Low Temperature District Heating

- Consequences for Existing Buildings.

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Brian Puggaard Thomsen

Abstract

Denne diplomprojektrapport belyser konsekvenserne for den eksisterende boligmasse, hvis fjernvarmesektoren implementerer lavtemperaturfjernvarme. Da op mod 40 % af enfamilieshusene i Danmark er bygget i perioden 1960-1979, er det denne boligtype, der fokuseres på i denne rapport.

Det bliver påvist i rapporten, at hvis temperatursættet for fjernvarmen ændres fra 70/40 °C til 55/30 °C bliver radiatoreffekten reduceret til 55 % af den oprindelige effekt. Det belyses desuden, hvordan der kan kompenseres for dette effekttab ved ændring i bygningens klimaskærm, og hvilke konsekvenser det får for bygningens opvarmning, hvis der ikke foretages ændringer af klimaskærmen.

På baggrund af de opstillede modeller og antagelser, konkluderes det i rapporten, at det kun er i en begrænset periode at varmebehovet for bygningen ikke er opfyldt. Der er således grundlag for at fjernvarmesektoren kan påbegynde overgangen til lavtemperatur, da det kun vil påføre forbrugerne ulemper i begrænset omfang.

This bachelor thesis addresses the consequences for existing buildings if the district heating sector implements low temperature district heating. Since up to 40 % of the existing single-family buildings have been built in the period 1960-1979 the emphasis of this report will be on that specific type of building.

It is shown in this report that if the district heating temperature set is changed from 70/40 °C to 55/30 °C the heating power output of the radiator is reduced to 55 % of the original heating output. Furthermore it is evaluated what changes there have to be done on the building envelope to compensate for this power loss, and what the consequences will be for heating of the building if nothing is done on the building envelope.

Based on the created models and assumptions in the report it is concluded, that it is only in a limited period that the heating demand of the building cannot be fulfilled. Thus, there is basis for the district heating sector to commence the transition to low temperature district heating, since it only will cause limited inconvenience to the consumer.

Preface

The background for this bachelor thesis is my internship at "Grøn Energi", a department of Danish District Heating Assosiation, where I was involved in an EU-project regarding retrofit of selected buildings in Aarhus, Denmark and in Växsjö, Sweden. The EU- project, READY (**R**esource **E**fficient cities implementing **AD**vanced smart cit**Y** solutions) is a very large an ambitious project addressing retrofit of both multi-storage and single-family buildings.

One part of the READY- project concerns the transition of an area of single-family houses to low temperature district heating. In the district heating sector there is also, in general, a focus on implementation of low temperature district heating. It was therefore relevant for me to address this issue and to elaborate the consequences of this transition.

It is my goal and hope that my analysis and conclusions can be an inspiration for the district heating sector in their pursuit for optimization of the district heating system.

I would like to thank Grøn Energi for giving me the opportunity to write my bachelor thesis as part of their team. It has been an inspiration.

Especially I would like to thank Nina Detlefsen and Kim Søgaard Clausen at Grøn Energi:

Nina for her help adjusting the tasks in READY for my bachelor thesis.

Kim for professional guidance in both my internship and regarding my bachelor thesis.

Brian Puggaard Thomsen
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Table of Contents

1 Low Temperature in District Heating

The development of District Heating over the years has generally been towards lower temperatures. This development can be divided into 4 different phases, or generations as it is often referred.

- 1st generation - Pressurized Steam approximately 200 °C.
- 2nd generation - Pressurized Water above 100 °C.
- 3rd generation - Water just below 100 °C.
- 4th generation - Water below 50-60 °C.

(1)

The fourth generation district heating has not yet been implemented full scale in Denmark, or anywhere else for that matter. There have been numerous projects and tests addressing this, but the major breakthrough is yet to come.

In Denmark the district heating network is well established. Approximately 52 % of the heated area in buildings has district heating as primary heat supply in 2014. (2) The distribution of heat sources in the buildings are shown in Figure 1.

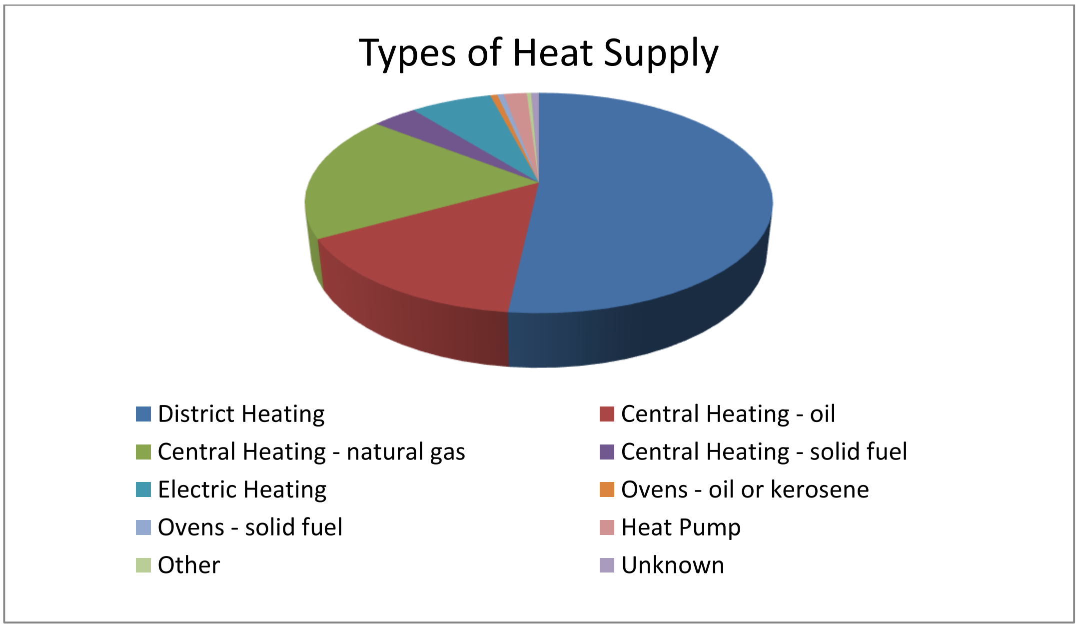

Figure 1: Distribution of the different heat supplies (2)

Even though district heating is one of the best ways to implement renewable energy sources in the energy system, it is still necessary to make the district heating system even more efficient. This is both due to a general concern to minimize energy loss, but also necessary in consideration of making district heating competitive to individual heat supplies in e.g. new low energy buildings.

A way to achieve this is by implementing 4th generation district heating, because lower temperature have some technical advantages, which can contribute to lower prices for the consumer. Some of these advantages will be elaborated in the following sections.

1.1 Heat Loss in Piping

A way to optimize the district heating network is to reduce the heat loss in the piping system. This can be done in several ways. One way is to increase the insulation of the pipes. This is however, a quite expensive solution and the implementation of this is depending of the renovation rate of the piping system. Another way to decrease the heat loss in the piping is to reduce the temperature of the water in the system. According to Newton's Law of Cooling the heat loss from the pipes is, simplified, directly proportional to the temperature difference between the water in the pipes and the surrounding soil. In equation 1.1 this relationship is shown.

$$Q_{Loss} = U \cdot A \cdot \Delta T \qquad \textbf{1.1}$$

(3)

Where

Q_{Loss}	$=$ Heat loss
U	$=$ Overall heat transfer coeffient
A	$=$ Surface area of the pipes
ΔT	$=$ Temperature difference $-$ water and soil

In an existing pipe system the overall heat transfer coefficient and the surface area of the pipes is constant. That means that the heat loss is only depending on ΔT. Since the temperature of the soil is also to be considered constant, the heat loss only depends on the water temperature in the pipes.

Regarding the assumption about the constant U-value it must be observed that, when the supply temperature is decreased, the return temperature is often not decreased by as much as the supply temperature and it will therefore be necessary with a higher flow in the pipes. This will affect the overall heat coefficient U in the system and by that the assumption of the U-value to be constant fails.

1.2 Flue or Exhaust Gas Cooling

Another advantage that can be gained by decreasing the return temperature in the pipes is a higher efficiency on the District Heating plant. The return water is used for cooling the exhaust or flue gas and a lower return temperature means the flue gas is cooled more and thereby more energy is subtracted from the flue gas. To obtain as much energy from the exhaust or flue gas it is crucial, that the gas is cooled to a temperature well below the gas´s dew point. In that way energy from condensation can be utilized.

As an example of that, combustion of natural gas in atmospheric air is considered. Natural gas consists primarily of methane, CH_4 and the reaction by combustion is:

$$CH_4 + 2(O_2 + 3{,}76N_2) \rightarrow CO_2 + 2H_2O + 7{,}52N_2 \quad \textbf{1.2}$$

By simulation in "Nasa-Glenn Chemical Equilibrium Program CEA2" it is found that the water content in the flue gas is approximately 70 g/kg air. This is however only the water from combustion, the water from the air supply is not included in the calculation since that can vary according to season, outside temperature etc. The output of this simulation can be seen in Annex 1 page 41. The simulation is performed under stoichiometric conditions, which is not the case in reality, but it gives a fairly accurate indication of the potential in the flue gas.

To quantify the energy that can be subtracted from the flue gas the program ESS (Engineering Equation Solver) is used to calculate the enthalpy change in the flue gas as a function of temperature. The advantage of using EES is that the program contains all relevant thermodynamic properties for fluids and gasses. The obtained data is processed in an Excel spreadsheet (Annex 1 on CD) and the results are visualized in Figure 2.

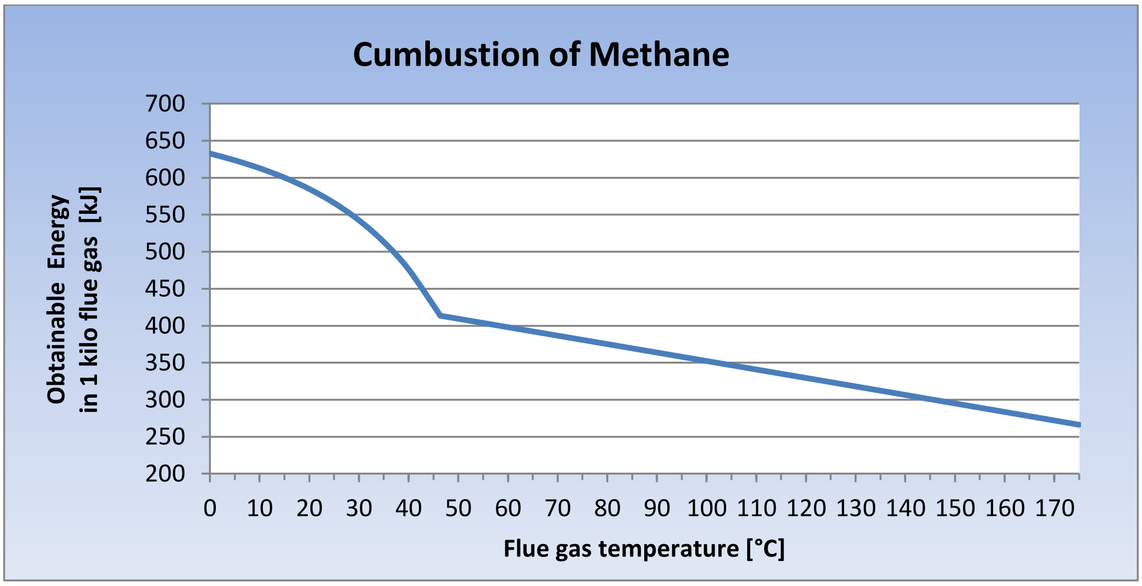

Figure 2: The potential by cooling flue gas with an initial temperature of 400 °C

As it can be seen from the curve, the obtainable energy increases significantly if the flue gas is cooled below approximately 45 °C. This is due to the water content of the flue gas. At the gas´ dew point the contained water will begin to condense. This energy of condensation is causing the increase in the obtainable energy. If the water content of the air used by combustion was included in the simulations it would have resulted in a higher dew point and the significant break of the curve would occur at a higher temperature. The break of the curve would also be at a higher temperature if the applied fuel has a higher water content, which would be the case for e.g. bio mass fuels.

Since the return water is used to cool the flue gas it is important to have low return temperature, definitely below the gas´ dew point, so there can be retrieved as much energy of condensation as possible.

