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ENERGY PERFORMANCE COMPARISON OF DECENTRALIZED VS. CENTRALIZED VENTILATION SYSTEMS

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Abstract

Ventilation in buildings is extremely important to create a healthy environment and prevent mold growth. The new building regulations demand an air-tight building envelope in order to improve its energy efficiency. Therefore, infiltrations and natural ventilation are often not sufficient to provide an adequate air change and mechanical ventilation becomes necessary.

Decentralized Ventilation systems (DV) with heat recovery are one of the best solutions for smaller flats, for historic buildings and for those protected by heritage. These systems are placed only in those rooms where outside air is needed. Each single unit functions fully by itself and can be individually controlled. They are normally mounted into or onto the façade and do not include long ventilation ducts. This saves space, energy for running the system as well as installation costs.

The main goal of this master's thesis is to evaluate the energy demand of a decentralized ventilation system with respect to a centralized ventilation system in a typical residential building using dynamic simulation in TRNSYS environment.

Sommario

La ventilazione negli edifici è di fondamentale importanza per garantire un ambiente confortevole e prevenire la formazione di muffe. Tuttavia le nuove normative in campo edilizio richiedono edifici impermeabili all'aria per incrementare l'efficienza energetica. Spesso, quindi, infiltrazioni e ventilazione naturale non sono più sufficienti a garantire un adeguato ricambio d'aria e diventa necessaria la ventilazione meccanica.

I sistemi di ventilazione decentralizzata con recupero termico sono una delle soluzioni migliori per piccoli appartamenti, edifici storici e vincolati. Tali sistemi sono installati solamente nelle stanze dove è necessario un ricambio d'aria esterna. Ciascuna unità è totalmente indipendente e può essere controllata individualmente. Sono spesso installate sulla facciata o al suo interno e non necessitano di lunghi condotti di ventilazione. Questo comporta un risparmio in termini di spazio, costi di funzionamento e di installazione. L'obiettivo principale di questa tesi è quello di valutare la richiesta energetica di un sistema di ventilazione decentralizzata confrontandola con quella di un sistema di ventilazione centralizzata in un tipico edificio residenziale, mediante simulazione dinamica con codice di calcolo TRNSYS.

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Introduction

Buildings in the European Union (EU) account for 40% of the EU's total final energy consumption and are responsible for about one third of global CO_2 emissions (Kamendere, 2014). By and large, heating, cooling and ventilation systems represent the majority of energy consumption of buildings, accounting for half the energy use (Moon Keun, 2014). In the EU, space heating accounts for about 26% of all final energy consumption. A study (Orme, 2001) about the energy impact of ventilation and air infiltration in residential buildings in 13 countries shows that the energy losses due to ventilation and air infiltration represent about 48% of the energy delivered for space heating.

The 1970's oil crisis caused the first wave of energy conservation campaigns in buildings. Since energy efficiency became an issue of concern in the developed countries, several regulations have been carried out. In order to reduce energy consumption of buildings, the European Performance of Building Directive (EPBD) was published in 2002. Along this, many European countries adopted rules at national level. For example, the Energieeinsparverordnung (EnEV) in Germany and the UNI/TS 11300 set in Italy have been established.

The EU has established an ambitious target that all new buildings must be nearly zero energy buildings by 2020 (Directive 2010/31/EU). If all the new buildings will be built according to the standards, the increase in energy demand will be reduced, but it would not reduce the existing demand. Only measures taken in existing buildings will have a significant effect on the total energy demands of building stock. Retrofitting of existing buildings offers a great opportunity for reducing energy consumption and greenhouse gas emissions. The first retrofit alternatives to be considered for existing residential buildings are improvement of the thermal insulation and air tightness. The usual standard renovation of multi-story apartment buildings includes facade insulation, end wall insulation, thermal insulation of the attic and basement ceiling and replacement of old windows, and heat substation renovation. To reduce energy consumption even more and improve the indoor climate, the heat distribution systems and ventilation systems have to be addressed.

In fact, heat losses by ventilation become the main source of heat losses after proper thermal insulation of the building envelope. As until the 1990s building were designed with natural ventilation, they become airtight buildings after renovation and the air infiltration rate is often not sufficient to maintain acceptable indoor air quality. Shortcuts between different ventilation ducts often become a problem after renovation, resulting in both high discomfort and energy losses. Indoor air quality is greatly influenced by the amount of fresh air that enters the building. This fresh air replaces the exhausted air, which contains biological pollutants, excess moisture, and volatile organic compounds released from building materials, carpets, furniture, and other household items as a result of aging, decomposition, or curing. A good solution in such buildings is mechanical ventilation with heat recovery to recover the heat from exhaust air and, in such way, to reduce the ventilation heat losses. A good strategy for maintaining good indoor climate and reduce energy consumption in a building would be airtight a building with a mechanical ventilation system and heat recovery. This solution may satisfy energy conservation goals in a cost-effective manner without reducing the indoor air quality.

Indoor climate control means satisfying some requirements on the following items, in order of decreasing priorities:

- Air quality

- Environmental temperature

- Air moisture

Developing more and more efficient ventilation techniques is a key issue, in the design of new “low” (and “zero”) energy buildings, as well as in the retrofit of existing ones.

The primary purpose of ventilation in dwellings is mainly to maintain good air quality by diluting air pollutants. The concentration of pollutant depends on the emission rate and on the ventilation rate. With a given emission rate, as ventilation rate increases, the steady state pollutant concentration is reduced. Therefore, an acceptable ‘comfort’ and ‘safe’ concentration of pollutant has to be identified, in order to maintain the concentration to or below this level. Buildings became, as Le Corbusier portended during the 1930s, of the same type: hermetically closed and controlled at a constant temperature in all climates. Improved airtightness and intensive weatherization actions considerably reduced the fresh air infiltration through the building envelope. As a consequence of this, the air infiltration rate is often below the minimum level to ensure good air quality. In the last decades, global epidemics of respiratory diseases break out unexpectedly: for example the smallpox outbreak in Meschede, Germany (1970), measles (1985), tuberculosis (1990), SARS (2003) and H1N1 (2009), with disastrous results in many countries (Cao, 2014). When buildings are built ever more air-tight, indoor air quality and health issues emerge. The wide variety of symptoms and physical discomfort were baptized ‘sick building syndrome’, with consequence on productivity in office buildings. Other health effects, such as inflammation, infections, asthma and allergic diseases also increased in recent years. In this contest, mechanical ventilation became more and more important, especially in new buildings.

Chapter 1

State-of-the-art

In this chapter, the state-of-the-art of mechanical ventilation systems is summarized, with a focus on the different configurations and the regulation at international and European level.

Ventilation in buildings is the process of replacing contaminated indoor air with fresh outside air. Today there is a variety of ventilation strategies in various European countries. As reported in a review on ventilation in European dwellings (Dimitroulopoulou, 2012), in some countries, uncontrolled air infiltration and window opening is often the only ventilation, while in others, passive stack ventilation systems are more or less used. In countries with colder climates, mechanical systems have been installed, with different configurations.

The former method is known as natural ventilation and it is widely used in southern Europe, where the climate conditions are warmer than in northern countries. In the latter method, i.e., mechanical ventilation, the airflow is distributed by means of fans and ductwork arrangement throughout the building and then distributed in the room via air terminal devices or diffusers. Characteristic air movement within a space can be described via two main principles of flow patterns: entrainment flow and stratified flow. Entrainment flow is known as mixing. When there is poor mixing in a room, short-circulating flow appears, as a result of leaving much of the supplied air from the room unmixed. Almost no mixing of the room air is achieved in a fully stratified flow (displacement) in the occupied zone, which is desirable for removing pollutants that have been generated in a room. Thus, it is necessary to know the characteristics of the ventilation for the purpose of design, in such a way that the occupants experience good thermal comfort and air quality. Thermal comfort and air quality are indoor environment parameters that are influenced by the air distribution system. An improper selection of the air distribution system can result in an unacceptable velocity, unacceptable temperature gradients and air stagnation in the occupied zone, which could lead to occupier discomfort. Based on the ASHRAE Standard 55 (2004), the occupied zone is generally considered the volume between the floor and a height of 1.8 m from the floor, with a distance of 1.0 m from the supply device and external wall (opposite wall). The occupied zone has a distance of 0.3 m from the internal walls (Janbakhsh, 2015). A study (Cao, 2014) classifies known ventilation systems used in different buildings types into eight ventilation methods based on concentration distribution, the location of the air supply/exhaust device and the use of natural and mechanical forces. Based on the concentration distribution, for example, ventilation methods can be classified into fully mixing system or non-uniform system, the latter can be further divided into piston, displacement ventilation, stratum ventilation and protected occupied zone ventilation.

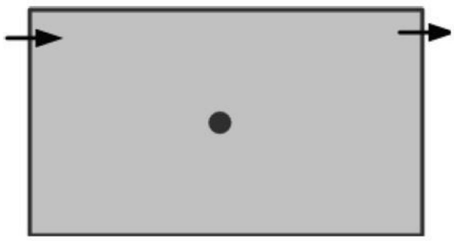

Figure 1.1: Principle of mixing ventilation.

The mixing ventilation, already illustrated at the end of 19th century, consists in diluting the contaminated room air by mixing the supplied fresh air with indoor air to lower the contaminant concentrations. The fresh air is supplied at a relative high velocity to the room and it is often distributed via a ceiling supply device. A high degree of mixing is created due to the entrainment of the room air in the supply jet. Mixing systems ideally maintain velocities of less than 0.25 m/s in the occupied zone. After many developments, the resulting temperature and contaminant concentration in the occupied zone should be uniform with a properly designed system. A proper design of a mixing ventilation can be used for cooling and heating as well as ventilation purposes. A mixing ventilation is the most common of the air distribution systems. One practical guidebook was recently published by REHVA (Müller, 2013). Figure 1.1 shows the principle of mixing ventilation.

A displacement ventilation system (Figure 1.2), unlike mixing ventilation, is based on the principle of displacing contaminated room air with fresh air from outside. Cool air is normally supplied at low velocity (typically < 0.5 m/s) at or near the floor to create an upward air movement (thermal plumes) as it is warmed up by heat sources in the room and it is removed at the ceiling level. This will normally create vertical gradients of air velocity, temperature and contaminant concentration. Displacement ventilation has two distinct zones that form in the room, which are called lower zone and upper zone. The zone with little or no recirculation (lower zone) has clean and fresh air, while the zone with recirculation (upper zone) is occupied with warm and more contaminant air. The system is a promising ventilation concept due to its high air-exchange efficiency and ventilation effectiveness. A limitation in displacement ventilation is that it might not be used for heating purposes (i.e., supplying an air temperature that is higher than the room air temperature). Displacement ventilation has a limited penetration distance in a room that has multiple buoyancy sources. It can be used in a cooling mode because buoyancy is the dominant force that drives air from the floor to the ceiling.

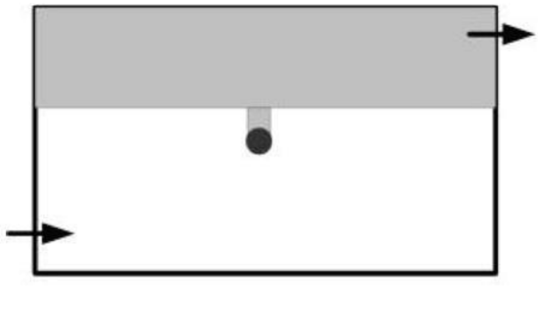

Figure 1.2: Principle of displacement ventilation.

A hybrid air supply system combines the characteristics of both mixing and displacement ventilation systems, using recently developed distribution systems based on high momentum jets, such as the impinging jet system and the confluent jets system. The former uses a duct or opening for supplying a jet of air down toward the floor so that it spreads over a large area of the floor. Impinging jet ventilation can combine the positive effects of both mixing and displacement systems. The jet it produces has a higher momentum than that for displacement ventilation and it can therefore spread more evenly over the floor. As a result, the system can provide a clean air zone in the lower part of the occupied zone like displacement ventilation but is capable of reaching further positions in the room than a displacement ventilation system. In addition, it is possible to use an impinging jet system in the heating mode as well as the cooling mode. In confluent jet system, a number of jets issuing from closely spaced slots or circular apertures in the same flow directions merge together a short distance downstream to form a single jet normally close to a room surface such as a wall or floor. The combined jets are then directed toward the floor to create a similar effect to that from an impinging jet system, thus producing a greater horizontal spread over the floor than a displacement jet system. The characteristics of confluent jet method are similar to the impinging jet in terms of being a higher momentum air supply rather than a buoyancy-driven flow as it is in the case of a displacement ventilation system.

During the last two decades, many studies have been performed to improve the air quality of a person's working environment via personalized ventilation. Using different types of air supply devices, this method can supply high-quality air directly to the exposure region. Up to 80% of inhaled air could consist of fresh personalized air with a supply flow rate less than 3.0 l/s. Moreover, a significant energy saving compared with mixing ventilation is obtained.

Stratum ventilation has been proposed to accommodate the elevation in room temperature. Besides thermal comfort, indoor air quality and energy consumption, one of the key issues in elevating the performance of stratum ventilation is weather this air distribution method offers significant performance advantages in combatting airborne infection.

A protected occupied zone ventilation has been developed to separate an office environment into several zones, by using a low turbulent plane jet. The plane jet diffusers creating the plane jet curtains are located above the height of the people working in the office. These plane jet curtains prevent the possibly polluted air from moving from one subzone to another.

Local exhaust ventilation (Figure 1.3) is primarily an extract ventilation system which is very effective in rooms where localized contaminant sources can be identified, such as in industrial premises or kitchens. Usually, an extractor hood is placed above the source to remove the pollution before it can spread into the room.

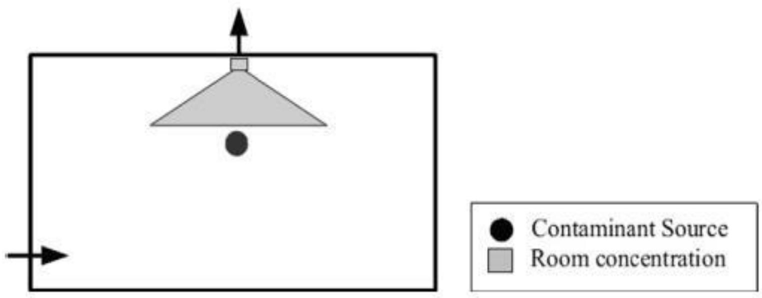

Figure 1.3: Principle of local exhaust ventilation.

In piston ventilation (Figure 1.4), air is supplied vertically or horizontally across the whole room at low velocity (typically 0.2-0.4 m/s) and turbulence to create

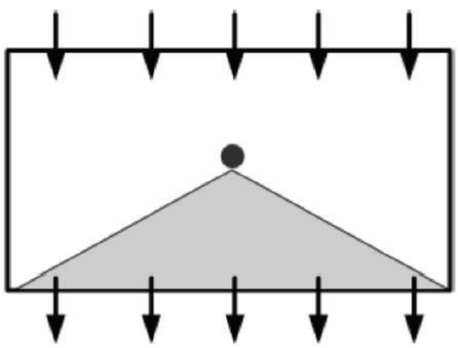

Figure 1.4: Principle of piston ventilation.

a “piston” type flow. This is a very effective way of removing contaminants from the room but it is also costly and requires a very high air change rate (200-600 ACH). Hence, it is only used in certain applications, such as clean room and hospital operating theatres.

1.1 Regulations on ventilation

On European level, there are two directives that relate to ventilation, but in an indirect way: the Construction Product Directive (CPD, 1989), and the Energy Performance of Building Directive (EPBD, 2002). The CPD requires construction product standards that relate to hygiene, health and the environment to be aligned within the member countries, whereas EPBD mentions that energy “requirements shall take account of general indoor climate conditions, in order to avoid possible negative effects such as inadequate ventilation”. The European Committee for Standardization (CEN) is the body responsible for most of the standards relating to ventilation. Each country has its own regulations, for example, the standard that is currently well known in Germany is the PassivHaus standard (Feist W., 2013). This was developed in Germany and its development was funded by EC between 1997 and 2002, in order to be a leading standard for energy efficient design and construction for cold-climate houses. The *PassivHaus Programme*, with over 5000 houses built prior to 2005, has promoted development and production of components such as low capacity compact HVAC units, windows and doors and innovative building systems. This standard is used in Germany, Austria and Ireland. The average expected ventilation rates, based on modelling work are given in the PassivHaus Planning Package (Feist W., 2013). It is recommended that in winter the ventilation rates should be lowered to maintain internal humidity values at acceptable levels. Other German standards related to ventilation are for example DIN 1946-6 (Building ventilation standard) and DIN 1807-3 (Ventilation in bathrooms and kitchen without windows).

In Italy the national standard related to ventilation are UNI/TS 11300-1 (Energy demand determination for air conditioning in summer and winter conditions) and UNI 10339 (Requirements of aeraulic systems for the purpose of welfare)

On European and international level, there are some standards related to thermal comfort and indoor air quality. These standards are EN ISO 7730 (Ergonomics of the thermal comfort), EN 15251 (Indoor environmental criteria), EN 13779 (Mechanical ventilation of non-residential buildings, Indoor air quality), EN 15242 (Airflow rates for building ventilation). American standards are also considered, such as ASHRAE 55 (Thermal environmental conditions for human occupancy) and ASHRAE 62.1 (Ventilation for acceptable indoor air quality).

The European standard EN 15251 is used to calculate the recommended design ventilation rates in the residential building. This standard states that indoor air quality achieved depends mainly on three criteria:

- Exhaust of pollutions in wet rooms (bathroom, kitchen, toilets).
- General ventilation of all rooms in the dwelling.
- General ventilation of all room in the dwelling with fresh air criteria in the main room (bed and living rooms).

It is admitted that some factors in these regulations have generally an impact on achieved indoor air quality. For instance, the criteria may be expressed by three different methods:

- Requiring exhaust in the wet rooms is necessary to remove local pollutions in these areas (depression is also necessary).
- Requiring a general ventilation (all rooms to be ventilated). This requirement is generally also allowing some transfer from main rooms (living and bed rooms) via corridors to wet rooms (bathroom, kitchen, toilet).
- Some regulations consider the overall ventilation rate in the building, others have added emphasize on the minimum supply air per bedroom and living

room. This addition allows for the same overall level a better indoor air quality because systems have to adapt and deliver in the appropriate room where the real occupation is.

The ventilation air flows of the bedrooms and living rooms for general ventilation in the dwelling are expressed as:

- An air change per hour for each room and/or outside air supply to achieve a requirement in the main rooms.
- Required exhaust rates (bathroom, toilets, and kitchen).

The supply air to kitchen, bathroom and toilet may be the transfer air from the bedrooms and living rooms.

Design ventilation airflow rates are presented in Annex B of the EN 15251. Assuming different criteria for the PPD-PMV (EN ISO 7730), different categories (Table 1.1) of the indoor environment are established. In this work, category II is considered. Recommended values for ventilation airflow rates in dwellings can be found in Table 1.2.

Table 1.1: Description of the applicability of the categories used.

Category	**Explanation**
I	High level of expectation and is recommended for spaces occupied by very sensitive and fragile persons with special requirements like some disabilities, sick, very young children and elderly persons, to increase accessibility.
II	Normal level of expectation
III	An acceptable, moderate level of expectation
IV	This category should only be accepted for a limited part of the year

The number of occupants in a dwelling can be estimated from the number of beds. When the values of any specific category in the table leads to different values of the ventilation depending on the number of occupants, floor area and number of wet rooms the following principle should be used:

- Calculate total ventilation rate for the dwelling based on floor area or air change per hour.
- Calculate supply ventilation rate for main rooms (bedrooms and living room)
- Select the higher value from above.
- Adjust the exhaust airflows from kitchen, bathroom and toilet accordingly: in dwellings with small floor area exhaust air flow rates become smaller and in large dwellings higher.
- Outdoor air should be supplied primarily to living rooms and bedrooms.

In residential buildings during unoccupied periods, i.e., when there is no ventilation demand, a minimum ventilation rate between 0.05 to 0.1 $l/s \cdot m^2$ is recommended, in order to provide an adequate air change to avoid problems to the building, e.g., mold growth.

Table 1.2: Design ventilation airflow rates for different categories with three methods (total, supply, exhaust). Ach values correspond to $l/s \cdot m^2$ when height of the rooms is 2.5 m.

Category	Total ventilation including air infiltration		Supply air flow for main rooms		Exhaust air flow, l/s		
	$l/s,m^2$	ach	$l/s \cdot$ per s	$l/s/m^2$	Kitchen	Bathrooms	Toilets
I	0.49	0.7	10	1.4	28	20	14
II	0.42	0.6	7	1.0	20	15	10
III	0.35	0.5	4	0.6	14	10	7

Chapter 2

Methodology

A first analysis of the literature is conducted in order to characterize the state-of-the-art of the ventilation systems. In this work, two different configurations of mechanical ventilation systems are considered and compared. Those are centralized and decentralized ventilation systems with heat recovery. Two cities are considered, in order to compare two different climate conditions and typical building characteristics, in terms of level of insulation. The first city is Stuttgart, Germany; the second one is Palermo, Italy. A typical residential building is considered for both countries, respectively. Models in TRNSYS environment are realized for the simulations. Weather condition, internal gains and occupancy patterns data are taken from the literature. Ventilation devices and duct systems are compared with real models on the market.

2.1 Configurations

Different configurations of mechanical ventilation systems are available on the market. A first distinction can be made based on the number of devices. Centralized ventilation systems have one device that supply/extract air to/from the whole building. Conversely, decentralized ventilation systems have one device in each room in which an air change is required. The former can be further divided into several configurations. Mechanical extract ventilation systems (MEV) creates a depression in the rooms by extracting air by means of a fan. Fresh air enters through openings (e.g., windows, grills) thanks to the pressure gradient. Figure 2.1 shows an example of a MEV system.

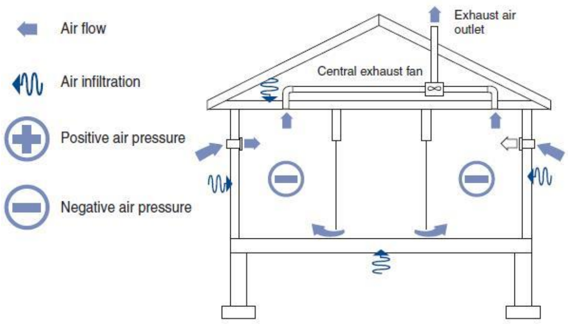

Figure 2.1: Mechanical extract ventilation system.

Supply-only ventilation systems (SOV) supply fresh air in the rooms by a fan and a duct system, creating a positive pressure in the building. This pressure gradient makes the exhaust air flow outside through openings in the envelope. Figure 2.2 shows the principle of a SOV system.

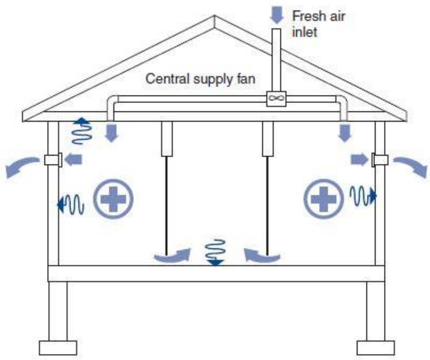

Figure 2.2: Supply-only ventilation system.

