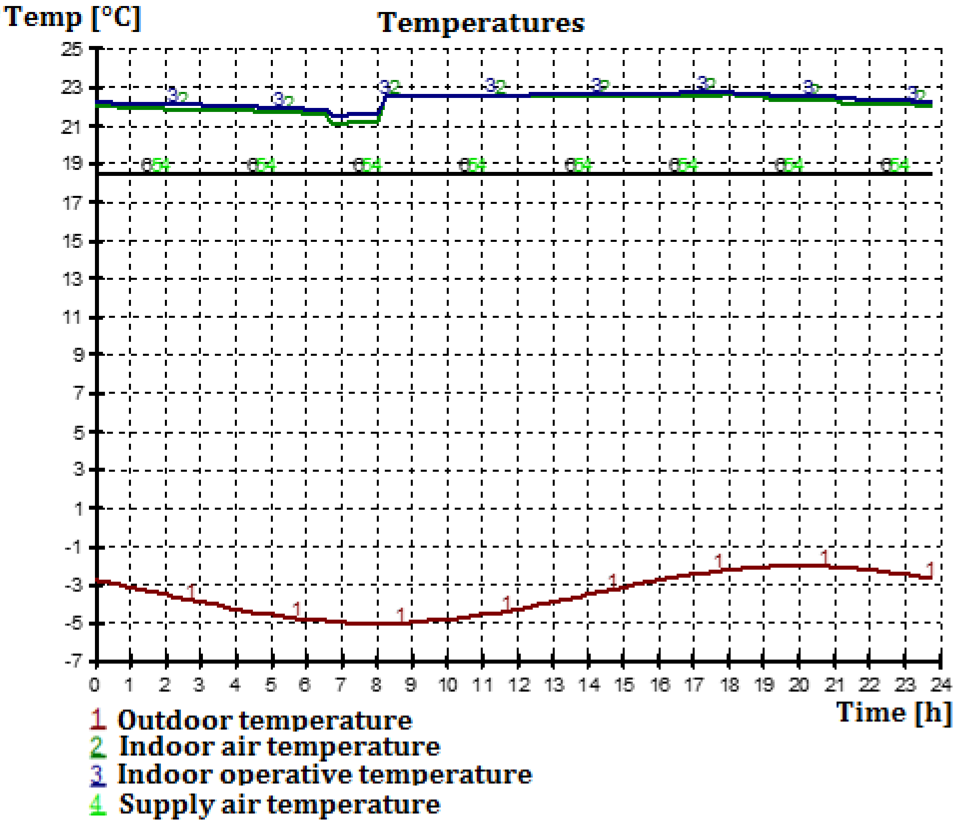

Figure 50 Simulated indoor temperature, northern zone Tuesday week 50

If the real building had less heating capacity than projected, it could explain the need of more power during the night. If that were the case, maximum power demand for room heating would be smaller. This because maximum power demand occur in the morning where the heating up capacity contributes a lot. Figure 48 indicates that this is not the case; the maximum power demand of the adjusted simulation is almost equal to the measured, which indicates that the heating capacity is correct. Anyway, this should be tested using a simpler control of the building, where the indoor setpoint temperature is not changed gradually. The adjusted graph is a rough estimate including the effect of gradually change of indoor temperature.

A reasonable explanation for some power demand during the night is that the power demand may not be equally distributed in every room, which is assumed in the SIMIEN simulation, which only use two zones. Rooms in corners and with big surfaces against outer walls will need more heat than average rooms. The temperature may drop below seitpoint temperature in these rooms and the radiators must switch on. With doors closed, even more difference between rooms will occur. Some rooms may not have internal load during the night, while other rooms have more. As the outdoor temperature was not as cold as wanted the days of measuring the indoor temperature do not necessarily have time to fall down to the temperature setpoint outside operational hours in rooms with more internal load than average. These rooms will again need less power at the start of the day, explaining low power use measured from 07:00-08:00. To test if this hypothesis could be the case, one will need measurements that are more accurate and preferably on room level. This could also be a part of the explanation why measured energy use is higher than simulated, average temperature in the building gets higher, which leads to more heat loss.

Even though the measured total energy use is higher than the simulated, design power demand matches quite well. During operational hours, the measured and simulated power demand and thereby energy use is quite similar. As this master thesis investigates design power demand, one assume that the simulation is reliable during operational hours.

## Total power demand

Total power demand is in this case the sum of the power demand used in the ventilation system and for room heating, not including domestic hot water. Figure 51 shows total power demand Tuesday in week 50, both simulated and measured. The simulated power demand for room heating is adjusted according to Table 18. The simulated power demand by the ventilation heating coil is adjusted according to heat release in the ventilation system. To simplify, it is assumed 4,8 kW constant reduction due to heat release during operational time. Notice that the measured values are smoother and the peak correspond best with the green graph, distributing the peak for room heating cause of gradually temperature change (as discussed earlier).

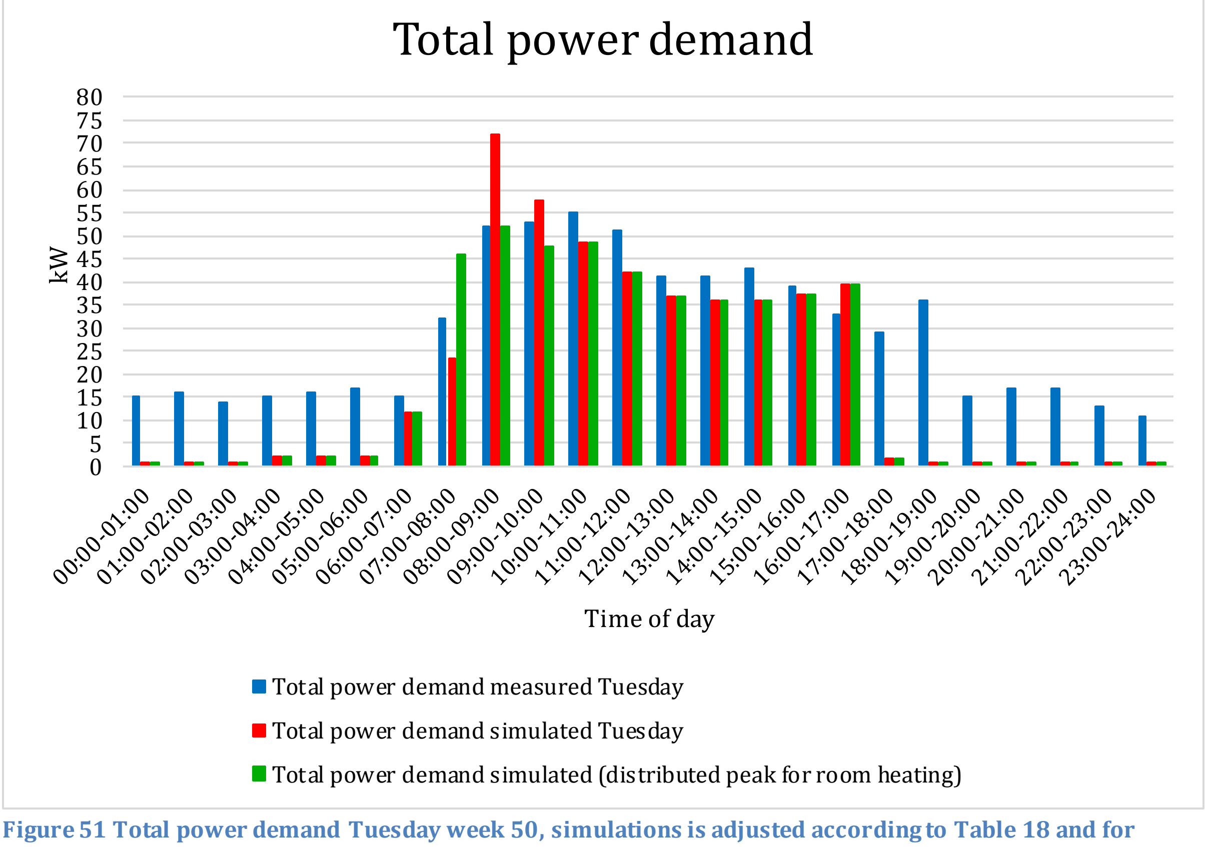


**Figure 51 Total power demand Tuesday week 50, simulations is adjusted according to Table 18 and for ventilation heat release**

### Theoretical calculated power demand week 50 compared to simulated and measured

Table 19 shows theoretical calculated power demand comparable to measurements from the actual building and the simulated values in Figure 49 and Figure 51. Maximum power demand occurs about 08:00 in the simulations. Measured maximum power demand is almost constant from 08:00 to 12:00. The internal load contribute more as more people arrive and theoretical heating up capacity gets huge the first hours after changing the indoor setpoit temperature. As the goal is to find a method to calculate the theoretical maximum power demand, values measured in the timespan 08:00-09:00 are used as input to Table 19.

Theoretical calculations assumes steady state conditions. It is no surprise that the theoretical calculated values matches the measured and simulated quite well in the ventilation system. Heating capacity in the ventilation system is small, there is less other uncertainties to consider, airflow and efficiency of the heat exchanger is the two most important input parameters. When designing the heating coil, one must have in mind that maximum power demand for ventilation occur from 07:00-08:00, but for a total hating system one can look at the timespan 08:00-09:00 as in Table 19.

**Table 19 Theoretical calculated power demand Tuesday week 50, based on *NS-EN 12831:2003***

| Total power demand Tuesday week 50 | | | |
|---|---|---|---|
| Part of the building | North [kW] | South[kW] | Total [kW] |
| *Total room heating demand (excl. heating up capacity)* | *25,9* | *25,9* | *51,8* |
| *- Internal load* | *16* | *14* | *30,0* |
| = Room heating (excl. heating up capacity) | 9,9 | 11,9 | 21,8 |
| Ventilation heating | 10,8 | 10,8 | 21,6 |
| Heating up capacity | | | |
| Reheat time: 1 hour, ΔT=2,2K , building mass: high | 36,2 | 36,2 | 72,3 |
| Reheat time: 2 hour, ΔT=2,2K , building mass: high | 31,6 | 31,6 | 63,2 |
| Reheat time: 3 hour, ΔT=2,2K , building mass: high | 25,6 | 25,6 | 51,3 |
| Reheat time: 4 hour, ΔT=2,2K , building mass: high | 22,8 | 22,8 | 45,6 |
| Sum | | | |
| With 1 hour reheat time | 56,8 | 58,9 | 115,7 |
| With 2 hour reheat time | 52,3 | 54,4 | 106,6 |
| With 3 hour reheat time | 46,3 | 48,4 | 94,7 |
| With 4 hour reheat time | 43,5 | 45,5 | 89,0 |

The power demand for room heating on the other hand, does not match that good to neither simulated or measured values. The indoor temperature in the SIMIEN simulation changes to the desired temperature in short time, approximately 15 min according to Figure 50. This indicate that it would be appropriate to compare the SIMIEN simulation with the theoretical values with 1 hour reheat time (actually 15 min).

Eventually one could compare the SIMIEN simulation with distributed peak with theoretical values with about 2 hours reheat time. Theoretical calculated power demand for room heating (incl. heating up capacity) is larger than the simulated and measured. The theoretical calculated values with a reheat time of 2 hours have a power demand for room heating of 21,8 kW + 63,2kW = 85kW, the simulated graph with distributed peak gives a power demand of 35,9kW. Without distributed peak, simulated power demand is 55,9kW, compared to 21,8kW + 72,3kW = 94,4kW using theoretical calculated values with 1 hour reheat time. It is a huge percentage difference between the calculation methods. The real measurements of maximum power for room heating are also lower than theoretical values, only 38kW maximum power demand. To obtain this low value for room heating using theoretical calculations, the heating up capacity must be 16,2 kW (assuming “room heating excl. heating up capacity” is correct).

Figure 49 shows that the measured power demand for room heating from 18:00-19:00 is 35 kW. At this time heating up capacity have no (negligible) effect. Figure 46 and Figure 47 shows that internal load is still about 24 kW, giving a total room heating power demand excl. heating up capacity and internal load of 59 kW. This could be directly compared to upper value in Table 19, *“Total room heating demand (excl. heating up capacity)”,* 51,8kW. Theoretical room heating excluding heating up capacity may be a little low compared to measured values. Many factors could contribute to more measured power demand for room heating. Anyways, one should have some safety margin. Making changes to the heating system after it is built could be very expensive.

Having such advanced and automatic control of the investigated building makes it hard to figure out which parameters that contributes to the difference from measured values. Making some tests using easier control, such as no night setback, and make changes at a certain times, not gradually, could eliminate some sources of error.

The adjusted SIMIEN simulation with distributed peak and less rooms having facades exposed to wind, shown in both Figure 49 and Figure 51, fits quite well with measured maximal power demand. The theoretical calculation are based on the same formulas and should be equal to the simulated values <u>if there were no heating up capacity</u>. Figure 52 shows a SIMIEN simulation using 22,5°C as constant indoor temperature to exclude heating up capacity, internal loads is also excluded. The northern and southern zone both have a power demand of about 32kW each, 64 kW total, minus 7,1kW according to Table 18 (one or more facades exposed to wind) gives a total room heating power demand of 56,9kW. Theoretical calculated value given in Table 19, 51,8kW, is quite a similar power demand. As room heating excluding heating up capacity seems to match quite well, it seems like the theoretical heating up capacity become too high in this case.

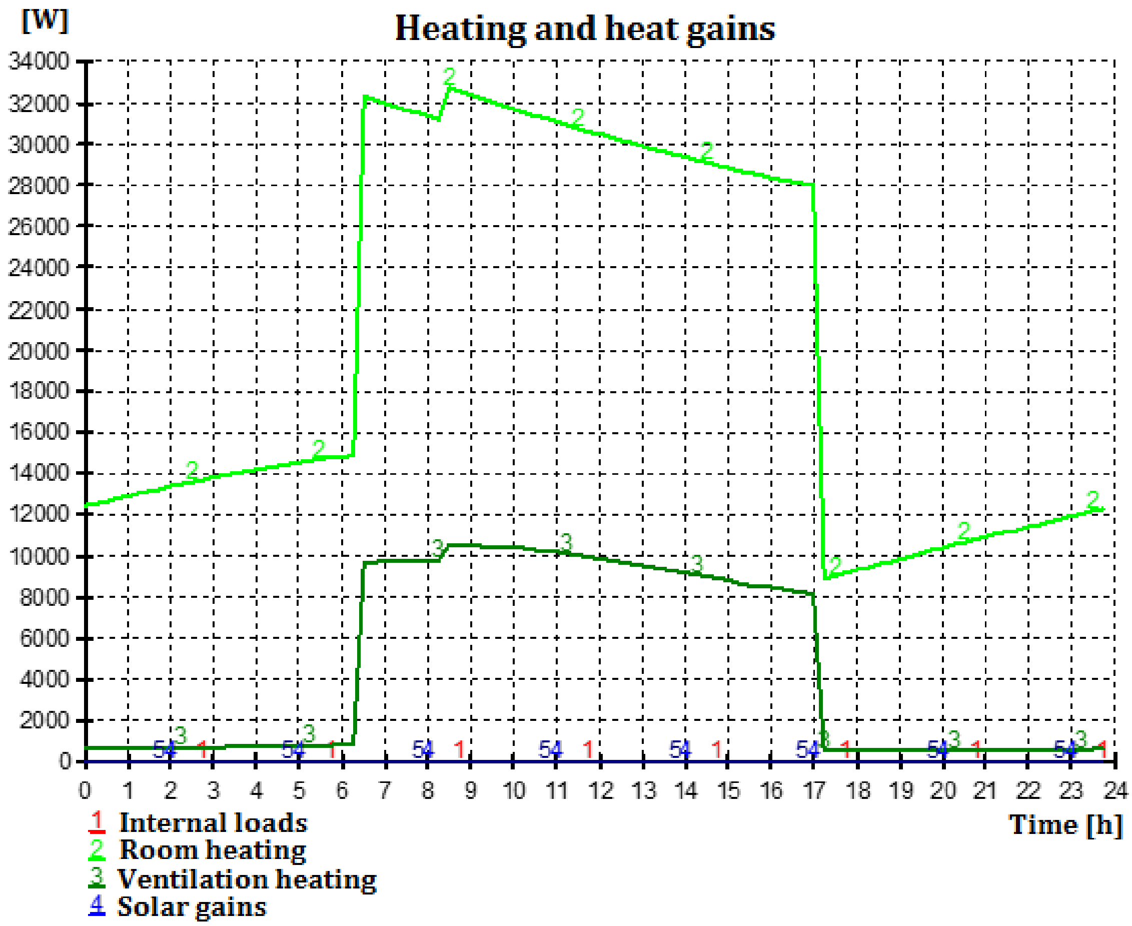


Figure 52 SIMIEN simulation using constant indoor temperature of 22,5°C and excluding internal loads and solar gains. Equal for the northern and the southern zone

## Design power demand

Looking at design power demand, one must remember that the supply air temperature changes according to the outdoor temperature according to the compensation curve given in Figure 26. Using a design outdoor temperature of -20 °C, the supply air temperature should be 21,5°C. This kind of control will as seen in Figure 53 and Figure 54 lead to higher power demand in the ventilation and less power demand for room heating, a relocation of power demand actually. This is probably an economical favorable control. One even larger heater is in most cases cheaper than lager ducts and radiators in every room. It is important when using high supply air temperature that the ventilation system distribute the supply air in a good way. High supply air temperature could lead to “short circuiting” (that supply air move along the ceiling, directly from supply to exhaust), especially with displacement ventilation.