It is however important to notice that a lowering of the supply temperature not automatically will result in a lower return temperature. The opposite can often be the case if the consumer installation does not meet the new requirements. This issue will be addressed in this report

1.3 Implementing Renewable Energy Sources

Many renewable energy sources as solar energy, surplus heat from industry and electrical heat pumps have a decreasing efficiency at increasing temperature. For that reason, low temperature district heating is more suitable for implementing these renewable energy sources.

1.3.1 Solar Panels

The efficiency of a solar panel is among other things depending on the temperature difference between the fluid in the panel and the surroundings. (4) It can be calculated by the empirically developed equation:

$$\eta = \eta_0 - \frac{a_1 \cdot (T_m - T_a)}{G} - \frac{a_2 \cdot (T_m - T_a)^2}{G} \tag{1.3}$$

(5)

Where

η	$=$ Efficiency of the solar panel $[-]$
η_0	$=$ Efficiency when there is no heat loss $[-]$
a_1	$=$ Efficiency factor for specific panel $[\mathrm{W/m^2K}]$
a_2	$=$ Efficiency factor for specific panel $[\mathrm{W/m^2K^2}]$
T_m	$=$ Mean temperature of the fluid in the panel $[°\mathrm{C}]$
T_a	$=$ Temperature of the surroundings $[°\mathrm{C}]$
G	$=$ Solar radiation $[\frac{\mathrm{W}}{\mathrm{m^2}}]$

Given the data from Arcon Sunmark's HT HEATstore 35/10 the efficiency of the solar panel can be shown as a function of the temperature difference between the fluid in the panel and the surroundings $T_m - T_a$. This is shown in Figure 3.

Arcon Sunmark's HT HEATstore 35/10:

$$\eta_0 = 0{,}827$$

$$a_1 = 1{,}18 \ [W/m^2 \cdot K]$$

$$a_2 = 0{,}032 \ [W/m^2 \cdot K^2]$$

$$G = 800 \left[\frac{W}{m^2}\right]$$

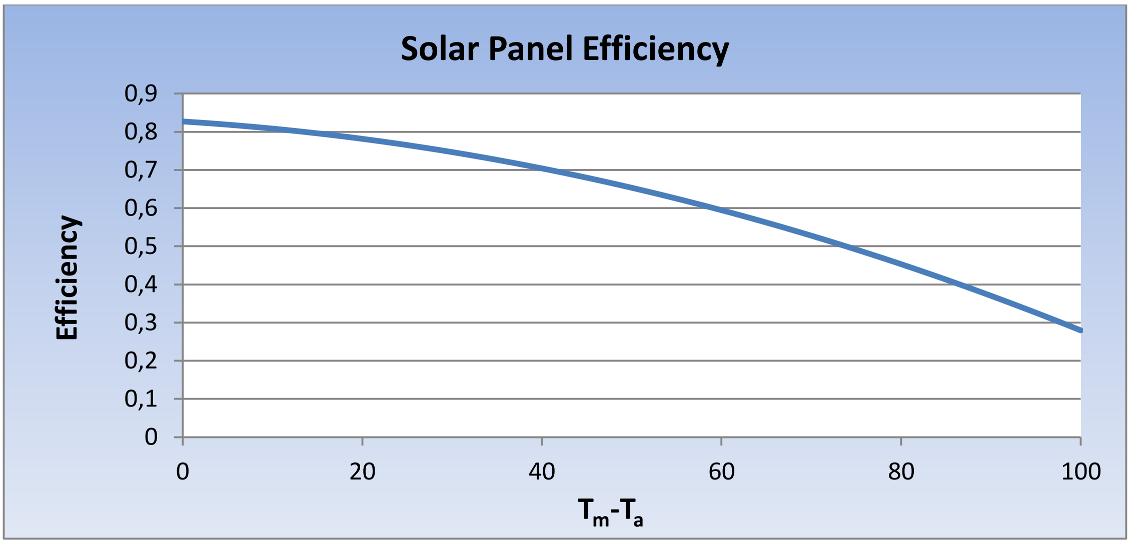

Figure 3: Efficiency of Arcon Sunmark Type HT HEATstore 35/10 as function of the temperature difference (5)

By looking at Figure 3 it is obvious that the efficiency of the solar panel decreases significantly with increasing temperature in the panel. To quantify the effect of lowering the temperature in the district heating system where sun panels are installed some examples where the temperature is lowered are regarded.

Using equation 1.3 page 4 the efficiency at different supply temperatures and a surrounding temperature of 20 °C are calculated. These values are shown in Table 1. The data and calculations can also be found as annex 2 on CD.

Supply Temperature [°C]	Return Temperature [°C]	Mean Temperature [°C]	Outside Temperature [°C]	Efficiency
90	40	65	20	0,680
80	40	60	20	0,704
70	40	55	20	0,726

Table 1: Efficiency of solar panels at different temperatures

As it can be seen in Table 1, the efficiency increases from 0,680 to 0,726 when the supply temperature is lowered from 90 °C to 70 °C . This is an improvement around 7 % on the solar panels efficiency. If it is possible to lower both the supply and return temperature an even higher improvement can be achieved.

To elaborate this, a simulation is run in energyPRO, which is a simulation tool used in the district heating sector. Two identical solar plants with the supply temperature as the only variable have been compared. It can be seen that the annual solar heat production have increased from 2661,0 to 3002, 1 MWh/year equivalent to 12,8 %, when the supply temperature is lowered from 90 °C to 70 °C. A print of the model and its results can be found in Annex 2 page 45.

1.3.2 Electrical Heat Pump

When an electrical heat pump is implemented in a district heating system it is important to design the system in a way where the highest COP (Coefficient Of Performance) is achieved. A way to do that is to minimize the rise of temperature the heat pump must perform.

From the definition of the COP the relation between the temperature rise and the heat pumps COP can be seen:

$$COP = \frac{Produced\ Heat}{Electric\ Consumption} = \frac{Q_{out}}{Q_{out} - Q_{in}} = \frac{1}{1 - \frac{Q_{in}}{Q_{out}}} \quad \mathbf{1.4}$$

(3)

Where:

Q_{out} = Energy supplied to the district heating system

Q_{in} = Energy absorbed from the heat source for the heat pump

W_{in} = Work of the kompressor

For a Carnot heat pump the COP-value can also be expressed by the temperature levels T_{Low} og T_{High}.

$$COP = \frac{1}{1 - \frac{T_{Low}}{T_{High}}} \qquad \mathbf{1.5}$$

(3)

Where:

T_{Low} = Temperature of the coolant at the heat source [K]

T_{High} = Temperature of the coolant at the district heating [K]

To evaluate the effect of lowering the temperature on the hot side of the heat pump equation 1.5 has been used to calculate the COP for different temperatures. The heat source for the heat pump is groundwater at 8 °C. The results of these calculations are shown graphically in Figure 4.

The calculations have all been made with a 3 °C ΔT over the heat exchangers, and a 65 % thermal efficiency of the heat pump. The calculations can be found as Annex 3 on CD.

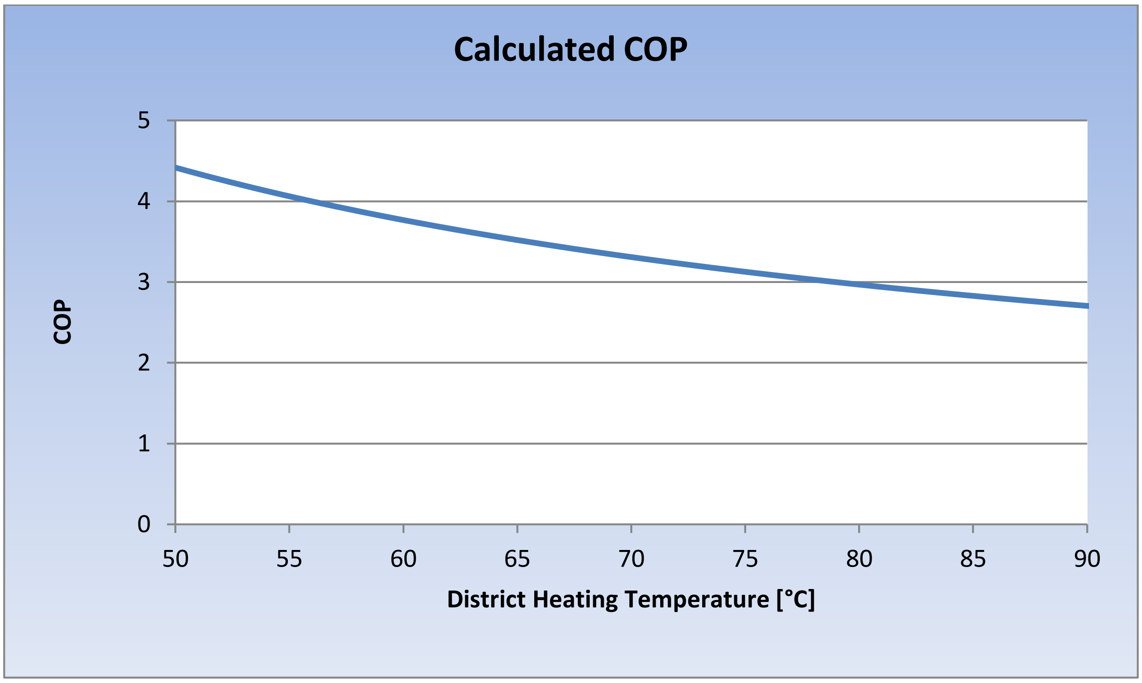

Figure 4: Calculated COP as a function of the District Heating supply temperature for a heat pump with 8 °C groundwater as heat source.

1.3.3 Surplus Heat from Industry

Industrial processes and industrial cooling often results in a surplus of heat. This surplus heat can however be difficult to retrieve for district heating purposes. In many cases the temperature of the heat source is lower than the supply temperature of the district heating system, so a direct heat exchange is therefore not possible. By lowering the district heating supply temperature a bigger part of the surplus heat can be retrieved by direct heat exchange.

In the cases where the temperature of the surplus heat source is too low for a direct heat exchange, the temperature can be boosted with a heat pump. As described in section 1.3.2 the heat pump operates with a significantly better COP if the demanded temperature rise is reduced.

2 Problem Definition

As described in chapter 1 there are many advantages in the district heating system by lowering the temperature. An issue however is; what are the consequences for the consumer if the temperature is lowered? This is what will be investigated in this report. The emphasis will be on single family houses and their household installations such as domestic hot water systems, heating systems, and the building envelope.

The main subject of this project will be the buildings heating system, and the domestic hot water system will therefore only be covered superficially.

The specific problem definition for this report is:

What are the consequences for existing single-family buildings if the District Heating temperature is lowered for low temperature district heating?

- ***What change in heating power does this low temperature cause?***
- ***What changes in the building envelope is necessary to compensate for the loss in heating power?***

These are the main topics which will be investigated in this report.

3 Methodology

This report is based on the curriculum for the education “Bachelor of Engineering in Sustainable Energy with specialization in Thermal Energy “

The knowledge collection for the report has been based on literature and material from previous courses during the study, literature from library, relevant published reports, and government guides and regulation regarding the subject - in printing or online documents. All literature, documents and web sites have been assessed regarding their validity, and if there have been any doubt, the information has been attempted verified by alternative sources.

In addition to that, the experience and knowledge retrieved during the previous internship have been a source of general understanding of the subject.

There will in this project not be made any laboratory work to demonstrate and verify that the calculations and conclusions in the report are correct. The verification of the results has instead been done by comparison of the calculated results with results and software from the district heating sector and related industry.

The simulation and modelling in this report have been performed with the following software:

- **Nasa-Glenn Chemical Equilibrium Program CEA2.**
  Software for simulation and analysis of combustion.

- **EES – Engineering Equation Solver**
  For modelling of the radiator, and other calculations, where fluid properties are needed EES is the preferred software.

- **Microsoft Excel**
  For general mathematical calculations and graphical representations, Excel spreadsheets have been the obvious choice.

- **Be10**
  The software have been developed by Aalborg University, Danish Building and Research, Sbi. In this Bachelor Thesis it is used for calculations of the heat loss and energy consumption in buildings. In Denmark, it is also a regulatory demand to use Be10 for calculation of a buildings energy consumptions to obtain a building permit.