Balanced ventilation uses a supply fan and an exhaust fan to regularly exchange indoor air; both fans move similar volumes of air, so indoor pressure fluctuates near neutral or “balanced” From a safety and health perspective, balanced pressure is better than negative indoor pressure, but not as beneficial as positive indoor pressure, which helps keep outdoor pollutants outdoors. This configuration can have a heat recovery to exchange thermal energy between the two airflows; for example, in winter conditions the exhaust air heats the fresh outside air, in order to save energy in terms of heating demand. Figure 2.3 shows an example of balanced ventilation system with heat recovery.

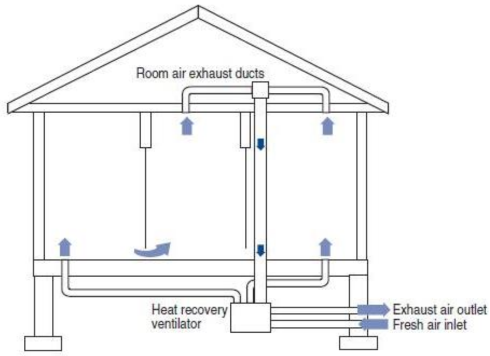

Figure 2.3: Balanced ventilation system with heat recovery.

A decentralized ventilation system counts several devices that act independently like balanced ventilation systems. Each device supplies and extracts air in a different room and can have a heat recovery to save energy as well as to improve the thermal comfort. This system does not include long ventilation ducts. An example of decentralized ventilation system can be found in Figure 2.4. In this work, just the last two configurations are considered, i.e., balanced centralized system and decentralized system.

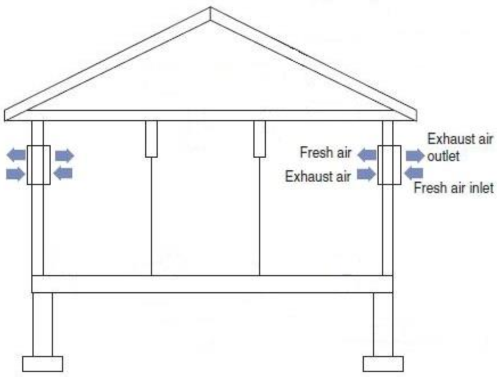

Figure 2.4: Decentralized ventilation system.

2.2 Comfort criteria

Providing a thermal comfort environment and acceptable air quality is one of the important parameters to be considered when designing a ventilation system. According to the European standard EN 13779, ventilation, air-conditioning or room-conditioning systems influence the following parameters:

- Thermal environment.
- Indoor air quality.
- Indoor air humidity.
- Acoustic environment.

2.2.1 Thermal environment

The most important design assumptions with respect to the thermal environment are the clothing and the activity of the occupants. Thermal comfort with given clothing and activity is therefore mainly due to the operative temperature and the air velocity. Further influences such as the vertical air temperature gradient, warm and cold floors and radiant asymmetry should be considered. A human being's thermal sensation is mainly related to the thermal balance of his or her body as a whole. This balance is influenced by parameters listed above. When these factors have been estimated or measured, the thermal sensation for the body as a whole can be predicted by calculating the predicted mean vote (PMV) and the predicted percentage dissatisfied (PPD), according to the international standard ISO 7730.

2.2.2 Predicted mean vote

The PMV is an index that predicts the mean value of a large group of people on the 7-point thermal sensation scale (see Table 2.1), based on the heat balance of the human body. Thermal balance is obtained when the internal heat production in the body is equal to the loss of heat to the environment. In a moderate environment, the human thermoregulatory system will automatically attempt to modify skin temperature and sweat secretion to maintain heat balance.

Table 2.1: PMV 7-point scale.

+3	Hot
+2	Warm
+1	Slightly warm
0	Neutral
-1	Slightly cool
-2	Cool
-3	Cold

Equations 1) to 4) are used to calculate the PMV:

$$\begin{aligned} PMV = {} & [0.303 \cdot \exp\ -0.036 \cdot M\ \ + 0.028] \cdot\ \ M - W\ \ - 3.05 \cdot 10^{-3} \\ & \cdot [5733 - 6.99 \cdot\ \ M - W\ \ - p_a] - 0.42 \cdot [\ M - W\ \ - 58.15] \\ & - 1.7 \cdot 10^{-5} \cdot M \cdot\ \ 5867 - p_a\ \ - 0.0014 \cdot M \cdot\ \ 34 - t_a\ \ - 3.96 \\ & \cdot 10?-8 \cdot f_{cl} \cdot [\ t_{cl} + 273\ ^4 -\ \ t_{mr} + 273\ ^4] - f_{cl} \cdot h_c \cdot (t_c \\ & - t_a) \end{aligned} \quad 1)$$

Where:

$$\begin{aligned} t_{cl} = {} & 35.7 - 0.028 \cdot\ \ M - W\ \ - I_{cl} \\ & \cdot \{3.96 \cdot 10^{-8} \cdot f_{cl} \cdot [\ t_{cl} + 273\ ^4 -\ \ t_{mr} + 273\ ^4] + f_{cl} \\ & \cdot h_c \cdot\ \ t_{cl} - t_a\ \ \} \end{aligned} \quad 2)$$

$$h_c = \begin{Bmatrix} 2.38 \cdot |t_{cl} - t_a|^{0.25} & for\ 2.38 \cdot |t_{cl} - t_a|^{0.25} > 12.1 \cdot \sqrt{v\_ar} \\ 12.1 \cdot \sqrt{v\_ar} & for\ 2.38 \cdot |t_{cl} - t_a|^{0.25} > 12.1 \cdot \sqrt{v\_ar} \end{Bmatrix} \quad 3)$$

$$f_{cl} = \begin{Bmatrix} 1.00 + 1.290 l_{cl} & for\ l_{cl} \leq 0.078\ m^2 \cdot K/W \\ 1.05 + 0.645 l_{cl} & for\ l_{cl} > 0.078\ m^2 \cdot K/W \end{Bmatrix} \quad 4)$$

Where:

M is the metabolic rate, in Watt per square meter (W/m^2);

W is the effective mechanical power, in Watt per square meter (W/m^2);

I_{cl} is the clothing insulation, in square meters Kelvin per Watt ($m^2 \cdot K/W$);

f_{cl} is the clothing surface area factor;

t_a is the indoor air temperature, in degree Celsius (°C);

t_{mr} is the mean radiant temperature, in degree Celsius (°C);

v_{ar} is the relative air velocity, in meters per second (m/s);

p_a is the water vapor partial pressure, in Pascal (Pa);

h_c is the convective heat transfer coefficient, in Watt per square meter Kelvin [$W/(m^2 \cdot K)$];

t_{cl} is the clothing surface temperature, in degree Celsius (°C).

The equations for t_{cl} and h_c may be solved by iteration. By setting $PMV = 0$, an equation is established which predicts combinations of activity, clothing and environmental parameters which on average will provide a thermally neutral sensation.

2.2.3 Predicted People Dissatisfied

The PPD predicts the mean value of the thermal votes of a large group of people exposed to the same environment. However, individual votes are scattered around this mean value and it is useful to be able to predict the number of people likely to feel uncomfortably warm or cool. Thermally dissatisfied people are those who will vote hot, warm, cool or cold on the seven-point thermal sensation scale given in Table 2.2.

With the PMV value determined, the PPD can be calculated using Equation 5, see Figure 2.1.

$$PPD = 100 - 95 \cdot \exp(-0.03353 \cdot PMV^4 - 0.2179 \cdot PMV^2) \qquad 5)$$

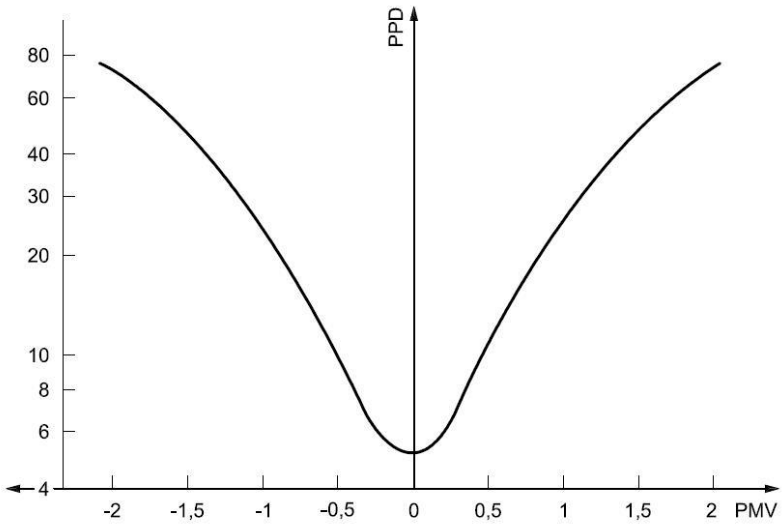

Figure 2.1: PPD as function of PMV.

Chapter 3

The model

3.1 Introduction to the simulation program TRNSYS

TRNSYS (Transient Systems Simulation) is a graphically based software environment used to simulate the behavior of transient systems. It was developed in 1975 at the University of Wisconsin in Madison for the simulation of solar thermal systems. Today the program is extended with modules and it is widely used in simulating the performance of thermal and electrical energy system, e.g., a multi-zone building. Engineers and researchers use it to validate new energy concepts, such as energy plants in buildings, integrating control strategies, behavior of occupants and renewable energy systems. One of the key factors of the program is its modular structure. A "black box" component approach is taken to develop and solve simulations: the outputs of one component are sent to the inputs of another component (transient successive substitution). The source code and the models of different components are open to the user. This simplifies the editing and extension of existing models, in order to make them suitable for the specific purposes of the user. Thanks to the architecture based on the file extension DLL, it is possible to add custom templates of components, using a common programming language (e.g., C, C++, Fortran, etc.…). In addition, TRNSYS can be easily connected with many other applications (e.g., Microsoft Excel, Matlab, Comis, etc.…). TRNSYS is made up of two parts. The first is an engine (called the kernel) that reads and processes the input file, iteratively solves the system, determines convergence, and plots system variables. The kernel also provides utilities that (among other things) determine thermophysical properties, invert matrices, perform linear regressions, and interpolate external data files. The second part of TRNSYS is an extensive library of components, each of which models the performance of one part of the system. The standard library includes approximately 150 models, called types, ranging from pumps to multi-zone buildings, wind turbines to electrolyzers, weather data processors to economics routines, and basic HVAC equipment to innovative technologies. Models are constructed in such a way that users can modify existing components or write their own, extending the capabilities of the environment. The version used in this work is TRNSYS 17, which operates with a graphical user interface in the Simulation Studio. The program generates a deck file to summarize in a text file the information from the Simulation Studio. For each component, parameters and inputs can be assigned. Parameters are fixed values that do not change over the course of the simulation. Input values change from one time step to another and are usually obtained from

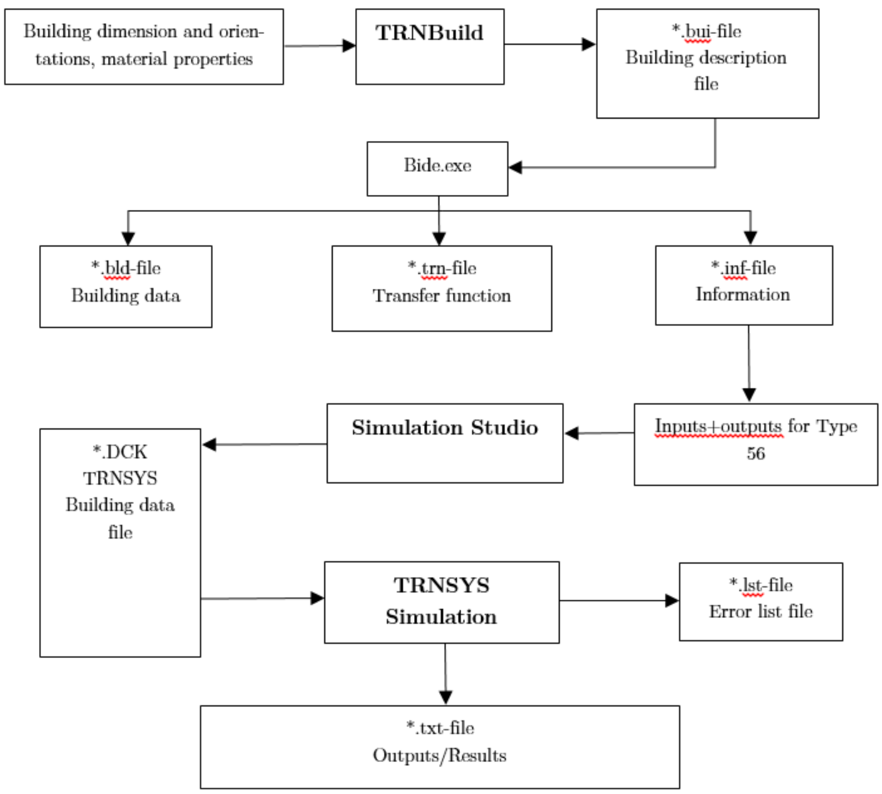

Figure 3.1: Interaction between the simulation environment and its interfaces.

the outputs of other components. The simulation time and the time step can be adjusted depending on the task. Plotter and printer can be used for the result output, in order to display the results easily and to write them down in a text file. Figure 3.1 shows the interaction between the simulation environment and its interfaces.

3.2 Building model

A typical German dwelling designed for four people is considered in this work. The building is a single-family detached home and it consists of two stories. In the ground floor, there are a kitchen, a guestroom, a living room, a toilet and a corridor. In the first floor, there are two children's bedrooms, one parent's bedroom, one bathroom and a corridor. Over the first floor, there is an uninhabitable attic. Each room, included the attic, is considered as a thermal zone with a single air node, so the model counts 11 thermal zones. The motivation of this choice should be found in the aim to consider each room thermally independent, with doors closed and, in simulations with decentralized ventilation, with independent control on ventilation flow rates. The building's walls face the four cardinal direction with a view factor of the sky of 0.5. The two roof pitches face the East and West directions with an inclination of 45° and have a view factor of the sky equal to 0.85. The height of the rooms is 2.5 meters. The windows are oriented towards the north, the south and the east. Each story has a floor surface of 69.1 m^2, for a total net floor surface of 138.2 m^2. The total volume of the dwelling is 329.1 m^3. The roof pitches are inclined at 45 degrees and they are oriented to the east and the west. Figure 4.1A and Figure 4.2A show the plan of the building model.

Visual interface TRNBuild, known as Type 56, is used to model the building. This Type is compliant with the requirements of ANSI/ASHRAE Standard 140-2001. The level of detail of Type 56 also meets the general technical requirements of the European Directive on the Energy Performance of Buildings. TRNBuild generates a building file (*.BUI) containing the required information, such as structure details, building material properties, heating, cooling and ventilation schedules, infiltration rate, internal heat gains, etc.

The exterior walls are made of calcareous sandstone with polystyrene as external insulation and plaster on both sides. The total thickness is 0.385 m and the U-value is 0.2 W/m^2K.

The interior walls are made of gypsum and mineral wool, with a total thickness of 0.126 m and U-value of 0.358 W/m^2K.

The floor layers, starting from the innermost one, are the following: tile, cement screed, polystyrene, reinforced concrete, and plaster (this one for first floor only). The total thickness is 0.455 m and the U-value is 0.199 W/m^2K.

The roof layers are the following: oriented strand board, polystyrene, a vapor barrier and roof tile. The total thickness is 0.225 m and the U-value is 0.218 W/m^2K.

The U-values are according to the German standard EnEV 2013 (Energieeinsparverordnung). Infiltration rate, set temperatures for heating and cooling, internal heat gains, solar radiation, and ventilation rate are taken as inputs from the Simulation Studio. The heating system has unlimited power with 30% radiative part and without humidification. The cooling system has unlimited power with dehumidification. Internal heat gains are divided into a radiative power component and convective power component. Airflow of the ventilation system is considered as mass flow rate (kg/s) and, as well as the temperature of air flow, is an input from the Simulation Studio. The humidity of airflow is the relative humidity of the outside air, but condensation inside the ventilation device is taken into account. Figure 3.2 shows the program interface.

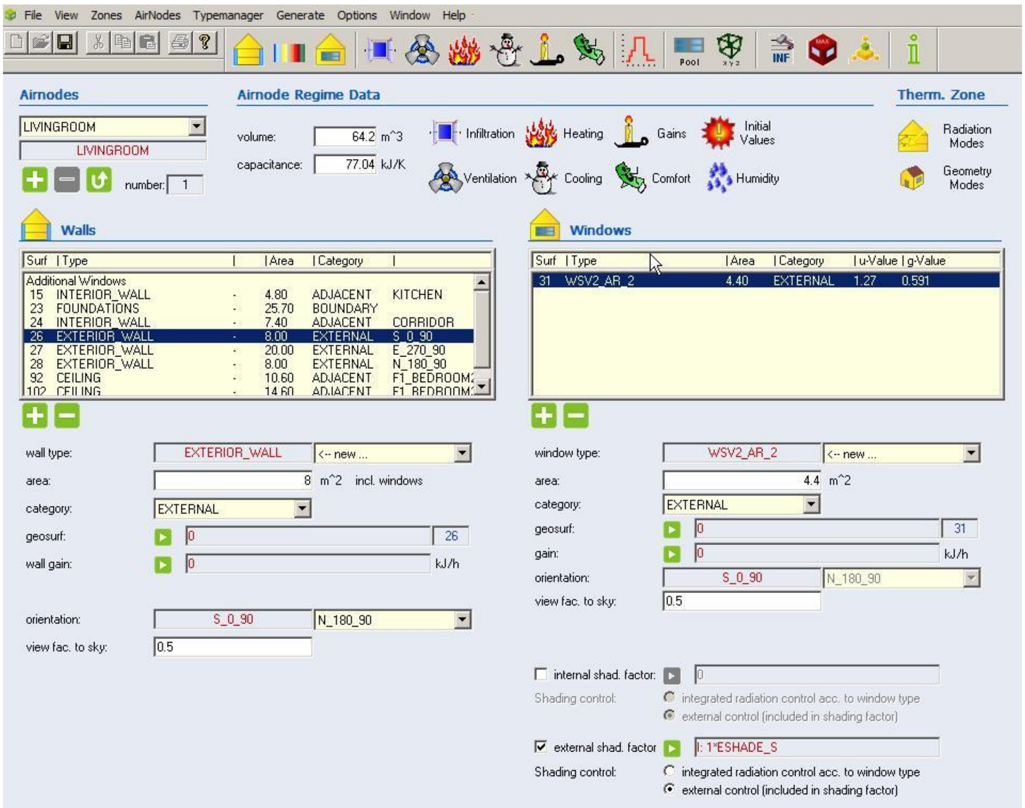

Figure 3.2: TRNBuild interface.

3.3 Simulation Studio

Simulation Studio is a complete simulation package containing several tools, from simulation engine programs and graphical connection programs to plotting and spreadsheet software. It is an integrated tool that can be used from the design of a project to its simulation. Simulation Studio reads inputs from Txt-files, equations, building and weather data, forcing functions and other external files. During the simulation, the program plots the results and write them down in a Txt-file. In Simulation Studio a library of types is available, with different models ready to be used. User-defined types can be created using a programming language or connecting an external program like Microsoft Excel or Matlab. Parameters have to be set in most types, as well as inputs from other types or equations. All the types of the model are connected using arrows that determine the direction of the information flow. In order to make the workspace more clear, the user can define macros that contain parts of the model. Figure 3.3 shows the scheme of the Simulation Studio's graphical user interface.

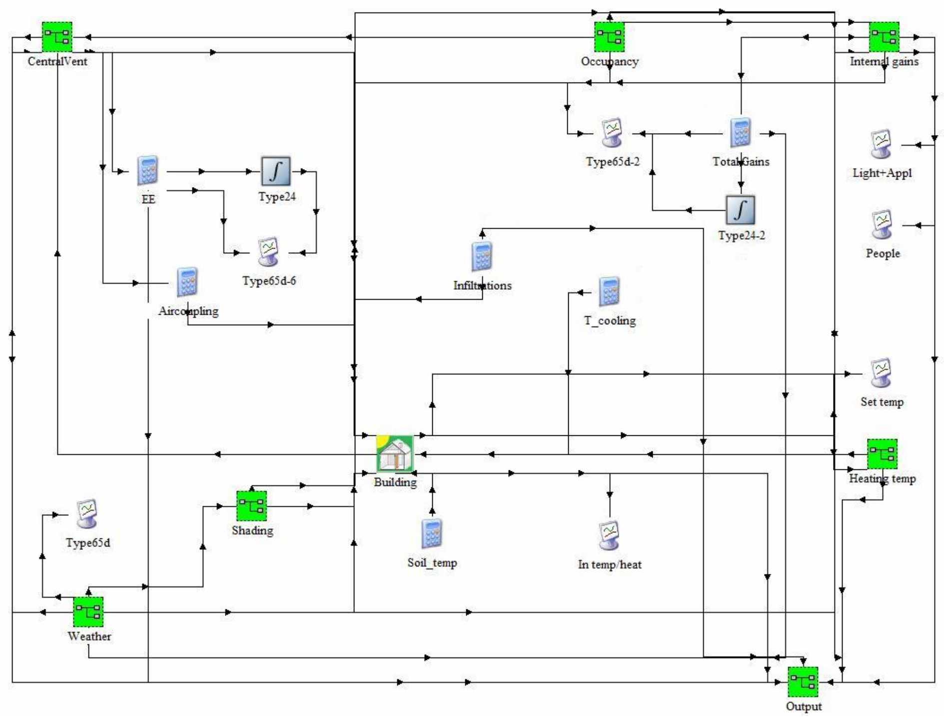

Figure 3.3: The Simulation Studio interface.

3.4 Weather data

Two different cities are considered in this work, in order to simulate the influence of the climate conditions on the behavior of the system: Stuttgart and Palermo.

Stuttgart is the capital of the state of Baden-Württemberg in southwest Germany. Stuttgart experiences an oceanic climate (Köppen climate classification: Cfb). On average, Stuttgart enjoys 1807 hours of sunshine per year. The average annual ambient air temperature is 10.2°C. The coldest month is January with an average temperature of 0°C. The summers have an average temperature of 20°C in the hottest months of July and August. A soil temperature of 10°C is assumed, according to the average annual air temperature in Stuttgart. Figure 3.4 shows the external temperature all over the year.

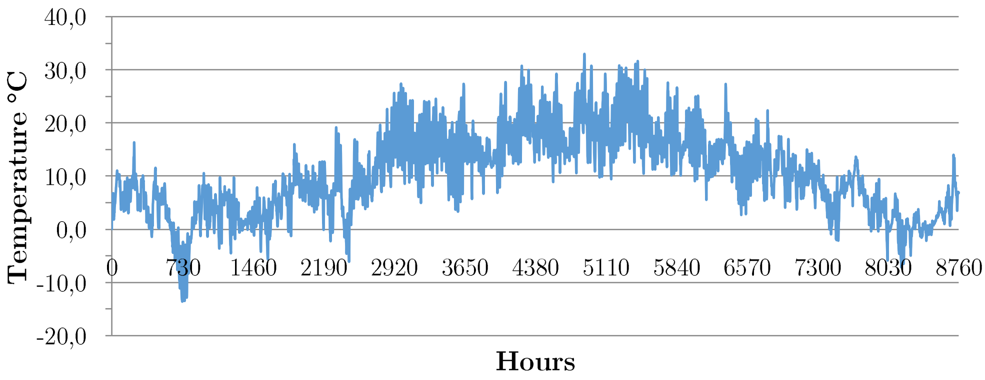

Figure 3.4: External temperature in Stuttgart.