One have to design the room heating system to manage maximum power demand, which according to the simulations is not when the outdoor temperature is -20°C because of the compensation curve and the relocation of power from room heating to the ventilation heating coil.

The ventilation heating system on the other hand have design power demand when the outdoor temperature is -20°C. The total power demand is also maximum, e.g. total design power demand occur at design conditions.

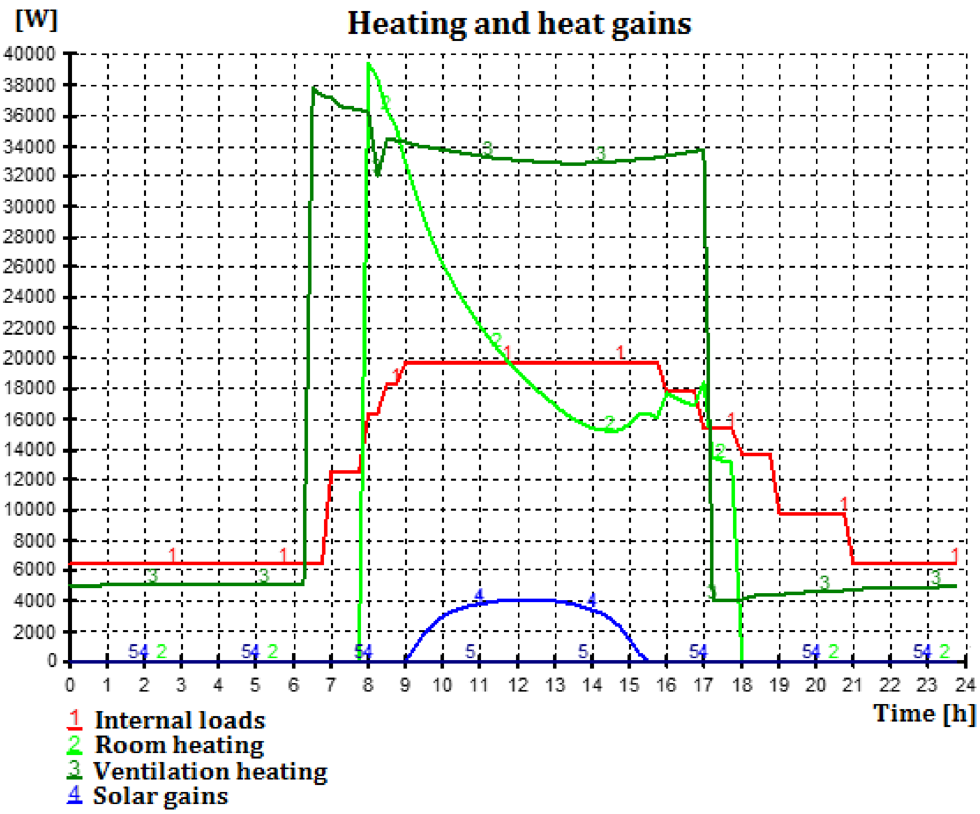


Figure 53 Simulated power use in the northern zone, outdoor temperature of -20°C

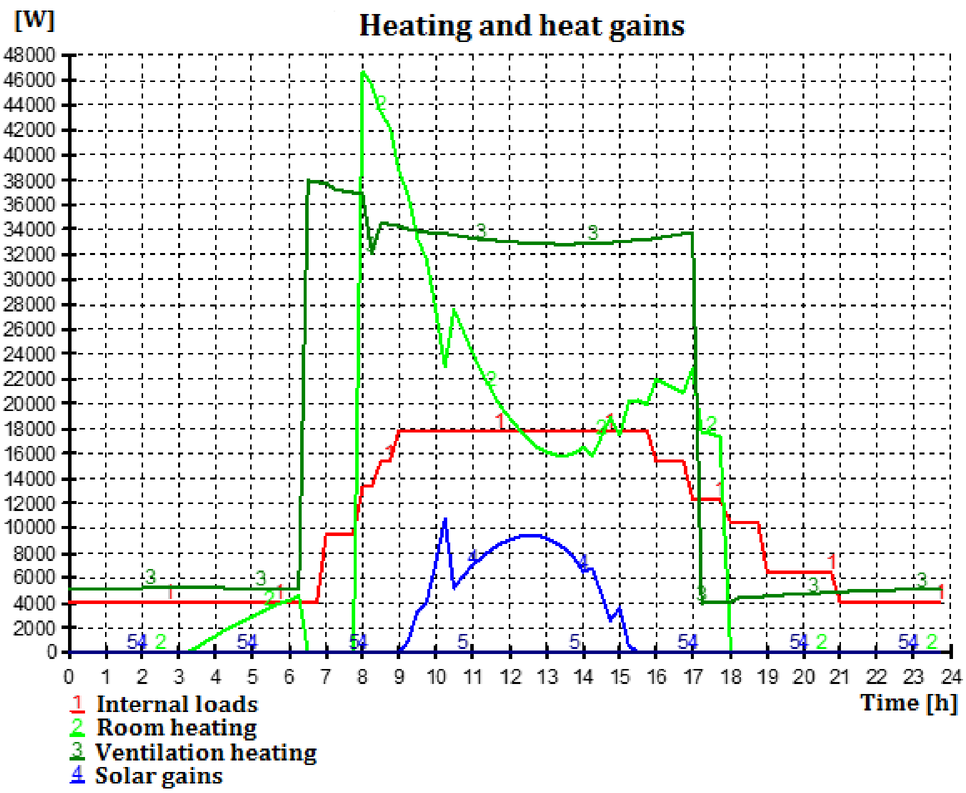


Figure 54 Simulated power use in the southern zone, outdoor temperature of -20°C

### Theoretical calculated design power demand for room heating

The design power demand for room heating given in Table 20 is the theoretical calculated power demand based on *NS-EN 12831:2003*. The difference from Table 18 is that the outdoor temperature used is -20°C and the supply air temperature is 21,5°C instead of 18,5°C. It turns out that the difference between that all zones have more than one facade exposed to wind and that only the corners (26%) have more than one façade exposed to wind is 11,9kW. Dividing the building in smaller parts is common to be able to pick the right sized radiators for each room. The value of "Sum of the rooms" from Table 20 are used in the theoretical calculations of total power demand, in Table 21. Table 21 also include 10% safety margin, which is common to use in practice. [46]

Table 20 Design power demand for room heating, theoretical calculated

| Design power demand room heating (using predicted real conditions, as simulated) | | | |
|---|---|---|---|
| Part of the building | North [kW] | South [kW] | Total [kW] |
| Rooms with one facade exposed to wind (74%) | 17,3 | 17,3 | 34,6 |
| Rooms with two facades exposed to wind (26%) | 8,1 | 8,2 | 16,3 |
| Sum of the rooms | 25,4 | 25,5 | 50,9 |
| Zones with two facades exposed to wind | 31,3 | 31,5 | 62,8 |
| Differance | 5,9 | 5,9 | 11,9 |

### Theoretical design power demand for ventilation heating – choice of airflow rate

Calculating the theoretical power demand for ventilation, shown in Table 21, one use the same formulas as described earlier in this chapter. Outdoor temperature and supply air temperature is of course changed to -20°C and 21,5°C. Another difference is that the ventilation system are running at maximum airflow, 26 000 m$^3$/h. This only fits with reality if the $CO_2$ concentration demands more airflow. During the measured week, the airflow rate never reached this level (Figure 25). It was not that cold outside in the measured week either, which means that the extra ventilation demand measured might as well be due to cooling demand. Using maximum ventilation airflow to estimate the power demand causes oversizing, which might not be necessary. There is a question of the necessity to cover unexpected user behavior and have a safety margin. Using only necessary estimated airflow rate, in our case 17 800$\frac{m^3}{h}$ (6,3$\frac{m^3}{m^2\,h}$) (assuming no extra airflow demand given by the $CO_2$ sensors), one <u>may</u> allow poorer indoor air quality if there is unexpected user behavior days with design conditions. There is probably a good idea to oversize somewhat as a safety margin, but using maximum airflow seems like an exaggeration.

### Theoretical heating up capacity

The theoretical heating up capacity in Table 21 is calculated based on *NS-EN 12831:2003* given in Table 3.

### Total design power demand

Total design power demand is the maximum power demand needed from both room heating and ventilation heating. In the SIMIEN model the total design power demand occurs 08:00 and depends a lot on the room heating part as this varies the most.

Total design power demand found in the SIMIEN simulation using steady state conditions were 162 kW (the sum of room heating and ventilation heating from 08:00 - 09:00 in Figure 53 and Figure 54, domestic hot water is not included). In this timespan, the internal load is about 30 kW.

Making a "yearly" SIMIEN simulation with actual climate data for the last year, but edit one day to -20°C constant, the design power demand become 145,5 kW (not including domestic hot water). The same power demand actually occurs using a yearly simulation with only some few hour with -20°C. SIMIEN take into account heat storage in the building and sudden changes and peaks in outdoor temperature would not influence too much. Because of high building mass, it is performed a simulation with 3 and 7 days with -20°C. The design power demand became 148 kW and 174,5 kW respectively.

**Table 21 Theoretical total design power demand (excl. domestic hot water)**

| Total design power demand (maximal airflow: 26000m3/h and safety margins) | | | |
|---|---|---|---|
| Part of the building | North [kW] | South [kW] | Total [kW] |
| Room heating plus 10% safety margin (excl. heating up capacity) | 29,2 | 29,3 | 58,5 |
| Ventilation heating (using max airflow of 26000 m3/h) | 48,5 | 48,5 | 97,0 |
| Heating up capacity | | | |
| Reheat time: 1 hour, ΔT=2,2K , building mass: high | 36,2 | 36,2 | 72,3 |
| Reheat time: 2 hour, ΔT=2,2K , building mass: high | 31,6 | 31,6 | 63,2 |
| Reheat time: 3 hour, ΔT=2,2K , building mass: high | 25,6 | 25,6 | 51,3 |
| Reheat time: 4 hour, ΔT=2,2K , building mass: high | 22,8 | 22,8 | 45,6 |
| Sum | | | |
| With 1 hour reheat time | 113,9 | 114,0 | 227,9 |
| With 2 hour reheat time | 109,3 | 109,5 | 218,8 |
| With 3 hour reheat time | 103,3 | 103,5 | 206,8 |
| With 4 hour reheat time | 100,5 | 100,6 | 201,1 |

The yearly SIMIEN simulation with 7 days of -20°C gives a result somewhat higher than the winter simulation with steady state conditions. The calculation method in SIMIEN is somewhat different in the two simulations. Theoretical calculated power demand according to *NS-EN 12831:2003* shown in Table 21 gives even higher results. *NS-EN 12831:2003* does actually not demand 10% safety margin, although it is common in practice.

Comparing the SIMIEN simulations with theoretical calculated values one must use the same conditions, predicted real design conditions. As there was no real measurements at design conditions, a SIMIEN simulation based on measurements week 50 is used as "predicted measurements". All adjustments discussed in this chapter is extracted from the design power demand of the SIMIEN simulation. These are reduction cause of heat release from the ventilation system, gradually change of indoor setpoint temperature and less area with more than one façade exposed to wind. One must then extract internal load from the theoretical calculation to compare results. Total design power demand in Table 22 are comparable to adjusted steady state results from SIMIEN.

The SIMIEN simulation actually gets more realistic by distributing the peak from 07:00-08:30. Indoor setpoint temperature changes gradually as people arrive gradually. This decrease the heating up capacity, eventually increase the reheat time. For this building, the reduction is approximately 30 kW based on distributing the peak load as done in Figure 48. One could also assume that the distributed peak correspond to 1,5 hours extra reheat time using theoretical calculation. Although that would not give the same reduction, it is the most reasonable approach. Since the change of temperature in the SIMIEN simulation happens quickly (about 15 min), one could say that the theoretical calculation should have approximately 2 hours reheat time comparing to the adjusted SIMIEN simulation. Table 23 shows a comparison of total design power demand using different methods.

**Table 22 Total design power demand, using simulation conditions (excl. domestic hot water)**

| Total design power demand, (using predicted real conditions, as simulated) | | | |
|---|---|---|---|
| Part of the building | North [kW] | South [kW] | Total [kW] |
| *Total room heating demand* | *25,4* | *25,5* | *50,9* |
| *- Internal load* | *16* | *14* | *30,0* |
| Room heating (excl. heating up capacity, airflow of 19250m3/h) | 9,4 | 11,5 | 20,9 |
| Ventilation heating (same airflow as simulated, 19250m3/h) | 35,9 | 35,9 | 71,8 |
| Heating up capacity | | | |
| Reheat time: 1 hour , ΔT=2,2K , building mass: high | 36,2 | 36,2 | 72,3 |
| Reheat time: 2 hour , ΔT=2,2K , building mass: high | 31,6 | 31,6 | 63,2 |
| Reheat time: 3 hour , ΔT=2,2K , building mass: high | 25,6 | 25,6 | 51,3 |
| Reheat time: 4 hour , ΔT=2,2K , building mass: high | 22,8 | 22,8 | 45,6 |
| Sum | | | |
| With 1 hour reheat time | 81,5 | 83,6 | 165,1 |
| With 2 hour reheat time | 76,9 | 79,0 | 156,0 |
| With 3 hour reheat time | 70,9 | 73,1 | 144,0 |
| With 4 hour reheat time | 68,1 | 70,2 | 138,3 |

Table 23 Comparison of total design power demand - Adjusted SIMIEN simulation vs Theoretical calculated

| Comparison of total design power demand (Adjusted SIMIEN simulation vs Theoretical calculation) | | | | | | |
|---|---|---|---|---|---|---|
| | SIMIEN simulation | | Theoretical calculation | | | |
| | | | Equal conditions as the (adjusted) SIMIEN simulation | | Incl. safety margins and max ventilation | |
| | Winter, -20°C steady state [kW] | Yearly, 7 days of -20°C [kW] | 2 hours reheat time [kW] | 4 hours reheat time [kW] | 2 hours reheat time [kW] | 4 hours reheat time [kW] |
| Total design power demand (results not adjusted) | 162 | 174,5 | 186 (Table 22 + internal load) | 168,3 (Table 22 + internal load) | **218,8** (Table 21) | **201,1** *(Table 21)* |
| ***Adjustments cause of:*** | | | | | | |
| - internal load | | | 30 | 30 | 30 | 30 |
| - Heat release from ventilation (Appendix I – Calculation of heat release from the ventilation system) | 4,8 | 4,8 | 4,8 | 4,8 | 9,9 | 9,9 |
| - gradually change of indoor setpoint temperature | 30 | 30 | Using 2h reheat time | Using 4h reheat time | Using 2h reheat time | Using 4h reheat time |
| - less area with more than one façade exposed to wind | 11,9 | 11,9 | - | - | - | - |
| **Adjusted results (adjusted simulation in SIMIEN illustrates the predicted measurements)** | **115,3** | **127,8** | **151,2** | **133,5** | **178,9** | **161,2** |

The theoretical calculation are higher than the predicted measurements found on basis of the SIMIEN simulation. 151,2 kW power demand with 2 hours reheat time. By using 4 hours reheat time instead; the theoretical power demand is almost equal the predicted measurements.