- **Rockwool Energy Design**
  The software is a design tool developed by Rockwool. It is in this report used for calculations of U-values of the different building parts.

4 Household Installations

When a building is supplied with District Heating there a generally two different ways the household installations can be connected to the district heating network – directly or indirectly

- **Directly connected**
  When the installations are directly connected, it is the water from the district heating network, which circulates in the buildings heating system.
- **Indirectly connected**
  With an indirect connection between the building and the district heating system, the household installations and the district heating network is connected through a heat exchanger. In that way the district heating water only circulates in the heat exchanger and the building has its own water to circulate in the heating system.

In this report, it will not be considered whether the heating system is direct or indirect. The differences between the two systems when looking at the household installations are insignificant. The overall heat loss in an indirect system is however higher than in a direct system because the temperature in the district heating system needs to be a few degrees higher than the temperature required by the household installations due to the necessary ΔT of the heat exchanger. This is not relevant for the topics of this report.

4.1 Domestic Hot Water

For the domestic hot water supply, there are two possible design options. There can either be a heat exchanger which heats the water instantly when the hot water taps are opened or there can be a hot water storage tank. The heat exchanger requires a rather high heating power compared to a hot water storage tank, which works like a buffer so the required heating power demand is distributed over time and thereby the heating power demand will be significantly smaller.

An issue that must be observed, when the temperature in the domestic hot water is lowered is the risk of legionella. The legionella bacteria are present in the mains water supply and if it is allowed to propagate it can cause illness, if the bacteria is inhaled with the water steam in the shower. The bacteria's living conditions is optimal around $35 - 46$ °C, but at temperatures above 55 °C the bacteria will die. (6)

According to the Danish Standard DS 439 (7) it is required that the domestic hot water can be heated to 60 °C to avoid any health hazard from development of legionella bacteria. There are however many studies, previous and ongoing, regarding legionella and although the authorities have not accepted it in Denmark, it seems that by using a heat exchanger solution, where the total volume of hot water in exchanger and piping, is less than 3 liters, there is no risk of legionella growth even at temperatures as low as 50 °C. (8)

From these investigations it must be concluded, that the lowest risk of legionella in the domestic hot water when the temperature is lowered is by use of a heat exchanger and to design the system with a total water volume less than 3 liters. If that is fulfilled, it should be safe to lower the district heating temperature to a temperature, depending on the ΔT over the heat exchanger, where the domestic hot water temperature can be maintained at $50\ ^{\circ}\mathrm{C}$.

The high power demand of the heat exchanger compared to a hot water storage tank for domestic hot water, however, means that there is a higher flow demand in the piping. A consequence of that is that the energy required for pumping will be higher and since the pump work is depending on the fluid velocity squared, the increase in the energy demand for pumping can be significant. The dimensioning of the piping system and the district heating supplier's technical terms must also be observed if a change from a domestic hot water storage tank to a heat exchanger is considered.

4.2 Floor Heating

When a house has floor heating installed for space heating the district heating water is mixed in a shunt valve to keep the temperature of the circulating water in the heating system around $30 - 40\ ^{\circ}\mathrm{C}$. The Danish Standard DS 469 - Heating and cooling systems in buildings - defines a design temperature for the floor surface as $27\ ^{\circ}\mathrm{C}$ and $35\ ^{\circ}\mathrm{C}$ maximum at specific areas. (9)

With that given it is obvious, that there will not be any problems regarding the floor heating system if the district heating temperature is lowered.

4.3 Radiators

As oppose to the floor heating system radiators are often designed to work at a significantly higher temperature. The Danish Standard DS 469 (9) defines a supply temperature of $60\ ^{\circ}\mathrm{C}$ and a return temperature of $40\ ^{\circ}\mathrm{C}$. This is however for new installations. Many radiators are designed to work at even higher temperatures e.g. a temperature set of 70/40/20. Meaning supply temperature of $70\ ^{\circ}\mathrm{C}$, return temperature of $40\ ^{\circ}\mathrm{C}$, and a room temperature of $20\ ^{\circ}\mathrm{C}$. Lowering the supply temperature will presumably result in a lower heat output.

The heat output from a radiator involves two heat transfer mechanisms:

- **Radiation**
  Some part of the heat transfer from the radiator is from radiation. The heat radiates from the radiators hot surface and is absorbed by the colder surfaces in the surroundings.
- **Natural convection**
  The hot surface of the radiator heats up the adjacent air. When air, or any gas, is heated it expands and thus its density decreases. Because of the buoyancy force the hot air rises and is replaced by new cold air, which the again is heated and so forth.

An illustration of how the radiator transfers heat can be seen in Figure 5.

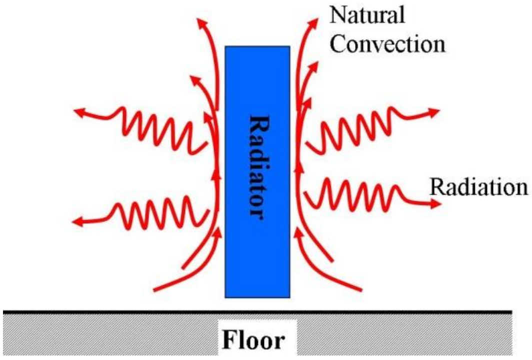

Figure 5: Radiator with heat transfer from both radiation and natural convection

For a conventional panel radiator, the magnitude of the heat transferred from radiation and natural convection are fairly equal. (10) There are several other radiator designs where the distribution between radiation and natural convection is different e.g. convectors, that are designed to be placed hidden, for instance, under a floor grate and therefore naturally only have heat transfer by convection.

In this report the emphasis is on conventional panel radiators. In Figure 6 a drawing of a single panel radiator is shown. This radiator type is also produced with more parallel panels, but it is the single panel radiator, that will be regarded in the calculations in this report.

Figure 6: 3D drawing of the modelled radiator created in Autodesk Inventor

To elaborate the temperature distribution in the radiator an x- and a y-axis is defined. This definition can be seen in Figure 7.

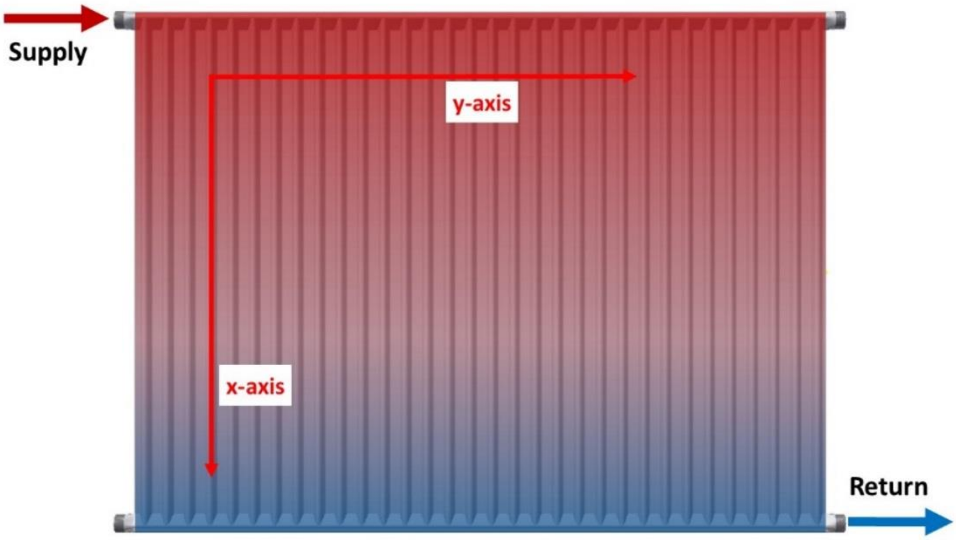

Figure 7: Definition of x and y-axis. The red and blue color shows the temperature distribution in the radiator.

Along the y-axis the temperature is, due to the design of the radiator, to be considered uniform. Along the x- axis the temperature development is as described in Figure 8.

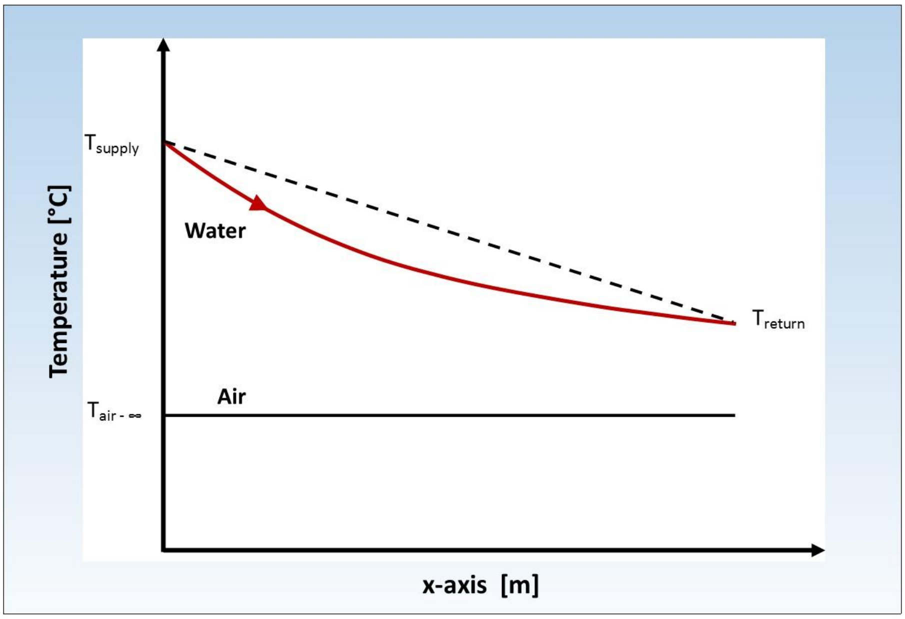

Figure 8: The red line shows the temperature development through the radiator. (10)

As it can be seen, the temperature drop along the x-axis is not linear. The reason for this is that, as the radiator's surface temperature drops, the heat transfer to the surrounding decreases due to the decreasing temperature difference between the radiator and the ambient air.

5 Radiator Calculations

To evaluate what impact a lowering of the district heating temperature has on the power output of a radiator an example will be calculated and modelled.

From a radiator there are, as previously stated, heat transfer from both radiation and natural convection. The radiator can, never the less, basically be regarded as a heat exchanger. In this approach, the two streams in the exchanger is:

- Inside - The water in the radiator, which have defined in- and output temperatures.
- Outside - The air across the radiator, which is assumed to have a uniform temperature. This temperature is considered as the temperature sufficiently far from the radiator surface where it is not affected by the radiator and is denoted T_∞.

The basic equation to calculate heat transfer through a heat exchanger is:

$$\dot{Q} = U \cdot A \cdot \Delta T_{lm} \qquad \textbf{5.1}$$

(3)

Where

U = Overall heat transfer coefficient

A = Surface area of the radiator

ΔT_{lm} = Log mean temperature difference

The overall heat transfer coefficient in a heat exchanger represents the combined heat transfer coefficients of both fluids and the connecting material. The heat transfer from radiation could also be included in the overall heat transfer coefficient, but in this report, the radiation will be calculated separately.

The correlation between them are:

$$\frac{1}{U} = \frac{1}{h_{in}} + \frac{1}{h_{out}} + R_{Wall} \qquad \textbf{5.2}$$

(3)

Where

h_{in} = Heat transfer coefficient of the water inside the radiator

h_{out} = Heat transfer coefficient of the air around the radiator

R_{Wall} = Thermal resistanse of the material in the radiator

When a radiator is regarded the thermal resistance of the material, R_{Wall} can be neglected and because of the significant difference in magnitude between the heat transfer coefficient on the inside and outside of the radiator, $h_{in} \gg h_{out}$, the part regarding h_{out} can be neglected as well.

Hence equation 5.2 can be reduced to:

$$\frac{1}{U} \approx \frac{1}{h_{out}} \qquad \textbf{5.3}$$

Therefore, the equation for calculating the heat transfer by convection from a radiator can be written as:

$$\dot{Q} = h \cdot A \cdot \Delta T_{lm} \qquad \textbf{5.4}$$

Where

h $= h_{out}$, the heat transfer coefficient of the air

A $=$ Surface area of the radiator

ΔT_{lm} $=$ Log mean temperature difference (radiator – air)

The reason why the log mean temperature difference is used instead of the arithmetic mean temperature difference is because of the nonlinear temperature development through the radiator shown in Figure 8 page 16. The arithmetic mean temperature difference would be on the dotted line in the figure – the log mean temperature will be closer to the red line, which describes the actual temperature development.