Palermo is a city in Insular Italy, the capital of the autonomous region of Sicily. Palermo experiences a hot-summer Mediterranean climate (Köppen climate classification: Csa). Winters are cool and wet, while summers are hot and dry. Palermo is one of the warmest cities in Europe, with an average annual ambient air temperature of 18.5°C. It receives approximately 2530 hours of sunshine per year. The coldest month is January with an average temperature of 12.1°C. The hottest month is August with an average temperature of 26.2°C. Figure 3.5 shows the external temperature all over the year.

The weather data are taken from EnergyPlus database. The weather file contains information about, temperatures, total and beam solar radiation on different

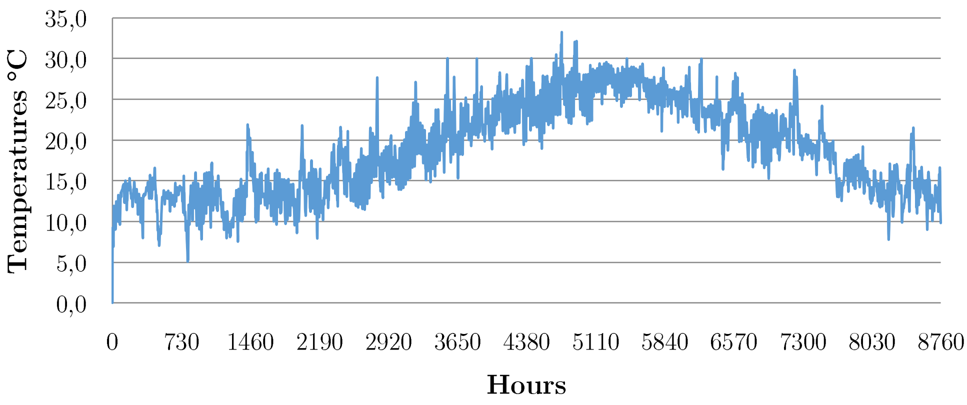

Figure 3.5: External temperature in Palermo.

surfaces, and relative humidity. Type 33 (Psychrometrics) calculates the dew point temperature and Type 69 calculates the fictive sky temperature.

3.5 Shading devices

Shading reduction factors are considered to calculate the contribution of solar radiation in terms of heat gains and to control the artificial lighting in the building. Type 2d (on/off differential controller) elaborates the solar radiation for each direction and generates a control function which can have a value of 1 or 0. The value of the control signal is chosen as a function of the difference between upper and lower radiations Rh and Rl, compared with two dead band radiation differences DRh and DRl. The new value of the control function depends on the value of the input control function at the previous time step. The controller is used with the input control signal connected to the output control signal, providing a hysteresis effect. If external temperature is greater than 15°C and if solar radiation differential controller is on, then the shading reduction factor is set equal to 0.9, else it is equal to zero. In the same conditions, a control signal respectively of one and zero is sent to the lighting system.

3.6 Building

Type 56 reads the building data from TRNBuild with the further following inputs:

- external temperature and relative humidity
- sky and ground temperature
- total and beam solar radiation on the 7 considered surfaces (horizontal, Nord 90°, South 90°, Est 90°, Est 45°, West 90°, West 45°)
- angles of incidence on the different surfaces
- infiltration rate
- ventilation rates and temperatures of each room
- heating temperatures for each room

- cooling temperature
- internal heat gains (convective and radiant)
- shading reduction factors
- air coupling rates between adjacent thermal zones

The outputs of Type 56 are the following:

- temperatures and relative humidity of each thermal zone
- sensible heating demand
- cooling demand (sensible and latent)
- mean radiant temperatures of each room
- ventilation and infiltration losses.

3.7 Infiltration

The infiltration rate is assumed because usually the accurate air change rate through infiltration depends on many factors like the outside air velocity, the pressure distribution around the building, the tightness of the windows, etc. For these reasons, it cannot be precisely predicted. The assumption is based on a referred building example in ASHRAE Handbook (ASHRAE, 2011). This example assigned infiltration rate of 0.1 h^{-1} depending the building tightens and insulation. Likewise the example, the studied building has the same degree of tightens and it is well insulated. This value is also in accordance with prEN ISO 13789. Thus, the infiltration rate is assigned as 0.1 h^{-1}. In summer, the infiltration rate should be higher, due to the uncertainty of occupant’s behavior for opening the windows. However, the same infiltration rate is considered all over the year.

3.8 Air conditioning

Heating and cooling devices are considered in this work. The cooling device is both sensible and latent, with a set point temperature of 26°C and dehumidification set point of 50% of relative humidity. Since cooling systems are not common in German dwellings, both cases are considered, with and without cooling devices.

The heating system (sensible) has a set point temperature of 20°C, according to DIN EN 15251:2012. It has also a 30% radiative component and 70% convective component. As shown in Figure 3.6, some rooms, such as kitchen and corridors, have a set point temperature of 16°C from 23.00 to 06.00 and a set point temperature of 20°C from 06.00 to 23.00, according to Annex A of DIN EN 15251:2012 for residential buildings in Category II. The heating temperature is calculated as follows:

$$t_H = \ 2 \cdot t_{setH} \ - t_{mr} \qquad 6)$$

Where

t_H is the heating temperature;

t_{setH} is the set point temperature for heating;

t_{mr} is the mean radiant temperature of the considered room.

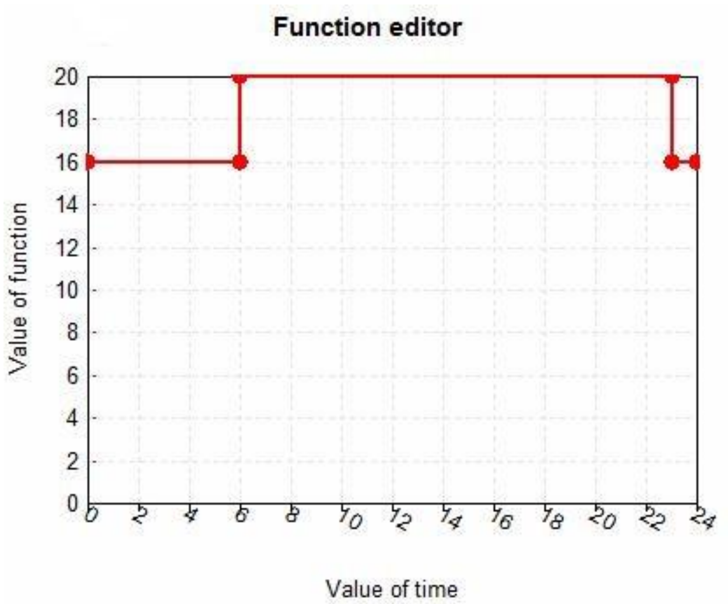

Figure 3.6: Heating set point temperature for "other rooms".

3.9 Occupancy

The building considered in this work is designed for a family of four people. Different occupancy patterns for each room are considered, according to a previous study on the same residential building (Zwiehoff, 2013) and a study from the literature (Martinaitis V., 2015). These patterns distinguish between weekdays and weekends. The time step considered for occupancy is half an hour. That is, every thirteen minutes a number of people between zero and four is assigned to each room. During the day, full occupancy as well as no occupancy in the entire building may occur. Type 14 is used to schedule the presence of people in each room, Type 41 merges weekdays and weekends information into one forcing function. Table 4.10A and Table 4.11A report the schedule used in this work.

3.10 Internal heat gains

Convective and radiative internal heat gains are considered in the simulations. For each room, the total internal heat gains is due to lighting, electrical appliances and people. The total thermal energy per year due to internal heat gains is 4255.4 kWh, or 30.8 kWh/m^2. Figure 3.7 show the internal heat gains all over the year.

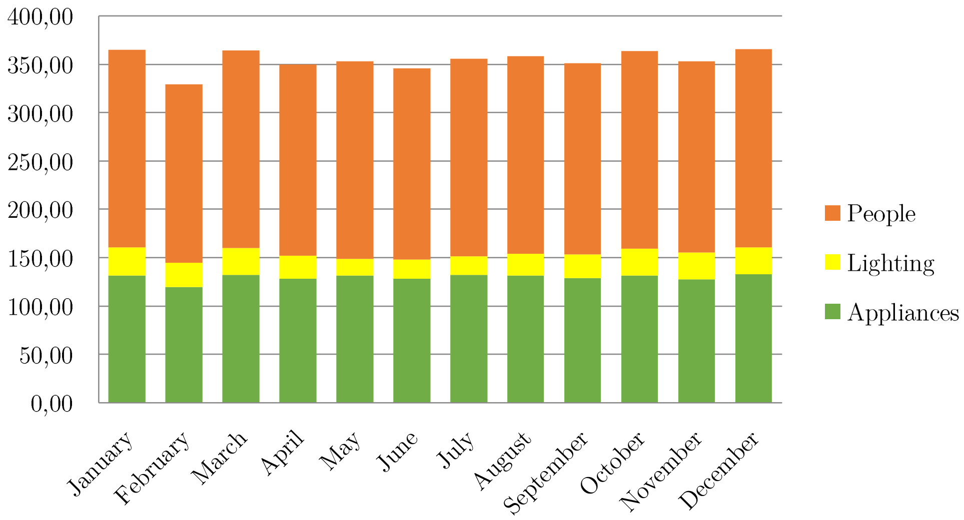

Figure 3.7: Internal heat gains during the year in kWh.

3.10.1 Lighting

The artificial lighting system is based on occupancy and solar radiation (for rooms with windows). Lighting is switched on when the following conditions are met:

- at least one person is in the room (occupancy greater than zero);

- global solar radiation on horizontal surface located outside is less than 150 W/m² when windows have no shading (shading control signal for lighting is zero);
- global solar radiation on horizontal surface located outside is less than 400 W/m² when windows have shading devices (shading control signal for lighting is one);
- time is between 6:30 and 23:30 in weekdays;
- time is between 8:00 and 23:30 in weekends;

For each room, electrical power for lighting is calculated according to a previous study (Zwiehoff, 2013) based on the standard VDI 2078:2012 and the values are summarized in Table 3.1.

Table 3.1: Power for lighting in each room.

Room	**Lighting power [W]**
Guest room	20
Kitchen	60
Livingroom	69
Toilet	28
Bedroom 1	27
Bedroom 2	35
Bedroom 3	27
Bathroom	42

3.10.2 Appliances

Electrical appliances for each room are considered in this work with a schedule of operation, distinguishing weekdays and weekends. For each electrical appliance, convective and radiative components are calculated. For the two corridors, a continuous operation power of 2W is assumed.

Table 3.1 and Table 4.12 summarize respectively the nominal power and the time schedule of each appliance.

Table 3.1: Nominal power of electrical appliances.

Room		Appliance	Power in W
GF	Living room	TV	160
		Music system	30
	Kitchen	Hob extractor	550
		Oven	1200
		Dishwasher	1150
		Fridge	42
	Toilet	Wash.machine	270
1F	Bedroom 2	PC	75

3.10.3 People

Internal heat gain due to presence of people is considered based on occupancy. According to the standard VDI 2078:2012, each person has a thermal power of 100 W, 50% convective and 50% radiative. Occupancy patterns can be found in Table 4.10A and Table 4.11A.

3.11 Air coupling

Each room is considered as a thermal zone with a single air node. In decentralized ventilation simulations, each room is considered independent from the others and airtight, unless the infiltration flow rate equal to 0.1 h^{-1}. In centralized ventilation mode, an air flow is assumed from some rooms to others. In the rooms in which the fresh air is supplied, a pressure gradient is established. In this way, an air flow passes from these rooms to those in which the air is extracted. The air passes through a grill in the bottom side of the doors. An air coupling flow rate is calculated for each room, according to the supply and extract flow rates. The air is supplied in the living room, in the guest room and in the three bedrooms. The air extractors are localized in the kitchen, in the toilet and in the bathroom. The two stories are considered coupled by an air flow from the corridor at the first floor to the corridor at the ground floor. The air coupling flow rates for the simulations with centralized ventilation are assumed as follows:

Guest room – corridor: equal to the supply flow rate in the guest room.

Living room – kitchen: equal to half the supply flow rate in the living room.

Corridor – kitchen: equal to the difference between the extract flow rate in the kitchen and the air coupling flow rate between living room and kitchen.

Living room – corridor: equal to half the supply flow rate in the living room.

Corridor – toilet: equal to the extract flow rate in the toilet.

Bedroom 1 – corridor 2: equal to the supply flow rate in the bedroom 1.

Bedroom 2 – corridor 2: equal to the supply flow rate in the bedroom 2.

Bedroom 3 – corridor 3: equal to the supply flow rate in the bedroom 3.

Corridor 2 – bathroom: equal to the extract flow rate in the bathroom.

Corridor 2 – corridor: equal to the difference between the sum of the supply flow rates in the bedrooms and the extract flow rate in the bathroom.

3.12 Ventilation system

3.12.1 Centralized ventilation

The centralized ventilation system is composed by an air handling unit (supply and extractor fans, heat recovery, auxiliary heater, filters), an inlet duct system from outside with silencer and wall terminal, an outlet duct system toward outside with silencer and wall terminal, a supply distribution duct system to main rooms and an exhaust duct system from wet rooms.

Airflow rates are calculated according to the standard DIN EN 15251:2012. The first step is to calculate the total ventilation airflow rate according to the net floor area:

$$Q_{f_{tot}} = 0.42 \frac{l}{s \cdot m^2} \cdot 138.2\ m^2 = 58 \frac{l}{s} \tag{7}$$

Where:

Q_f is the total ventilation for the building in l/s according to floor area;

0.42 is the ventilation airflow rate in l/s · m² recommended for category II;

138.2 is the total floor area of the dwelling in m².

Assuming a mean density of the air of 1.204 kg/m³, the airflow rate becomes 251.6 kg/h.

After that, the supply ventilation airflow rates are calculated for main rooms (living room and bedrooms) according to the design number of people for each room; namely, number of beds are considered for each room: one bed in bedroom 1, two beds in bedroom 2 and one bed in bedroom 3. For living room, a design number of

Table 3.1: Supply airflow rates for main rooms.

Room	**Design people**	**Supply airflow in l/s**	**Supply airflow in kg/h**
Living room	4	28	121.36
Guestroom	1	7	30.3
Bedroom 1	1	7	30.3
Bedroom 2	2	14	60.7
Bedroom 3	1	7	30.3
Total		63	273.1

four people is considered. Table 3.1 reports the supply airflow rates in the main rooms.

The supply ventilation airflow rate Q_s = 273.1 kg/h is chosen as nominal airflow rate in occupied periods, in order to provide the nominal airflow rates in each main room. During unoccupied period, i.e., when nobody is in the building, a minimum airflow rate Q_u of 0.1 l/s · m² is considered according to total floor area:

$$Q_u = 0.1\frac{l}{s\cdot m^2}\cdot 138.2\ m^2 = 13.8\frac{l}{s} = 59.9\frac{kg}{h} \qquad 8)$$

According to the standard EN 15251, the design airflow rates to be exhausted from the wet rooms can be found in Table 3.2.

Table 3.2: Extract airflow rates from wet rooms.

Room	Design airflow in l/s	Design airflow in kg/h	Room volume in m³	Exhaust airflow in m³/h	Exhaust airflow in kg/h
Kitchen	20	86.7	29.5	94.4	113.6
Toilet	10	43.3	16.5	52.8	63.5
Bathroom	15	65	24.9	79.7	95.9
Total		195	70.9	226.8	273.1

Since the total extract airflow rate is less than the supply one, an adjustment has to be made. Hence, the total exhaust airflow rate is assumed equal to the supply airflow rate and this value is weighted on rooms' volume, according to the following equation:

$$Q_e = Q_s \cdot \frac{V_{room}}{V_{tot}} \qquad 9)$$

Where:

Q_e is the exhaust airflow rate in m^3/h;

Q_s is the supply airflow rate in m^3/h;

V_{room} is the volume of the considered room;

V_{tot} is the total volume of the wet rooms (kitchen, bathroom and toilet).

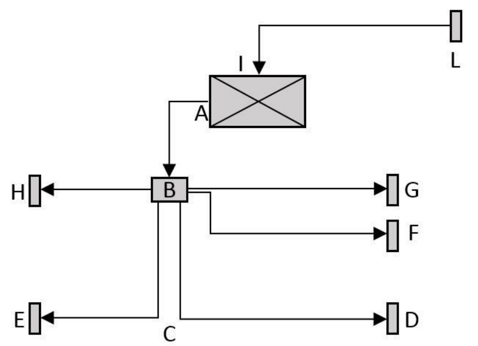

Figure 3.8: Centralized supply duct system.

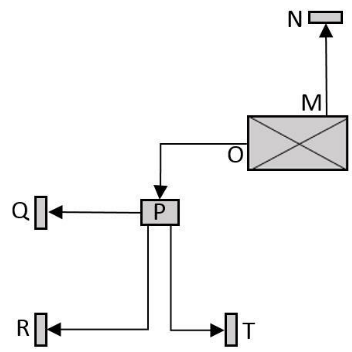

Figure 3.9: Centralized extract duct system.

Considering the disposition of rooms, a duct system is modeled. Figure 3.8 and Figure 3.9 show respectively the supply and the exhaust system graphs. Letters L and N represent respectively the inlet and outlet grills on the roof of the building. Letters B and P represent the Plenum for air distribution. Letters D, E, F, G, H represent the grills to supply the air respectively in living room, guestroom, bedroom

Table 3.3: Estimated lengths of supply and extract duct system.

Duct portion	Length in m	Duct portion	Length in m
EC	4.6	IL	1.9
CB	2.9	RS	1.5
CD	0.15	SP	2.9
HB	2.4	ST	1.4
BG	0.7	QP	0.8
BF	1.8	PO	3.0
AB	1.1	MN	2.0

2, bedroom 3 and bedroom 1. Letters Q, R, T represent the grills to exhaust the air respectively in bathroom, toilet and kitchen.

All the estimated lengths of the duct portions are reported in Table 3.3.

The lengths of the duct portions are used to calculate the pressure losses of the duct system. The internal diameter of a single duct is fixed to 0.063 m, according to the technical catalogue of the manufacturer AERMEC S.p.A. (AERMEC, 2015). Maximum airflow rate of 35 m^3/h is assumed in a single duct, in order to maintain the airflow velocity below 2.5 m/s. Therefore, number of parallel ducts for each portion of the distribution system has to be calculated by dividing the airflow needed in the portion of duct by the maximum airflow per single duct, rounded up. Vertical and horizontal bends are also considered to calculate the pressure drop. The pressure drop Δp for each duct is calculated with the following formula:

$$\Delta p = Q^2 \cdot \frac{Z}{1000} \qquad 10)$$

Where:

Δp = pressure drop in Pascal;

Q = airflow rate passing throw the considered duct portion in m^3/h;

Z = parameter characteristic of the considered duct portion for pressure loss calculation, assuming an air density of 1.2 kg/m^3. Z-values are according to AERMEC technical manual.

Table 4.13 and Table 4.15 report the calculation of pressure drop in each duct portion of the distribution system.

Table 4.14 reports the calculation of pressure drop in the inlet and outlet ducts (from and to outside).

According to the results of calculations, the pressure heads of the duct systems, supply and exhaust, are respectively of 48.7 Pa and 30.8 Pa. The pressure drops due to filters and heat recovery are both assumed equal to 110 Pa.

Therefore, the total Δp of supply and extract fans are respectively of 258.7 Pa and 240.8 Pa.

The model of the mechanical ventilation system in the Simulation Studio is realized with types of the TESS Library (Thermal Energy System Specialists of Madison, Wisconsin) and it has the following components:

- Supply fan
- Extract fan
- Air-to-air heat recovery
- Auxiliary heater
- Duct system

Supply and extract fans have respectively a nominal Δp of 278.7 Pa and 260.8 Pa.

According to the "Recommendations for calculation of energy consumption of air handling unit" (Eurovent, 2005), the electrical power consumption of fans is calculated with the following formula:

$$P_{el} = \frac{m \cdot \Delta p_{fan}}{\eta_{vent} \cdot 1000} \qquad 11)$$

Where:

P_{el} = electrical power in kW;

$\dot{m}$ = airflow rate in m^3/s;

Δp_{fan} = pressure head of the fan for the corresponding flow rate in Pa;

η_{vent} = ventilation efficiency (motor fan's efficiency and ventilation losses).

The actual pressure drop is calculated with the following formula:

$$\Delta p_{fan} = \eta_{vent} \cdot \Delta p_{fan,nom} \cdot \left(\frac{m}{m_{max}}\right)^2 \qquad 12)$$

Where:

$\Delta p_{fan,nom}$ = nominal pressure drop in Pa;

m = actual flow rate in kg/h;

m_{max} = nominal flow rate in kg/h.

The heat recovery (Type 760a) models the heat exchange between the fresh air from outside and the exhaust air from rooms. It is supposed to operate in counter flow with sensible effectiveness equal to 0.8, according to the actual market (CasaClima, 2015). The Type requires information about temperatures, flow rates and relative humidity of the two streams that enter the component. Rated power and control mode are both set to zero. This device is used to heat up the cold air from outside in winter conditions and to cool down the warm air from outside in summer conditions, using the thermal capacity of the exhaust air flow. A control signal is used to switch the heat recovery in on/off mode. Mode On is set when the following conditions are achieved:

$$\begin{gathered}\{[\ T_{ext} < 20\ \cup\ T_{ext} > T_{room}\ ] \cap (T_{room} < 26)\} \\ \cup \{[\ T_{ext} < 15\ \cup\ T_{ext} > T_{room}\ ] \cap\ T_{room} > 26\ \}\end{gathered} \qquad 13)$$

Where:

T_{ext} = external temperature in °C;

T_{room} = temperature of the room in °C.

The conditions of the exhaust air are calculated by mixing the extract flow rates from wet rooms. Type 33 (Psychrometrics) is used to calculate the humidity ratio of each air flow, knowing the dry bulb temperature and percent relative humidity of the air flow, and assuming an atmospheric pressure of 1 atm. Type 11 is used to mix the air flows providing temperature, flow rate and humidity ratio of the mixing flow.

The auxiliary heater (Type 6) goes in operation when the temperature of the fresh air reach 0°C. It uses the Joule's effect to heat up the air, in order to prevent frost. If the wet exhaust air is cooled by the fresh air below the saturation temperature, condensation occurs. In order to prevent condensate from frosting, the temperature of the heat recovery device should never goes below zero. A maximum heating rate of 1000 W is assumed. Set point temperature is 0°C. The efficiency of the heater is unitary, because all the electrical power is supposed to turn into heat.