In practice, the design power demand is calculated room by room to ensure enough power in every room. Usually only rooms with outer wall gets radiators to reduce ducts length and thereby costs. Inner rooms are often corridors and toilets, rooms one are spending small continuous time. It is also common to include an extra 10 %. Then one try to standardize the order to the company selling radiators to obtain quantity discounts and reduce the possibility of incorrect installation (switching of radiator sizes between rooms etc).[46]

The heating system heats water going to the radiators, ventilation system and domestic hot water. It is usually the sum of room heating including 10% extra, ventilation heat needed for maximal airflow and a relatively constant power demand for domestic hot water (small in an office building).

Theoretical calculated total design power demand including safety margins and maximal ventilation airflow is compared to the adjusted SIMIEN simulation and the theoretical calculation corresponding to the adjusted SIMIEN simulation in Table 23. According to *NS-EN 12831:2003* one should not extract internal load or heat release from the ventilation system. Then the theoretical calculated design power demand become 218,8 kW using 2 hours reheat time and 201,1 kW using 4 hours reheat time. This is about 67 kW higher than the theoretical calculated design power demand using equal conditions as in the adjusted SIMIEN simulation, extracting internal load and heat release in the ventilation (30kW + 4,8kW). The difference become even higher compared to the adjusted SIMIEN simulation. Comparing to the theoretical calculation using the same conditions as in the simulation, but not extracting internal load or heat release in the ventilation system, the difference is about 32kW.

Even by extracting internal load and heat release from the ventilation system, all the theoretical calculated results of design power demand are higher than the simulated. One cannot be sure that the simulated values would be equal measured under same conditions, design conditions. Nevertheless, the results of the coldest day measured (Tuesday week 50) also indicates the same tendencies, the theoretical total heating demand gets too high. Ventilation heating power demand seems to match quite good extracting heat release from the ventilation system, but the room heating gets too high. It seems like the theoretical heating up capacity does not match as there is a gradually change of indoor temperature, which may not give the same result as using longer theoretical reheat time. It seems fear to extract internal load from the power demand on building level, but at room level one may be more careful as internal load may not occur equally distributed. As knowing exactly how well a building actually is insulated could be hard, one should have a safety margin. Knowing exactly how much internal load there is could also be challenging. Tuesday in week 50 one get equal power demand by extracting most of the internal load measured.

In the winter, sun only contributes from about 09:00 to 15:00. Design total power demand in our case occur from 08:00-09:00, which indicates that solar heat gains should not be included in this building.

**Comparison to *FEBY 12***

The comparison in this chapter use -20°C as outdoor temperature, decided in the preliminary project. As discussed in chapter 5.1, *FEBY 12* give a design outdoor temperature of -18,2°C. Assuming that *FEBY 12* is the most accurate way of estimate the design outdoor temperature, the power demand in both simulations and in theoretical calculations at design conditions include a safety margin by using -20°C. Table 24 shows total design power demand using the outdoor temperature decided according to *FEBY 12*. Comparing to the design power demand using -20°C as design outdoor temperature (Table 21) shows that room heating and ventilation heating decrease by 2,1kW and 4,5kW respectively. This correspond to a 3,5% decrease in power demand for room heating and 4,6% decrease for ventilation heating. The heating up capacity on the other hand, will not change using *FEBY 12*.

**Table 24 Total design power demand using outdoor design temperature according to *FEBY 12***

| Total design power demand *FEBY 12* (design outdoor temperature: -18,2°C) (maximal airflow 26000m3/h and safety margins included) | | | |
|---|---|---|---|
| Part of the building | North [kW] | South [kW] | Total [kW] |
| Room heating plus 10% safety margin (excl. heating up capacity) | 28,2 | 28,3 | 56,5 |
| Ventilation heating (using max airflow of 26000 m3/h) | 46,3 | 46,3 | 92,5 |
| Heating up capacity | | | |
| Reheat time: 1 hour , ΔT=2,2K , building mass: high | 36,2 | 36,2 | 72,3 |
| Reheat time: 2 hour , ΔT=2,2K , building mass: high | 31,6 | 31,6 | 63,2 |
| Reheat time: 3 hour , ΔT=2,2K , building mass: high | 25,6 | 25,6 | 51,3 |
| Reheat time: 4 hour , ΔT=2,2K , building mass: high | 22,8 | 22,8 | 45,6 |
| Sum | | | |
| With 1 hour reheat time | 110,6 | 110,7 | 221,3 |
| With 2 hour reheat time | 106,0 | 106,2 | 212,2 |
| With 3 hour reheat time | 100,1 | 100,2 | 200,3 |
| With 4 hour reheat time | 97,2 | 97,3 | 194,6 |

A reduction of 6,6 kW in total using -18,2°C instead of -20°C as design outdoor temperature is not that much compared to the reduction of including internal load at design conditions, 30kW, but it is more than the 4,8kW heat release from the ventilation system. Together these factors contributes with 41,4kW, which is a lot compared to the total power demand theoretically calculated in Table 21, 200-230kW (depending on chosen reheat time). Without night setback, the percentage reduction will be even higher. Night setback leads to a heating up capacity that contribute a lot to the power demand, 45,6kW – 72,3 kW theoretically, depending on chosen reheat time.

## 5.4 Design of heating systems

The aim of this thesis is to achieve improvements in the calculation method of power demand for heating, both on room and building level. Therefore, it is important to be aware of how heating systems work and how an over- or under-sizing of the heating system affects energy and economics.

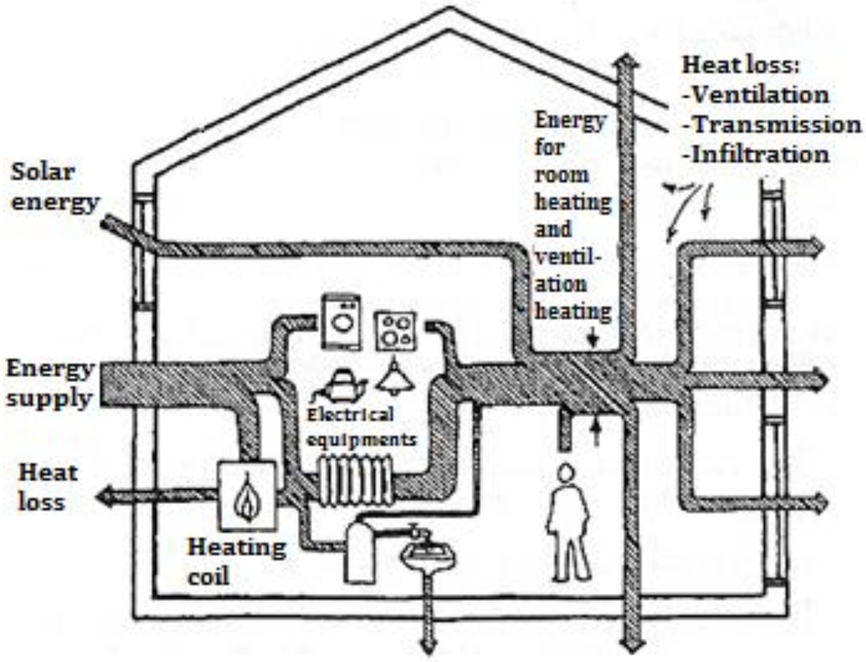


Figure 55 Example of energy flow in a building[47]

### 5.4.1 Base load and peak load

One design the base load to cover most of the annual heat demand in an energy system in the most profitable way. Typically, the base load is 70-90% of the annual heating demand of a building. Hence, the efficiency is very important. In many cases the efficiency decrease at partial load, the efficiency variation depends on the selected heating system. To achieve the most profitable operation, compliance between calculated and real power demand is important. The base load system usually entails large investment costs, but are inexpensive to operate.

The peak load is usually covered by a heating system which is cheap to install but often expensive in operation. The balance between base load and peak load should be cost optimized with regard to investment and operational costs. One have to fulfill the current requirements for energy supply given in the *TEK10* as well. Passive or low energy buildings must fulfill the requirements of the associated their standards.

It is not desirable to oversize the base load unnecessarily because of the high investment cost. One should find the optimal balance between base load and peak load from net power demand calculations as net power demand should equal real power demand theoretically. The energy consumption reduces significantly in passive and low energy buildings, which again reduce the profitability of large investments in base load heaters. The requirements in *TEK10* that 40% or 60% of the building shall be covered by a different energy supply than direct-acting electricity or fossil fuels are not always profitable. Passive residential buildings are therefore exempt from the requirement if it is not profitable.

The standards for calculating design power demand calculates gross power demand and does not include internal heat gains (including heat from the ventilation system) or solar heat gains. One could say that gross power demand includes safety margins. The investigated building described in chapter 0 is an example of that. One must remember that internal loads varies, meaning that the included safety margin by using gross power demand also varies.

Usually one design the peak load so that the heating system totally covers gross design power demand calculated according to *NS-EN 12831:2003*. The peak load heating system cover the difference between power supplied by the base load heating system and the gross design power demand. Peak load will also cover extra safety margins, if that is desirable. The peak load can typically be an electrical heating system or an oil boiler, both of which are cheap to install. Typical power-duration curve shown earlier in the thesis shows that it is important to have a peak load with good efficiency over the entire power range, since peak load mostly operates on partial load. This becomes especially important for offices, since the base load covers a relatively small part of the power demand in many cases. This because of low hot water demand and that the building is only in use during the day. Night setback of the indoor temperature is common as well. Improved efficiency reduces the energy consumption and saves thus both costs and environment, provided use of equal energy sources.

### 5.4.2 Consequences of miscalculation of power demand

Miscalculation of the net power demand will change the optimal balance between base load and peak load and leads to an under- or over-sized heating system. Using the duration curve of the gross power demand to find the optimal division between base load and peak load could lead to more part load operation of the base load system than necessary.

Undersized systems is unfortunate as one can experience that the thermal comfort can be so bad that one has to insert additional heaters or need to install more power in the system afterwards. That will result in additional costs and not least dissatisfaction among users. Reduced performance may also occur during periods when installed capacity is not big enough. Poor performance is particularly unfavorable in commercial buildings and causes financial losses.

Usually builders do not take any chances and oversizing of installed power in the heating system is common. The investment cost increases due to oversizing and operating costs may also increase, depending on the type of base load and peak load. The efficiency may decrease as a result of more partial load operation. Oversized heating systems therefore leads to increased total cost. The optimal balance between base load and peak load does not have change if the heating system is oversized intentionally and only more peak load heaters are installed. Hence, it would be advantageous to calculate the power demand accurate, e.g. net power demand, and rather have the ability to install additional peak load as a security.

Covering the entire load using a heat pump may be a better environmental alternative than divide into base load and peak load heaters. Probably also better economically in some cases. This will depend on several factors, the alternative peak load heater, chosen heat pump technology (air-to-air, ground source, etc.) and the change of efficiency of the heat pump at other loads than design load. One should calculate different solutions to find the optimal. The shape of the duration curve and chosen safety margin will be decisive as well. If the optimal solution become a heat pump covering the entire heat load, one should use the duration curve for net power demand to calculate the optimal point of operation.

#### 5.4.3 Oversized or undersized heating systems

Results given by district heating and electricity suppliers gives variable answers, but do they say something about what is usual, oversized or undersized systems?

*BKK* had to install additional power on *Løvåshagen* cooperatives in Bergen because the circuit breaker tripped. It is reasonable to assume that the power demand for heating the building can be involved, but not necessarily as BKK delivers electricity not only to heating purposes. According *Norsk fjernvarme* it is difficult to obtain information that states that passive and low energy buildings use more energy than intended, it is bad publicity as it is advertised with low energy consumption. Since energy is the integral of power use over time, this information may provide an indication of that the power demand in passive may be undersized.

The installed power in Løvåshagen cooperatives were designed with the assumption of low need of power. They did not say if they used *NS-EN 12831:2003* or not, but they did not install enough power. Many factors may play a part here. One of them may be that the power consumption in the passive houses not shrink as much as the energy consumption does. Another factor may be poor work in the construction, so that the cooperative is not as sealed or isolated as planned. Users may use the building completely different than intended. They may have many electrical equipment using a lot of power. Many people are used to ventilate by opening windows. This will lead to excessive heat loss in winter, as the heat exchanger is not utilized. In passive houses and other buildings with heat recovery it is intended that the airflow shall increase, instead of to ventilate through windows.

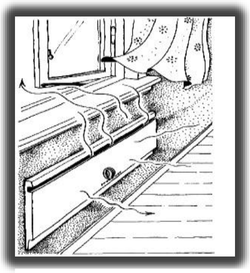
Figure 56 Ventilation through window[1]

According to *Hafslund Nett*, installed capacity is generally oversized for all types of buildings. Installers want to ensure that the circuit breaker does not get tripped. *Agder Energi* can report for their district heating deliveries, that they cannot remember increasing the capacity.

According to *Statkraft* district heating, it would not matter too much if room and ventilation heating were oversized or even matter doubled. Power demand for room heating and heating of ventilation air is very small compared with the power requirement for domestic hot water. Power demand is high by direct heating of domestic hot water. Apparently, it is not an accumulating tank in this case.

Answers from the electricity and district heating companies does not give any obviously answers, but it seems like it is most usual to oversize the heating system.

## 5.5 Control of the heating system

### 5.5.1 Heating up capacity – night setback of the temperature

Decreasing the indoor temperature at night saves energy, which leads to reduced operational costs and more environmental friendly operation. On the other hand, it requires heating up capacity, an additional power demand for reheating of the indoor temperature before operational hours. The heating up capacity gives increased investment cost. How much the temperature is lowered and the chosen time of reheating determines the size of the heating up capacity and is part of the calculation of gross power requirement as discussed in chapter 0. One must calculate how much one saves cause of reduced energy consumption compared to extra investment cost because of more installed power to find out what is profitable.

The power duration curve will change with night setback and thus will design power demand for base load as well as peak load also change. As previously mentioned, the efficiency is changed for some combustion systems and especially heat pumps when there are more part-load. The efficiency of the heating system should therefore be included in the calculation.

Let us say that it is desirable to operate with night setback, as in the example building from chapter 0, but simultaneously drop or reduce heating up capacity. One possibility is to drop the night setback the coldest days when there is a chance that installed capacity will not be large enough to cover reheating of the building or rooms. The profitability can be calculated in each case, as there are large differences between different heating systems in accordance to the influence of reduced efficiency, investment costs, operational costs, etc.