To determine the total power output of a radiator it is, as previously mentioned, necessary to evaluate both the contribution from radiation and convection.

5.1 Radiation

When the heat transfer from radiation is to be calculated it is necessary to state some basic assumptions:

- Both the radiator and the surroundings are considered as "black bodies", meaning that both the emissivity and absorptivity are ideal.
- The effect of view factors is disregarded
- The surroundings have a uniform surface temperature equal to the air temperature.
- The surface temperature of the radiator is uniform, and based on the log mean temperature difference. ($T_{surf} = \Delta T_{lm} + T_{\infty}$)

Based on these assumptions the heat transfer contribution from radiation can be determined by Stefan Boltzmann's Law:

$$\dot{Q} = \varepsilon \cdot A \cdot \sigma \cdot (T_{surf}^4 - T_{surr}^4) \quad \textbf{5.5}$$

Where

$$\varepsilon \quad = \text{Emissivity factor of the radiator's surface}$$

$$A \quad = \text{Surface area of the radiator}$$

$$\sigma \quad = \text{Stefan Boltzmann's constant}$$

$$T_{surf} \quad = \text{Surface temperature of the radiator}$$

$$T_{surr} \quad = \text{Temperature of the surroundings}$$

(3)

The calculations of this have been performed by EES (Engineering Equation Solver), where a model of a radiator has been built.

5.2 Convection

As previously stated the heat transfer by convection from a radiator can be determined by equation 5.4:

$$\dot{Q} = h \cdot A \cdot \Delta T_{lm} \quad \textbf{5.6}$$

Where

$$h \quad = \text{Heat transfer coefficient of the air}$$

$$A \quad = \text{Surface area of the radiator}$$

$$\Delta T_{lm} \quad = \text{Log mean temperature difference (radiator − air)}$$

To obtain a value for the heat transfer coefficient, h for natural convection, a number of variables have to be calculated initially.

One of these variables needed is the Grashof number. This is a dimensionless parameter, which represents the ratio between the buoyancy force and the viscous force.

The Grashof number can be calculated from the equation:

$$Gr = \frac{g \cdot \beta \cdot (T_{surf} - T_\infty) \cdot L_{ch}^3}{v^2} \qquad \textbf{5.7}$$

(3)

Where

g	$=$ gravitational acceleration
β	$=$ Volume expansion coefficient
T_{surf}	$=$ Surface temperature of the radiator
T_∞	$=$ Temperature of the air sufficiently far away from the radiator
L_{ch}	$=$ Characteristic length
v	$=$ Kinematic viscosity

In equation 5.7 another new variable, β, appears. β is the volume expansion coefficient which defines the net buoyancy force with respect to the temperature difference:

$$\beta = \frac{1}{\rho_{surf}} \cdot \frac{\rho_\infty - \rho_{surf}}{T_\infty - T_{surf}} \qquad \textbf{5.8}$$

(3)

Where

ρ_{surf}	$=$ Density of the air at the surface of the radiator
ρ_∞	$=$ Density of the air sufficiently far away from the radiator
T_{surf}	$=$ Surface temperature of the radiator
T_∞	$=$ Temperature of the air sufficiently far away from the radiator

The volume expansion coefficient is valid for fluids under constant pressure. A large β –value means that a temperature change gives a large change in density of the fluid. This results in a high velocity of the air across the surface and thus a high heat transfer coefficient.

The heat transfer coefficient, which is to be determined can also, be found from the Nusselt number using the equation:

$$Nu = \frac{h \cdot L_{ch}}{k} \qquad \textbf{5.9}$$

Where

h	$=$ Heat transfer coefficient
L_{ch}	$=$ Characteristic length
k	$=$ Thermal conductivity of the air

(3)

To obtain the heat transfer coefficient from equation 5.9 the Nusselt number must be determined. The Nusselt number is depending on the geometry and the orientation of the surface. The calculated radiator is assumed to be a vertical flat surface. There have been developed some empirical correlations for the Nusselt number on different geometries and orientations. For a vertical flat surface there are in "Fundamentals of Thermal-Fluid Scienses" by Yunus A. Cengel et. Al. three equations that describe this correlation. Two of them have a limitation in the range of Rayleigh number, where they are valid. The third is more general, and is valid in the entire range of Rayleigh's number.

The Rayleigh number, Ra, is simply the product of the Grashoff number, Gr, and the Prandtl number, Pr:

$$Ra = Gr \cdot Pr \qquad \textbf{5.10}$$

(3)

To ensure that the calculations of the heat transfer will be valid in the entire range of the Rayleigh number the correlation for the Nusselt number, covering the entire range is applied in this report:

$$Nu = \left\{ 0{,}825 + \frac{0{,}387 Ra^{1/6}}{\left(1 + \left(\frac{0{,}492}{Pr}\right)^{9/16}\right)^{8/27}} \right\}^2 \qquad \textbf{5.11}$$

(3)

These equations, 5.5 to 5.11, are used simultaneously, in combination with the thermodynamic properties of the air to calculate the radiator's heat transfer.

5.3 Modelling

To simulate the performance of the radiator's heating power output a model has been created in EES, where the heat transfer from both radiation and natural convection are calculated.

The radiator model built in EES is based on some general assumptions:

- The radiator is calculated as a vertical flat plate.
- Both the radiator and the surroundings are considered as "black bodies", meaning that both the emissivity and absorptivity are ideal.
- The effect of view factors is disregarded
- The surroundings have a uniform surface temperature equal to the air temperature.
- The surface temperature of the radiator is uniform, and based on the log mean temperature difference. ($T_{surf} = \Delta T_{lm} + T_{\infty}$)
- The heat transfer from the edges is neglected.
- There is no forced convection, only natural.
- The radiator is placed, so there is no disturbance in the airflow to or from the radiator.
- There is radiation and convection from both sides of the radiator.
- The relative humidity of the air is 40 %, and the atmospheric pressure is 101,3 kPa

Based on these assumptions a simulation of the radiator has been made at different supply and outlet temperatures. The heating power output of the radiator has been calculated for the temperature range:

- Supply temperature:
  - $50 - 80\ ^\circ\mathrm{C}$
- Outlet temperature:
  - $25 - 50\ ^\circ\mathrm{C}$

Applying these temperature spans with steps of 1 °C results in 806 calculations. By setting up a parametric table in EES, these calculations are done for all values. The results of the calculations are plotted in the three-dimensional plots in Figure 9. The colors from blue to red show the different heating power output from the radiator. Blue is a very low power output and red is a high power output. The EES model used for the simulation can be seen in Annex 3 and is also available as annex 4 on CD.

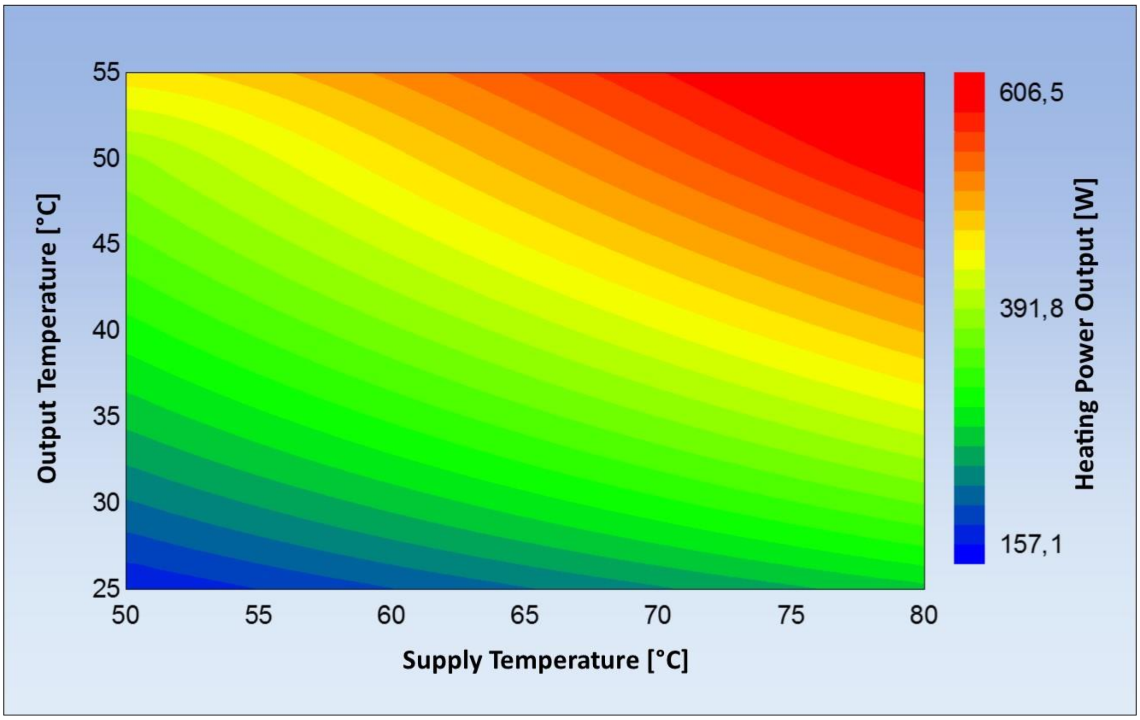

Figure 9: 3D plot of the power output of the radiator with respect to different supply and output temperatures calculated in EES.

From Figure 9 it can be seen that the power output is increasing as the temperature difference between the radiators mean surface temperature and the room's temperature increases. This is not surprising when regarding equation 5.5 and 5.6 page 19, where it is obvious that the heat transfer rate is proportional to the temperature difference.

The heat transfer coefficient in the calculated temperature range is however not constant either. Since the volume expansion coefficient also increases as the temperature increases the heat transfer coefficient also increases with rising temperature difference.

5.4 Evaluation

The data from the simulation in EES are analyzed furthermore in an Excel spreadsheet. Assuming that the design temperature set is 70/40/20, as mentioned in section 4.3 page 14, the heating power output has been evaluated using this temperature set's heating output as index 100. The indexes have been calculated at selected temperature sets, and are shown in Table 2.

Outlet temperature \ Supply temperature	50	55	60	65	70	75	80
25	36	40	45	49	53	58	62
30	49	55	60	66	71	77	82
35	60	67	74	80	86	93	99
40	71	78	86	93	100	107	114
45	80	89	97	105	113	121	129
50	90	99	108	116	125	134	142

Table 2: Calculated indexes at different temperatures for a radiator designed for 70/40/20 °C

From these data it can be seen that if the temperature of the district heating is changed from 70 °C supply temperature and 40 °C return temperature, which the system has been designed for, to 55 °C supply temperature and 30 °C return temperature the heating power output of the radiator drops to 55 % of the designed heating power output. This is a significant loss of heating power, and it can cause severe problems regarding the domestic heating when implementing low temperature district heating.

To verify that the obtained data in Table 2 are valid the calculations for the selected temperature sets have been performed in an Excel spreadsheet used by the radiator manufacturer Ribe Jernindustri A/S. The spreadsheet is a design tool available at their web page. The results of these calculations are shown in Table 3.

Outlet temperature \ Supply temperature	50	55	60	65	70	75	80
25	40	48	56	64	73	82	91
30	48	56	64	73	82	91	100
35	56	64	73	82	91	100	110
40	64	73	82	91	100	110	119
45	73	82	91	100	110	119	129
50	82	91	100	110	119	129	139

Table 3: Calculated indexes for a Rio Panel at different temperatures. Calculations performed by use of the Excel spreadsheet available at the manufacturer's homepage. 70/40/20 °C used as design point. (11)

Even though there are some differences the tendency is clearly the same. If the same change from 70 ℃ supply temperature and 40 ℃ return temperature to 55 ℃ supply temperature and 30 ℃ return temperature is evaluated again, using the commercial data set, the efficiency drops to 56 %, which is quite close to the previously calculated 55 %.