3.12.2 Decentralized system

The decentralized ventilation system is composed by: exhaust fan, supply fan, heat recovery, auxiliary heater and filters. One device is installed in each room in which air change is required, i.e., every room but corridors. In the building considered in this work the total number of devices is eight. Each single unit functions fully by itself and can be individually controlled. They are mounted onto the façade and do not include long ventilation ducts. This saves space, energy for running the system as well as installation costs. The airflow rates are determined according to EN 15251:2007. During occupied periods, the flow rate is calculated as follows:

$$Q_{dec,occ} = 7 \cdot N_P \qquad 14)$$

Where:

$Q_{dec,occ}$ = airflow rate during occupied periods in l/s;

7 = airflow rate per person recommended by the norm in l/(s · pers);

N_P = number of people in the room.

Assuming a density of the air of $\rho_{air} = 1.204$ kg/m^3, the flow rate per person is equal to 30.34 kg/(h · pers).

During unoccupied periods, the minimum flow rate is determined according to floor area with the following formula:

$$Q_{dec,un} = 0.1 \cdot A_f \quad 15)$$

Where:

$Q_{dec,un}$ = airflow rate during unoccupied periods in l/s;

0.1 = minimum airflow rate in l/(s · m^2);

A_f = floor area of the considered room in m^2.

Assuming a density of the air of $\rho_{air} = 1.204$ kg/m^3, the minimum flow rate is equal to 0.43 kg/(h · m^2).

The two fans are modeled in an equation window with the following formulae:

$$W = \frac{\Delta p \cdot m}{\rho_a \cdot \eta_{vent}} \quad 16)$$

Where:

W = electrical power consumption of the considered fan in Watt;

Δp = pressure drop of the considered fan in Pa;

m = airflow rate in kg/s;

ρ_a = density of the air in kg/m^3;

η_{vent} = ventilation efficiency of the fans, assumed to be equal to 0.7.

$$\Delta p = 0.3 \cdot \Delta p_{max} + 0.7 \cdot \Delta p_{max} \left(\frac{m}{m_{max}}\right)^2 \quad 17)$$

Where:

Δp_{max} = maximum pressure drop of the considered fan, due to heat recovery, filter, grills, and assumed to be equal to 260 Pa;

m_{max} = maximum flow rate assumed to be equal to 120.4 kg/h.

As well as the centralized ventilation system, the heat recovery is modeled by Type 760a with control mode set to zero. This means that outlet temperatures are not controlled. Sensible effectiveness is assumed to be equal to 0.8 as a mean value of the actual market (CasaClima, 2015).

The auxiliary heater is modeled by Type 6 and it is an electrical device with a nominal power of 1000 W. The set point temperature is 0°C, so the control signal is one when the external temperature goes below zero. The efficiency of the heater is

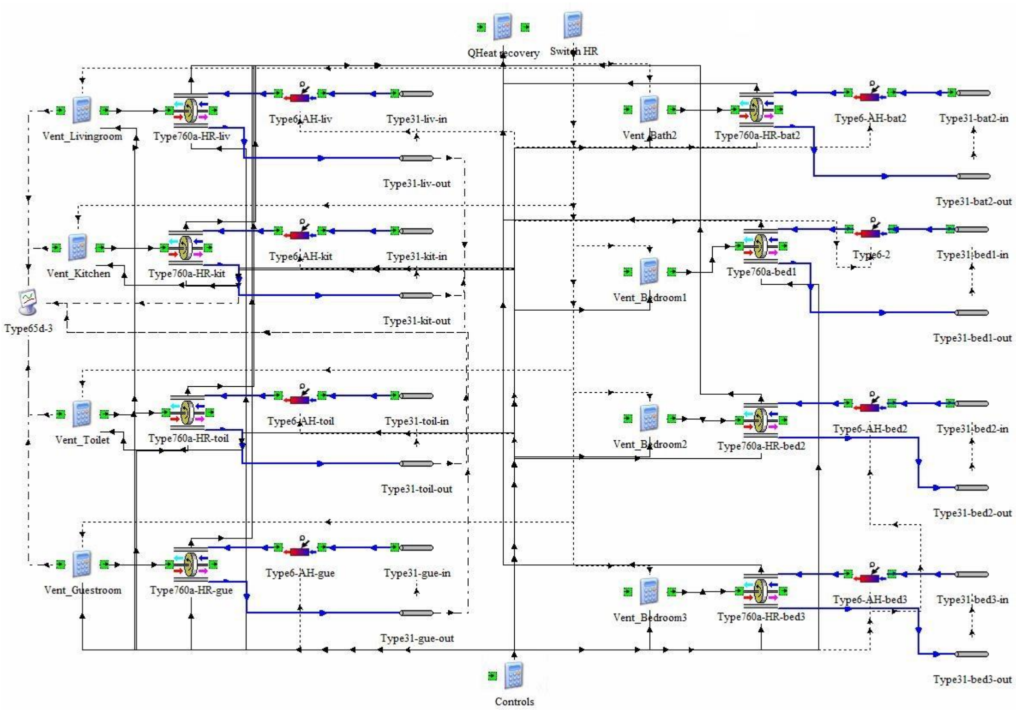

Figure 3.10: Model of the decentralized ventilation system.

unitary, because all the electrical power is supposed to turn into heat. Figure 3.10 shows the decentralized ventilation system as it appears in the program.

3.13 Control strategy

The centralized ventilation system considered in this work has two airflow rate levels, minimum and nominal, respectively of 49.8 m^3/h and 226.8 m^3/h. Minimum airflow rate is used when nobody is in the dwelling with the purpose of ensuring adequate air change to the building, i.e., to avoid condensation-related problems, such as mold growth. The nominal airflow rate is calculated in relation to the occupancy, viz. the presence of people in the building. Because of the duct system, during occupied periods the ventilation device has to provide the nominal airflow rate in each room, since a regulation on the distribution terminals would unbalance the flow rate in the rooms. Therefore, two-speed level fans are considered for the centralized device.

The decentralized devices have different levels of airflow rate, they are each one independent from the others and room-based. When nobody is in the room, the minimum airflow rate ($0.1\ l/(s \cdot m^2) \cdot A_{f,room}$) according to floor area $A_{f,room}$ is supplied to the considered room. Otherwise, airflow rate according to number of people N_p the room ($7\ l/(s \cdot pers) \cdot N_p$) is provided.

The presence of people can be detected by CO_2 sensors installed in each room where a ventilation device is set up. CO_2 concentration is proportional to the number of people in the room. According to the standard EN 13779:2007, the default CO_2

Table 3.4: CO_2 concentrations limits.

Category	CO_2-level above level of outdoor air in ppm	
	Typical range	**Default range**
I	<400	350
II	400 – 600	500
III	600 – 1000	800
IV	>1000	1200

emission rate is equal to 20 l/(h · pers). Furthermore, the norm gives reference values of CO_2 concentrations above outdoor concentration for different categories of indoor environment, as can be found in Table 3.4. The default outside CO_2 concentration is assumed equal to 400 ppm.

Chapter 4

Simulations and results

Several case studies are selected for the simulations, based on the city, the heat recovery, the air-conditionings, and the type of building. Concerning the city, Stuttgart and Palermo are chosen as representative of two different climate conditions, viz. oceanic and Mediterranean climate. Simulations with and without heat recovery are conducted in order to estimate the impact of heat recovery on energy performance. Since summer cooling is not common in German dwellings, due to cooler weather conditions than Italy, both cases with and without cooling system are considered. Lastly, two different building envelopes are considered. The first one is according to EnEV values of thermal transmittance for opaque and glazed surfaces, while the second one is according to Appendix B, Annex 1 of the Italian

Table 4.1: Transmittances of the envelopes considered.

Envelope component	**U (W/m^2K)**	
	German envelope	**Italian envelope**
External walls	0.20	0.45
Roof	0.22	0.34
Floor	0.20	0.48
Glazing components	1.27	3.20

D.M. 26 giugno 2015 "Requisiti minimi", for the climate zone B, as reported in Table 4.1. However, beside the different thermal transmittance values, the same building is considered, i.e., layout of the rooms, heat gains, ventilation and thermal plants are the same in all simulations.

4.1 Case 1

In the first simulation, the following conditions are considered:

- Stuttgart
- Heat recovery
- Cooling system
- German building

Figure 4.1 and Figure 4.2 show the heating and cooling demand (sensible and latent) all over the year, respectively for centralized and decentralized ventilation systems.

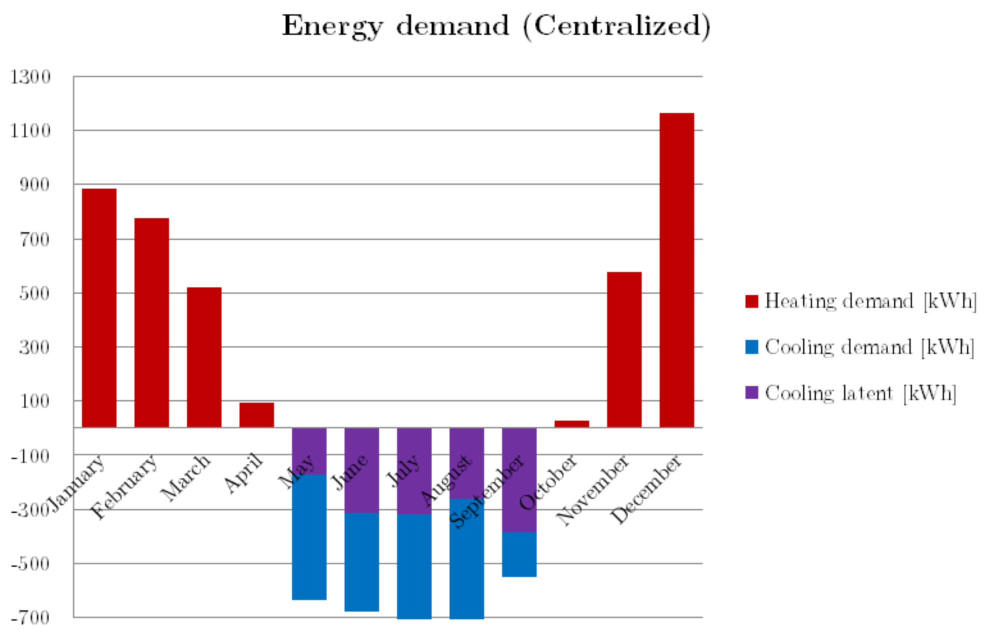

Figure 4.1: Energy demand with centralized system, case 1.

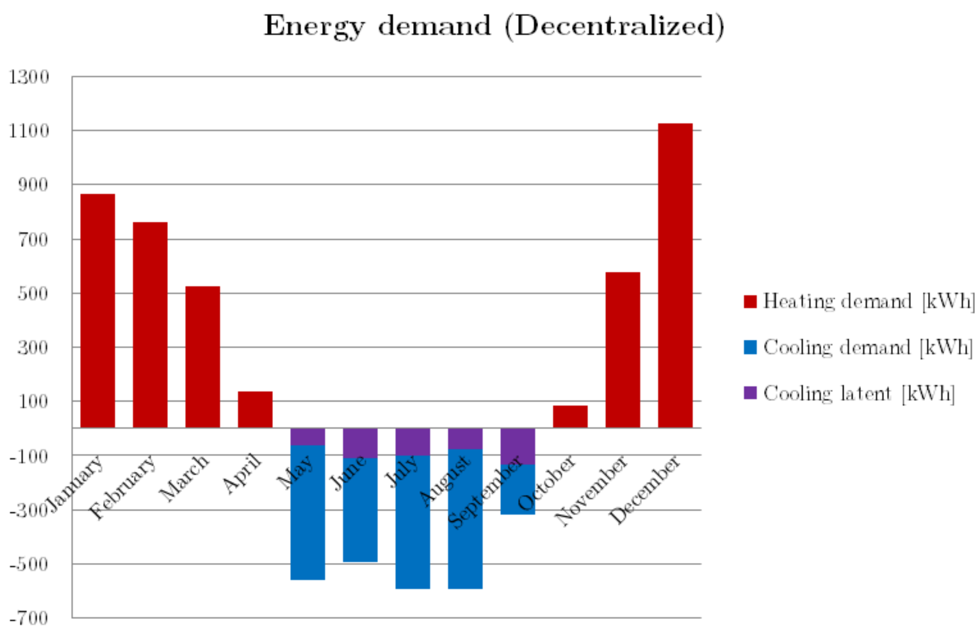

Figure 4.2: Energy demand with decentralized system, case 1.

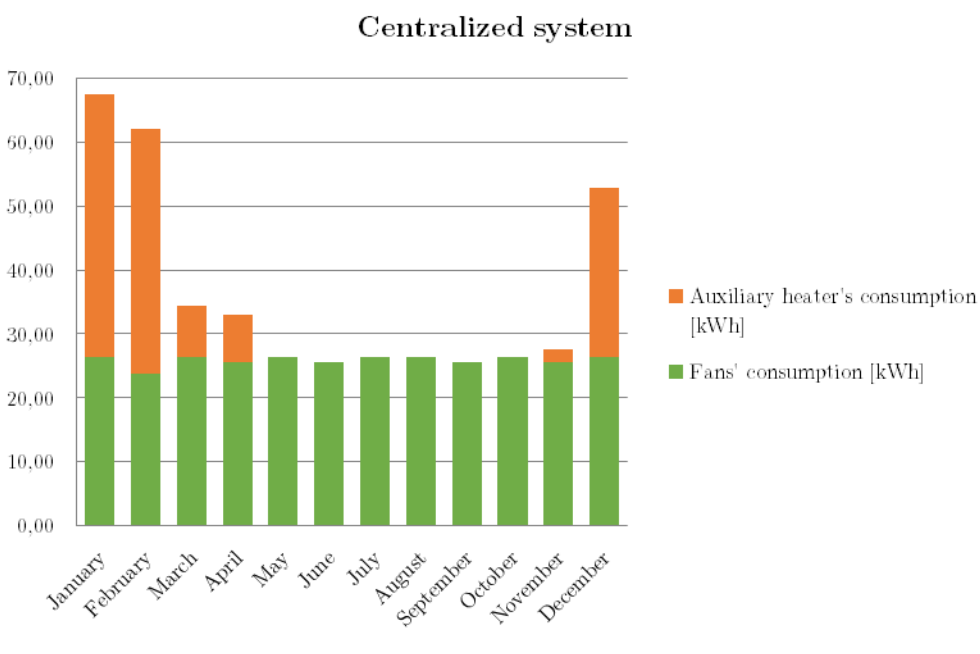

Figure 4.3: Electrical consumptions with centralized system, case 1.

The heating demand is mostly the same for the two configurations. Sensible cooling demand is slightly less with the centralized system, while latent cooling demand is less with the decentralized system. Electrical consumptions of fans and auxiliary heater are reported in Figure 4.3 and Figure 4.4. The decentralized system absorbs 56% less power than the centralized, because the latter is in operation for a higher number of hours. For the same reason, the heat recovered with centralized system is 46% more than that with decentralized system, as can be seen in Figure 4.5. Table 4.16A summarizes the results of this first simulation.

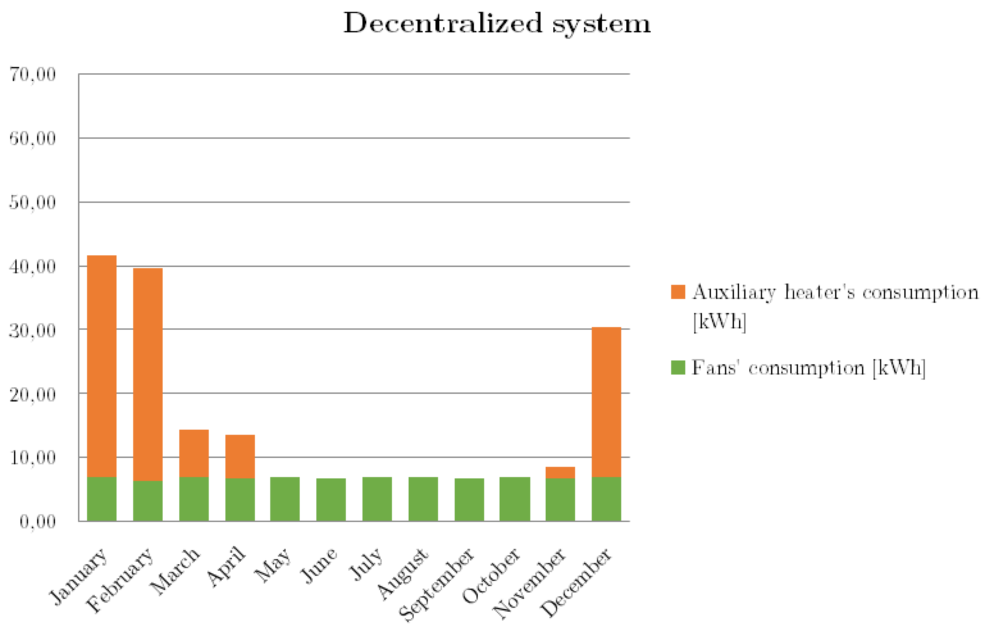

Figure 4.2: Electrical consumptions with decentralized system, case 1.

In order to achieve a better comparison of the two systems, the primary energy demands (EP) in kWh are calculated with the following formula:

$$EP = W_e \cdot f_{p,e} + \left(\frac{Q_H}{\eta_g}\right) \cdot f_{p,t} + \left(\frac{Q_C}{SEER}\right) \cdot f_{p,e} \tag{18)}$$

Where:

W_e = electrical consumptions of the ventilation system in kWh;

$f_{p,e}$ = primary energy factor for electricity, assumed equal to 2.2;

Q_H = energy demand for heating in kWh;

η_g = generation efficiency of the heating system, assumed equal to 0.9;

$f_{p,t}$ = primary energy factor for thermal energy, assumed equal to 1.05;

Q_C = energy demand for cooling in kWh;

SEER = Seasonal Energy Efficiency Ratio, assumed equal to 3.

For the centralized system, EP = 7171.1 kWh; for the decentralized system, EP = 6693.4 kWh.

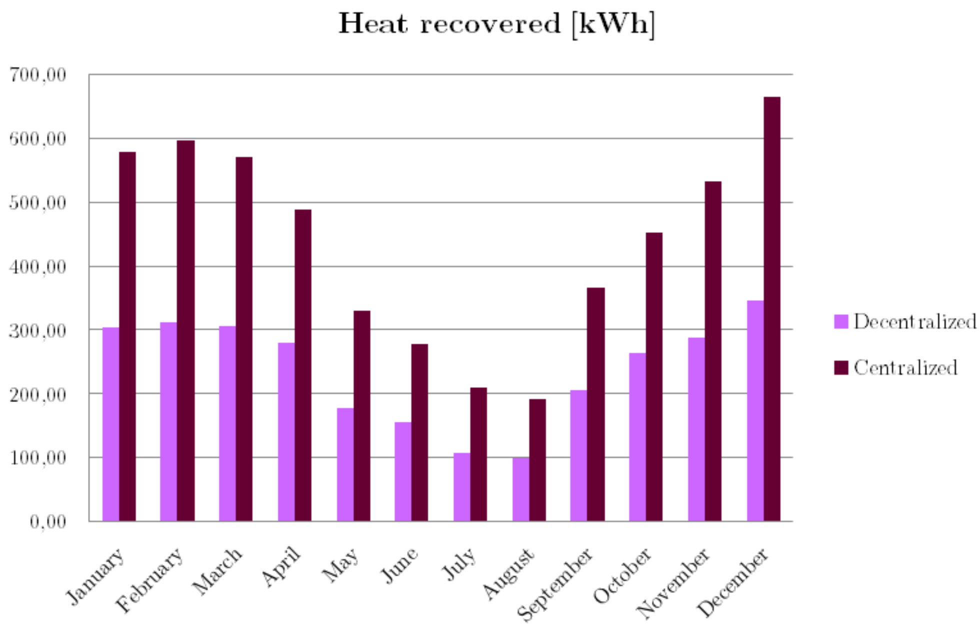

Figure 4.3: Comparison of heat recovery units, case 1.

4.2 Case 2

In the second simulation, the following conditions are considered:

- Stuttgart
- No heat recovery
- Cooling system
- German building

In this configuration, the heating and cooling (sensible) demands with decentralized ventilation system are respectively 24% less and 23% more than with centralized system; latent demand for cooling is 75% less. Since the centralized system could recover more heat, because it is more in operation, the decentralized system becomes more competitive without heat recovery. The electrical consumptions are the same of the previous case. Table 4.17A summarizes the results. Primary energy demand for centralized and decentralized system, calculated using Equation 18, are respectively $EP_c = 10444.8$ kWh and $EP_d = 8121.5$ kWh.

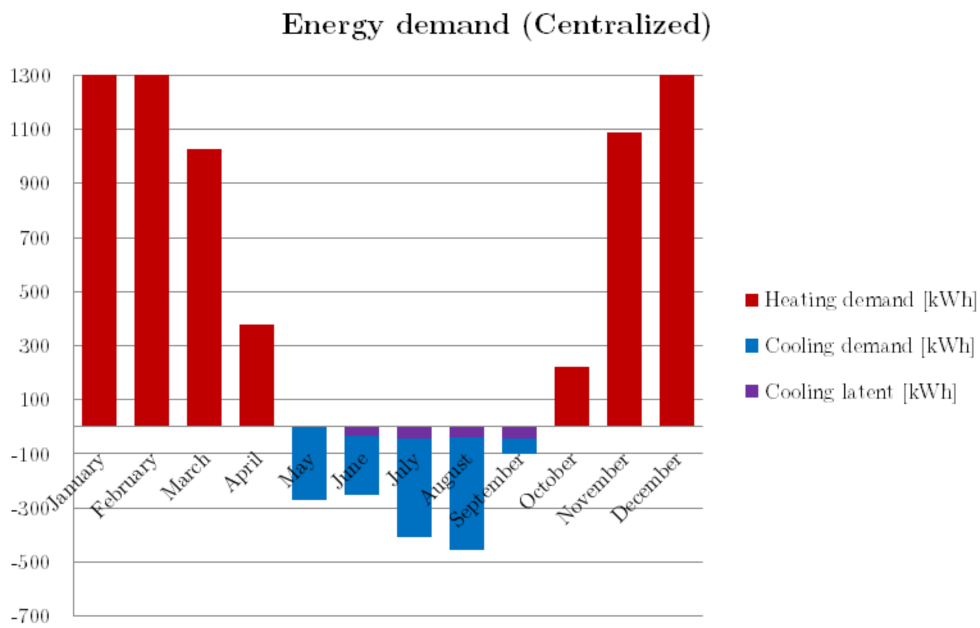

Figure 4.4: Energy demand with centralized system, case 2.