Analyzes of the office building with night setback of the temperature, given in Chapter 0, shows that both gross and net power demand for heating occurs at the beginning of the workday. Without night setback, it is not given that design gross power demand for heating is greatest at night even though outside temperature is generally lower at night. The ventilation airflow is adjusted to a minimum, at least in office buildings. This reduces the heat loss and thereby the power demand. Simulations of the office building in chapter 0 using constant temperature shows that the net power demand of this building occurs at night, but gross power demand occurs at the beginning of the workday.

Passive and low energy buildings generally have low heat loss because of good insulation, tight building and high efficiency heat exchanger. Night reduction of the indoor temperature will therefore provide less energy savings in passive and low energy buildings than in buildings with higher heat loss coefficient. The extra heating up capacity required to heat the air in the building the necessary degrees within a specified time, will be equal regardless of the building standard. It only varies with the indoor temperature change (assuming that the heating capacity of the building is not changed). Therefore, night setback of the indoor temperature is less profitable for passive and low energy buildings than for buildings of poorer standards. The heat loss in a building is the temperature difference between indoor and outdoor temperature multiplied by the heat transfer coefficient for the building. To achieve equal temperature the next morning, all heat loss must be replaced. The heat loss equals the heating demand in terms of energy. The better building standards, the slower the temperature falls to the minimum setpoint temperature at night setback and the average temperature difference between inside and outside becomes higher. Heating will take just as long time in every building no matter building standard, assuming it is equal heating capacity. The result will be that one save less energy in passive and low energy buildings, but must invest in in equally large additional power. Economically it will always depend on calculations, but it is a smaller chance that night setback will pay off in buildings with low heat loss.

#### 5.5.2 Reduced indoor climate

It is always desirable to have a good indoor climate. Both thermal climate and atmospheric climate are factors that affect the power demand and energy consumption. Selected indoor temperature determines the temperature difference between inside and outside and thus affect the heat loss and power demand for heating. Ventilation is an important factor for achieving good air quality, but also results in heat loss and power demand.

**Reduced ventilation airflow**

Indoor air quality depends on many factors, which can be divided into the following main categories:

- Emissions from humans and their activities
- Emissions from the building, furnishing and ground
- Emissions from the HVAC system (Depending on outdoor air quality, filtration, treatment and any moisture damage in the ventilation system)

It is important to provide a good indoor climate, how to achieve the good indoor climate actually does not matter. Pollutants are removed by air replacement, but it is obvious that inside air quality depends not only on ventilation airflow. It will be equally important to eliminate or reduce the sources of pollution as having large ventilation quantity. In low emitting buildings, there is a need for smaller ventilation airflow, which reduce the need of energy and power to heat the ventilation air.

The ventilation airflow rate should not only satisfy the health requirements, but also the users comfort requirements. Since comfort usually is dimensioning for the ventilation airflow rate, reducing the airflow at design conditions is a possibility as long as one ensure that the health factors are covered. Reduced airflow will reduce human performance and might therefore be more relevant for residential buildings. One do not necessarily produce anything or need to concentrate to the same extent as in commercial buildings. For commercial buildings, one can make a calculation depending on the loss of productivity and salary compared to savings of having a smaller ventilation system. Looking at how the performance varies with the ventilation airflow rate, one can see that halving of the ventilation airflow rate reduce the performance with about 1.8%.

Number of hours (or days) that will have design conditions in a year depends on outdoor climate. Reduction of the ventilation airflow in this short period can be an economical solution, especially in buildings without heat exchanger (typical partial renovated residential buildings).

There is a requirement that passive houses must have a heat recovery system with minimum 80% temperature efficiency and low energy buildings (Class 1) at least 70%. A reduction of the airflow is obviously less profitable when heat is recovered.

**Reduced temperature**
Figure 11 shows that the performance falls on both sides of 21,5°C operative temperature. The performance do not change that rapid near the vertex, were the temperature is about 21,5°C. The graph has approximately exponential shape. A temperature change have less importance near the vertex. A small temperature reduction can be considered in the coldest period. Performance will for example fall by 2% if the operational temperature drops to about 18,5°C. There may be some complaints about the comfort at 18,5°C and in both residential and commercial buildings people are usually willing to pay extra for better comfort. The profitability of such a control is calculated in the same way as reduced performance due to reduced ventilation.

*Example: Cost analysis of a commercial building*
*Let's say that the temperature drops to between 21,5°C and 18,5°C in 4 working days a year giving an average performance degradation of 1%.*
*Loss pr. person with the hourly cost of 400NOK and 8-hour workday:*

$$400 \frac{NOK}{h * pers} * (8 * 4) \frac{h}{year} * 0{,}01 = \mathbf{128} \frac{\boldsymbol{NOK}}{\boldsymbol{pers * year}}$$

*Let us say that a person has an office of about 20 m² and it is a passive house with:*

- $0{,}40 \frac{W}{m^2 K}$ *heat loss from transmission and infiltration (according to the passive standard)[6]*
- *Ventilation according to TEK10 requirements:* $26 \frac{m^3}{pers*h} + 2{,}5 \frac{m^3}{m^2*h} * 20m^2 = 76 \frac{m^3}{h}$, *which gives* $1{,}27 \frac{W}{m^2 K}$ *using air density:* $1{,}2 \frac{kg}{m^3}$*, heat capacity:* $1 \frac{kJ}{kg*K}$*) [4].*
  *Heat recovery: 80%, gives* $1{,}27 \frac{W}{m^2 K} * 0{,}2 = 0{,}25 \frac{W}{m^2 K}$

*Then energy savings using an average temperature reduction of 1.5K during the day is:*

$$(0{,}40 + 0{,}25) \frac{W}{m^2 K} * 20 \frac{m^2}{pers} * 1{,}5K * (24 * 4) \frac{h}{year} = 1{,}87 \frac{kWh}{pers * yeay} \approx \mathbf{1{,}87} \frac{\boldsymbol{NOK}}{\boldsymbol{pers} * \text{å}\boldsymbol{r}}$$

*<u>To make it profitable, reduced installation costs cause of lower installed power must be higher than the cost cause of reduced performance minus the energy savings:</u>*

$$128 \frac{NOK}{person * year} - 1{,}87 \frac{NOK}{person * year} \approx \mathbf{126{,}13} \frac{\boldsymbol{NOK}}{\boldsymbol{person * year}}$$

*In a commercial building with offices this size and with for example 400 employees, the total cost is:*

$$126{,}13 \frac{NOK}{person * year} * 400\ persons = 50\ 452 \frac{NOK}{year}$$

*<u>Based on the result in this example with a passive house, it is unlikely profitable to reduce installed capacity and thus temperature in commercial buildings. Saved installation costs of peak load are typically small since installation of peak load tends to be cheap!</u>*

In passive and low energy buildings, it is generally cheaper to maintain good indoor air quality. There is less heat loss through the construction, which makes it cheaper to maintain the temperature at selected level. Heat recovery system with high efficiency makes it cheaper to keep the airflow rate high and ensure good air quality.

### 5.5.3 Utilize internal load

Passive and low energy buildings are better insulated and has a low heat loss compared to buildings of older standard or for that matter *TEK10* standard. Utilization of internal load for heating could constitute a larger share of the heating demand. Table 2 shows that the ratio between net and gross design power demand is in the order of 0.90 to 0.95 for passive and low-energy residential buildings, depending on where in Norway the calculation is done. Passive houses obviously have lower ratios than low-energy buildings.

The building investigated in chapter 0 is a commercial building and had bigger difference between net and gross design power demand. In the morning between 08:00 and 09:00 gross power demand were simulated to be 162kW (Table 23) with winter simulation and internal load were measured to 30kW. This gives a ratio of 0,82 between net and gross design power demand. Taking into account 4,8kW heat release from the ventilation system as well, the ratio become 0,79.

For commercial buildings, the ratio is generally lower than for residential buildings as it is larger internal load in this type of building and less use of domestic hot water (assuming use of accumulating tanks). The ratio is in the order 0.60 to 0.80. Since these calculations are for design conditions, solar gains is a minimum, actually none in at the investigated building.

The question become; is it wise to include internal load to reduce peak load at design conditions? Internal load comes from different kinds of sources, not to mention of heat from people. Number of people cannot be controlled, while the use of equipment can be difficult to control. In an office building at night, one cannot assume that people are present. Lights can however be controlled, either with presents control as in the analyzed building or with other control methods. Lights, people and computers contributes with internal load in the beginning of operational time in the investigated building. If design power demand occur in the beginning of operational hours as well, as in the investigated building. The simulations in chapter 0 indicates that it is smart to include internal load at building level to get results equal to real measurements. One may only include internal load at building level, not at room level, as internal load may not occur in every room. Some internal load from people and light have to occur in rooms in use though. Using a safety margin seems wise, as simulations not always fits with real measurements, user behavior is often an uncertainty and the building is not always built as predicted.

If one are using electricity to cover the peak load, lights could be switched on regardless of presents during the coldest periods to cover a part of the peak load. Lights use electricity, so the energy price is equal and investment costs is saved. Note that the market are moving towards use of LED lights and lamps with low energy consumption, which reduce the effect of this control. Computers will contribute in the same way, but it is generally up to the user if it is turned on or off. Electricity is valuable energy, but covering peak load it is often a good economical option to use direct acting electricity.

Overall, it is useful to include predicted internal load to calculate net power demand in the most accurate way and obtain a net power demand duration curve of a year. Then one can optimize the design of the heating system covering the base load. An optimization is especially important using heat pumps or combustion plants that have lower efficiency at part load.

The power demand of the heat source covering the peak load will depend on whether one include internal load or not. The peak load should cover the rest of the power demand not covered by the base load heating system. Actually, the peak load heating system only need to cover the net design power demand in theory. In practice it would be far too risky, it is expensive to change an already installed system and not desirable to get explains from the users. Therefore, a safety margin is a good solution. This safety margin should be added to the peak load heating system, which is cheap to install.

#### 5.5.4 Utilization of installed power intended to cover domestic hot water

Heating systems have to heat domestic hot water in addition to heat the building through ventilation and room heating. Power demand for heating domestic hot water is not considered in this thesis, as it was not measured in the investigated building. It is assumed constant power demand for domestic hot water, which is the case when using a large accumulation tank.

However, it is interesting to look at the possibilities of a smart management of the power demand estimated for domestic hot water. One possibility is to control the system so that in periods for which additional power to heat the building is needed, use of power intended for domestic hot water can be used to heat the building.

This kind of control becomes relevant in the investigated building, as the need of heating up capacity after night setback is huge in a relatively small period. Available power calculated to heat the building in other periods of the day could then be used to heat the domestic hot water.

Higher degree of utilization of the system will lead to higher use of base load and less part load operation, resulting in a possible increase in efficiency of the heating system. Ventilation heating, room heating and heating of domestic hot water may have different requirements for temperature, which change conditions of the heating control panel. If the total heating system is designed to deliver the highest temperature required, it will influence the efficiency of the heating system in a negative direction compared to delivering the required temperature to each of the three different parts. This applies in particular using heat pumps, which get poorer efficiency with higher temperature lifts. Electric heating however will have the same efficiency. Utilize power intended to cover domestic hot water will be profitable if saved investment costs together with the eventually change in efficiency gives a saving.

Residential buildings use a larger proportion of the total energy consumption to heat domestic hot water than commercial buildings. Improved building standards makes the share even higher, as shown in Figure 4. Passive residential buildings typically use 40-70% of the annual energy for heating of domestic hot water. Even though one assume constant power demand to heat the domestic hot water, there is a potential in a smart management, especially in residential buildings.

## 5.6 Comparison of *NS-EN 12831:2003* and *ASHRAE 2013* *also taking FEBY 12 into account*

### 5.6.1 Simultaneity

The power calculations in the standards is conservative and calculates gross power demand by:

- Ignore heat gains from the sun
- Ignore internal heat gains
- Ignore the heat capacity of the building (steady state conditions)

In most cases, this provides an embedded safety margin as all these assumptions usually will not occur simultaneously. At room level there is more likely that all these assumptions occurs simultaneously than at building level. The calculation of gross power demand can therefore be a good solution at room level, as long as one does not use internal load actively as a heating supplement. At building level, it is not likely that the worst possible situation occurs at the same time in every room. Often rooms are unused and may have periodically lowered temperature, which will reduce the power demand. Predicting the level of simultaneity is difficult and some buildings may have users that use all rooms simultaneously. Users often change throughout the lifetime of a building as well.

### 5.6.2 Heat storage

Design power demand is calculated using the assumption of steady state conditions, heat storage is ignored. In heavy constructions with large heat storage, temperature will drop slower and raise slower as heat storage acts as a buffer against temperature fluctuations. Passive houses and low energy buildings often have thicker walls and ceilings with a greater mass than other buildings because they must achieve low heat losses. Often this means more heat storage as well, depending on materials and exposed materials. Taking into account the heat storage in the control of the heating system in passive and low energy buildings, need of installed power will become smaller. The Swedish Passive House Standard, *FEBY 12*, take into consideration heat storage when choosing design outdoor winter temperatures. Using *FEBY 12*, which is described in chapter 5.1, gives an outdoor design winter temperature of -18,2° C, instead of -20° C when using *NS-EN 12831:2003*. Assuming that *FEBY 12* is most realistic approach gives a potential of reducing the power demand in heavy buildings. A drawback occur if one want to use night setback of the temperature, heavy buildings need more heating up capacity than light building (Table 3). This again increase the design power demand, as in the investigated building. Heating up capacity is included in the Norwegian standard. It seems fear to include the *FEBY 12* method to estimate design outdoor winter temperature as well.

Another question could be; how big temperature drop should be allowed under extremely cold conditions? That will depend on how important the thermal comfort is, compared the costs of installing more power.

One possible control of the heating system will be to turn on more power and thus set up the temperature with for example one degree in advance of an announced cold period. Another way to control the system, which will give approximately the same result, is to turn up the heating system even more when it is already running at a certain percentage, say 95% of maximum capacity. Such regulation in a building with a high heating capacity has the potential to lower the power demand. The temperature can be kept at the highest part of the performance curve, 20-22,5°C, which is perceived as good thermal comfort. On the other hand, increased indoor temperature lead to higher energy consumption since the difference between the indoor and outdoor temperature increases during this period. Again, this is an optimization question that will vary with climate profiles, energy prices, installation costs and operational costs.

#### 5.6.3 Design values

To find design heat loss, design values for indoor and outdoor climate must be known or estimated. When comparing the Norwegian and American standards one assume that the same climate data is collected (temperature, humidity and wind direction and speed, etc.). It is also assumed that design indoor values are based on the same comfort requirements.

*Design outdoor temperature*

Design outdoor temperature can be calculated in several ways. If 99.6% is used instead of 99%, the dimensioning outdoor temperature becomes lower and estimated power demand increases. Using the “n-day mean air temperature” method, results vary a lot depending number of days used. One could use one-day, four-day or twelve days average. Estimated power demand will be highest using the daily average value.

It is not that simple concluding which method is the best. Heat storage of some degree occur in all buildings and a significant change in temperature will take time, long time in buildings with much heat storage. It is reasonable that both the Norwegian, Swedish and American standard adapts to this phenomenon. Should it be used an average of the coldest days, then how many days? Or cut 0.4 to 1% of the coldest periods?

The Swedish passive house Standard, *FEBY 12*, which specifies how many day averages should be used as a function of heat storage is perhaps the best method. Then one adapts the method in relation to heat storage in each building. One can utilize the heat storage in heavy buildings by using an average temperature over many days, 12 days if the building is extra heavy.