To further investigate the validity of the calculated values in Table 2 two output temperature series have been compared to the commercial values from Ribe Jernindustri A/S. The results of this are graphically displayed in Figure 10. As it can be seen the index curves for the calculated and the commercial values are fairly equal.

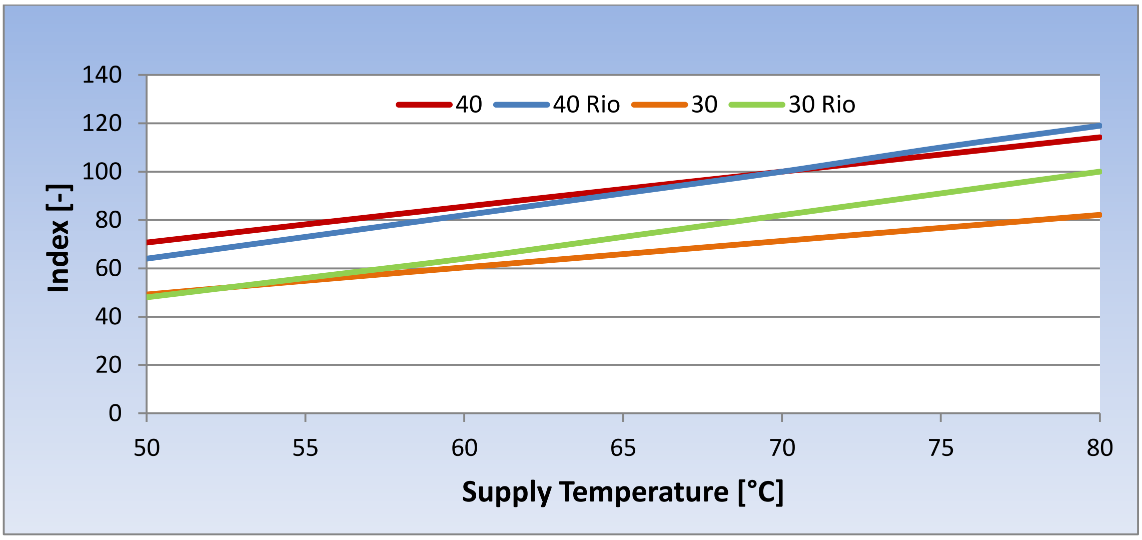

Figure 10: Comparison of two temperature series. Variable supply temperature and 30℃ and 40℃ output temperature repectively

Some different options can compensate for the loss in heating power output of the radiators when the district heating temperature is lowered. One option is to change the radiators, so the surface area is increased to fulfill the power demand. This can however be a rather expensive and comprehensive solution regarding both interior design and economy.

Another solution can be to change the buildings envelope and by that reducing the heating demand to match the power output of the radiators.

6 Building Envelope

The general purpose of the heating system in a building is to keep the temperature in the building on a comfortable level. To do this it must be able to compensate for the heat that is lost through the building envelope to the surroundings. The heat that is lost from a building can basically escape in two ways:

- It can escape through the building materials – this is regarded as transmission loss.
- It can escape along different building parts e.g. foundation and connection between windows and walls – this is regarded as line loss.

There can also be some ventilation losses, but that will not be addressed in this report, since it is regarded as unchanged in all the scenarios for the building envelope.

To evaluate the effect of the reduced heating power output from the radiators a house is being modelled using the software:

- **Rockwool Energy Design** – A design tool created by Rockwool where heat loss and transmission coefficients, U-values, can be calculated. (12)
- **Be10** – Software developed by Aalborg University, Danish Building and Research. The software is required by legislation for calculating energy consumption in buildings. (13)

In these models different scenarios will be analyzed to see if a reasonable change in the buildings envelope can compensate for the loss in heating power, which will be the consequence if the district heating converts to low temperature district heating.

6.1 Model assumptions

The modelled house is a house, which is characteristic for a house built in the sixties or seventies. The reason for this choice of building is that this is the most common single-family building type in Denmark. The distribution of single-family buildings according to their year of construction is shown in Figure 11.

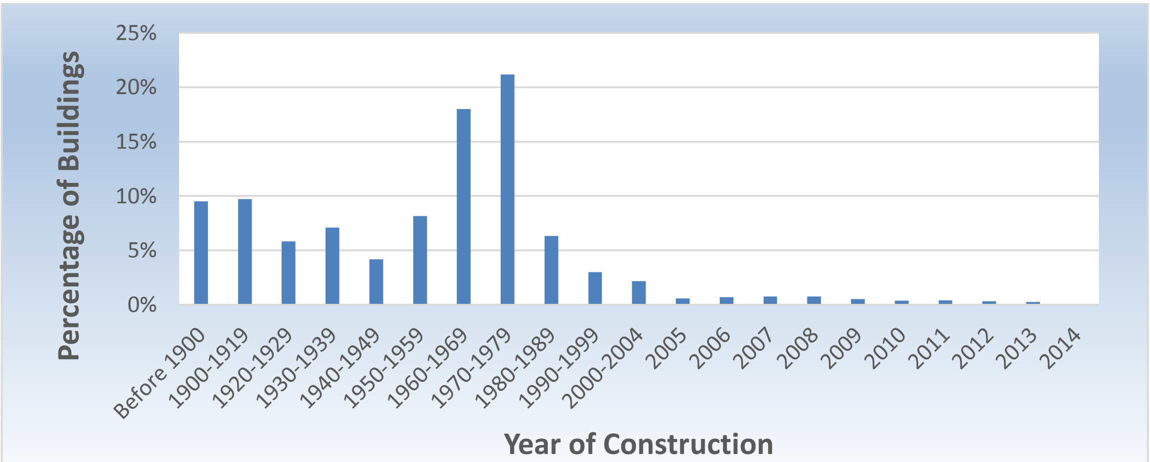

Figure 11: The distribution of single-family houses according to their year of construction. (14)

As it can be seen from the Figure 11 almost 40 % of the existing single-family houses in Denmark are built in the period from 1960 to 1979. It is therefore relevant to focus on the buildings from this specific period. (14)

It is assumed that the only energy renovation that previously has been done on the modelled house is that the windows have been replaced in the nineties, due to an expected lifespan of windows on approximately 30 years. (15) Roof, walls, and insulation are original and no renovation has been done since the house has been built.

From these general considerations the following assumptions regarding the modelled house are:

- Rectangular building.
- 140 m^2 inside the outer walls.
- One-floor building.
- 30° roof angle.
- Outer walls are hollow brick walls.
- Floor is wooden floorboards.
- Insulation level satisfies regulatory demands at year of construction.
- Windows and doors are 1990's standard.

6.2 Heat Loss

To calculate the heat loss of a building component it is necessary to know the transmission coefficient, the U-value, for the given component or material. In general, the U-values for any component can be determined from the equation for the thermal resistance network:

$$\frac{1}{U} = R_{in} \cdot R_{out} \cdot \sum_{i=1}^{n} R_i \qquad \textbf{6.1}$$

(16)

Where

R_{in} = Thermal resistance by convection on the inside

R_{out} = Thermal resistance by convection on the outside

$\sum_{i=1}^{n} R_i$ = Sum of thermal resistance by conduction in the layers

For any practical use of the U-value, it has to be a corrected for e.g. cracks and binders in the insulating layers and potential rain on the outside surface. (16)

When the U-values are known, the heat loss through a component or material can be calculated by Newton´s Law of Cooling:

$$\dot{Q} = U \cdot A \cdot (T_{in} - T_{out}) \quad \textbf{6.2}$$

(3)

Where

U	$=$ Corrected transmission coefficient
A	$=$ Surface area of the component or material
T_{in}	$=$ Dimensioning temperature inside the building
T_{out}	$=$ Dimensioning temperature outside the building

The corrected U-value has been empirically determined for a number of standard materials, and for others it is supplied by the manufacturer of the component or material. The software Rockwool Energy Design also have a built in function to calculate U-values for building components made of standard material. (12) This software is used in this report to determine some of the relevant U-values. The calculations regarding the heat loss are in this report performed by the SBi software Be10, which is the standard program used for documentation to the building authorities in Denmark. (13) This software calculates the heat loss and consumption according to design temperature values specified by legislation.

6.3 Reference scenario

The reference scenario of the modelled house is based on the assumption in section 6.1 page 27. It is assumed that the building comply with the demands in the Building Regulation at the time of construction, hence the regulatory demands can be used to determine the heat loss of the building.

In the Danish building regulation for 1966-1972 the following maximum U-values have been defined. The windows and doors have however, as previously mentioned, been replaced in the 1990’s, and is therefore complying with the Danish Building Regulation of 1985:

	U-values [W/m2K]
Floor over low cellar	0,60
Walls	1,00
Ceiling/Roof	0,45
Windows incl. line loss	2,90
Doors incl. line loss	2,00

Table 4: Maximum U-values according to the Danish Building Regulation of 1966-1972. Windows and doors are according to Danish Building Regulation of 1985 (17) (18)

The specifications of the modelled house in the reference scenario can be seen in Table 5 to Table 9. There have been no regulatory demands to maximum line loss along the foundation in the previous Building Regulations, so the ψ-value for the line loss is from the building Regulation of 2010.

Windows			
Orientation	Quantity [-]	Total Area [m^2]	U-value [W/m^2K]
North	3	5,04	2,90
South	4	6,72	2,90
East	1	0,36	2,90
West	1	1,68	2,90
Total	9	13,8	

Table 5: Number of windows and their corresponding areas and U-values

Glass Doors			
Orientation	Quantity [-]	Total Area [m^2]	U-value [W/m^2K]
South	1	4,4	2,90
Total	1	4,4	

Table 6: Number of glass doors and their corresponding areas and U-values

Doors			
Orientation	Quantity [-]	Total Area [m^2]	U-value [W/m^2K]
North	1	2,64	2,00
East	1	2,09	2,00
Total	1	4,73	

Table 7: Number of doors and their corresponding areas and U-values

Floor, walls, and ceiling		
	Total Area [m^2]	U-value [W/m^2K]
Floor	156,73	0,60
Ceiling	156,73	0,45
Walls	110,97	1,00

Table 8: Area and U-values of the floor, ceiling, and walls.

Line losses		
	Total Length [m]	ψ-value [W/m K]
Foundation	53,56	0,40

Table 9: Line loss along the foundation. There are no specific demands in the Building Regulation of 1966, so the value for the Building Regulation BR10 is used for all calculations.

When modelling the house with these data, a reference for the total heat loss for the building can be estimated. Thus the heating power demand and heat consumption can be determined. The model of the reference house has been created with emphasis on the building envelope and therefore energy consumption for light, domestic hot water etc. have been disregarded. The model can be found as Annex 5 on CD and a printout of the key figures can be found in Annex 4 page 53.

From the model it can be seen that there is a maximum heat demand on 11,2 kW at the design temperatures -12 °C outside and 20 °C inside. To ensure that the heat demand is met in the entire building at all times it is assumed, that the heating system have been designed with a surplus of heating power on 20 %. That means that the heating system has been designed for a heating power output of 13,4 kW.

6.4 Potential Improvements

When the district heating temperature set is changed from the original design set $70/40$ °C to $55/30$ °C it has been proven in section 5 page 17 that the heating output decreases to 55 % of the designed output. This reduced heating output of the heating system can by the new temperature set only provide 7,4 kW heating power – a 6,0 kW reduction

If this significant decrease in heating power has to be compensated by changes in the building envelope, the potential in the different building parts will be evaluated. In Figure 12 the heat loss from the different building parts is shown.