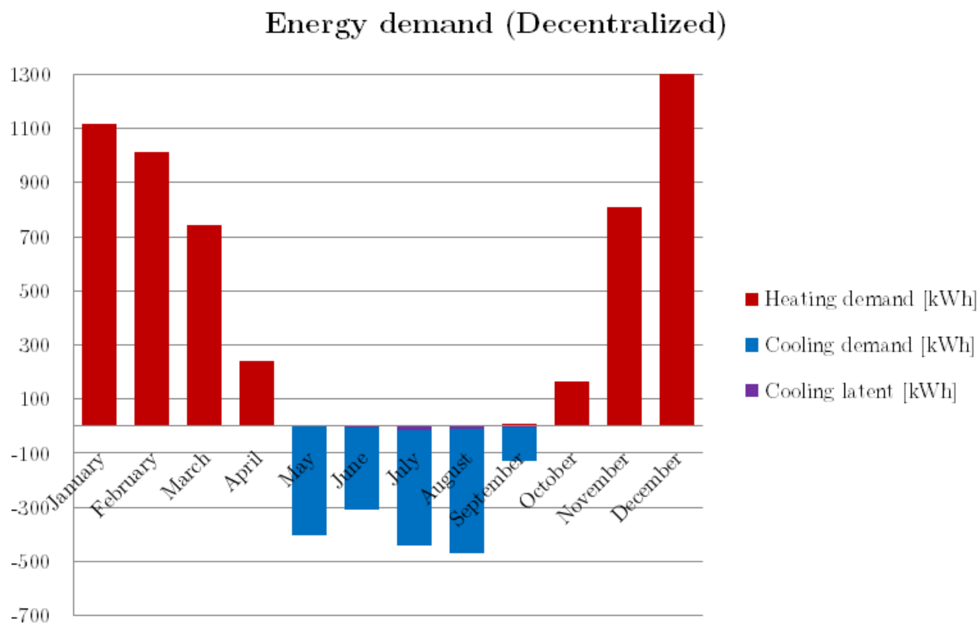

Figure 4.5: Energy demand with decentralized system, case 2.

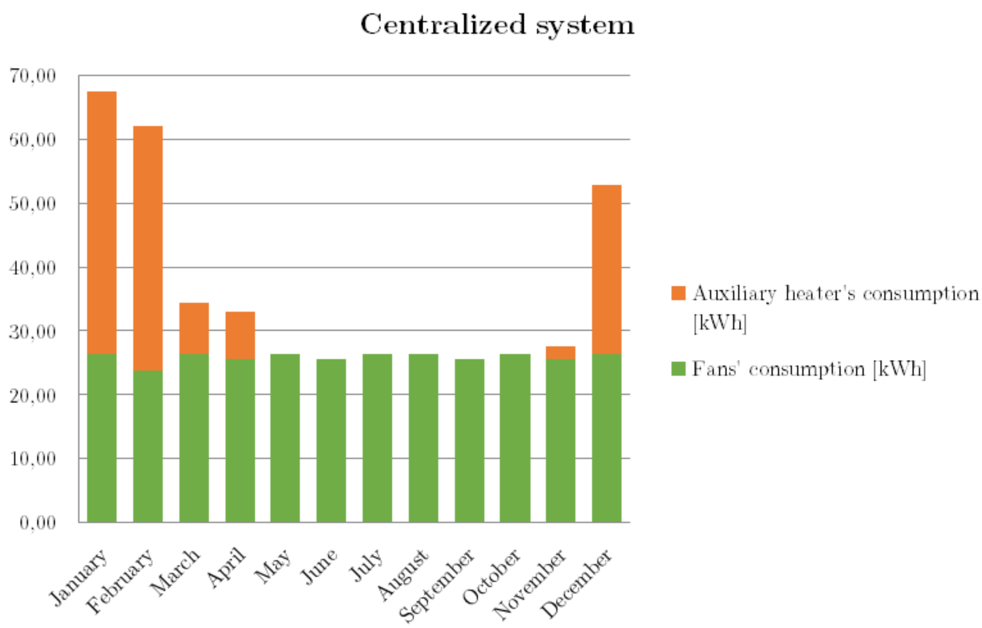

Figure 4.6: Electrical consumptions with centralized system, case 2.

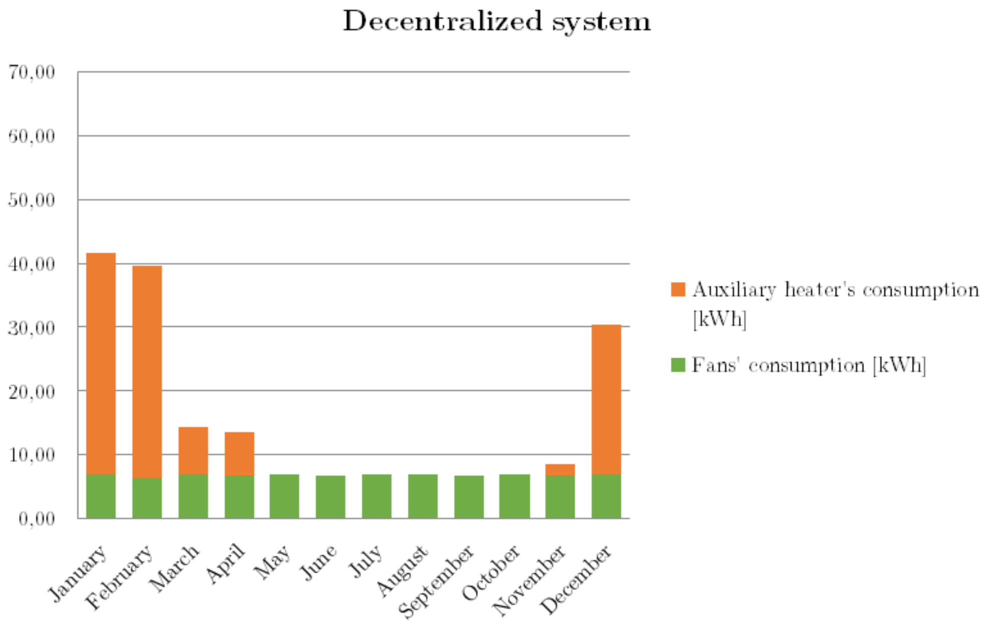

Figure 4.6: Electrical consumption with decentralized system, case 2.

4.3 Case 3

In the third simulation, the following conditions are considered:

- Stuttgart
- Heat recovery
- No cooling system
- German building

In this configuration, the heating demands with decentralized and centralized ventilation systems are similar, respectively equal to 4191 kWh and 4138 kWh, like the first case. The reason for this has to be search in the airflow rates: in a specific period, the centralized system elaborates more air and so recovers more heat; conversely, the decentralized system elaborates less air saving energy from ventilation losses. As can be seen in Figure 4.8, the electrical consumptions are very similar to the previous case. Primary energy demand for centralized and decentralized system are respectively $EP_c = 5782.5$ kWh and $EP_d = 5308.4$ kWh.

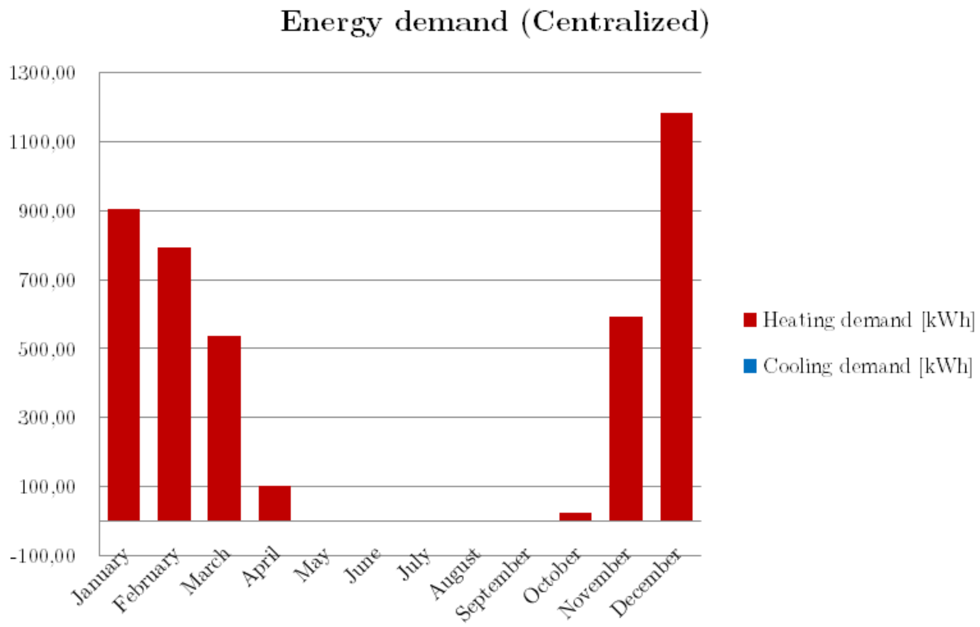

Figure 4.7: Energy demand with centralized system, case 3.

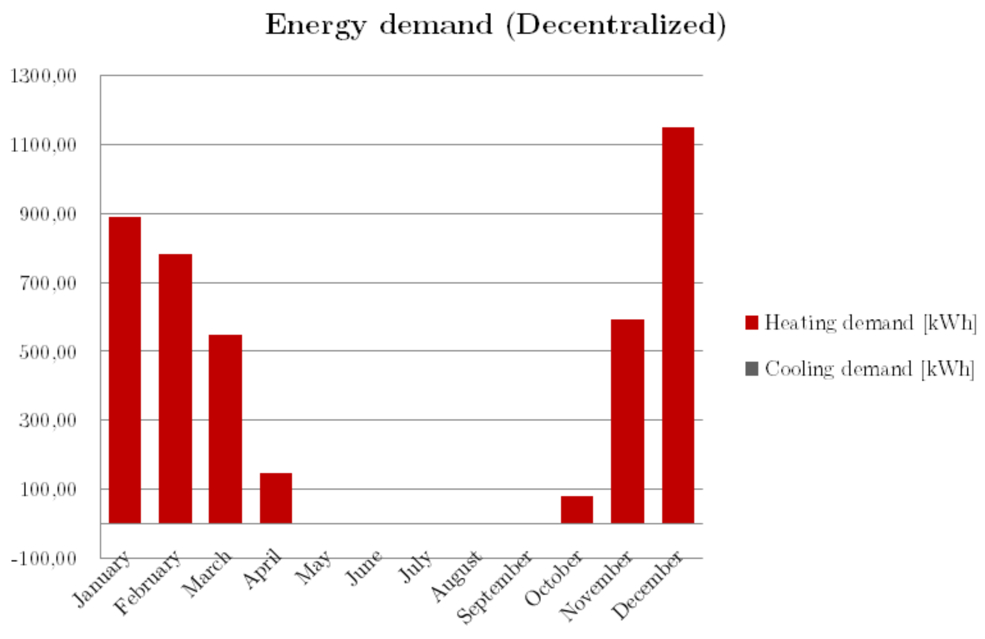

Figure 4.9: Energy demand with decentralized system, case 3.

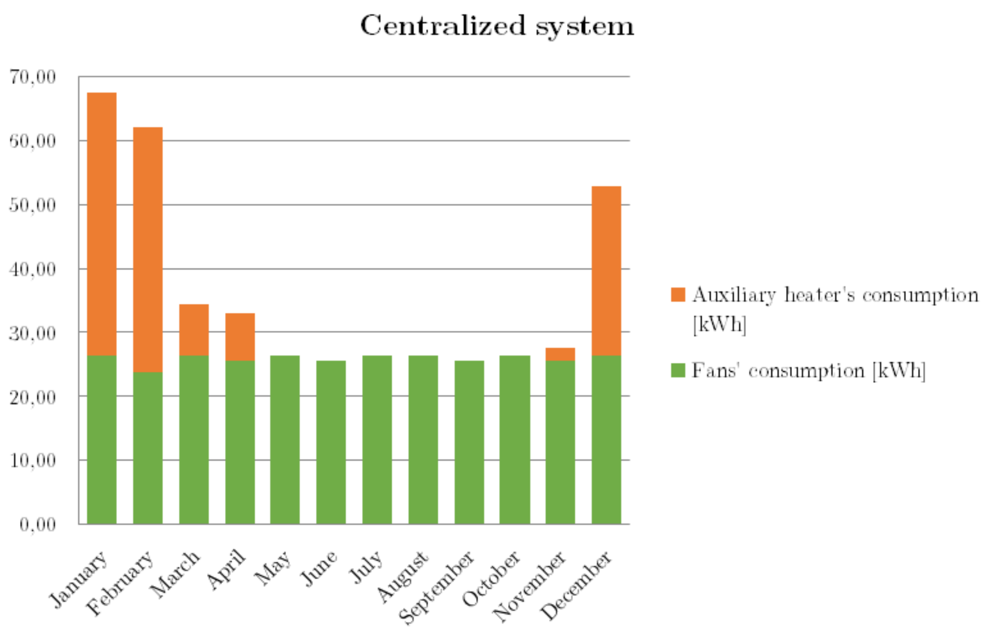

Figure 4.8: Electrical consumptions with centralized system, case 3.

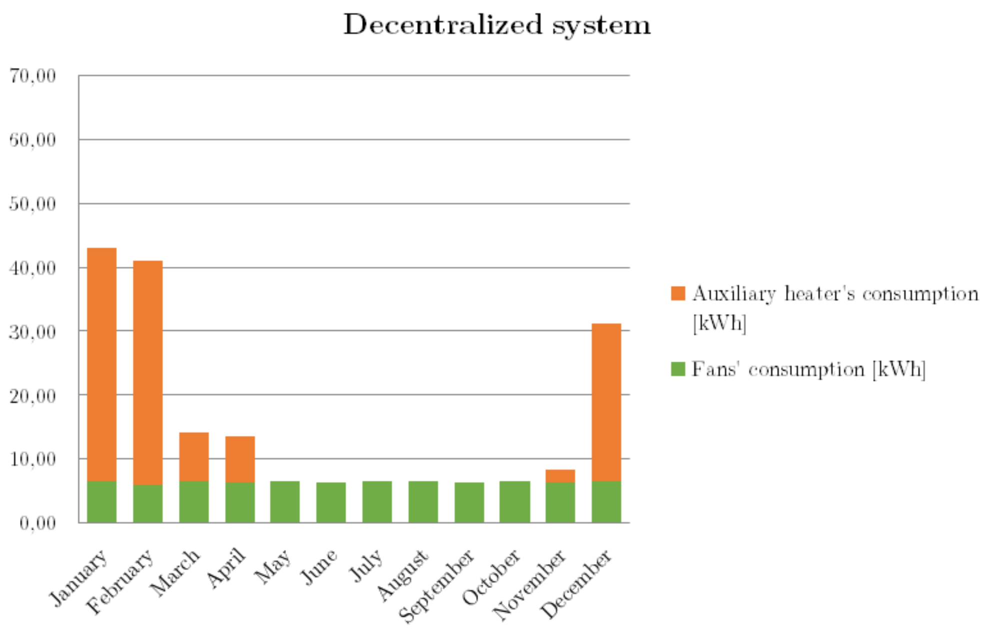

Figure 4.10: Electrical consumptions with decentralized system, case 3.

4.4 Case 4

In the fourth simulation, the following conditions are considered:

- Stuttgart
- No heat recovery
- No cooling system
- German building

In this configuration, with the decentralized ventilation system the heating demand is 24% less than with centralized system; the electrical consumption is 56% less. Primary energy demand for centralized and decentralized system are respectively $EP_c = 9442.2$ kWh and $EP_d = 6821.6$ kWh.

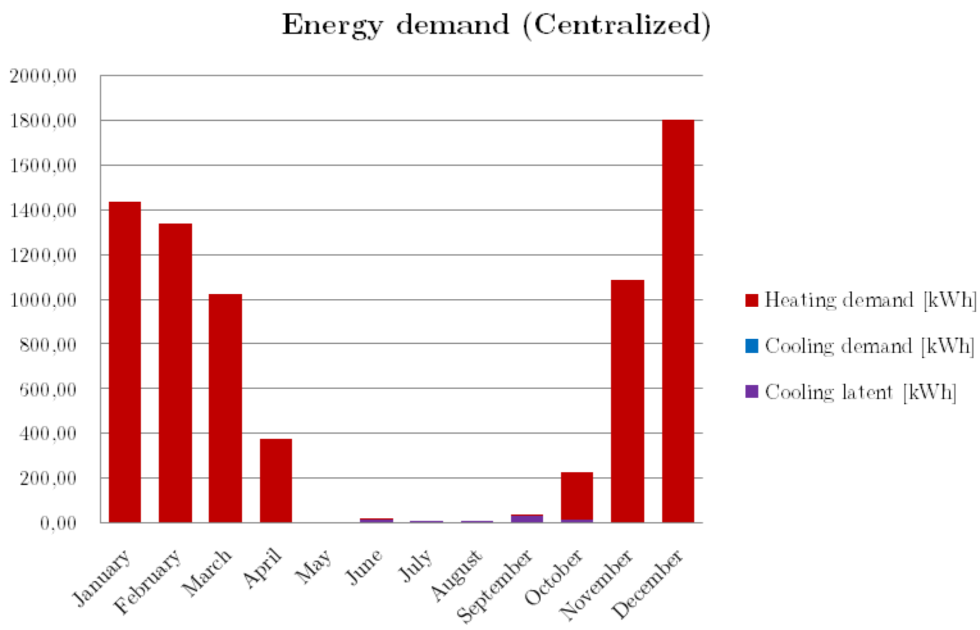

Figure 4.11: Energy demand with centralized system, case 4.

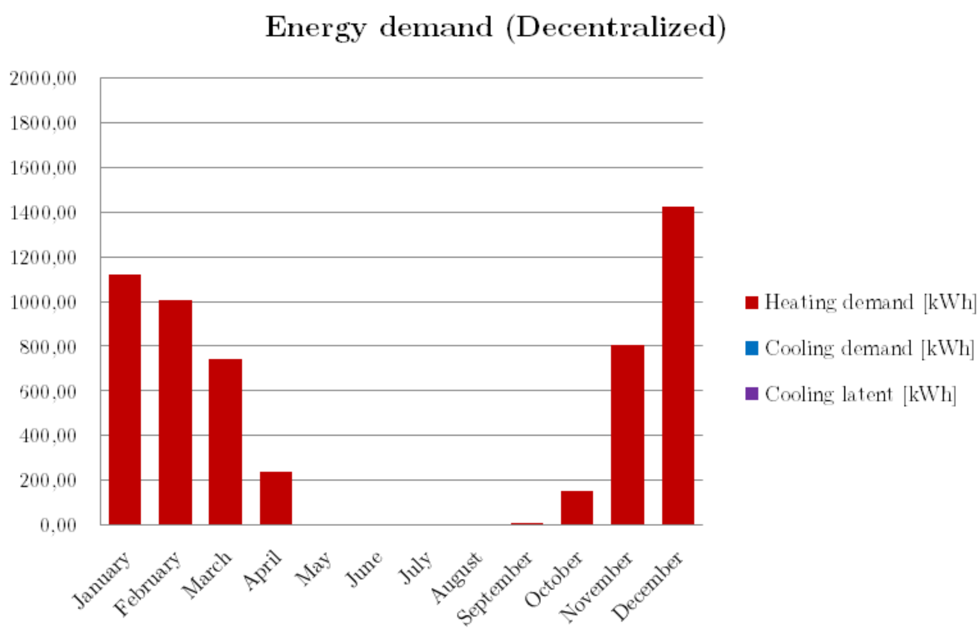

Figure 4.12: Energy demand with decentralized system, case 4.

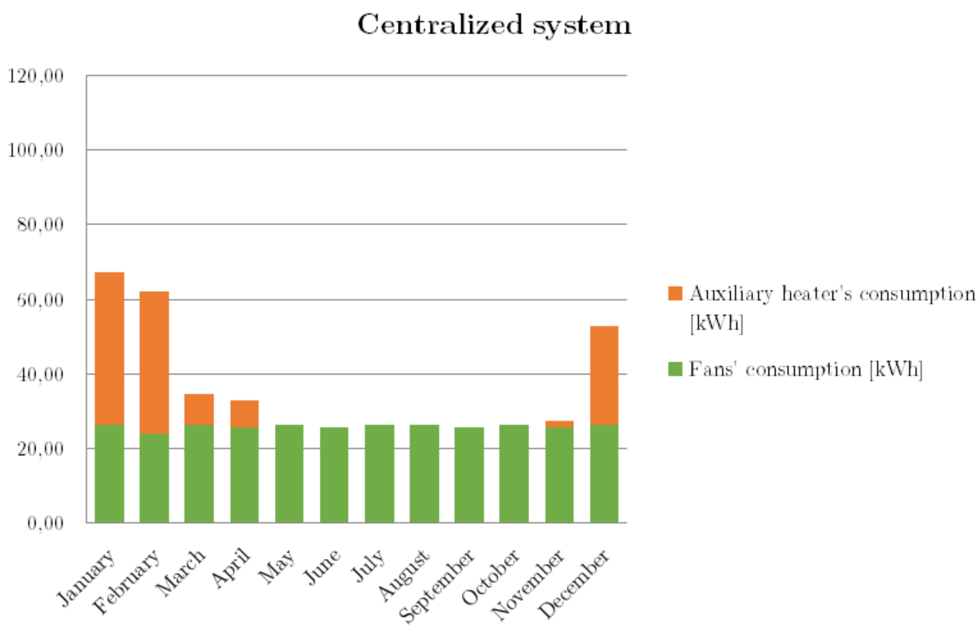

Figure 4.14: Electrical consumption with centralized system, case 4.

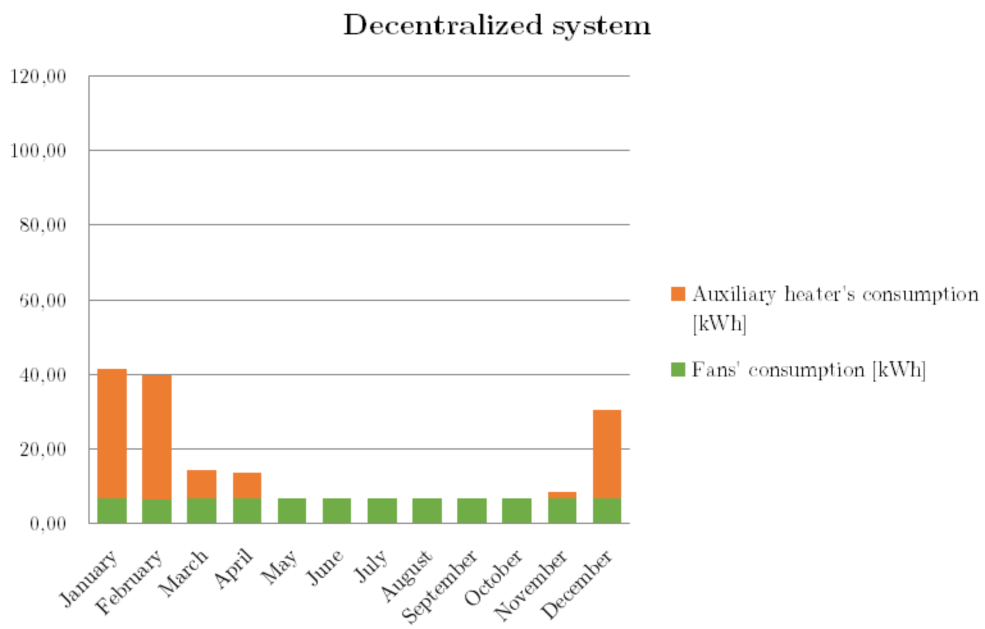

Figure 4.13: Electrical consumption with decentralized system, case 4.