If it is desirable to hedge against power shortage, perhaps it is best to use the one-day average or 99.6% method, but is it necessary? That will probably depend on the temperature profile, heat storage and not the least if the builder want even higher safety margin to ensure enough power even if it costs extra. On the other hand, heaters covering peak load are generally cheap to install. For this reason, it may be advantageous to ensure good comfort and use *FEBY 12* in order to find net power duration curve and optimal distribution between base load and peak load.

#### 5.6.4 Transmission

U-values are slightly different in the different models. In the American standard, the thermal bridging effect is included in the U-value, which is not the case in Norwegian standard. The Norwegian standard includes the thermal bridging effect when calculating heat transfer coefficient, thus there is no difference in the calculation of heat loss and power demand. Both standards corrects for climate impacts. For the sake of comparison of the standards, one must assume equal corrections.

When calculating the power demand it is essential that the U-values used in the calculations correspond with real U-values. Are the building built in line with expectations? And not the least; is the cold bridges in the real building equal as used in the calculations? The building might be have higher U-values in reality than what it calculated if poor work were performed at site.

<u>*Losses to unheated rooms*</u>

It is of course easiest if the temperature in the cold room is given and one can calculate in the same way as calculating losses to the outside. The temperature is not given when a new building it is to be built and heat losses must therefore be calculated or obtained using a standard. The result has potential to be best if heat losses are calculated, but it requires that the input values are correct.

The calculation method for heat loss to unheated rooms in the Norwegian standard involves multiplying by the factor $b_u$. This factor takes into account the heat transfer coefficient of the inner wall and the wall to the outside, including infiltration losses. It could seem like a good estimation, but let us look closer at the formula:

$$b_u = \frac{H_{ue}}{H_{iu} - H_{ue}}.$$

If the heat transfer coefficient from the unheated room to the outside, $H_{ue}$, is as large or larger than the heat transfer coefficient from the heated room to the unheated room, $H_{iu}$, the denominator will become respectively zero or negative. The inner wall cannot be nearly as good as the outer wall if the formula is to function as intended. Inner walls are usually poorly insulated and leaks through the inner doors are much larger than exterior doors. This will therefore most likely not be a problem, especially not in passive and low energy buildings, which have extra good insulated exterior walls.

The formula used in *ASHRAE 2013* estimates the temperature of the unheated room and includes loss from distribution facilities and solar gains. Solar gains shall not be included in power demand calculations for heating, but applies for cooling systems. Heat loss from distribution facilities in unheated rooms are still interesting and can make the calculation more accurate. Ventilation/infiltration of outside air and transmission from heated rooms is also included in the calculation of the temperature in the unheated room. If one expects that the infiltration from heated rooms are included in the formula, although it is not specified, this is a more detailed method than what is used in the Norwegian standard.

#### 5.6.5 Losses to the ground

There is most likely some differences between the standards when it comes to losses to the ground, at least if one use simplified methods. Both standards have taken into account the same parameters, although they are prepared in different ways. There is a more detailed method in *NS-EN ISO 13370* and *ASHRAE 2013* refers to several other detailed methods. The main principles are still the same, and there is reason to believe that the results of the calculations will be similar.

#### 5.6.6 Infiltration

There is a great advantage if the building is pressure tested so that one can use these values directly. If the building is not yet built, one usually base the calculations on table values that is developed based on requirements or experience. There is requirements for maximum leakage in the passive and low-energy standards and for both commercial buildings and residential, but is the building actually built as sealed as the requirements indicate? Nails going through the plastic and make many small holes could be a source of error leading to higher infiltration than desired. It should however be measured leakage as a check that one achieve the requirements.

The main features of the two standards is that:

- In *NS-EN 12831:2003* it is stated that one use an airflow at 50 Pa pressure difference, $n_{50}$, to calculate natural infiltration.
    - Correction in relation to the wind by means of a shielding coefficient and a correction factor for height is included in the standard.
    - Correction with respect to an increase in pressure difference with height (chimney effect) is <u>not</u> included.
    - Mechanical infiltration equal mechanical exhaust airflow and is added as a supplement, distributed according to the permeability of the different rooms.

- In *ASHRAE 2013,* one add the pressure differences that arise because of natural and mechanical infiltration instead of adding the airflows together.
    - The pressure differences due natural infiltration is calculated with respect to the chimney effect and wind pressure.
    - It is taken into account the shielding of wind and that wind speed increases with height.
    - It is taken into account that the chimney effect also changes with altitude.

*ASHRAE 2013* state that the volume flow can be found by $Q=c(\Delta p)^n \left[\frac{m^3}{s}\right]$. Experiments have shown that n typically will be 0,65. It is a non-linear relationship between pressure difference and infiltration. When using *NS-EN 12831:2003* it is assumed that all the air removed mechanically is additional as extra infiltration. This can lead to over-dimensioning of the infiltration.

All air that is removed must be replaced. Leaks can be regarded as a resistance air must squeeze through, the natural infiltration will therefore decrease by having mechanical extract air according to the American standard. Total infiltration will of course increase, but not as much as the Norwegian standard states.

For commercial buildings, *ASHRAE 2013* also takes into account the infiltration (air exchange) through exterior doors, as automatically rotating doors etc. *NS-EN 12831:2003* does not.

It does not necessarily matter that *NS-EN 12831:2003* use an airflow at 50Pa pressure difference, $n_{50}$, for natural infiltration, while *ASHRAE 2013* calculates the pressure differences. The correction factors in the Norwegian standard could actually consider this, but correcting for height with respect to pressure difference that occurs because of the chimney effect is not considered in *NS-EN 12831:2003*.

*<u>Example: The chimney effect</u>*

*Let us say that $T_{indoor}$=21 °C and $T_{outdoor}$=-20 °C at design conditions. The ASHRAE 2013 method gives $P_T$=9,81*1,2*41/293=1,65Pa/m. The pressure difference varies with 1,65 Pa pr meter height, due to the chimney effect.*

It is perhaps not so much, but if the building high; this would contribute a great deal. It seems like *ASHRAE 2013* use a more detailed method to calculate the infiltration.

In the American standard, the power demand to evaporation of water by humidification is also included. In Norwegian climate, it is rarely needs of humidification and one can often omit that part.

At the room level, the Norwegian standard take into consideration that all infiltration can occur at on one side of the building and lead to a doubling of power demand compared to if the infiltration were evenly distributed. The same is mentioned in chapter of residential buildings in *ASHRAE 2013*. There are reasons to believe that this also applies for commercial buildings in *ASHRAE 2013*.

*NS-EN 12831:2003* states that this doubling of infiltration at room level not apply to the mechanical infiltration, which instead is distributed by the permeability of each room. The pressure difference over the outer wall increases with mechanical infiltration. In principle infiltration increases the most in rooms where exhaust air is withdrawn, but if the wind is strong enough, it could be just as reasonable to assume that all mechanical infiltration could occur on one side of the building as well. A possible improvement of the Norwegian standard may be to include a doubling of mechanical infiltration at room level to cover the worst-case scenario and take into account that the extra infiltration is not equal to mechanical exhaust because of non linear relationship between pressure difference over the exterior wall and infiltration airflow.

#### 5.6.7 Ventilation

The airflow supplied to each room depends upon number of persons, pollution from materials and pollution from activities. It also depends on how stringent indoor air quality requirements that is desired. The airflow can vary from room to room and with time, depending on whether it is selected displacement or stirring ventilation and if there is variable, demand-controlled or constant airflow. To compare standards, one must assume that these requirements and choices are equal. The ventilation system covers the airflow required and which is not supplied by infiltration.

The power demand at room level is reduced if the supply air is heated, either through a heat recovery system, in the ventilation heating coil or comes as overflow from an adjacent warm room. Overflow from an adjoining room may lead to increased power demand in the room the ventilation air enters. Heating the ventilation air to a specific temperature at design conditions, can reduce installation costs by using a central heating rather than distributing the power demand in every rooms. At building level, it is possible to take into account that every room may not be in operation at the same time and one may multiply with a simultaneity factor.

The power demand to heat the supplied ventilation air is not included on room level, according to *ASHRAE 2013*. In the USA it is normal that the air is heated to room temperature levels in the central heating system. This would not mean anything as one assume equal conditions in the comparison. If the supply air is not heated to indoor air temperature in the central heating system, one must take into account the power required at room level. Many ventilation systems need lower air temperatures than the room temperature to function as intended and avoid that supply air goes directly to exhaust.

*Heat release in the ventilation system*

The heat release coming from the ventilation systems and heat loss cause of transmission and leakage from the ventilation ducts are included in *ASHRAE 2013*, but not in *NS-EN 12831:2003*. Energy use for fans in the ventilation system often represent a significant share of the total energy use in passive and low energy buildings, which also was the case in the building investigated in chapter 0. Even though fans are not necessarily running on maximum in the winter, energy is released and the power demand for room heating will decrease as the supply air temperature increase. It is not desirable with large heat gains in the ventilation, as it will lead to energy loss and danger of overheating in the summer. Heat release in the ventilation system can be considered as internal load that supplies heat by using electricity. Heat will occur in the engine and because of friction in the fan and the channels. Not all energy consumption in the exhaust can be regarded as heat gains to the building, but a lot can be reclaimed in the heat exchanger.

Chapter 0 shows that heat release in the ventilation system represents a significant power gain in the investigated building. Let us still look upon a simple example to be certain of the potential of including heat release in the ventilation in the Norwegian standard.

*Example: passive house*

*Let us say that the SFP factor is constant just below the passive house requirements; SFP=*$1\frac{kW}{m^3/s}$. *To simplify the calculation even more, it is assumed that all heat release in the ventilation system contributes heat the building. (PS: calculations of the investigated building shows that about 63% of the heat release enters the building, the rest is lost. Therefore, one may adjust the results depending on ventilation system and its composition, but the point proven is equal)*

- *The temperature increase become* $\Delta T = \frac{SFP}{C_p * \rho} = \frac{1}{1{,}2*1}$*=0,833K*
    - *Density of air:* $\rho \approx 1{,}2\frac{kg}{m^3}$
    - *Heat capacity of air:* $C_p \approx 1\frac{kJ}{kg*K}$

*Let us assume an office where it is desired to fulfill best category, category 1, in the Norwegian standard for indoor air quality.[12] It is assumed a "low pollutant building".*

- *Office area:* $10m^2$
- *Number of persons at the office in operational hours: 1person*

- *Ventilation airflow:*
  - *In operational hours:* $36\frac{m^3}{pers*h}*1pers+3{,}6\frac{m^3}{m^2*h}*10m^2=72\frac{m^3}{h}$
  - *Outside operational hours:* $0{,}7\frac{m^3}{m^2*h}*10m^2=7\frac{m^3}{h}$
    *(TEK10 requirements)*
  - *Average ventilation:* $0{,}35*72\frac{m^3}{h}+0{,}65*7\frac{m^3}{h}\approx 30\frac{m^3}{h}$
    *Assume an average of 35% operational time in a year, taking vacations, weekends and nights into consideration.*

*Supplied power to the office in operational time:* $1\frac{kW}{m^3/s}*\frac{72}{3600}\frac{m^3}{s}=20W$

*Supplied power to the office outside operational time:* $1\frac{kW}{m^3/s}*\frac{7}{3600}\frac{m^3}{s}\approx 2W$

*Supplied power to the office in average:* $1\frac{kW}{m^3/s}*\frac{30}{3600}\frac{m^3}{s}\approx 8{,}3W$

*Taking into consideration that the passive house requirement for energy use is* $20\frac{kWh}{m^2*år}$*, it is interesting to look at the energy use of the fan system in the ventilation in a year. It is assumed that all the electricity consumption is supplied the office as heat.*

*Supplied energy as heat release in one year:* $8{,}3W*8760h=72{,}7kWh=7{,}27\frac{kWh}{m^2}$
*This represents 36.4% of all energy demand.*

*Passive house requirement for energy consumption looks at energy use after the heat from the fan is supplied, in this sense it could have been compared with 27,27*$\frac{kWh}{m^2*år}$ *and not* $20\frac{kWh}{m^2*år}$*. The point is that the heat release from the ventilation system contributes a great deal, even though a lot of the energy appears in the summer when one heat is not needed.*

The SFP factor is in reality not constant, but reduces with lower airflows. Friction reduces and the power supplied to the fan per amount of air (SFP) become lower as well. At design conditions in the winter, one only ventilate to achieve the desired air quality and not for cooling, as in the summer. SFP is most likely lower in the winter when it is possible to utilize the heat release. Nevertheless, this example demonstrates that the inclusion of power from fans and friction in the ventilation has the potential to improve the power calculations, even in the winter. Fresh air is always required and it is in particular economical in passive and low energy buildings where it is cheaper to maintain good indoor air quality. 20W in operational hours or 8,3 W in average for this 10m$^2$ office do not seems that much, compared to for example light bulbs. Most light bulbs have higher power use and heat release. The power duration curve will however be lowered throughout the heating season. Net power duratio n curve can be improved by include heat/power release and the balance between base load and peak load can be further optimized.

Leakage and transmission heat losses from the ducts will, unlike heat gains from the fan and friction, increase the power demand for heating depending on how seal and isolated the channels are. Leakage to the rooms where it is not desirable with more fresh air, such as cold unused rooms, leads to increased total ventilation airflow in order to achieve the same air quality. This would lead to higher power demand. Transmission heat loss will lead to increased power demand and may particularly occur if the channels are poorly isolated and passes through an unheated attic or basement.

# 6 Conclusion

## 6.1 Optimal operation

To obtain the most profitable operation of the heating system one should find the optimal distribution between base load and peak load. Because of environmental concerns, *TEK10* and the passive and low energy standards require use of energy sources that is not direct acting electricity or fossil fuels in most cases. Often this part of the heating system cover the base load. Passive houses use so little energy that the requirements of *TEK10* may entail large expenses; therefore, passive residential buildings could be exempt from this requirement.

In the design of a heating system, it is important to seek correspondence between calculated and real power demand, then both calculated gross and net power demand will be more realistic. Hence, net power duration curve become more realistic and the most profitable distribution between base load and peak load can be optimized economically. An extra safety margin in addition to net design power demand is necessary, *NS-EN 12831:2003* are using gross design power demand. It is normal to add a safety margin on top of the gross power demand as well. Peak load heaters should cover the extra power demand cause of these safety margins. Investing in peak load heaters is cheap and would not affect the operational costs noteworthy as these peak load heaters is rarely used.

Oversizing the heating system results in increased installation costs, but may also lead to decreased efficiency of the heating system because of increased part load operation. This is also the reason that dividing into base load heaters and peak load heaters often is profitable. Economically, one should calculate the optimal division.

Covering the entire load using a heat pump may be a better environmental alternative than divide into base load and peak load heaters. Probably also better economically in some cases. This depends on several factors, the alternative peak load heater, chosen heat pump technology (air-to-air, ground source, etc.) and the change of efficiency of the heat pump at other loads than design load. One should calculate different solutions to find the optimal. The shape of the duration curve and chosen safety margin will be decisive as well. If the optimal solution become a heat pump covering the entire heat load, one should use the duration curve for net power demand to calculate the optimal point of operation.

Information provided from district heating and electricity suppliers shows that oversizing of heating systems are probably most common, installers want to ensure that enough power is installed. Passive houses may have greater heat loss in reality than intended because of poor workmanship in the construction phase or because users do not use the building as intended. Both over- and under sizing might as well be a result of an inaccurate calculation method for design power demand. The method is conservative, which in most cases leads to over-sizing of the heating system.