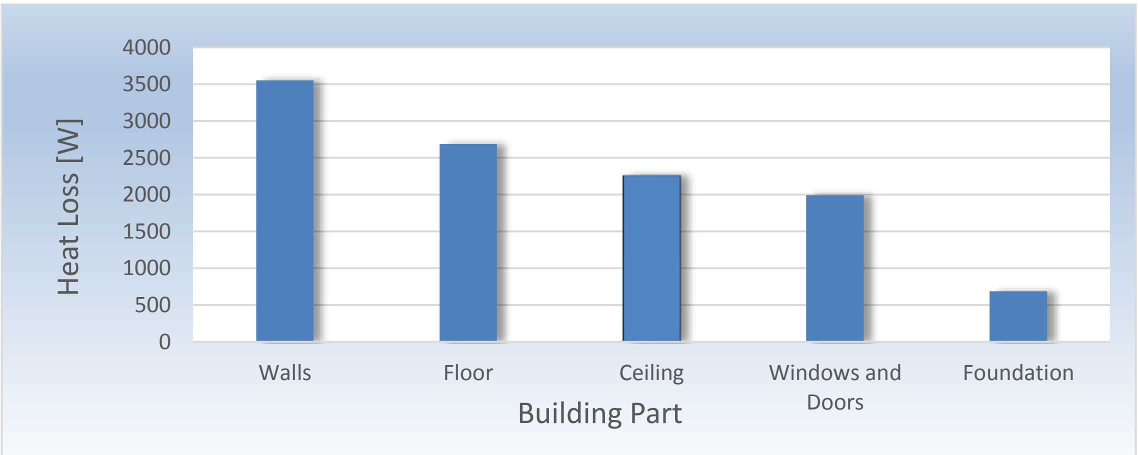

Figure 12: Heat Loss from the different building parts

As it can be seen the largest heat loss from the building is from the walls. The walls on the modelled house are hollow brick walls with no insulation in the cavity. It will be considered to fill the cavity with insulating material such as granulated mineral wool. An illustration of this change is shown in Figure 13.

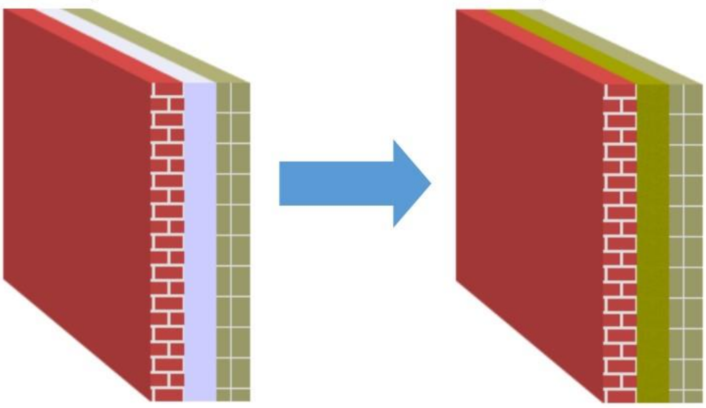

Figure 13: To the left a hollow brick wall without any insulation. To the right a wall insulated with granulated mineral wool. (12)

This insulation of the hollow brick wall will reduce the U-value of the wall from 1,00 W/m²K to 0,33 W/m²K. Modelling the house with the new U-value for the walls shows that the total heat loss by this is reduced by 2,4 kW.

The second largest heat loss from the building is from the floor. The floor has due to the Building Regulation at the time of construction a U-value of 0,60 W/m²K. To reduce this insulation is added as shown in Figure 14.

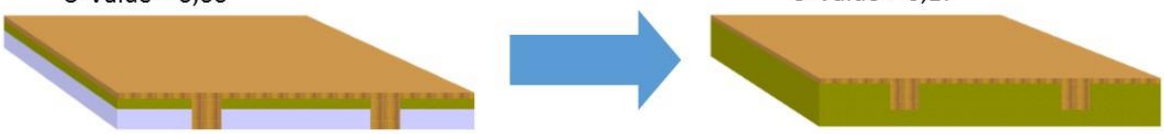

Figure 14: To the left the original floor with 50 mm insulation. To the right there has been added 200 mm insulation. (12)

The insulation of the floor has been improved with 200 mm insulation in addition to the existing insulation. This reduces the U-value from 0,60 W/m²K to 0,17 W/m²K. When modelled in Be10 it shows a reduction of the total heat loss by 1,9 kW.

Even though the improvements in the insulation of the walls and floor have brought the heating power demand below the available heating output of the heating system it is still relevant to investigate the potential in both the floor and the doors and windows to get an overview of the saving potentials in the entire building.

The ceiling of the modelled house has originally a U-value of 0,45 W/m^{2}K. This can be reduced by adding insolation as shown in Figure 15.

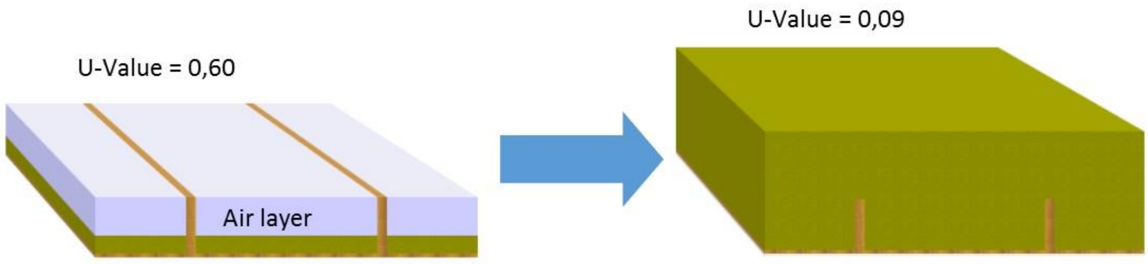

Figure 15: To the left the original ceiling with 50 mm insulation. To the right the ceiling with the added 400 mm granulated mineral wool. (12)

If the ceiling is insulated with 400 mm granulated mineral wool the U-value is lowered from 0,60 to 0,09 W/m^{2}K, which actually satisfies the demand in the 2010 Building Regulation, BR10. When the model is made in Be10 it shows a reduction of the total heat loss by 1,8 kW.

There has in the period since the 1990's where the modelled house's windows and doors are manufactured been an ongoing research and development to improve their U-values. This is also reflected in the Building Regulations. In BR-S 85, which was applicable from 1985-1995, the U-values of doors and windows were 2,0 W/m^{2}K and 2,9 W/m^{2}K respectively. In BR 10 the demands are 1,4 W/m^{2}K for doors and 1,5 W/m^{2}K for doors containing glass. For windows the demands contains calculations that are more specific and based on the windows´ total energy supply to the building. These demands will not be treated in this report, and therefore the U-value demand for doors containing glass will be applied for windows as well.

Modelling the house with new windows, glass doors and doors shows a reduction of the total heat loss by 0,9 kW.

The potential reduction in heat loss caused by the line loss along the foundation of the house will not be considered in this report. This is primarily because there were no demands regarding the line loss along the foundation in the building regulation at the time of construction. It can therefore be difficult to make a valid assumption of the magnitude of the line loss in the reference house. Another reason to disregard it is that the magnitude of the line loss also will be affected by the improvement of the insulation in walls and floor, and it can therefore be difficult to calculate the line loss separately.

The heat losses from the different building parts are listed in Table 10. From these data it can be seen, that the heat loss in the different building parts can be reduced significantly, when the building parts are upgraded to contemporary standards.

Building part	Initial heat loss [kW]	Heat loss after reduction [kW]	Percent reduction [%]
Walls	3551	1172	67
Floor	2688	762	72
Ceiling	2257	451	80
Windows and Doors	1992	1086	45
Foundation	686	686	0
Total	11173	4156	63

Table 10: Reduction in heat loss from the building parts

In Figure 16 the same data is displayed in a bar diagram. When they are regarded the different saving potential can be compared.

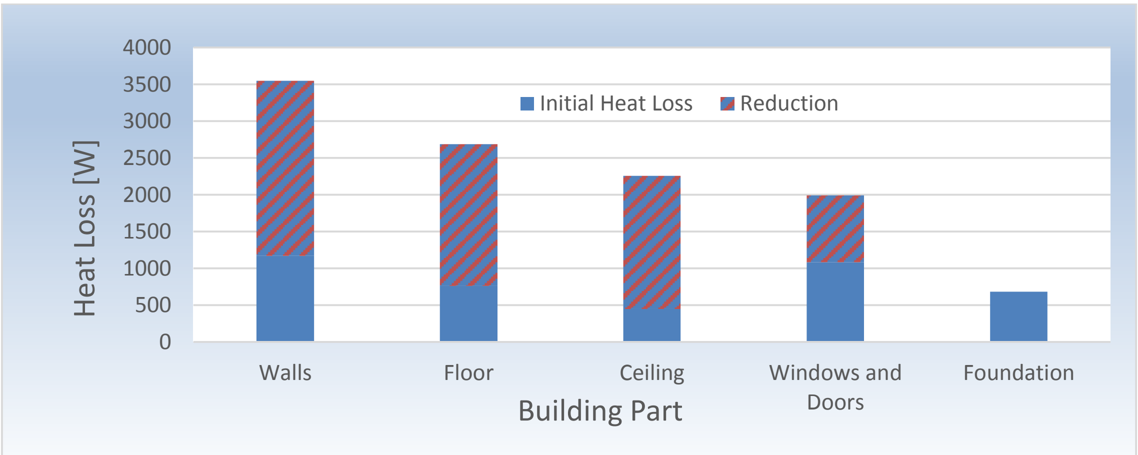

Figure 16: Bar diagram showing the initial heat loss and the reduction.

To evaluate which insulation improvement to choose must depend on some economical calculations, which are not included in this report. It is however obvious, that there are some significant saving potentials in upgrading the insulation standard.

6.5 With no Improvements

The Be10 software is using the data from the Design Reference Year (DRY-data) from the Danish Meteorological Institute (DMI). (19) This data set is created from hourly weather data from Denmark in the period 2001 to 2010. Based on these data a "typical" year for the Danish climate is created. The data set contains information about temperature, wind, relative humidity and solar radiation. The data set is developed for technical dimensioning of e.g. solar plants and buildings.

To elaborate what the consequences would be if nothing were done to compensate for the loss in heating power the DRY- data is used. The data for temperature is used to calculate the heating power demand for the reference house on an hourly basis. From these data a duration curve can be drawn. A duration curve shows the heating power demand with respect to the number of hours where it is required. The duration curve for the reference house is shown in Figure 17.

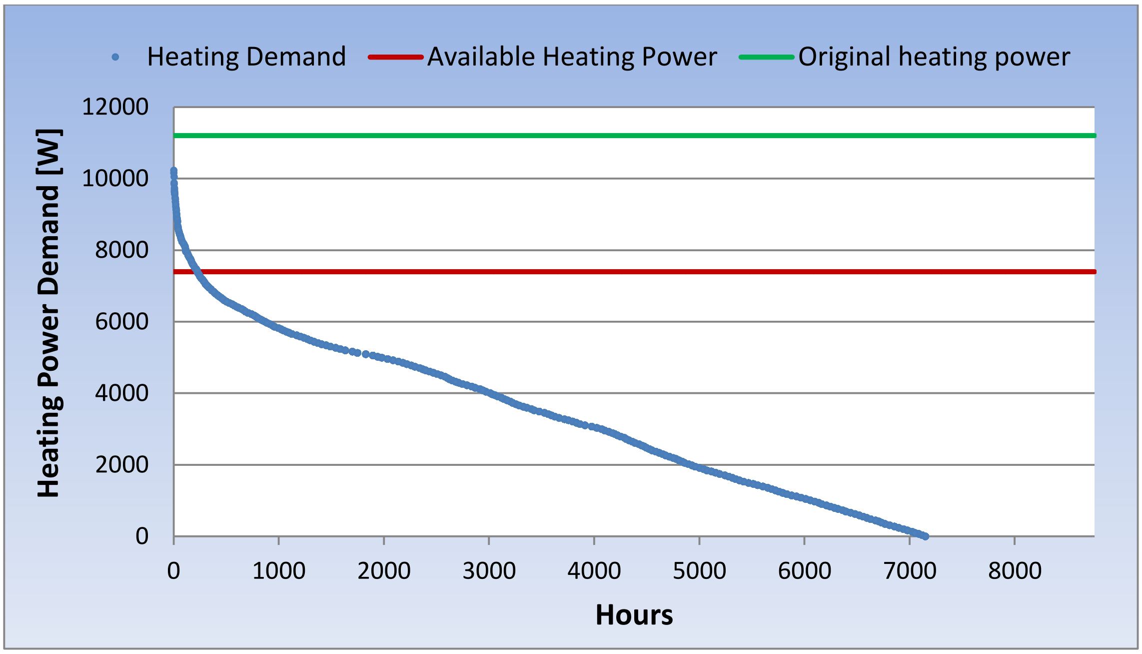

Figure 17: Duration curve for the reference building. The red line shows the capacity of the heating system at low tempereture district heating. The green line is the original capacity of the heating system

As it can be seen there is only a few hours of the year where the heating power demand exceeds the available heating power supply. From the data where Figure 17 is extracted it can be stated, that there is 228 hours where there is insufficient heating power available. The highest deficit in heating power is 2,8 kW and the total deficit energy over these 228 hours represents 178 kWh.