4.5 Case 5

In the fifth simulation, the following conditions are considered:

- Palermo

- Heat recovery
- Cooling system
- Italian building

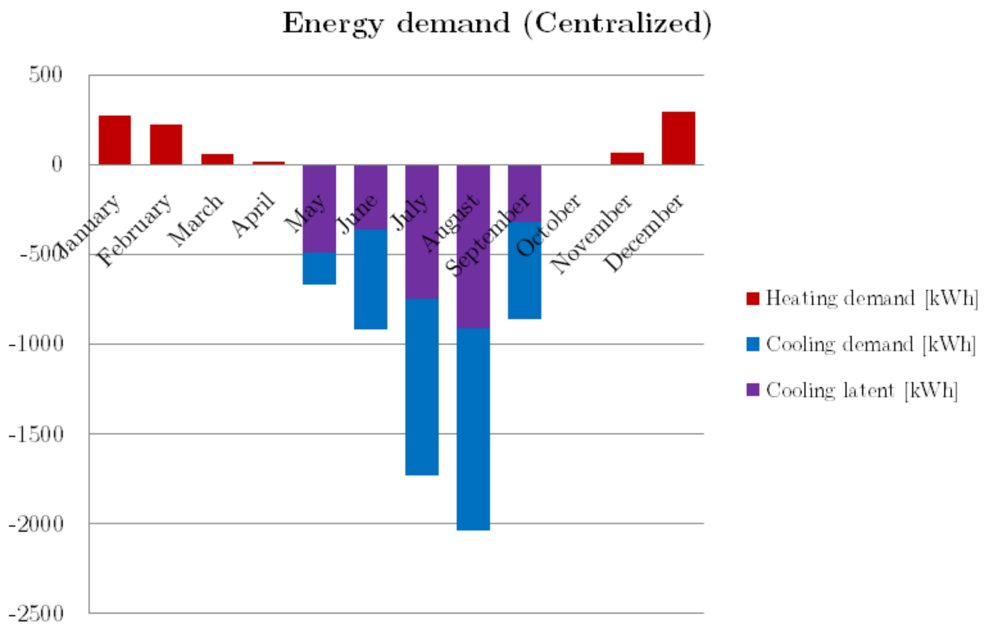

Figure 4.15: Energy demand with centralized system, case 5.

In this configuration, with the decentralized ventilation system the heating demand is 28% more than with centralized system; the sensible and latent cooling demand are respectively 9% less and 67% less. The electrical consumptions with

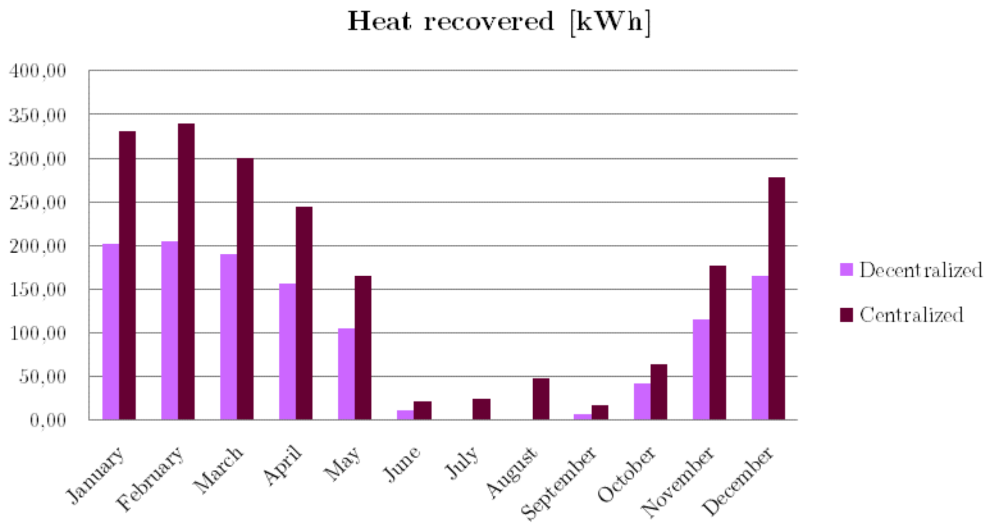

Figure 4.16: Heat recovered with both systems, case 5.

the decentralized system are 74% less. Conversely, the centralized system recovers 42% more heat, as can be seen in Figure 4.16. Table 4.19A summarizes the results. Primary energy demand for centralized and decentralized system are respectively $EP_c = 4244.3$ kWh and $EP_d = 3932.6$ kWh.

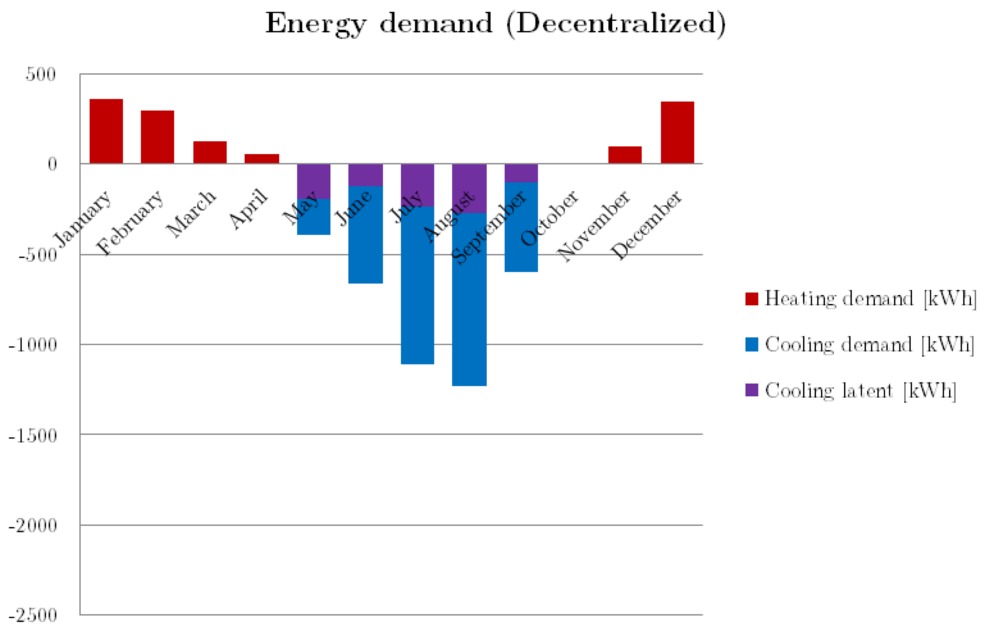

Figure 4.18: Energy demand with decentralized system, case 5.

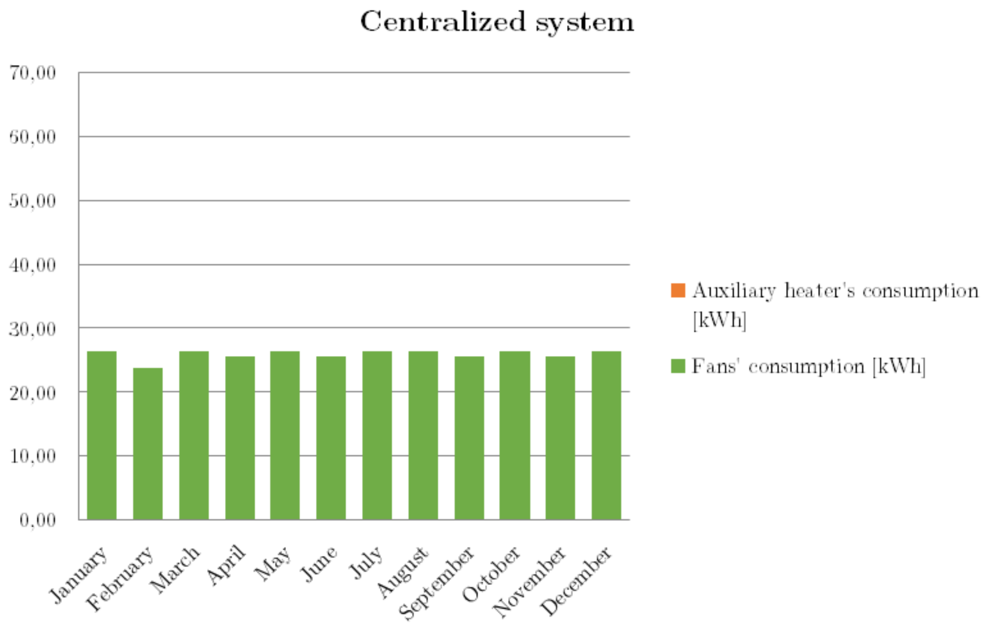

Figure 4.17: Electrical consumption with centralized system, case 5.

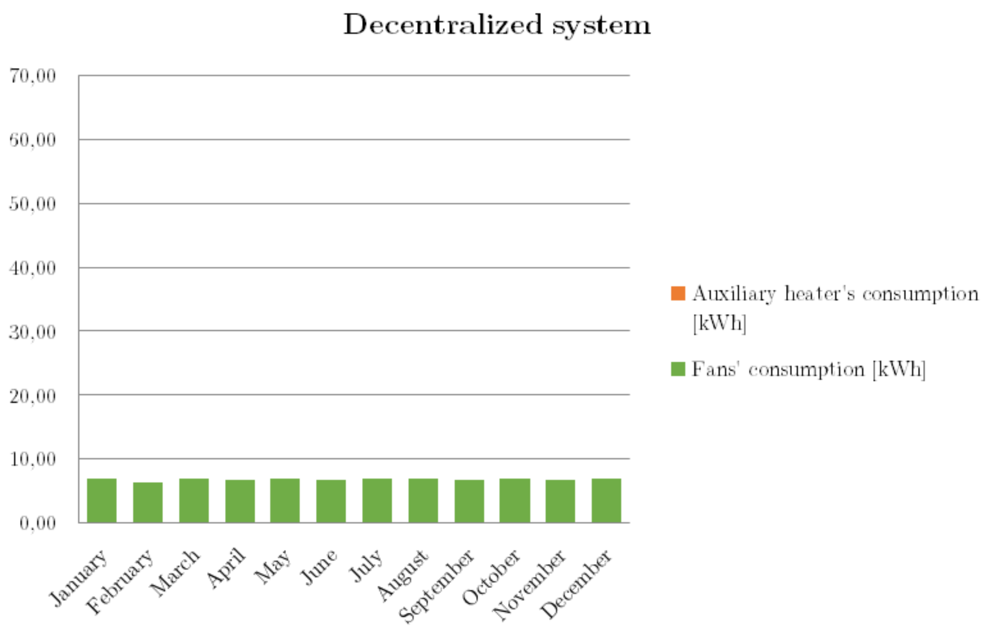

Figure 4.19: Electrical consumption with decentralized syststem, case 5.

4.6 Case 6

In the sixth simulation, the following conditions are considered:

- Palermo
- No heat recovery
- Cooling system
- Italian building

In this configuration, the heating demand is 9% less with decentralized system, the cooling demand is 8% less (sensible) and 66% less (latent). The electrical consumption is 74% less with decentralized system. Table 4.20A summarizes the results. Primary energy demand for centralized and decentralized system are respectively $EP_c = 5168.3$ kWh and $EP_d = 4298.2$ kWh.

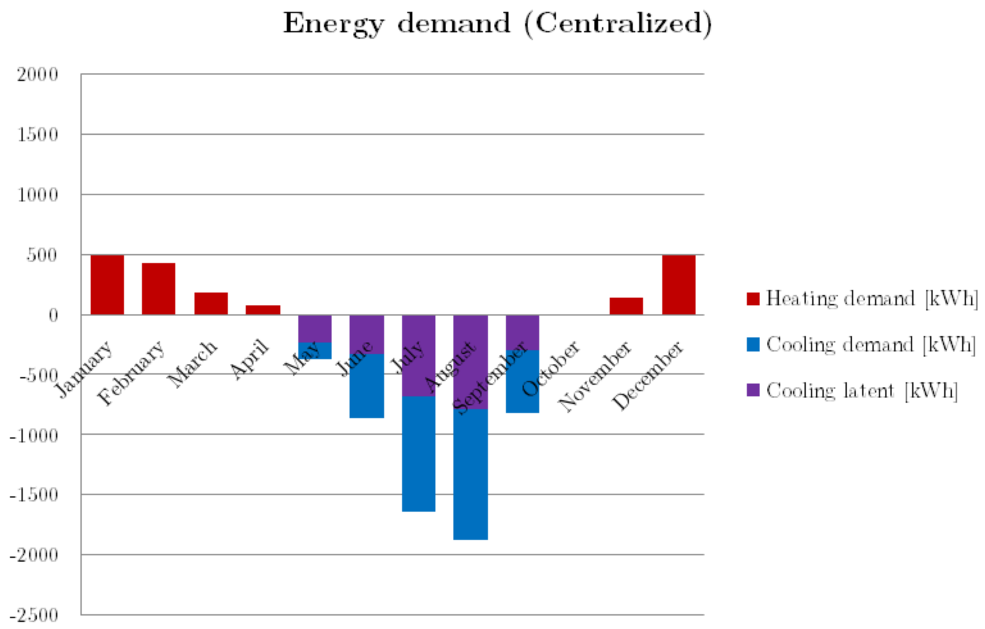

Figure 4.20: Energy demand with centralized system, case 6.

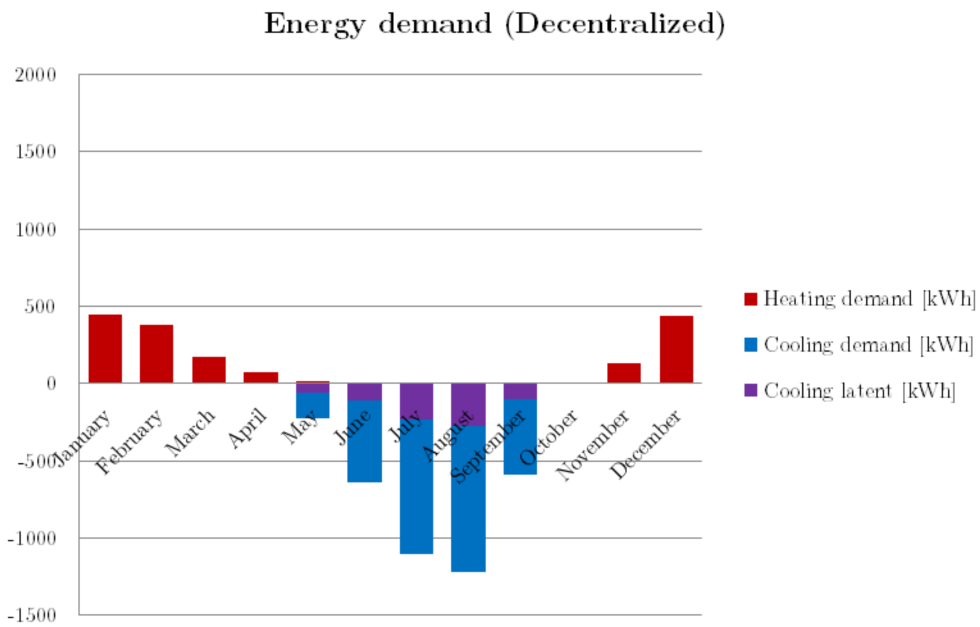

Figure 4.21: Energy demand with decentralized system, case 6.

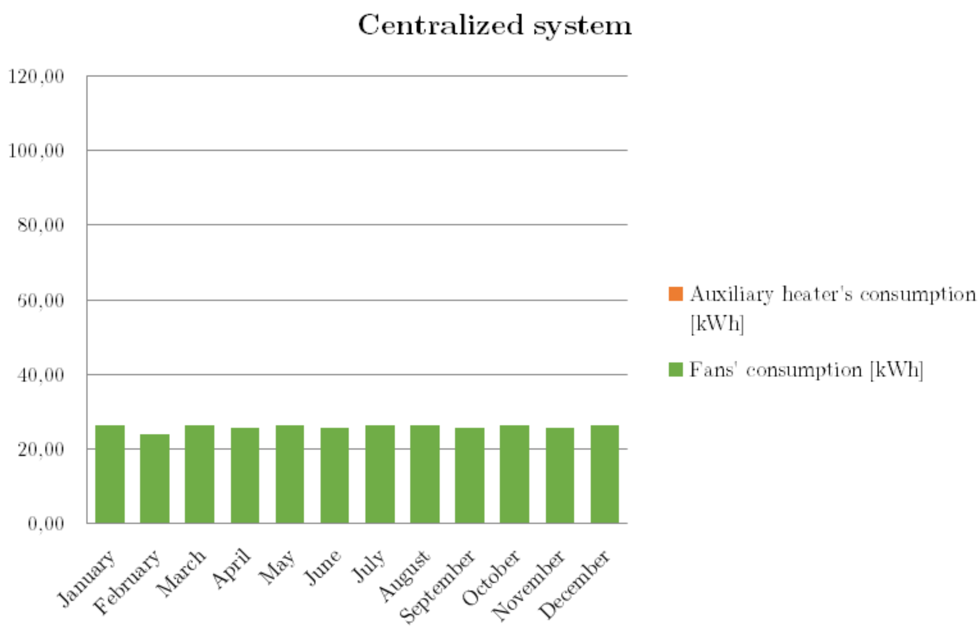

Figure 4.22: Electrical consumption with centralized system, case 6.

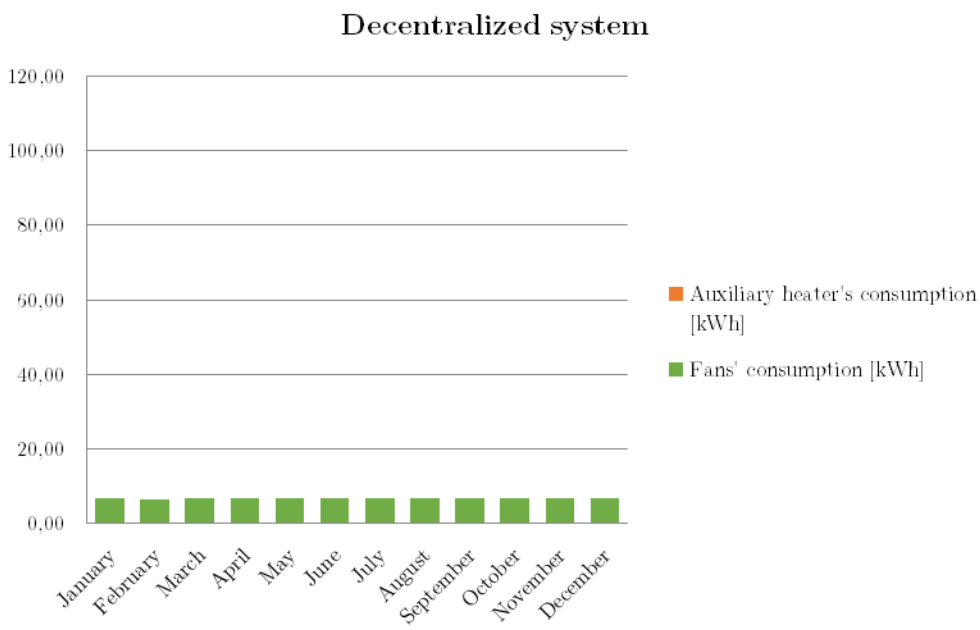

Figure 4.23: Electrical consumption with decentralized system, case 6.

4.7 Case 7

In the seventh simulation, the following conditions are considered:

- Palermo
- Heat recovery

- Cooling system
- German building

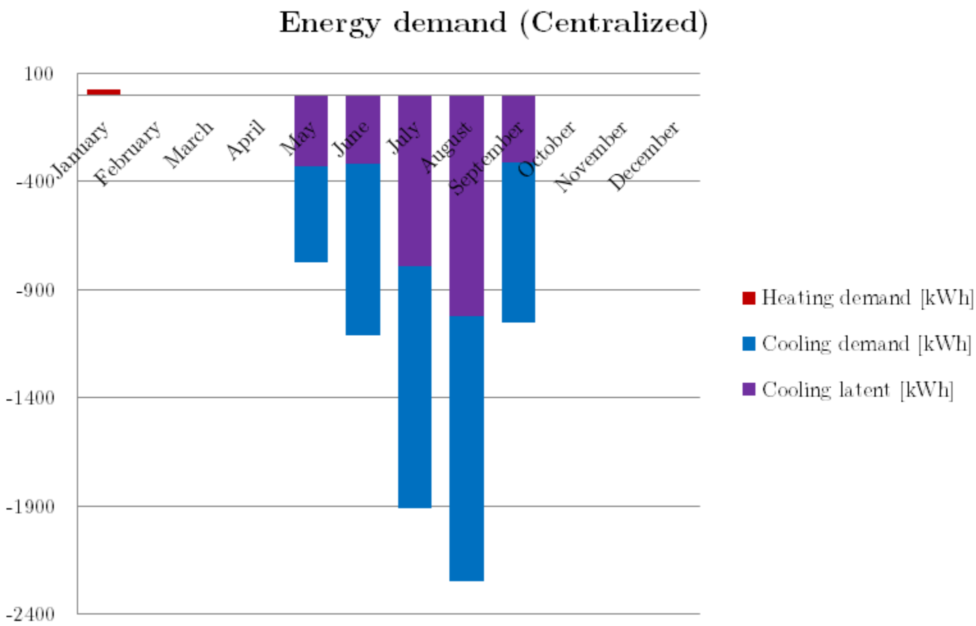

Figure 4.24: Energy demand with centralized system, case 7.

In this configuration, the heating demand is 85% more with decentralized ventilation system; the sensible and latent cooling demands are respectively 12% less

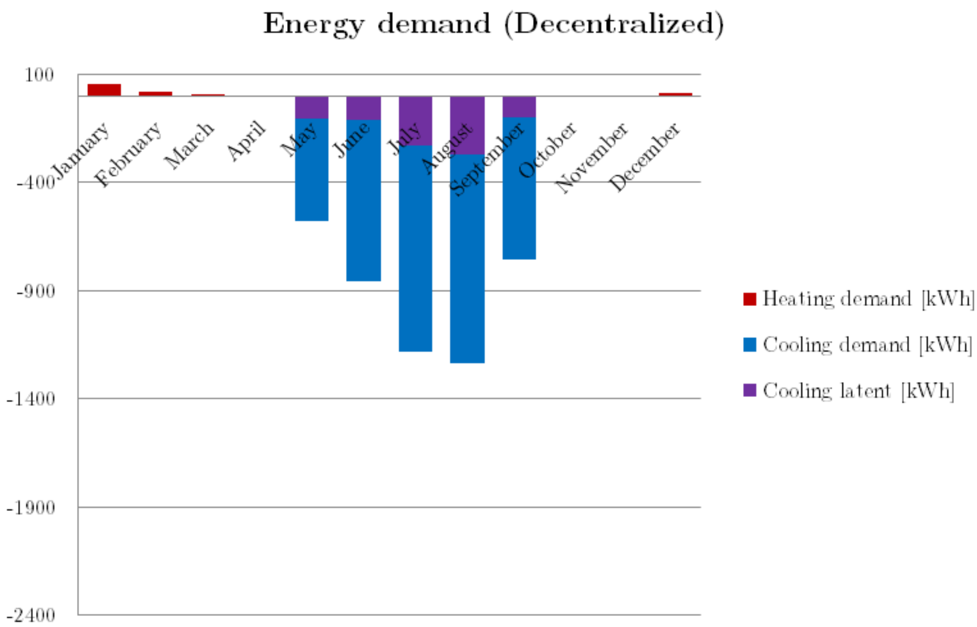

Figure 4.25: Energy demand with decentralized system, case 7.

and 70% less. The electrical consumption is 46% less than with centralized system. Table 4.22A summarizes the results. Primary energy demand for centralized and decentralized system are respectively $EP_c = 3878.8$ kWh and $EP_d = 3057.7$ kWh.