#### 6.1.1 Compensation curve

The control principle is essential for the design of the heating system. The compensation curve used in the investigated building, which control the temperature of the supply air based on outdoor temperature, actually move power demand from the room heating system to the ventilation heating system the coldest days. This kind of control leads to higher power demand in the ventilation heating system and less in the room heating system. Economically, this control is a good choice, as a huge heating coil in the ventilation in most cases is cheaper than an increased room heating system.

In the investigated building, maximum power demand for room heating actually occurred when it was warmer outside then the design outdoor temperature because of the compensation curve. It is vital to check for this opportunity using a compensation curve. If the power demand for room heating is not maximum at design conditions, one actually could install less capacity in the ventilation system by compensate less and cover this power demand by the room heating system.

#### 6.1.2 Night setback and heating up capacity

Passive and low energy buildings generally has low heat loss because of good insulation, sealed building and high efficiency heat exchanger. Hence, night setback of the indoor temperature will provide less energy savings in passive and low energy buildings compared to buildings with higher heat loss coefficient. The indoor temperature falls slower and the temperature difference between inside and outside is higher in average. The extra heating up capacity needed will be equal regardless of building standards (provided equal heat capacity) and will only vary with the change in indoor temperature.

If the heat capacity in the building become higher in a passive or low energy building, the heating up capacity will also increase. In the investigated building, the heating up capacity cause of night setback constituted a significant part of the total power demand, actually 45,6kW – 72,3 kW theoretically, depending on chosen reheat time. This is a lot compared to the total power demand of the building, predicted to be about 120kW at design conditions or theoretically calculated to about 170kW-220kW (depending on reheat time and inclusion of safety margins). Night setback cannot be depreciated in passive or low energy, but the chances of profitability reduces. In each case, one should calculate to find out what is optimal. One can also look at the possibility of reducing the extra installed power by dropping night setback the coldest days.

### 6.1.3 Indoor climate

In passive houses and low energy buildings, it is cheaper to obtain a good indoor climate. The heat loss coefficient is low and the heat recovery system have high efficiency. Reducing the indoor climate rarely pays off economically, neither thermal nor atmospheric air quality. Especially in commercial buildings, where a few percent drop in performance may constitute a major cost in wages for non-productive time. In residential buildings it is different, one have to balance the importance of good indoor climate and economy. Passive and low energy buildings increases the chance that one will choose a good indoor climate significantly.

### 6.1.4 Control of internal load

Central control and utilization of light and other internal loads could reduce the need for installed peak load heaters. Internal load usually use direct acting electricity, which is high quality energy, but electricity is often a good economical option as peak load. Nevertheless, LED lights and internal loads with low power demand is an advantage, although it reduces the possibilities of utilization in winter. Problems with overheating and need of cooling in summer decreases as well as the total energy use.

### 6.1.5 Control of installed power for domestic hot water

In general, residential buildings use more domestic hot water than commercial buildings. Residential buildings have thus the greatest potential to utilize the installed power intended for the domestic hot water heating system, to other heating purposes. In passive residential buildings, the energy use of the domestic hot water typically constitutes 40-70% of the annual energy for heating purposes. Even though the power demand for domestic hot water is regarded as constant in this thesis, there is a potential in a smart management and particularly in residential buildings. Smart management of installed power requires a large accumulating tank as a buffer and a heating system adapted for alternating usage. The profitability should be calculated taking eventual change in efficiencies of heating systems into account. Both cause of the potential of less part load of the total heating system and cause of the possible change of temperature requirements of the heating system.

### 6.1.6 Control and utilization of heat capacity

To achieve low heat loss, passive houses and low energy buildings often have thicker walls and ceilings with greater mass compared to other buildings. In many cases, that entails greater heat storage. One possible control that utilize the heat storage is to increase the temperature, e.g. increase the power demand ahead of an announced cold period or when the heating system approaches maximum capacity, for example, 95% of maximum capacity. This control will keep the temperature at the highest part of the performance curve (20-22,5°C). One achieve good indoor air quality by using less installed capacity.

## *6.2* Potential of improvement of *NS-EN 12831:2003*

As seen in chapter 6.1, the choices of operation control are many and there are many ways to increase or decrease the power demand depending on operation choices. In this chapter, the conclusion regarding the method of calculating the power demand is given and improvements proposed.

Many interesting improvements are proposed and there is no doubt that there is opportunities for improvements of the Norwegian standard, especially in passive and low energy buildings. The improvements proposed should be tested in several buildings before implementing them in the Norwegian standard, *NS-EN 12831:2003*. The investigated building had advanced control, which is positive energy vise. Nevertheless, using simpler control of the building and have the opportunity to play with the control parameters will make it easier to test improvements.

This master thesis have compared simulations in SIMIEN, theoretical calculated values and measured values in one investigated office building. There are many uncertainties associated with a building; many parameters could be sources of error. Several parameters did not match as good as predicted. Measured operational hours did not always match as expected based on the measured input values used in simulations and calculations. This was strange, and there may be some errors in the measuring system, either in the energy measurements or in input values, such as airflows, operational time, etc. A fatal error found in the investigated building occurred in the ventilation. It was measured more exhaust airflow than supply airflow, which would lead to more infiltration, mechanical infiltration. Based on measured power use and the control principle of the ventilation, unbalanced ventilation did not match at all. Simulated and calculated power demand were double of measured during operation. The conclusion became that it had to be balanced ventilation and measurement errors in the measuring of airflow. Measurement errors could hence result in huge errors. Despite all these uncertainties and by making some assumptions, measured power demand became comparable to simulated and calculated values.

The preliminary project assumed a heat recovery efficiency of 85% (average). Measured efficiency was 72% during operation, when maximum power demand occurred. Using 85% efficiency in the planning could lead to lack of power at design condition, depending on other included safety margins off course.

### 6.2.1 Total power demand oversized at the investigated building

Even by extracting internal load and heat release from the ventilation system, theoretical calculated design power demand are higher than the adjusted simulation (predicted measurements) at design conditions. One cannot be sure that the adjusted simulated values would be equal measured at design conditions. Nevertheless, the results of the coldest day measured (Tuesday week 50) also indicates the same tendencies, the theoretical calculated total power demand gets higher than measured.

### 6.2.2 Ventilation

The heat release in the ventilation system and heat loss cause of transmission and leakage from ventilation ducts are included in *ASHRAE 2013* may be improvements to the Norwegian standard. Heat loss cause of transmission and leakage from the ventilation ducts will vary from building to building. In the investigated building, the ducts were insulated and sealed well, hence one assume negligible heat loss. The calculated ventilation power demand seems to match the measured values best extracting heat release. Heat release in the ventilation system constitutes quite a small part of the total power demand at design conditions. 4,8kW in the investigated part of the building is about 6,7% of the power demand for ventilation heating at design conditions.

The ventilation airflow allowed at design conditions is often much higher than necessary in ordinary use. In the investigated building, one could allow maximum airflow, 26 000m$^3$/h, at design conditions. One could also do the opposite, only allowing an airflow of approximately 17 800 m$^3$/h ($6,3\frac{m^3}{m^2\,h}$) This airflow is determined based on predicted number of people present and require no need for cooling or extra ventilation because of high $CO_2$ levels. During the days of measuring, it was not that cold outside and it is unknown if the extra ventilation airflow that occurred the measured day is due to cooling demand or high $CO_2$ levels. Hence, a recommendation will be uncertain. Cooling demand in the summer often determine the maximum airflow. One rarely need cooling at design winter conditions, but $CO_2$ levels must be considered. Using only necessary estimated airflow rate, one <u>may</u> allow poorer indoor air quality if there is unexpected user behavior days with design conditions. There is probably a good idea to oversize somewhat as a safety margin, but using maximum airflow seems like an exaggeration.

### 6.2.3 Internal load

In most buildings, one can predict the user behavior roughly, but it is hard to know exactly how the user behavior is going to be. How many users present, when they arrive and leave and which rooms they use is not possible to know for sure. There may be different kind of users throughout the lifetime of a building and it is not certain that all of them use the building as intended. One may install some extra power as a safety margin because of these factors. Users should learn to use the building as intended and the safety margin should not include too much improper use.

Based on the investigated building it seems fear to extract internal load from the power demand on building level, but one may be more careful at room level as internal load may not occur equally distributed. Knowing exactly how much internal load occurring could be challenging. Tuesday in week 50 one could extract most of the internal load measured in order to get equal results as measured. Measured internal load were 30 kW at the time of design power demand, in the start of the working day. This is quite a lot compared predicted design power demand of about 120kW (adjusted SIMIEN simulation at design conditions). Based only on this investigated building the recommendation would be to extract internal load on building level to ob tain the most correct net duration curve and the best possible distribution between base and peak load. Because of many sources of error, as internal load and poorer heat loss confessions one should include a safety margin added to the peak load.

At room level, one should be more careful including internal load unless one manage lights etc. as peak load. In the investigated building, offices may have one person and the lights on when calculating design power demand at room level. This because the building is controlled due to presents of people. D esign power demand occur when people arrive cause of the night setback of the temperature.

### 6.2.4 Heating up capacity

The calculated design power demand for room heating is higher than the measured in the investigated building. It seems like room heating extracting heating up capacity and internal load matches measured values quite well. Theoretical calculated heating up capacity is therefore too high in in the investigated building. It is hard to recommend anything based on the investigated building though. There was a gradually change of the indoor temperature as people arrive, which may not correspond to the theoretical calculation using longer reheat time. To test the theoretical heating up capacity against measured one should control the building in a simpler way, making changes of parameters at a certain time.

### 6.2.5 Heat gains from the sun

In the winter, measurements shows that sun only contributes from about 09:00 to 15:00 at the location of the investigated building. Total design power demand in our case occur from 08:00-09:00, which indicates that heat gains from sun would not make any difference.

In most cases maximum net power demand for heating either occur in the morning cause of night setback or at night cause of small internal load. In both cases, no solar heat gains will occur in the winter at the location investigated. If design power demand should occur in a time where sun contributes, the recommendation will be to include solar gains at building level and in the calculation of the net duration curve to optimize the distribution between base and peak load. One should not to exaggerate, but use worst-case scenario as basis, which actually would be a cloudy day.

### 6.2.6 Simultaneity

At room level, it is natural that the worst possible situation is taken into account. At building level, simultaneity factors could be included in some buildings, but simultaneity varies in every case and should be evaluated more closely at in each case. It provides a built-in safety margin assuming that all rooms are in operation at the same time, but that is actually the case in many buildings. The building may also change users several times during its lifetime. Some users may use all the rooms, depending on building category etc. In the office building investigated, some of the companies used more or less all of the rooms at the same time. The recommendation would be to take into account worst possible situation and dimensioning as all rooms are in use simultaneously, as *NS-EN 12831:2003* are doing.

### 6.2.7 Design outdoor temperature

The design outdoor temperature is of great importance when design power demand is calculated. There is several methods to estimate the design outdoor temperature. None of the methods use the absolute lowest outdoor temperature ever measured. That is logical, knowing that the outdoor temperature varies and extreme values equalize because of heat storage in the building.

The Swedish standard, *FEBY 12*, seems to have the most detailed approach of how to choose design outdoor temperature. One finds the number of days in the n-days mean temperature method by calculating the heat storage in the actual building. *FEBY 12* provides a method of grading the choice of design outdoor temperature based on heat storage in the building.

In this master thesis, the outdoor temperature was not very cold the days of measuring and investigate whether or not the *FEBY 12* method matched reality would not give results of value. Recommendation of using *FEBY 12* and the n-days average outdoor temperature method is based on theory and the assumption that *FEBY 12* fits reality.

Using *FEBY 12* to calculate the design power demand in the investigated building reduced the outdoor temperature from -20°C to -18,2°C, which resulted in a reduction of 6,6kW. About 4% of the theoretical calculated power demand excluding heating up capacity in this building.

### 6.2.8 Heat loss to unheated rooms

If the temperature in an unheated room is unknown, it appears that the method used in *ASHRAE 2013* for calculating the temperature is more detailed than in *NS-EN 12831:2003*. Hence, also the *ASHRAE 2013* method of calculating the heat loss to unheated rooms if the temperature is unknown is more detailed compared to the method in *NS-EN 12831:2003*. The method in *ASHRAE 2013* will probably give a better result, but is also more complex.

#### 6.2.9 Infiltration

The American standard, *ASHRAE 2013* seems to calculate infiltration in the most accurate way. The pressure difference due to the different phenomena, chimney effect, wind and mechanical infiltration, is calculated in a way that makes sense. *ASHRAE 2013* take greater consideration to variables that affect the pressure difference than *NS-EN 12831:2003*.

*ASHRAE 2013* refers to several studies that concludes that the airflow is not linear with the pressure difference, but with the pressure difference raised to 0.65. The Norwegian standard is based on air exchange at a pressure difference of 50 Pa and multiplies with correction factors for the wind. It not taken into account that the chimney effect varies with room height! The mechanical infiltration airflow is added to the natural infiltration airflow. Not as an additional pressure difference as in *ASHRAE 2013*. This will provide more infiltration than in *ASHRAE 2013*, where airflow, as mentioned, is not linear with the pressure difference.

Both standards are taking into account that all natural infiltration may enter on one side of the building at room level. *NS-EN 12831:2003* distributes mechanical infiltration based on permeability and does not take into account that all mechanical infiltration may also enter on one side of the building. *ASHRAE 2013* include the possibility of double total infiltration airflow at worst-case scenario, not only natural infiltration. The calculation method for infiltration in *ASHRAE 2013* seems like the most accurate method. Changing to this method is a potential for improvement of the Norwegian standard.

# 7 Recommendations for further work

A building is complex and there could be many reasons why a building does not always have the same power demand in reality as calculated. Many possibilities could be tested furthermore. It will be much easier if one have a simpler control principle than in the investigated office used in this master thesis. Being able to play with the control of the building will also make it a lot easier to eliminate sources of error. Several buildings should be examined in order to conclude that the recommendations given in this thesis are the best options. Nevertheless, the main principles found in this master thesis should be considered reliable.

It would also be an advantage having colder outdoor temperature the days of measuring. The lowest outdoor temperature the measured week in the investigated office building occurred Tuesday and varied between -2°C and -6°C. Results will be most lifelike if there is design temperature outside. Colder outdoor temperature will eliminate the possibility of cooling demand in most cases.

The opportunity to exploit domestic hot water is an interesting control opportunity that should be investigated further. Unfortunately, this was not possible in the investigated building, which shared domestic hot water with another building.

It would be advantageous to use a more advanced simulation software than SIMIEN, especially if one wants to examine power demand at room level. Then one can divide the building into rooms and compare measurements, theoretical calculations and simulations in each room.