7 Conclusion

Based on the considerations in chapter 4 page 13 the heating power output of the heating system is not affected, when the temperature in the district heating system is lowered, if there is floor heating installed.

For radiator based heating system it is in chapter 5 page 17 calculated that the heating power output of a single panel traditional radiator is reduced to 55 % of its designed output, if the temperature set is changed from $70/40/20\,^{\circ}\mathrm{C}$ to $55/30/20\,^{\circ}\mathrm{C}$. Assuming that the results for the single panel radiator is applicable for any other traditional radiators installed in a building from 1960-1979 a model of a house according to the standard of that time have been created.

The house has been modelled in Be 10 to evaluate where improvements could be made to reduce the heat loss. It was shown in section 6.4 page 31 that the largest heat loss was from the walls followed by the floor, ceiling, and windows and doors. On that background, different scenarios were modelled and evaluated:

- Insulation of the cavity in the walls.
- Additional insulation of the floor.
- Additional insulation of the ceiling.
- Replacing the windows and doors.

The insulation level of the different building parts have been brought up, either to contemporary standard or to the maximum level possible within the buildings limitations. The models have shown that it is possible to reduce the total heat loss from the building by 63 %, if all the insulation improvements are implemented.

The heating demand of the reference building when all the improvements have been implemented is reduced from 11,2 kW to 4,2 kW. This is only 37 % of the original heating demand, so the conclusion is that it is possible to change the buildings envelope to compensate for the loss in heating power caused by the reduction in the district heating temperature.

In section 6.5 page 34 it has been evaluated what the consequence for the heating of the reference house would be, if no improvements of the building would be implemented. It is shown that the decreased district heating temperature only causes problems in 228 hours of the year and the maximum need of auxiliary heating power required is 2,8 kW, with a total energy consumption of 178 kWh.

It must however, be noted that the calculations are based on the theoretical design temperatures, and that consumer behavior or comfort level requirements have not been taken into account.

8 Perspectives

A number of advantages in low temperature district heating has been pointed out in this report. It is in everybody´s interest to grasp those advantages. An implementation of lower temperatures in the district heating system will in the end result in decreasing heating prices for the consumer, which will make district heating even more competitive to individual heating systems. It is crucial to maintain and strengthen the position of district heating in the Danish energy system to ensure and continue the transition to more sustainable energy sources

One of the main obstacles regarding implementation of low temperature district heating has been that, it has been unclear how this change will affect the existing buildings.

In this report, the emphasis has been on buildings constructed in 1960-1979. The reason for this is that almost 40 % of the existing single-family buildings in Denmark were built in that period, and it is expected that buildings, built before that have been energy renovated in some extent.

The work throughout this report has shown that if the temperature in the district heating system is lowered, it only causes problems for the selected building type in a limited period. The problems can be solved by implementing a rather limited improvement in the insulation of the building or alternatively using an auxiliary heat supply until necessary renovation is required anyhow. If the number of consumers who experiences inconvenience caused by the low temperature district heating, the district heating plant instead, can prevent it by increasing the supply temperature in the coldest periods.

The conclusion of this report is nevertheless unambiguous; the temperature in the district heating system can be lowered with limited inconvenience to the consumer.

9 Annex 1

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         NASA-GLENN CHEMICAL EQUILIBRIUM PROGRAM CEA2, MAY 21, 2004
                   BY  BONNIE MCBRIDE AND SANFORD GORDON
      REFS: NASA RP-1311, PART I, 1994 AND NASA RP-1311, PART II, 1996

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   oxid=Air moles=1 t,k=298.15
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     plot mw
 end

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 P,BAR =     1.001300

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          C  1.00000  H  4.00000
 O: Air                1.000000  -0.150977E+02   298.15  0.0000
          N  1.56168  O  0.41959  AR 0.00937  C  0.00032

  SPECIES BEING CONSIDERED IN THIS SYSTEM
 (CONDENSED PHASE MAY HAVE NAME LISTED SEVERAL TIMES)
  LAST thermo.inp UPDATE:    9/09/04

  g 3/98  *Ar              g 7/97  *C               tpis79  *CH
  g 4/02  CH2              g 4/02  CH3              g11/00  CH2OH
  g 7/00  CH3O             g 8/99  CH4              g 7/00  CH3OH
  srd 01  CH3OOH           g 8/99  *CN              g12/99  CNN
  tpis79  *CO              g 9/99  *CO2             tpis91  COOH
  tpis91  *C2              g 6/01  C2H              g 1/91  C2H2,acetylene
  g 5/01  C2H2,vinylidene  g 4/02  CH2CO,ketene     g 3/02  O(CH)2O
  srd 01  HO(CO)2OH        g 7/01  C2H3,vinyl       g 9/00  CH3CN
  g 6/96  CH3CO,acetyl     g 1/00  C2H4             g 8/88  C2H4O,ethylen-o
  g 8/88  CH3CHO,ethanal   g 6/00  CH3COOH          srd 01  OHCH2COOH
  g 7/00  C2H5             g 7/00  C2H6             g 8/88  CH3N2CH3
  g 8/88  C2H5OH           g 7/00  CH3OCH3          srd 01  CH3O2CH3
  g 7/00  CCN              tpis91  CNC              srd 01  OCCN
  tpis79  C2N2             g 8/00  C2O              tpis79  *C3
  n 4/98  C3H3,1-propynl   n 4/98  C3H3,2-propynl   g 2/00  C3H4,allene
  g 1/00  C3H4,propyne     g 5/90  C3H4,cyclo-      g 3/01  C3H5,allyl





                                          1
```

FileEditor:CombustionCH4.out

```
  g 2/00  C3H6,propylene   g 1/00  C3H6,cyclo-      g 6/01  C3H6O,propylox
  g 6/97  C3H6O,acetone    g 1/02  C3H6O,propanal   g 7/01  C3H7,n-propyl
  g 9/85  C3H7,i-propyl    g 2/00  C3H8             g 2/00  C3H8O,1propanol
  g 2/00  C3H8O,2propanol  srd 01  CNCOCN           g 7/88  C3O2
  g tpis  *C4              g 7/01  C4H2,butadiyne   g 8/00  C4H4,1,3-cyclo-
  n10/92  C4H6,butadiene   n10/93  C4H6,1butyne     n10/93  C4H6,2butyne
  g 8/00  C4H6,cyclo-      n 4/88  C4H8,1-butene    n 4/88  C4H8,cis2-buten
  n 4/88  C4H8,tr2-butene  n 4/88  C4H8,isobutene   g 8/00  C4H8,cyclo-
  g10/00  (CH3COOH)2       n10/84  C4H9,n-butyl     n10/84  C4H9,i-butyl
  g 1/93  C4H9,s-butyl     g 1/93  C4H9,t-butyl     g12/00  C4H10,n-butane
  g 8/00  C4H10,isobutane  g 6/01  C4N2             g 8/00  *C5
  g 5/90  C5H6,1,3cyclo-   g 1/93  C5H8,cyclo-      n 4/87  C5H10,1-pentene
  g 2/01  C5H10,cyclo-     n10/84  C5H11,pentyl     g 1/93  C5H11,t-pentyl
  n10/85  C5H12,n-pentane  n10/85  C5H12,i-pentane  n10/85  CH3C(CH3)2CH3
  g 2/93  C6H2             g11/00  C6H5,phenyl      g 8/00  C6H5O,phenoxy
  g 8/00  C6H6             g 8/00  C6H5OH,phenol    g 1/93  C6H10,cyclo-
  n 4/87  C6H12,1-hexene   g 6/90  C6H12,cyclo-     n10/83  C6H13,n-hexyl
  g 6/01  C6H14,n-hexane   g 7/01  C7H7,benzyl      g 1/93  C7H8
  g12/00  C7H8O,cresol-mx  n 4/87  C7H14,1-heptene  n10/83  C7H15,n-heptyl
  n10/85  C7H16,n-heptane  n10/85  C7H16,2-methylh  n 4/89  C8H8,styrene
  n10/86  C8H10,ethylbenz  n 4/87  C8H16,1-octene   n10/83  C8H17,n-octyl
  n 4/85  C8H18,n-octane   n 4/85  C8H18,isooctane  n10/83  C9H19,n-nonyl
  g 3/01  C10H8,naphthale  n10/83  C10H21,n-decyl   g 8/00  C12H9,o-bipheny
  g 8/00  C12H10,biphenyl  g 6/97  *H               g 6/01  HCN
  g 1/01  HCO              tpis89  HCCN             g 6/01  HCCO
  g 6/01  HNC              g 7/00  HNCO             g10/01  HNO
  tpis89  HNO2             g 5/99  HNO3             g 4/02  HO2
  tpis78  *H2              g 5/01  HCHO,formaldehy  g 6/01  HCOOH
  g 8/89  H2O              g 6/99  H2O2             g 6/01  (HCOOH)2
  g 5/97  *N               g 6/01  NCO              g 4/99  *NH
  g 3/01  NH2              tpis89  NH3              tpis89  NH2OH
  tpis89  *NO              g 4/99  NO2              j12/64  NO3
  tpis78  *N2              g 6/01  NCN              g 5/99  N2H2
  tpis89  NH2NO2           g 4/99  N2H4             g 4/99  N2O
  g 4/99  N2O3             tpis89  N2O4             g 4/99  N2O5
  tpis89  N3               g 4/99  N3H              g 5/97  *O
  g 4/02  *OH              tpis89  *O2              g 8/01  O3
  n 4/83  C(gr)            n 4/83  C(gr)            n 4/83  C(gr)
  g11/99  H2O(cr)          g 8/01  H2O(L)           g 8/01  H2O(L)

 O/F =  17.238522

                      EFFECTIVE FUEL     EFFECTIVE OXIDANT       MIXTURE
 ENTHALPY                 h(2)/R              h(1)/R               h0/R
 (KG-MOL)(K)/KG       -0.55928246E+03     -0.52123615E+00     -0.31157558E+02

 KG-FORM.WT./KG            bi(2)               bi(1)                b0i
  *C                   0.62334580E-01      0.11013248E-04      0.34281523E-02
  *H                   0.24933832E+00      0.00000000E+00      0.13670972E-01
  *N                   0.00000000E+00      0.53915890E-01      0.50959735E-01
  *O                   0.00000000E+00      0.14486046E-01      0.13691790E-01
  *Ar                  0.00000000E+00      0.32331996E-03      0.30559264E-03

 POINT ITN      T            C           H           N           O
                  AR
   1    26    2223.980     -21.492     -12.726     -13.830     -17.271
                -26.259
```

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```
        THERMODYNAMIC EQUILIBRIUM COMBUSTION PROPERTIES AT ASSIGNED

                                  PRESSURES

 CASE =

             REACTANT                        MOLES         ENERGY      TEMP
                                                         KJ/KG-MOL       K
 FUEL        CH4                           1.0000000     -74600.000    298.150
 OXIDANT     Air                           1.0000000       -125.530    298.150

 O/F=   17.23852  %FUEL=  5.482900  R,EQ.RATIO= 1.000000  PHI,EQ.RATIO= 1.000000

 THERMODYNAMIC PROPERTIES

 P, BAR              1.0013
 T, K               2223.98
 RHO, KG/CU M     1.4909-1
 H, KJ/KG           -259.06
 U, KJ/KG           -930.67
 G, KJ/KG          -22132.4
 S, KJ/(KG)(K)       9.8352

 M, (1/n)            27.533
 (dLV/dLP)t        -1.00246
 (dLV/dLT)p          1.0739
 Cp, KJ/(KG)(K)      2.1876
 GAMMAs              1.1859
 SON VEL,M/SEC        892.4

 MOLE FRACTIONS

 *Ar                0.00841
 *CO                0.00896
 *CO2               0.08542
 *H                 0.00039
 *H2                0.00358
 H2O                0.18284
 *NO                0.00185
 *N2                0.70061
 *O                 0.00021
 *OH                0.00318
 *O2                0.00454

  * THERMODYNAMIC PROPERTIES FITTED TO 20000.K

    PRODUCTS WHICH WERE CONSIDERED BUT WHOSE MOLE FRACTIONS
    WERE LESS THAN 5.000000E-06 FOR ALL ASSIGNED CONDITIONS

 *C                *CH               CH2               CH3               CH2OH
 CH3O              CH4               CH3OH             CH3OOH            *CN
 CNN               COOH              *C2               C2H               C2H2,acetylene
 C2H2,vinylidene   CH2CO,ketene      O(CH)2O           HO(CO)2OH         C2H3,vinyl
```