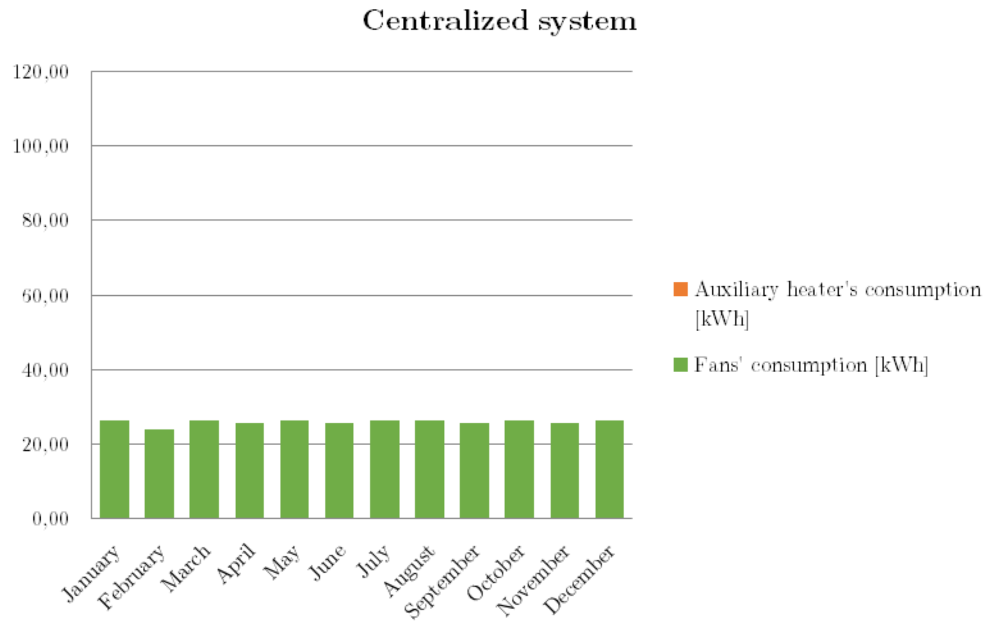

Figure 4.26: Electrical consumption with centralized system, case 7.

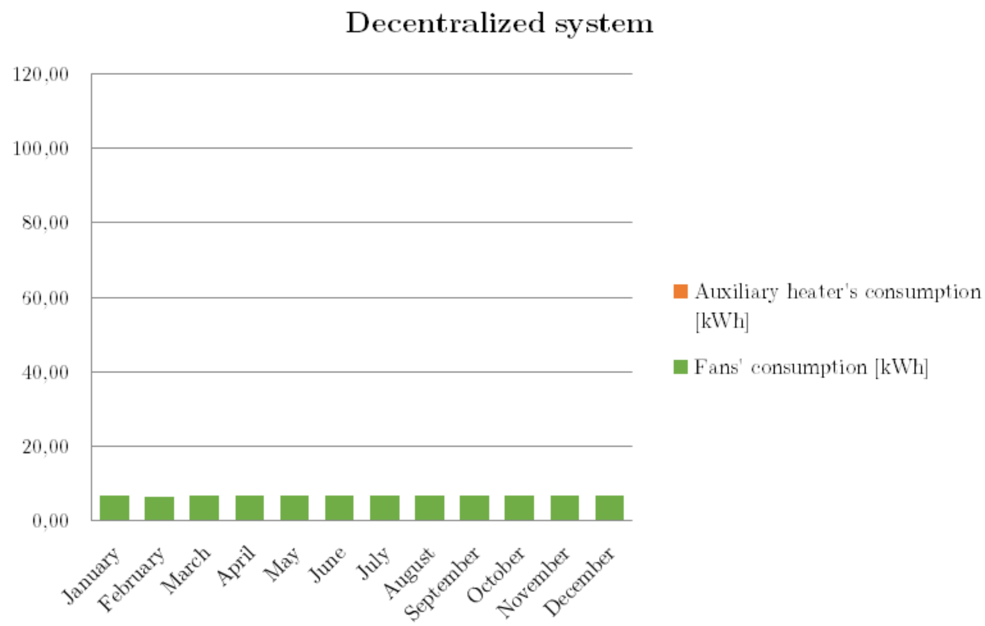

Figure 4.27: Electrical consumption with decentralized system, case 7.

4.8 Case 8

In the eighth simulation, the following conditions are considered:

- Palermo
- No heat recovery
- Cooling system
- German building

In this configuration, the heating demand with decentralized ventilation system is 62% more than with centralized system; the cooling demand is 9% less (sensible) and 65% less (latent). The electrical consumption is 74% less with decentralized system. Table 4.21A summarizes the simulation's results with this configuration. Primary energy demand for centralized and decentralized system are respectively $EP_c = 3741.9$ kWh and $EP_d = 3115.2$ kWh.

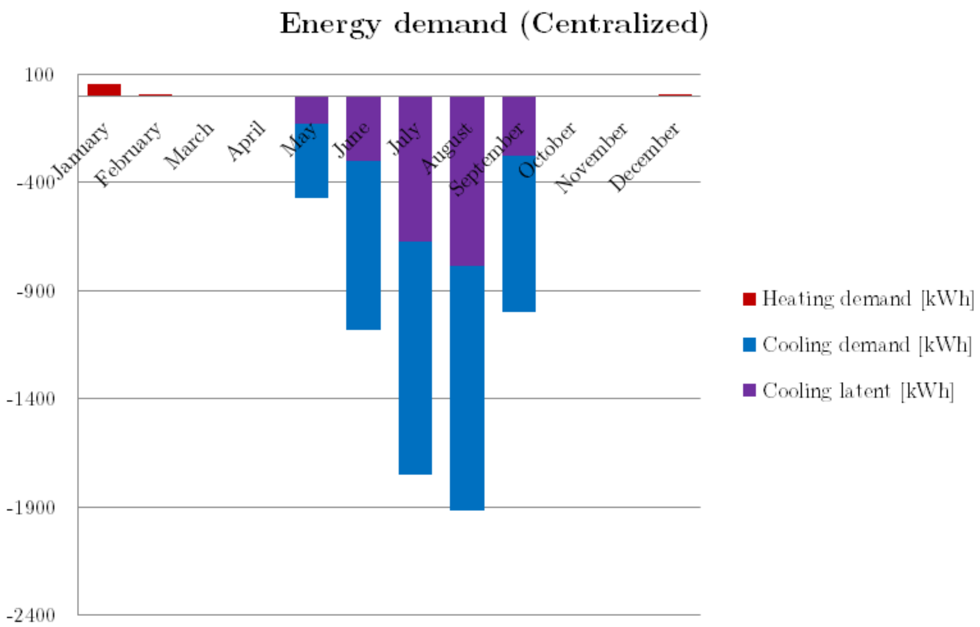

Figure 4.28: Energy demand with centralized system, case 8.

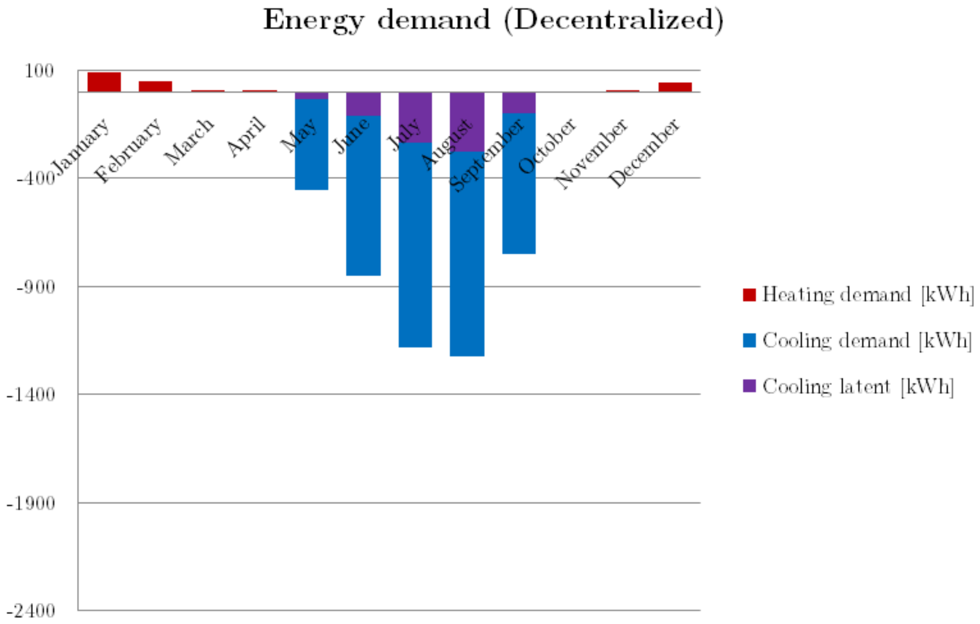

Figure 4.29: Energy demand with decentralized system, case 8.

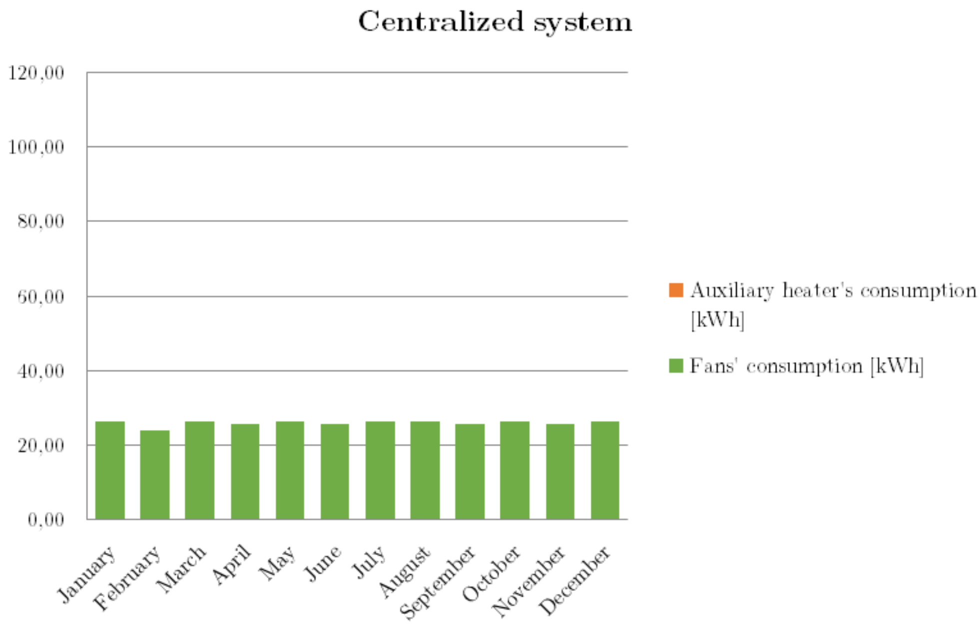

Figure 4.30: Electrical consumption with centralized system, case 8.

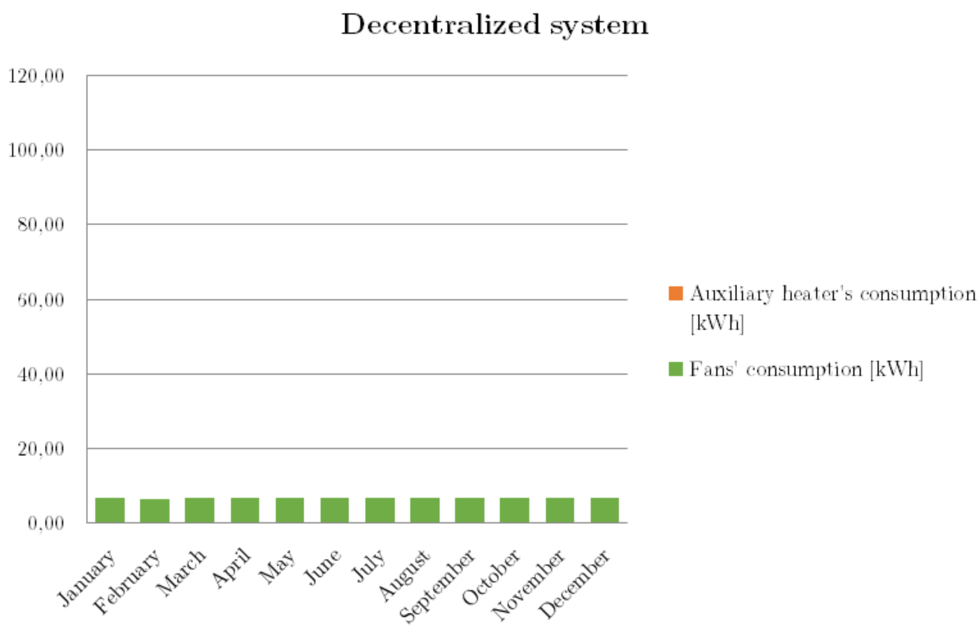

Figure 4.31: Electrical consumption with decentralized system, case 8.

Conclusions

Two different mechanical ventilation systems, centralized and decentralized, were considered in this work, in order to compare the energy performance. In the first chapter, a study of the state-of-the-art in literature of mechanical ventilation systems was conducted, concerning the aim of ventilation in dwellings, the problems related to new air-tight buildings and the kind of ventilation. In the second chapter, the methodology of work was explained, with a focus on the two considered configurations of the ventilation system and the thermal environment. In the third chapter, the software TRNSYS and the model of the residential building considered were presented, concerning the settings and the assumptions made. In the fourth chapter, the following results of different case of studies were presented:

- The centralized system requires higher airflow rates, in order to supply the design airflows in each room. Therefore, it recovers more energy than the decentralized system, but the former is more ‘energivorous’.
- In colder climate with heat recovery, the energy demand for heating and sensible cooling does not change sensibly in the two configurations. Energy demand for latent cooling and electrical consumption (fans and auxiliary heaters) are sensibly lower in decentralized system.
- In colder climate without heat recovery, lower energy demand for heating, latent cooling and electrical consumption is required with decentralized system. However, it requires higher energy demand for sensible cooling.

- In warmer climate, the decentralized system requires more energy for heating and less for cooling and electrical consumption with respect to the centralized system.

The decentralized ventilation system appears to be more convenient in terms of energy demand for heating and cooling than the centralized system in most of the cases, since the latter has to supply an higher airflow rate to provide the design airflow in each room. However, other factors (e.g., installation and O&M costs) could influence the convenience of this configuration. This issue could be an interesting continuation of this work.

Appendix A

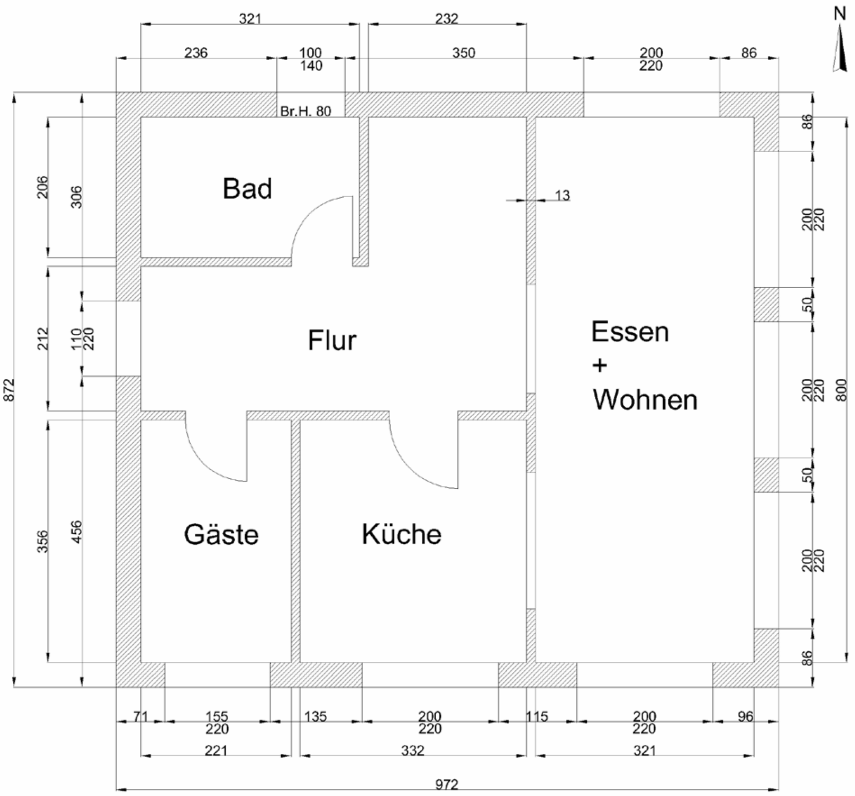

Figure 4.1A: Ground floor plan.

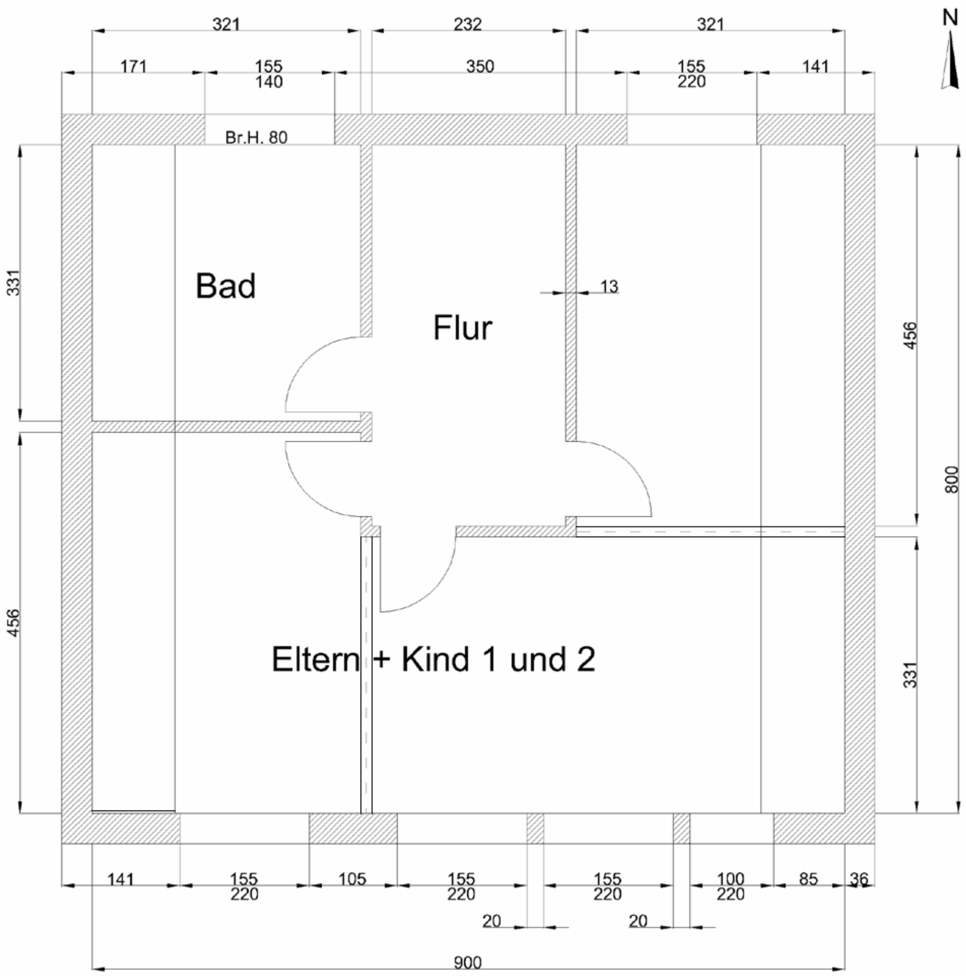

Figure 4.2A: First floor plan.

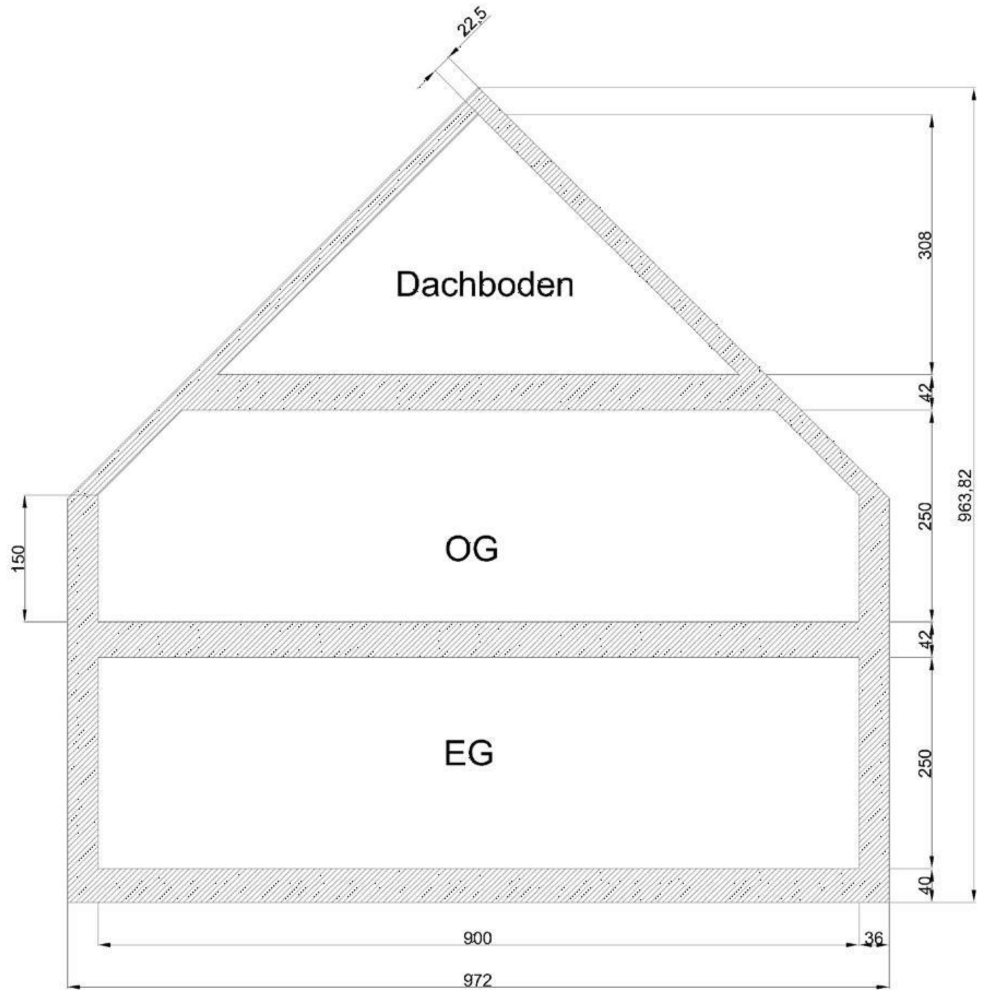

Figure 4.3A: Vertical section of the building.