# 8 References


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# 9 Appendix 1 – Calculation of heat release from the ventilation system

| | | | | | | |
|---|---|---|---|---|---|---|
| ***Design point*** : | | | | | | |
| Aggregate: | DVN 100 | | | | | |
| Airflow: | 26000 | m3/h | | | | |
| External pressure loss: | 250 | Pa | | | | |
| Aggregate: | DVN 40 | | | | | |
| Airflow: | 10000 | m3/h | | | | |
| External pressure loss: | 250 | Pa | | | | |
| Source: Systemair report | | | | | | |
| given by Erichsen & Horgen AS | | | | | | |
| ***In operating time*** : | | | | | | |
| **Aggregate:** | **DVN 100** | | | | | |
| **Airflow:** | **26000** | **m3/h** | | | | |
| External pressure loss: | 250 | Pa | | | | |
| Results for simulation | | | | | | |
| Power from the main supply, including frequency converter, SUPPLY AIR | 8,36 | kW | | | | |
| Power from the main supply, excluding frequency converter, SUPPLY AIR | 7,94 | kW | | | | |
| Power from the main supply, including frequency converter, EXHAUST AIR | 7,46 | kW | | | | |
| Power from the main supply, excluding frequency converter, EXHAUST AIR | 7,08 | kW | | | | |
| SFP, with clean filter including frequency converter | 2,01 | kW/(m3/s) | | | | |
| SFP, with clean filter excluding frequency converter | 1,91 | kW/(m3/s) | | | | |
| SPF, at design pressure drop in the filter including frequency converter | 2,19 | kW/(m3/s) | | | | |
| SPF, at design pressure drop in the filter excluding frequency converter | 2,08 | kW/(m3/s) | | | | |
| Efficiency of heat exchanger | 74,2 % | | | | | |
| Efficiency of the engine | 88 % | | | | | |
| Efficiency of the fan | 79 % | | | | | |
| Sypply air: | | | | | | |
| Total pressure | 767 | Pa | | | | |
| Exhaust air: | | | | | | |
| Dynamic pressure (total pressure - static pressure) | 49 | Pa | | | | |
| Pressure losses after heat exchanger (in the aggregate incl heat exchanger) | 241 | Pa | | | | |
| Pressure losses before heat exchanger (in ducts and aggregate) | 393 | Pa | | | | |
| Heat release [kW] | | | | **Heat release entering the building** | | |
| SUPPLY AIR | | | | | | Comment: |
| Frequency converter | 0,42 | kW | | 0 | kW | palced outside the aggregate |
| Engine | 0,95 | kW | | 0,95 | kW | |
| Fan | 1,47 | kW | | 1,47 | kW | |
| Loss in ducts , aggregate parts and dynamic energy in the air supplied to rooms | 5,54 | kW | P [W]=P[Pa]*Q[m3/s] | 5,54 | kW | |
| SUM | 8,38 | kW | equal to power supply | 7,96 | kW | |
| EXHAUST AIR | | | | | | |
| Frequency converter | 0,38 | kW | | 0 | kW | palced outside the aggregate |
| Engine | 0,85 | kW | | 0 | kW | placed after the heat exhanger |
| Fan | 1,31 | kW | | 0 | kW | placed after the heat exhanger |
| Dynamic energy in the exhaust air | 0,33 | kW | P [W]=P[Pa]*Q[m3/s] | 0 | kW | |
| Losses in the aggregate after heat exchanger | 1,62 | kW | P [W]=P[Pa]*Q[m3/s] | 0 | kW | |
| Losses in ducts and aggregate before heat exchanger | 2,65 | kW | P [W]=P[Pa]*Q[m3/s] | 1,97 | kW | |
| SUM | 7,14 | kW | equal to power supply | 1,97 | kW | |
| **TOTAL HEAT FROM VENTIALTION** | **9,92** | **kW** | | | | |
| ***Given in % of power use:*** | ***64 %*** | | | | | |

| | | | | | | |
|---|---|---|---|---|---|---|
| ***In operating time*** : | | | | | | |
| **Aggregate:** | **DVN 100** | | | | | |
| **Airflow:** | **24264** | **m3/h** | | | | |
| External pressure loss: | 218 | Pa | | | | |
| | | | | | | |
| Results for simulation | | | | | | |
| Power from the main supply, including frequency converter, SUPPLY AIR | 6,68 | kW | | | | |
| Power from the main supply, excluding frequency converter, SUPPLY AIR | 6,34 | kW | | | | |
| Power from the main supply, including frequency converter, EXHAUST AIR | 5,93 | kW | | | | |
| Power from the main supply, excluding frequency converter, EXHAUST AIR | 5,63 | kW | | | | |
| | | | | | | |
| SFP, with clean filter including frequency converter | 1,69 | kW/(m3/s) | | | | |
| SFP, with clean filter excluding frequency converter | 1,61 | kW/(m3/s) | | | | |
| SPF, at design pressure drop in the filter including frequency converter | 1,87 | kW/(m3/s) | | | | |
| SPF, at design pressure drop in the filter excluding frequency converter | 1,78 | kW/(m3/s) | | | | |
| | | | | | | |
| Efficiency of heat exchanger | 74,2 % | | | | | |
| Efficiency of the engine | 88 % | | | | | |
| Efficiency of the fan | 79 % | | | | | |
| | | | | | | |
| Sypply air: | | | | | | |
| Total pressure | 656 | Pa | | | | |
| Exhaust air: | | | | | | |
| Dynamic pressure (total pressure - static pressure) | 42 | Pa | | | | |
| Pressure losses after heat exchanger (in the aggregate incl heat exchanger) | 184 | Pa | | | | |
| Pressure losses before heat exchanger (in ducts and aggregate) | 354 | Pa | | | | |
| | | | | | | |
| Heat release [kW] | | | | **Heat release entering the building** | | |
| SUPPLY AIR | | | | | | Comment: |
| Frequency converter | 0,34 | kW | | 0 | kW | palced outside the aggregate |
| Engine | 0,76 | kW | | 0,76 | kW | |
| Fan | 1,17 | kW | | 1,17 | kW | |
| Loss in ducts , aggregate parts and dynamic energy in the air supplied to rooms | 4,42 | kW | P [W]=P[Pa]*Q[m3/s] | 4,42 | kW | |
| SUM | 6,69 | kW | equal to power supply | 6,35 | kW | |
| | | | | | | |
| EXHAUST AIR | | | | | | |
| Frequency converter | 0,3 | kW | | 0 | kW | palced outside the aggregate |
| Engine | 0,68 | kW | | 0 | kW | placed after the heat exhanger |
| Fan | 1,04 | kW | | 0 | kW | placed after the heat exhanger |
| Dynamic energy in the exhaust air | 0,28 | kW | P [W]=P[Pa]*Q[m3/s] | 0 | kW | |
| Losses in the aggregate after heat exchanger | 1,24 | kW | P [W]=P[Pa]*Q[m3/s] | 0 | kW | |
| Losses in ducts and aggregate before heat exchanger | 2,39 | kW | P [W]=P[Pa]*Q[m3/s] | 1,77 | kW | |
| SUM | 5,93 | kW | equal to power supply | 1,77 | kW | |
| **TOTAL HEAT FROM VENTIALTION** | **8,12** | **kW** | | | | |
| ***Given in % of power use:*** | **64 %** | | | | | |

**In operational time**

| | | | | | | |
|---|---|---|---|---|---|---|
| **Aggregate:** | **DVN 100** | | Simulation of 24264 m3/h adjusted by theory | | | |
| **Airflow:** | **20000** | **m3/h** | | | | |
| External pressure loss: | 148 | Pa | Adjusteed by airflow squared | | | |
| | | | | | | |
| Results for simulation | | | | | | |
| Power from the main supply, including frequency converter, SUPPLY AIR | 3,98 | kW | Based | | | |
| Power from the main supply, excluding frequency converter, SUPPLY AIR | 3,79 | kW | on | | | |
| Power from the main supply, including frequency converter, EXHAUST AIR | 3,53 | kW | the SFP | | | |
| Power from the main supply, excluding frequency converter, EXHAUST AIR | 3,36 | kW | factors | | | |
| | | | | | | |
| SFP, with clean filter including frequency converter | 1,22 | kW/(m3, | Adjusted using the figure | | | |
| SFP, with clean filter excluding frequency converter | 1,16 | kW/(m3, | showing the relation | | | |
| SPF, at design pressure drop in the filter including frequency converter | 1,35 | kW/(m3, | between flow and SFP | | | |
| SPF, at design pressure drop in the filter excluding frequency converter | 1,29 | kW/(m3, | asuming "good" control | | | |
| | | | | | | |
| Efficiency of heat exchanger | 72,0 % | | *Measured | | | |
| Efficiency of the engine | 88 % | | *Not changed by this | | | |
| Efficiency of the fan | 79 % | | small adjustment in airflow | | | |
| | | | (based on systemairCAD) | | | |
| Sypply air: | | | | | | |
| Total pressure | 446 | Pa | Adjusteed by airflow squared | | | |
| Exhaust air: | | | | | | |
| Dynamic pressure (total pressure - static pressure) | 29 | Pa | Adjusteed by airflow squared | | | |
| Pressure losses after heat exchanger (in the aggregate incl heat exchanger) | 125 | Pa | Adjusteed by airflow squared | | | |
| Pressure losses before heat exchanger (in ducts and aggregate) | 241 | Pa | Adjusteed by airflow squared | | | |
| | | | | | | |
| Heat release [kW] | | | | **Heat release entering the building** | | |
| SUPPLY AIR | | | | | | Comment: |
| Frequency converter | 0,19 | kW | | 0 | kW | palced outside the aggregate |
| Engine | 0,45 | kW | | 0,45 | kW | |
| Fan | 0,70 | kW | | 0,70 | kW | |
| Loss in ducts , aggregate parts and dynamic energy in the air supplied to rooms | 2,48 | kW | P [W]=P[Pa]*Q[m3/s] | 2,48 | kW | |
| SUM | 3,82 | kW | equal to power supply | 3,63 | kW | |
| | | | | | | |
| EXHAUST AIR | | | | | | |
| Frequency converter | 0,17 | kW | | 0 | kW | palced outside the aggregate |
| Engine | 0,40 | kW | | 0 | kW | placed after the heat exhanger |
| Fan | 0,62 | kW | | 0 | kW | placed after the heat exhanger |
| Dynamic energy in the exhaust air | 0,19 | kW | P [W]=P[Pa]*Q[m3/s] | 0 | kW | |
| Losses in the aggregate after heat exchanger | 0,84 | kW | P [W]=P[Pa]*Q[m3/s] | 0 | kW | |
| Losses in ducts and aggregate before heat exchanger | 1,62 | kW | P [W]=P[Pa]*Q[m3/s] | 1,17 | kW | |
| SUM | 3,85 | kW | equal to power supply | 1,17 | kW | |
| **TOTAL HEAT FROM VENTIALTION** | **4,80** | **kW** | | | | |
| ***Given in % of power use:*** | **63 %** | | | | | |

| | | | | | | |
|---|---|---|---|---|---|---|
| **Aggregate:** | **DVN 40** | | **Projected airflow - can not be simulated - assumptions is made** | | | |
| Aggregate: | DVN 40 | | **Use results from this simulation and correct for lower airflows** | | | |
| Airflow: | 3600 | m3/h | **(lowest possible airflow and pressure simulated)** | | | |
| External pressure loss: | 137 | Pa | (32 Pa was calculated) | | | |
| | | | | | | |
| **Airflow:** | **3000** | **m3/h** | (projected during operation hours) | | | |
| External pressure loss: | 22 | Pa | (calculated) | | | |
| | | | | | | |
| Results for simulation | | | | | | |
| Power from the main supply, including frequency converter, SUPPLY AIR | 0,40 | kW | Wighted because of 352 Pa pessure loss | | | |
| Power from the main supply, excluding frequency converter, SUPPLY AIR | 0,38 | kW | for supply air and | | | |
| Power from the main supply, including frequency converter, EXHAUST AIR | 0,36 | kW | 336 Pa pressure loss for exhaust air | | | |
| Power from the main supply, excluding frequency converter, EXHAUST AIR | 0,35 | kW | at design conditions | | | |
| | | | | | | |
| SFP, with clean filter including frequency converter | 0,76 | kW/(m3/s) | | | | |
| SFP, with clean filter excluding frequency converter | 0,72 | kW/(m3/s) | | | | |
| SPF, at design pressure drop in the filter including frequency converter | 0,92 | kW/(m3, | an increase of 118Pa from 570Pa total | | | |
| SPF, at design pressure drop in the filter excluding frequency converter | 0,87 | kW/(m3, | gives: SFPdes= SFPclean*(688/570) | | | |
| | | | | | | |
| Efficiency of heat exchanger | 90 % | | Can be used, if anything it is a little better | | | |
| Efficiency of the engine | ? | | because lower airflow | | | |
| Efficiency of the fan | ? | | | | | |
| | | | | | | |
| | | | | | | |
| Supply air | | | | | | |
| Total pressure loss | 130 | Pa | corrected for lower airflow | | | |
| Exhaust air: | | | than at | | | |
| Dynamic pressure (total pressure - static pressure) | 6 | Pa | lowest simulation | | | |
| Pressure losses after heat exchanger (in the aggregate incl heat exchanger) | 36 | Pa | | | | |
| Pressure losses before heat exchanger (in ducts and aggregate) | 79 | Pa | | | | |
| | | | | | | |
| | | | | | | |
| Heat release [kW] | | | | **Heat release entering the building** | | |
| SUPPLY AIR | | | | | | |
| Frequency converter | 0,02 | kW | | 0 | kW | palced outside the aggregate |
| Engine and Fan | 0,27 | kW | Power supply - other he | 0,27 | kW | placed inside the aggregate |
| Loss in ducts , aggregate parts and dynamic energy in the air supplied to rooms | 0,11 | kW | P [W]=P[Pa]*Q[m3/s] | 0,11 | kW | |
| SUM | 0,40 | kW | equal to power supply | 0,38 | kW | |
| | | | | | | |
| | | | | | | |
| EXHAUST AIR | | | | | | |
| Frequency converter | 0,02 | kW | | 0 | kW | palced outside the aggregate |
| Engine and Fan | 0,24 | kW | Power supply - other he | 0 | kW | placed after the heat exhanger |
| Dynamic energy in the exhaust air | 0,01 | kW | P [W]=P[Pa]*Q[m3/s] | 0 | kW | placed after the heat exhanger |
| Losses in the aggregate after heat exchanger | 0,03 | kW | P [W]=P[Pa]*Q[m3/s] | 0 | kW | |
| Losses in ducts and aggregate before heat exchanger | 0,07 | kW | P [W]=P[Pa]*Q[m3/s] | 0,06 | kW | |
| SUM | 0,36 | kW | equal to power sypply | 0,06 | kW | |
| **TOTAL HEAT FROM VENTILATION** | **0,44** | **kW** | | | | |
| ***Given in % of power use:*** | **57 %** | | | | | |

| | | | | | | |
|---|---|---|---|---|---|---|
| ***Outside operational hours:*** | | | | | | |
| Aggregate: | DVN 100 | | **Use results from this simulation and correct for lower airflows** | | | |
| Airflow: | 9360 | m3/h | **(lowest possible airflow and pressure simulated)** | | | |
| External pressure loss: | 133 | Pa | (32 Pa was calculated) | | | |
| Aggregate: | DVN 40 | | | | | |
| Airflow: | 3600 | m3/h | **(lowest possible airflow and pressure simulated)** | | | |
| External pressure loss: | 32 | Pa | (calculated) | | | |
| **Aggregate:** | **DVN 100** | | **Projected airflow - can not be simulated - assumptions is made** | | | |
| **Airflow:** | **4320** | **m3/h** | (projected airflow) | | | |
| External pressure loss: | 7 | Pa | (calculated) | | | |
| Results for simulation | | | | | | |
| Power from the main supply, including frequency converter, SUPPLY AIR | 0,57 | kW | Wighted because of 352 Pa pessure loss | | | |
| Power from the main supply, excluding frequency converter, SUPPLY AIR | 0,54 | kW | for supply air and | | | |
| Power from the main supply, including frequency converter, EXHAUST AIR | 0,52 | kW | 335 Pa pressure loss for exhaust air | | | |
| Power from the main supply, excluding frequency converter, EXHAUST AIR | 0,49 | kW | at design conditions | | | |
| SFP, with clean filter including frequency converter | 0,75 | kW/(m3/s) | | | | |
| SFP, with clean filter excluding frequency converter | 0,71 | kW/(m3/s) | | | | |
| SPF, at design pressure drop in the filter including frequency converter | 0,91 | kW/(m3, | an increase of 118Pa from 569Pa total | | | |
| SPF, at design pressure drop in the filter excluding frequency converter | 0,86 | kW/(m3, | gives: SFPdes= SFPclean*(688/569) | | | |
| Efficiency of heat exchanger | 89,0 % | | Can be used, if anything it is a little better | | | |
| Efficiency of the engine | ? | | because lower airflow | | | |
| Efficiency of the fan | ? | | | | | |
| Supply air | | | | | | |
| Total pressure loss | 41 | Pa | corrected for lower airflow | | | |
| Exhaust air: | | | than at | | | |
| Dynamic pressure (total pressure - static pressure) | 7 | Pa | lowest simulation | | | |
| Pressure losses after heat exchanger (in the aggregate incl heat exchanger) | 12 | Pa | | | | |
| Pressure losses before heat exchanger (in ducts and aggregate) | 24 | Pa | | | | |
| Heat release [kW] | | | | **Heat release entering the building** | | |
| SUPPLY AIR | | | | | | |
| Frequency converter | 0,03 | kW | | 0 | kW | palced outside the aggregate |
| Engine and Fan | 0,49 | kW | Power supply - other he | 0,49 | kW | placed inside the aggregate |
| Loss in ducts , aggregate parts and dynamic energy in the air supplied to rooms | 0,05 | kW | P [W]=P[Pa]*Q[m3/s] | 0,05 | kW | |
| SUM | 0,57 | kW | equal to power supply | 0,54 | kW | |
| EXHAUST AIR | | | | | | |
| Frequency converter | 0,03 | kW | | 0 | kW | palced outside the aggregate |
| Engine and Fan | 0,44 | kW | Power supply - other he | 0 | kW | placed after the heat exhanger |
| Dynamic energy in the exhaust air | 0,01 | kW | P [W]=P[Pa]*Q[m3/s] | 0 | kW | placed after the heat exhanger |
| Losses in the aggregate after heat exchanger | 0,01 | kW | P [W]=P[Pa]*Q[m3/s] | 0 | kW | |
| Losses in ducts and aggregate before heat exchanger | 0,03 | kW | P [W]=P[Pa]*Q[m3/s] | 0,03 | kW | |
| SUM | 0,52 | kW | equal to power sypply | 0,03 | kW | |
| **TOTAL HEAT FROM VENTILATION** | **0,57** | **kW** | | | | |
| ***Given in % of power use:*** | **52 %** | | | | | |

| | | | | | | |
|---|---|---|---|---|---|---|
| **Aggregate:** | **DVN 40** | | **Projected airflow - can not be simulated - assumptions is made** | | | |
| **Airflow:** | **250** | **m3/h** | | | | |
| External pressure loss: | 0,2 | Pa | | | | |
| Results for simulation | | | | | | |
| Power from the main supply, including frequency converter, SUPPLY AIR | 0,03 | kW | Wighted because of 352 Pa pessure loss | | | |
| Power from the main supply, excluding frequency converter, SUPPLY AIR | 0,03 | kW | for supply air and | | | |
| Power from the main supply, including frequency converter, EXHAUST AIR | 0,03 | kW | 336 Pa pressure loss for exhaust air | | | |
| Power from the main supply, excluding frequency converter, EXHAUST AIR | 0,03 | kW | at design conditions | | | |
| SFP, with clean filter including frequency converter | 0,76 | kW/(m3/s) | | | | |
| SFP, with clean filter excluding frequency converter | 0,72 | kW/(m3/s) | | | | |
| SPF, at design pressure drop in the filter including frequency converter | 0,92 | kW/(m3, | an increase of 118Pa from 570Pa total | | | |
| SPF, at design pressure drop in the filter excluding frequency converter | 0,87 | kW/(m3, | gives: SFPdes= SFPclean*(688/570) | | | |
| Efficiency of heat exchanger | 90 % | | Can be used, if anything it is a little better | | | |
| Efficiency of the engine | ? | | because lower airflow | | | |
| Efficiency of the fan | ? | | | | | |
| Supply air | | | | | | |
| Total pressure loss | 1,0 | Pa | corrected for lower airflow | | | |
| Exhaust air: | | | than at | | | |
| Dynamic pressure (total pressure - static pressure) | 6 | Pa | lowest simulation | | | |
| Pressure losses after heat exchanger (in the aggregate incl heat exchanger) | 0 | Pa | | | | |
| Pressure losses before heat exchanger (in ducts and aggregate) | 0,6 | Pa | | | | |
| Heat release [kW] | | | | **Heat release entering the building** | | |
| SUPPLY AIR | | | | | | |
| Frequency converter | 0,00 | kW | | 0 | kW | palced outside the aggregate |
| Engine and Fan | 0,03 | kW | Power supply - other he | 0,03 | kW | placed inside the aggregate |
| Loss in ducts , aggregate parts and dynamic energy in the air supplied to rooms | 0,00 | kW | P [W]=P[Pa]*Q[m3/s] | 0,00 | kW | |
| SUM | 0,03 | kW | equal to power supply | 0,03 | kW | |
| EXHAUST AIR | | | | | | |
| Frequency converter | 0,00 | kW | | 0 | kW | palced outside the aggregate |
| Engine and Fan | 0,03 | kW | Power supply - other he | 0 | kW | placed after the heat exhanger |
| Dynamic energy in the exhaust air | 0,00 | kW | P [W]=P[Pa]*Q[m3/s] | 0 | kW | placed after the heat exhanger |
| Losses in the aggregate after heat exchanger | 0,00 | kW | P [W]=P[Pa]*Q[m3/s] | 0 | kW | |
| Losses in ducts and aggregate before heat exchanger | 0,00 | kW | P [W]=P[Pa]*Q[m3/s] | 0,00 | kW | |
| SUM | 0,03 | kW | equal to power sypply | 0,00 | kW | |
| **TOTAL HEAT FROM VENTILATION** | **0,03** | **kW** | | | | |
| ***Given in % of power use:*** | ***50 %*** | | | | | |

**Method of calculations:**
Systemair is the company delivering the ventilation system. Their program is used in the calculations at different airflows; SystemairCAD.

Since the aggregate used in the actual building is not from the newest collection, an older version of the program had to be used, «SystemairCAD C2010.02». This version calculates among other thing power demand, efficiencies and pressure loss in the aggregate. Extern pressure loss (loss in ducts etc.) is needed as input to the program to get a proper result and must be calculated using Bernoulli's formulas for each airflow. The newest version of the program «SystemairCAD C2014» use Bernoulli's formulas, which makes it easier to calculate operation at lower air flows. «SystemairCAD C2014» verified the assumptions made, that Bernoulli's formulas are leading to the equation:

$$\Delta P_{low\,flow} = \Delta P_{dim}(\frac{Q_{low\,flow}}{Q_{dim}})^2$$

It turned out that «SystemairCAD C2010.2» has a minimum limit for external pressure loss and airflow. The lowest airflow that is possible to simulate is 36% of design airflow. Because of safety margins used of Systemair when picking out aggregate, there were no opportunities in «SystemairCAD C2010.2» to set the extern pressure loss as low as in real life when looking at low airflows. This limitation is solved by using the formula for $\Delta P_{low\,flow}$ when calculating the extern pressure loss. The same formula is used to calculate the pressure drop in the aggregate at lower airflows then simulated.

A comparison of results from SystemairCAD indicates that the ventilation systems approximately follows the curve for «Good» control in Figure 17 from r=0,36 to 1. One must assume that this also yields using even lower airflows. This assumption is realistic considering how NS3031: 2007 calculates SFP at reduced airflow. This implies that the SPF factor is approximately constant in the range of r = 0.1 -r = 0.36 following the curve for "Good" control. At lower airflows, the pressure loss in the ducts (external pressure loss) becomes smaller and the fan system efficiency (fan, motor, frequency converter) becomes lower. This result in relatively more heat release from the aggregate components compared to the ducts.

The fact that the aggregates is not able to run simulations with the real pressure loss at the lowest possible airflow does not seem to change SFP values significantly assuming that the system follows the green curve for "good" control in Figure 17.

The calculations of useful heat are using the formulas given in chapter 3.5. The calculations simplifies dynamic pressure assuming equal dynamic pressure in the outlet and in the aggregate. «SystemairCAD C2010.2» gives both total pressure and static pressure in the aggregate, which makes it easy to find the dynamic pressure. Dynamic pressure can be written as; total pressure – static pressure.

# 10 Appendix 11 – Løvåsen cooperatives

enova

forbilde prosjekter
Rapport 2008:1

Forbildeprosjektene viser gode eksempler på boliger og bygninger med fremtidsrettede løsninger for å oppnå lavt energibruk og for bruk av fornybare energikilder.

**For informasjon om støtteprogram for Energibruk i Bolig, bygg og anlegg, se: www.enova.no/?pageId=3003**

Løvåshagen Borettslag FOTO: MIR AS

## Løvåshagen Borettslag

# Banebrytende Energisparing

**I løpet av fjerde kvartal 2008 starter overlevering av 28 passivhus i Fyllingsdalen utenfor Bergen. Ca. 3 måneder senere ferdigstilles 52 lavenergiboliger. Prosjektet Løvåshagen Borettslag består av 80 leiligheter hvor man tar energisparing på alvor. Mulig energibesparelse pr år er beregnet av SINTEF Byggforsk til ca kWh 550.000, i forhold til normal bebyggelse.**

**Beskrivelse av prosjektet**

ByBo AS bygger Løvåshagen i Fyllingsdalen, 4 km fra Bergen sentrum. Løvåshagen består av fire hus med til sammen 80 leiligheter. Mens 52 av leilighetene er lavenergiboliger, blir energibesparelsene dradd enda et hakk lengre i de 28 enhetene som bygges som passivhus.

Alle husene ligger vest og syd-vest vendt, noe som gir mye lys, lite sjenanse fra nabolaget og god kontakt med naturen.

Leilighetene er fordelt på 3-, 4- og 5-roms og varierer i størrelse fra 50 m2 til 95 m². Gjennomsnittlig leilighetsstørrelse er 80 m². Det er lagt stor vekt på universell utforming og alle leilighetene har livsløpsstandard, heis til boligetasjen og innendørs parkeringsanlegg.

»

Enova SF 1-2008

FOTO: MIR AS

Også i utearealene er det tatt ekstra hensyn til dem som trenger enkel fremkommelighet. Det er blant annet lagt opp til ledelinjer langs fortau og på den åpne piazzaen, slik at alle skal kunne bruke utearealene.

### Målsetting

Målsettingen med Løvåshagen har vært å vise at man kan bygge energibesparende boliger og at kravene til slike kan ivaretas i større prosjekter. For passivhusene er det krav om et maksimalt oppvarmingsbehov på 15 kWh/m²år mens det for lavenergiboligene siktes mot ca. 25 kWh/m²år.

Dette krever nøye utarbeidelser av bygningsmessige detaljer, både for hovedkonstruksjoner og for den enkelte leilighet. Løsningene skal være funksjonsdyktige og ikke unødvendig kostnadsdrivende. Derfor er det brukt mye og nødvendig tid på utforming av detaljer.

### Bygningskonstruksjon

Det er gjort tiltak i omtrent hele konstruksjonen for å oppnå energimålene. Til tross for at noen hovedgrep er gjort, er det summen av alle tiltakene som bidrar til å nå målet.

Kuldebroer er redusert ved å isolere inn betongkonstruksjonene og redusere gjennomføringer til et minimum.

Vinduene er av typen NorDan N-tech 3 lags vindu med argongass, isolerende avstandslister og isolert karm. U-verdien er 0,7-0,8 W/m²K. Ytterdører og balkongdører holder seg også under en U-verdi på ca 1,0 W/m²K.

Yttervegger/langvegger består av dobbeltvegg-konstruksjon med 350 mm isolasjon.

Gavlvegger har 400 mm isolasjon. Yttertak består av I-profil bjelker bygget som luftet åstak med 500 mm isolasjon. Gulv mot grunn har 350 mm isolasjon på kultlag med 100 mm påstøp. Det er brukt dobbel vindtetting med tape i alle skjøter for å minimere luftlekkasjer.

### Tekniske system

- Alle leilighetene får balansert ventilasjon med høyeffektiv roterende varmegjenvinner fra Flexit. Virkningsgrad på ca. 80 % og spesifikk vifteeffekt: SFP < 2,0 kW/(m³/s).

- Det er tilrettelagt for "inne-ute" brytere som setter boligen i "hvilemodus" når man er ute. Denne kobler ut unødvendige strømkretser når det ikke er behov for disse.

- Alle passivhusleilighetene får to solfangere hver på taket som primært gir varmtvann, men som også dekker noe av gulvvarme på bad, og varme til en radiator i stuen.

- Lavenergiboligene har elektrisk VV bereder, varmekabler på bad og en panelovn i stuen på ca. 1000 W.

Bygningene ligger godt solvendt som vil bidra til gode lys- og solforhold. Sammen med gode vinduer vil dette også bidra til deler av oppvarmingen.

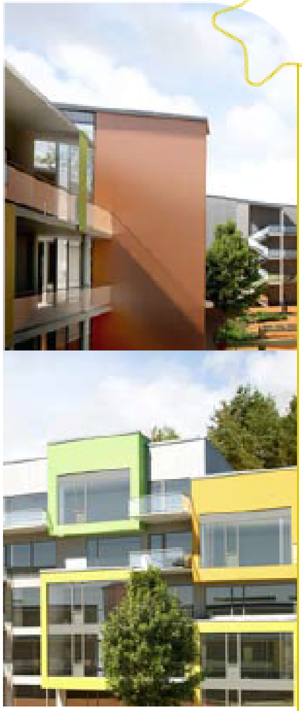
FOTO: MIR AS

### Energiytelse
Totalt netto energibehov for passivhusene er beregnet til 91 kWh/m²år, og levert energi (elektrisitet) er beregnet til 74 kWh/m²år. Solvarme d ekker 17 kWh/m²år.

Netto energibehov for lavenergiboligene er beregnet til 101 kWh/m²år, og levert energi (elektrisitet) er beregnet til det samme, 101 kWh/m²år. All energitilførsel er elektrisk.

### Planleggingsverktøy
Energiberegninge er gjort med simulerings-verktøyet SCIAQ 2.0 fra ProgramByggerne (www.programbyggerne.no). Det er brukt beregningsregler i den nye energiberegnings-standard: NS 3031 Beregning av bygningers energiytelse.

### Kostnader og lønnsomhet
Dette er ikke analysert enda da nødvendig grunnlag p.t. ikke er kjent.

### Finansiering
Løvåshagen er blitt en realitet gjennom et samarbeid mellom ByBo, Sintef Byggforsk og Husbanken. Boligene har også fått støtte som forbildeprosjekt fra Enova .

Husbanken gir kjøpere av leilighet i Løvåshagen helt spesielle lånevilkår med en nedbetalingstid på 50 år, inkludert 8 års avdragsfrihet. I tillegg har Husbanken bidratt med støtte til utviklingen av boligene.

### Fremdrift
Leilighetene vil være innflyttingsklare i andre halvdel av 2008.

### Energiytelse

| | Passivhus | Lavenergihus |
|---|---|---|
| Totalt netto energiforbruk: | 91 kWh/m²år | 101 kWh/m²år |
| Oppvarming av rom og ventilasjon: | 15 kWh/m²år | 25 kWh/m²år |
| Tappevannsoppvarming: | 30 kWh/m²år | 30 kWh/m²år |
| Vifter og pumper: | 6 kWh/m²år | 6 kWh/m²år |
| Belysning: | 17 kWh/m²år | 17 kWh/m²år |
| Teknisk utstyr: | 23 kWh/m²år | 23 kWh/m²år |
| Kjøling: | 0 kWh/m²år | 0 kWh/m²år |

### Prosjektteam
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- Arkitekt: Arkitektkontoret ABO AS, Os
- Prosjektledelse: Jan Kavlie-Jørgensen AS,
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