Table 4.1A: Building materials characteristics.

Building Materials	ρ in kg/m³	c_p in kJ/(kg·K)	λ in W/(m·K)	Thickness in mm	U-value in W/(m²·K)
External wall					
Plaster	1200	0,84	0,35	15	
Calcareous sandstone	1600	1	0,79	175	
Polystyrene	15	1,25	0,039	175	
Plaster	1200	0,84	0,35	20	
Total				385	0,200
Interior wall					
Gypsum	900	1	0,21	13	
Mineral wool	60	1	0,04	100	
Gypsum	900	1	0,21	13	
Total				126	0,358
Ceiling					
Tile	2000	1	1	20	
Cement screed	2000	1	1,4	120	
Polystyrene 025	15	1,25	0,025	115	
Reinforced concrete	2400	0,84	2,2	180	
Plaster	1200	0,84	0,35	20	
Total				455	0,199
Floor slab					
Tile	2000	1	1	20	
Cement screed	2000	1	1,4	120	
Polystyrene 025	15	1,25	0,025	115	
Reinforced concrete	2400	0,84	2,2	200	
Total				455	0,201
Roof					
OSB	650	1,88	0,13	15	
Polystyrene	15	1,25	0,039	150	
Vapor barrier	700	1	0,13	20	
Roof tile	530	0,9	0,14	40	
Total				225	0,218

Table 4.2A: Door and windows properties.

Building materials	ρ in kg/m³	c_p in kJ/(kg·K)	λ in W/(m·K)	Thickness in mm	U-value in W/(m²·K)
Door	600	2,1	0,097	40	
Total				40	1,720

	g-value	U-value in W/(m² · K)
Window		
Heat insulated glass	0,598	0,86
Frame	-	2,269
Total		1,071

Table 4.3A: Guest room details.

Floor area in m²	7,9	**Guest room GF**			
Maximum room height in m	2,5				
Room volume in m³	19,7				
Wall	Length in m	Width in m	Height in m	Area in m²	Orientation
Floor slab	3,6	2,2		7,9	Horizontal
Exterior wall	2,2		2,5	5,5	South
Window				3,4	South
Exterior wall	3,6		2,5	8,9	West
Ceiling	3,6	2,2		8,2	Horizontal
Internal wall	3,6		2,5	8,9	East
Internal wall	2,2		2,5	5,7	North

Table 4.4A: Kitchen details.

Floor area in m^2	11,8	**Kitchen GF**			
Maximum room height in m	2,5				
Room volume in m^3	29,5				
Wall	**Length in m**	**Width** in m	Height **in m**	**Area** in m^2	**Orientation**
Floor slab	3,6	3,3		11,8	Horizontal
Exterior wall	3,3		2,5	8,3	South
Window				4,4	South
Interior wall	3,6		2,5	8,9	West
Ceiling	3,6	3,3		11,7	Horizontal
Ceiling	2,4	0,2		0,4	Horizontal
Interior wall	3,6		2,5	9,1	East
Interior wall	3,3		2,5	8,5	North

Floor area in m^2	25,7	Living room GF			
Maximum room height in m	2,5				
Room volume in m^3	64,2				
Wall	Length in m	Width **in m**	Height **in m**	**Area** in m^2	**Orientation**
Floor slab	8,0	3,2		25,7	Horizontal
Interior wall	3,6		2,5	9,1	West
Exterior wall	3,2		2,5	8,0	South
Window				4,4	South
Exterior wall	8,0		2,5	20,0	East
Window				13,2	East
Exterior wall	3,2		2,5	8,0	North
Window				4,4	North
Ceiling	8,0	3,2		26,0	Horizontal
Interior wall	4,4		2,5	10,8	West

Table 4.5A: Corridor on ground floor and toilet details.

Floor area in m²	17,1	**Corridor GF**			
Maximum room height in m	2,5				
Room volume in m³	42,7				
Wall	**Length in m**	Width **in m**	**Height** in m	**Area** in m²	**Orientation**
Floor slab	5,7	3,0		17,1	Horizontal
Exterior wall	2,1		2,5	2,9	West
Interior wall	3,4		2,5	8,5	South
Interior wall	4,3		2,5	10,8	East
Exterior wall	2,3		2,5	5,8	North
Ceiling	3,3	0,9		3,0	Horizontal
Interior wall	2,3		2,5	5,7	South
Door				2,4	West
Ceiling	4,3	2,4		10,1	Horizontal
Interior wall	3,3		2,5	8,0	North
Interior wall	2,2		2,5	5,2	West
Ceiling	3,3	1,2		3,8	Horizontal

Floor area in m²	6,6	**Toilet GF**			
Maximum room height in m	2,5				
Room volume in m³	16,5				
Wand	**Length in m**	Width **in m**	**Height** in m	**Area** in m²	**Orientation**
Floor slab	3,2	2,1		6,6	Horizontal
Exterior wall	2,1		2,5	5,2	West
Exterior wall	3,2		2,5	8,0	North
Window				1,4	North
Ceiling	3,2	2,1		6,8	Horizontal
Interior wall	3,2		2,5	8,0	South
Interior wall	2,1		2,5	5,2	East

Table 4.6A: Bedroom 1 details.

Floor area in m^2	14,6	**Bedroom 1 1F**			
Maximum room height in m	2,5				
Room volume in m^3	29,18				
Wall	Length **in m**	**Width** in m	**Height** in m	**Area** in m^2	**Orientation**
Exterior wall	4,56		1,5	6,8	West
Exterior wall	3,21		2,5	7,8	South
Window				3,4	South
Interior wall	2,21		2,5	7,5	North
Interior wall	1,12		2,5	0,8	East
Interior wall	3,31		2,5	8,3	East
Roof	1,4	4,56		6,5	West
Floor	2,21	3,56		7,9	Horizontal
Floor	0,87	3,56		3,1	Horizontal
Floor	0,87	3,21		2,8	Horizontal
Ceiling	1,76	4,56		8,0	Horizontal

Table 4.7A: Bedroom 2 details.

Floor area in m^2	18,7	**Bedroom 2 1F**			
Maximum room height in m	2,5				
Room volume in m^3	46,84				
Wall	**Length in m**	**Width in m**	**Height in m**	**Area in m^2**	**Orientation**
Exterior wall	5,7		2,5	13,7	South
Exterior wall	3,31		1,5	5,0	East
Window				9,0	South
Interior wall	2,32		2,5	3,8	North
Interior wall	3,21		2,5	7,5	North
Interior wall	3,31		2,5	8,3	West
Roof	1,4	3,31		4,6	East
Floor	2,32	3,31		7,7	Horizontal
Floor	3,21	3,31		10,6	Horizontal
Ceiling	4,21	3,31		13,9	Horizontal

Table 4.8A: Bedroom 3 details.

Floor area in m^2	14,6	**Bedroom 3 1F**			
Maximum room height in m	2,5				
Room volume in m^3	29,18				
Wall	Length **in m**	**Width** in m	**Height** in m	**Area** in m^2	**Orientation**
Exterior wall	4,56		1,5	6,8	East
Exterior wall	3,21		2,5	7,8	North
Window				3,4	North
Interior wall	2,21		2,5	7,5	West
Interior wall	1,12		2,5	0,8	West
Interior wall	3,31		2,5	8,3	South
Roof	1,4	4,56		6,5	East
Floor	2,21	3,56		7,9	Horizontal
Floor	0,87	3,56		3,1	Horizontal
Floor	0,87	3,21		2,8	Horizontal
Ceiling	1,76	4,56		8,0	Horizontal

Table 4.9A: Corridor on first floor and bathroom details.

Floor area in m^2	10,6	**Corridor 1F**			
Maximum room height in m	2,5				
Room volume in m^3	26,4				
Wall	**Length in m**	Width **in m**	**Height** in m	**Area** in m^2	**Orientation**
Ceiling	4,6	2,3		10,6	Horizontal
Interior wall	1,2		2,5	3,0	West
Exterior wall	2,3		2,5	5,8	North
Interior wall	4,6		2,5	11,4	East
Interior wall	2,3		2,5	6,5	South
Interior wall	3,3		2,5	8,3	West
Floor	2,3	0,2		0,4	Horizontal
Floor	4,4	2,3		10,1	Horizontal

Floor area in m^2	10,6	**Bathroom 1F**			
Maximum room height in m	2,5				
Room volume in m^3	24,9				
Wall	**Length in m**	Width **in m**	**Height** in m	**Area** in m^2	**Orientation**
Ceiling	3,3	2,2		7,3	Horizontal
Exterior wall	3,3		1,5	5,0	West
Exterior wall	3,2		2,5	7,7	North
Window				2,2	North
Roof	3,3	1,4		4,7	West 45°
Floor	3,2	1,2		3,8	Horizontal
Floor	3,2	2,1		6,8	Horizontal
Interior wall	3,3		2,5	8,3	East
Interior wall	3,2		2,5	7,5	South

Table 4.11A: Occupancy pattern for weekdays.

	Guest room	Kitchen	Livingroom	Toilet	Corridor	Bedroom 1	Bedroom 2	Bedroom 3	Corridor 1	Bathroom
23:30						1	2	1		
23:00						1	2	1		
22:30			2			1		1		
22:00			1			1		1		1
21:30			3	1						
21:00			3	1						
20:30			2							
20:00		4								
19:30		3		1						
19:00		1	2	1						
18:30			2							1
18:00			2							1
17:30			2							
17:00			2							
16:30			2							
16:00										
15:30										
15:00										
14:30										
14:00										
13:30										
13:00		2								
12:30		2								
12:00		2								
11:30										
11:00										
10:30										
10:00										
09:30										
09:00										
08:30		4								
08:00		4								
07:30		2		1						1
07:00		2		1						1
06:30						1		1		
06:00						1	2	1		
05:30						1	2	1		
05:00						1	2	1		
04:30						1	2	1		
04:00						1	2	1		
03:30						1	2	1		
03:00						1	2	1		
02:30						1	2	1		
02:00						1	2	1		
01:30						1	2	1		
01:00						1	2	1		
00:30						1	2	1		
00:00						1	2	1		

Table 4.10A: Occupancy pattern for weekends.

	Guest room	Kitchen	Livingroom	Toilet	Corridor	Bedroom 1	Bedroom 2	Bedroom 3	Corridor 1	Bathroom
23:30						1	2	1		
23:00			2			1		1		
22:30	1		2							
22:00	1		2							1
21:30			4							
21:00			4							
20:30			4							
20:00		1	2	1						
19:30		4								
19:00		4								
18:30			2							1
18:00			2	1						1
17:30			2							
17:00			2							
16:30										
16:00										
15:30										
15:00				1						
14:30										
14:00										
13:30										
13:00		4								
12:30		4								
12:00		4								
11:30		2								
11:00		2	2							
10:30			4							
10:00			3	1						
09:30		4								1
09:00		4								1
08:30						1	2	1		1
08:00						1	2	1		
07:30						1	2	1		
07:00						1	2	1		
06:30						1	2	1		
06:00						1	2	1		
05:30						1	2	1		
05:00						1	2	1		
04:30						1	2	1		
04:00						1	2	1		
03:30						1	2	1		
03:00						1	2	1		
02:30						1	2	1		
02:00						1	2	1		
01:30						1	2	1		
01:00						1	2	1		
00:30						1	2	1		
00:00						1	2	1		

Table 4.12A: Appliances' schedule.

23:30				1				
23:00				1				
22:30				1		1		
22:00				1	1			
21:30				1	1			
21:00				1	1			1
20:30				1	1			
20:00				1	1			
19:30				1	1			
19:00				1				
18:30	1			1				1
18:00				1	1	1		
17:30				1	1	1		
17:00				1				1
16:30				1				1
16:00				1				
15:30				1				
15:00				1				
14:30			1	1				
14:00			1	1				
13:30				1				
13:00				1				
12:30				1				
12:00	1	1		1				
11:30	1			1				
11:00				1				
10:30				1				
10:00				1				
09:30				1			1	
09:00				1			1	
08:30				1			1	
08:00				1			1	
07:30				1				
07:00				1				
06:30				1				
06:00				1				
05:30				1				
05:00				1				
04:30				1				
04:00				1				
03:30				1				
03:00				1				
02:30				1				
02:00				1				
01:30				1				
01:00				1				
00:30				1				
00:00				1				
	Hob extractor	Oven	Dishwasher	Fridge	TV	Music system	Washma-	PC

Table 4.13A: Calculation of pressure drop in each duct portion of the distribution system.

Room	Airflow needed m^3/h	flow/ flow max 1duct	n° ducts	flow per duct m^3/h	line lenght m	total ducts lenght m	n° horiz bends	n° vertical bends	Z-value duct	Z-value horiz bends	Z-value vertical bends
Livingroom	100,8	2,88	3	33,6	3,1	9,3	0	3	20,925	0	2,9358
Guestroom	25,2	0,72	1	25,2	7,5	7,5	2	1	16,875	1,9572	0,9786
Bed1	25,2	0,72	1	25,2	2,4	2,4	0	0	5,4	0	0
Bed2	50,4	1,44	2	25,2	1,8	3,6	2	0	8,1	1,9572	0
Bed3	25,2	0,72	1	25,2	0,7	0,7	1	0	1,575	0,9786	0
Kitchen	94,4	2,696192	3	31,5	4,3	12,9	0	3	29,025	0	2,9358
Toilet	52,8	1,508039	2	26,4	4,4	8,8	0	2	19,8	0	1,9572
Bathroom	79,7	2,275769	3	26,6	0,8	2,4	3	0	5,4	2,9358	0

Table 4.15A: Calculation of pressure drop in each duct portion of the distribution system.

Room	duct	duct lenght [m]	Z-value duct	Z-value collectors+ nozzles	Z-value total	Δp duct [Pa]	Z-value flow regulator	Total flow rate [m³/h]	Δp plenum [Pa]
Livingroom	A	3,1	6,97748	18,5152	26,47128	29,885016	0	226,8	29,88502
	B	3,1	6,97748	18,5152	26,47128	29,885016	0		
	C	3,1	6,97748	18,5152	26,47128	29,885016	0		
GueStroom	A	7,5	16,881	1,0204	20,8372	13,232455	26,22285		
Bed1	A	2,4	5,40192	1,0204	6,42232	4,0784301	40,63773		
Bed2	A	1,8	4,05144	18,5152	23,54524	14,952169	23,51481		
	B	1,8	4,05144	18,5152	23,54524	14,952169	23,51481		
Bed3	A	0,7	1,57556	1,0204	3,57456	2,2699886	43,48549		
Kichen	A	4,3	9,67844	9,9036	20,56064	20,343786	0	226,8	20,34379
	B	4,3	9,67844	9,9036	20,56064	20,343786	0		
	C	4,3	9,67844	9,9036	20,56064	20,343786	0		
Toilet	A	4,4	9,90352	9,9036	20,78572	14,476601	8,424192		
	B	4,4	9,90352	9,9036	20,78572	14,476601	8,424192		
Bathroom	A	0,8	1,80064	9,9036	12,68284	8,9405931	16,17621		
	B	0,8	1,80064	9,9036	12,68284	8,9405931	16,17621		
	C	0,8	1,80064	9,9036	12,68284	8,9405931	16,17621		

Table 4.14A: Calculation of pressure drop in inlet and exhaust ducts.

	flow rate [m³/h]	duct lenght [m]	Z-value duct (Z/m · m)	n° bends	Z-value bends	Z-value wall terminals	Z-value silencer	Z-value total	Δp [Pa]	Total Head [Pa]
Inlet air	226,8	2,2	0,022	3	0,18	0,14	0,0245	0,3665	18,852115	48,7371312
Exhaust air	226,8	2,7	0,027	2	0,12	0,0328	0,0245	0,2043	10,508832	30,8526186

Table 4.16A: Summary of the results, case 1.

	Decentralized	**Centralized**
Heating demand [kWh/y]	4078,24	4042,79
Specific heating demand [kWh/m^{2}y]	29,51	29,25
Cooling demand [kWh/y]	-2070,62	-2046,36
Specific cooling demand[kWh/m^{2}y]	-14,98	-14,81
Ventilation losses [kWh/y]	-2530,79	-2715,81
Specific ventilation losses [kWh/m^{2}y]	-18,31	-19,65
Heat recovered by HR [kWh/y]	2845,14	5259,02
Cool recovered by HR [kWh/y]	0,57	2,79
Total energy recovered by HR [kWh/y]	2845,72	5261,81
Fans' consumption [kWh$_e$/y]	81,70	310,42
Auxiliary heater [kWh$_e$/y]	107,85	123,15
Total electrical consumption [kWh$_e$/y]	189,55	433,57
Specific fan power [Wh$_e$/m^3]	0,09	0,20
(Heat recovered $\triangle$Q)/ (Electrical consumption E)	15,01	12,13

Table 4.17A: Summary of the results, case 2.

	Decentralized	**Centralized**
Heating demand [kWh/y]	5526,93	7301,66
Specific heating demand [kWh/m^{2}y]	39,99	52,83
Cooling demand [kWh/y]	-1713,33	-1325,92
Specific cooling demand[kWh/m^{2}y]	-12,40	-9,59
Ventilation losses [kWh/y]	-5007,95	-7459,46
Specific ventilation losses [kWh/m^{2}y]	-36,24	-53,98
Heat recovered by HR [kWh/y]	0,00	0,00
Cool recovered by HR [kWh/y]	0,00	0,00
Total energy recovered by HR [kWh/y]	0,00	0,00
Fans' consumption [kWh$_e$/y]	81,70	310,42
Auxiliary heater [kWh$_e$/y]	107,85	123,15
Total electrical consumption [kWh$_e$/y]	189,55	433,57
Specific fan power [Wh$_e$/m^3]	0,09	0,20
(Heat recovered ΔQ)/ (Electrical consumption E)	0,00	0,00

Table 4.18A: Summary of the results, case 3.

	Decentralized	**Centralized**
Heating demand [kWh/y]	4191,46	4138,83
Specific heating demand [kWh/m^{2}y]	30,33	29,95
Cooling demand [kWh/y]	0,00	0,00
Specific cooling demand[kWh/m^{2}y]	0,00	0,00
Ventilation losses [kWh/y]	-3037,33	-3532,48
Specific ventilation losses [kWh/m^{2}y]	-21,98	-25,56
Heat recovered by HR [kWh/y]	3087,49	5067,57
Cool recovered by HR [kWh/y]	0,39	0,35
Total energy recovered by HR [kWh/y]	3087,88	5067,93
Fans' consumption [kWh$_e$/y]	76,70	310,42
Auxiliary heater [kWh$_e$/y]	113,46	123,15
Total electrical consumption [kWh$_e$/y]	190,16	433,57
Specific fan power [Wh$_e$/m^3]	0,08	0,20
(Heat recovered ΔQ)/ (Electrical consumption E)	16,24	11,69

Table 4.19A: Summary of the results, case 5.

	Decentralized	**Centralized**
Heating demand [kWh/y]	1289,69	927,74
Specific heating demand [kWh/m^{2}y]	9,33	6,71
Cooling demand [kWh/y]	-3065,83	-3380,59
Specific cooling demand[kWh/m^{2}y]	-22,18	-24,46
Ventilation losses [kWh/y]	-1460,56	-1212,89
Specific ventilation losses [kWh/m^{2}y]	-10,57	-8,78
Heat recovered by HR [kWh/y]	1203,50	2011,23
Cool recovered by HR [kWh/y]	-26,88	4,51
Total energy recovered by HR [kWh/y]	1176,62	2015,74
Fans' consumption [kWh$_e$/y]	81,70	310,42
Auxiliary heater [kWh$_e$/y]	0,00	0,00
Total electrical consumption [kWh$_e$/y]	81,70	310,42
Specific fan power [Wh$_e$/m^3]	0,09	0,20
(Heat recovered $\triangle$Q)/ (Electrical consumption E)	14,73	6,48

Table 4.20A: Summary of the results, case 6.

	Decentralized	**Centralized**
Heating demand [kWh/y]	1645,83	1802,33
Specific heating demand [kWh/m^{2}y]	11,91	13,04
Cooling demand [kWh/y]	-2997,71	-3249,08
Specific cooling demand[kWh/m^{2}y]	-21,69	-23,51
Ventilation losses [kWh/y]	-2521,15	-2968,04
Specific ventilation losses [kWh/m^{2}y]	-18,24	-21,48
Heat recovered by HR [kWh/y]	0,00	0,00
Cool recovered by HR [kWh/y]	0,00	0,00
Total energy recovered by HR [kWh/y]	0,00	0,00
Fans' consumption [kWh$_e$/y]	81,70	310,42
Auxiliary heater [kWh$_e$/y]	0,00	0,00
Total electrical consumption [kWh$_e$/y]	81,70	310,42
Specific fan power [Wh$_e$/m^3]	0,09	0,20
(Heat recovered $\triangle$Q)/ (Electrical consumption E)	0,00	0,00

Table 4.22A: Summary of the results, case 7.

	Decentralized	**Centralized**
Heating demand [kWh/y]	84,55	24,25
Specific heating demand [kWh/m^{2}y]	0,61	0,18
Cooling demand [kWh/y]	-3790,03	-4319,50
Specific cooling demand[kWh/m^{2}y]	-27,42	-31,26
Ventilation losses [kWh/y]	-2110,19	-1777,44
Specific ventilation losses [kWh/m^{2}y]	-15,27	-12,86
Heat recovered by HR [kWh/y]	1381,57	2508,40
Cool recovered by HR [kWh/y]	-25,69	0,36
Total energy recovered by HR [kWh/y]	1355,88	2508,77
Fans' consumption [kWh$_e$/y]	81,70	310,42
Auxiliary heater [kWh$_e$/y]	0,00	0,00
Total electrical consumption [kWh$_e$/y]	81,70	310,42
Specific fan power [Wh$_e$/m^3]	0,09	0,20
(Heat recovered ΔQ)/ (Electrical consumption E)	16,91	8,08

Table 4.21A: Summary of the results, case 8.

	Decentralized	**Centralized**
Heating demand [kWh/y]	187,62	71,52
Specific heating demand [kWh/m^{2}y]	1,36	0,52
Cooling demand [kWh/y]	-3704,42	-4057,52
Specific cooling demand[kWh/m^{2}y]	-26,80	-29,36
Ventilation losses [kWh/y]	-3260,95	-3757,32
Specific ventilation losses [kWh/m^{2}y]	-23,60	-27,19
Heat recovered by HR [kWh/y]	0,00	0,00
Cool recovered by HR [kWh/y]	0,00	0,00
Total energy recovered by HR [kWh/y]	0,00	0,00
Fans' consumption [kWh$_e$/y]	81,70	310,42
Auxiliary heater [kWh$_e$/y]	0,00	0,00
Total electrical consumption [kWh$_e$/y]	81,70	310,42
Specific fan power [Wh$_e$/m^3]	0,09	0,20
(Heat recovered ΔQ)/ (Electrical consumption E)	0,00	0,00

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