3

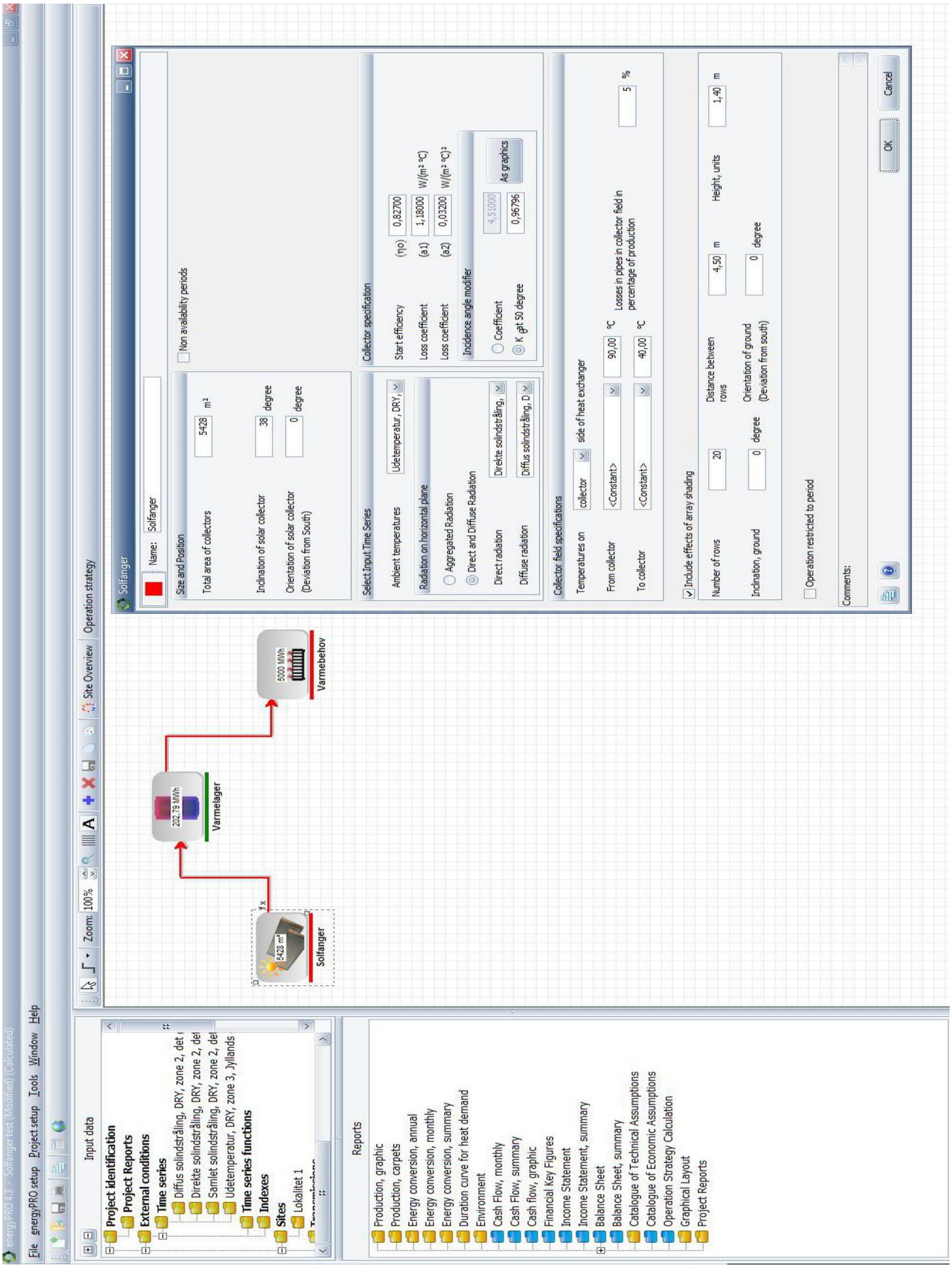
nov. 27, 2015 11:53 AM

FileEditor:CombustionCH4.out

```
CH3CN           CH3CO,acetyl    C2H4            C2H4O,ethylen-o CH3CHO,ethanal
CH3COOH         OHCH2COOH       C2H5            C2H6            CH3N2CH3
C2H5OH          CH3OCH3         CH3O2CH3        CCN             CNC
OCCN            C2N2            C2O             *C3             C3H3,1-propynl
C3H3,2-propynl  C3H4,allene     C3H4,propyne    C3H4,cyclo-     C3H5,allyl
C3H6,propylene  C3H6,cyclo-     C3H6O,propylox  C3H6O,acetone   C3H6O,propanal
C3H7,n-propyl   C3H7,i-propyl   C3H8            C3H8O,1propanol C3H8O,2propanol
CNCOCN          C3O2            *C4             C4H2,butadiyne  C4H4,1,3-cyclo-
C4H6,butadiene  C4H6,1butyne    C4H6,2butyne    C4H6,cyclo-     C4H8,1-butene
C4H8,cis2-buten C4H8,tr2-butene C4H8,isobutene  C4H8,cyclo-     (CH3COOH)2
C4H9,n-butyl    C4H9,i-butyl    C4H9,s-butyl    C4H9,t-butyl    C4H10,n-butane
C4H10,isobutane C4N2            *C5             C5H6,1,3cyclo-  C5H8,cyclo-
C5H10,1-pentene C5H10,cyclo-    C5H11,pentyl    C5H11,t-pentyl  C5H12,n-pentane
C5H12,i-pentane CH3C(CH3)2CH3   C6H2            C6H5,phenyl     C6H5O,phenoxy
C6H6            C6H5OH,phenol   C6H10,cyclo-    C6H12,1-hexene  C6H12,cyclo-
C6H13,n-hexyl   C6H14,n-hexane  C7H7,benzyl     C7H8            C7H8O,cresol-mx
C7H14,1-heptene C7H15,n-heptyl  C7H16,n-heptane C7H16,2-methylh C8H8,styrene
C8H10,ethylbenz C8H16,1-octene  C8H17,n-octyl   C8H18,n-octane  C8H18,isooctane
C9H19,n-nonyl   C10H8,naphthale C10H21,n-decyl  C12H9,o-bipheny C12H10,biphenyl
HCN             HCO             HCCN            HCCO            HNC
HNCO            HNO             HNO2            HNO3            HO2
HCHO,formaldehy HCOOH           H2O2            (HCOOH)2        *N
NCO             *NH             NH2             NH3             NH2OH
NO2             NO3             NCN             N2H2            NH2NO2
N2H4            N2O             N2O3            N2O4            N2O5
N3              N3H             O3              C(gr)           H2O(cr)
H2O(L)
```

4

10 Annex 2


Solfanger
Varmelager
Varmebehov
Input data
Reports
Size and Position
Total area of collectors
Collector specification
Collector field specifications
OK
Cancel

energyPRO 4.3.155

Solfanger test.epp

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Spring 2015

January 2015 to August 2015

Energy conversion, annual

Calculated period: 01-2015 - 12-2015

Heat demands:

Varmebehov	5.000,5 MWh	
Max heat demand	0,6 MW	

Heat productions:

Solfanger	2.661,0 MWh/year	100,0 %

Peak electric production:

Hours of operation:

	Total [h/Year]	Of annual hours
Solfanger	1.875,0	21,4%
Out of total in period	8.760,0	

Turn ons:

Solfanger	321

energyPRO is developed by EMD International A/S, Niels Jernesvej 10, DK-9220 Aalborg Ø, Tlf. +45 96 35 44 44, Fax +45 96 35 44 46, Homepage: www.emd.dk energyPRO 4.3.155

Solfanger test.epp

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Spring 2015

January 2015 to August 2015

Energy conversion, annual

Calculated period: 01-2015 - 12-2015

Heat demands:

Varmebehov	5.000,5 MWh	
Max heat demand	0,6 MW	

Heat productions:

Solfanger	3.002,1 MWh/year	100,0 %

Peak electric production:

Hours of operation:

	Total [h/Year]	Of annual hours
Solfanger	2.100,0	24,0%
Out of total in period	8.760,0	

Turn ons:

Solfanger	311

energyPRO is developed by EMD International A/S, Niels Jernesvej 10, DK-9220 Aalborg Ø, Tlf. +45 96 35 44 44, Fax +45 96 35 44 46, Homepage: www.emd.dk

11 Annex 3

File:Radiator Calculation.EES 30-12-2015 14:52:39 Page 1
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Radiator Calculations

Inputs

$P = 101{,}3$ *Air pressure*

$rh = 0{,}4$ *Relative humidity*

T_{in}=60

Temperature of the water into the radiator

T_{out}=30

Temperature of the water from the radiator

$T_{inf} = 20$ *Temperature of the room*

$H_{rad} = 0{,}555$ *Height of the radiator*

$W_{rad} = 1$ *Width of the radiator*

$A_{rad} = H_{rad} \cdot W_{rad} \cdot 2$ *The surface area of the radiator*

Temperature calculations

$T_m = \frac{T_{in} + T_{out}}{2} - T_{inf}$ *Arithmetic mean temperature difference*

$Arg = \frac{T_{in} - T_{inf}}{T_{out} - T_{inf}}$

$T_{lm} = \frac{T_{in} - T_{out}}{\mathbf{ln}\left[Arg \right]}$ *Log mean mean temperature difference*

$T_{surf} = T_{lm} + T_{inf}$ *Surface temperature of the radiator*

$T_{film} = \frac{T_{surf} + T_{inf}}{2}$ *Temperature of the film used in calculations*

Volume expansion coefficient calculations

$\rho_{surf} = \rho\left[AIRH2O ; P = P ; T = T_{surf} ; R = rh \right]$ *Density of the air near the surface of the radiator*

$\rho_{inf} = \rho\left[AIRH2O ; P = P ; T = T_{inf} ; R = rh \right]$ *Density of the air in the room*

$\rho_{film} = \rho\left[AIRH2O ; P = P ; T = T_{film} ; R = rh \right]$ *Density of the air in the film*

$\beta = -\frac{1}{\rho_{surf}} \cdot \left[\frac{\rho_{inf} - \rho_{surf}}{T_{inf} - T_{surf}} \right]$ *Volume expansion coefficient*

Heat Transfer Coefficient calculations

$L_c = H_{rad}$ *Characteristic length*

$\mu = \mathbf{Visc}\left[AIRH2O ; T = T_{film} ; R = rh ; P = P \right]$ *Dynamic viscosity*

File:Radiator Calculation.EES 30-12-2015 14:52:41 Page 2
EES Ver. 9.937: #2180: For use only by students and faculty at Aalborg University, Denmark

$v = \frac{\mu}{\rho_{film}}$ *kinematic viscosity of the air*

$g = 9{,}8$ *Force of gravity*

$Pr = \mathbf{Pr}\left[AIRH2O\ ; T = T_{film}\ ; R = rh\ ; P = P\right]$ *Prandtl number*

$Gr = \frac{g \cdot \beta \cdot \left[T_{surf} - T_{inf}\right] \cdot L_c^{3}}{v^{2}}$ *Grashoff number*

$Ra = Pr \cdot Gr$ *Rayleigh number*

$k_{air} = \mathbf{k}\left[AIRH2O\ ; P = P\ ; T = T_{film}\ ; R = rh\right]$ *Conduktivity of air*

$$Nus = \left[0{,}825 + \frac{0{,}387 \cdot Ra^{(1/6)}}{\left(1 + \left[\frac{0{,}492}{Pr}\right]^{\left[\frac{9}{16}\right]}\right)^{\left(\frac{8}{27}\right)}}\right]^{2}$$ *Nusselt number for a vertical plate*

$h = \frac{Nus \cdot k_{air}}{L_c}$

$Q_{conv} = h \cdot A_{rad} \cdot \left[T_{surf} - T_{inf}\right]$

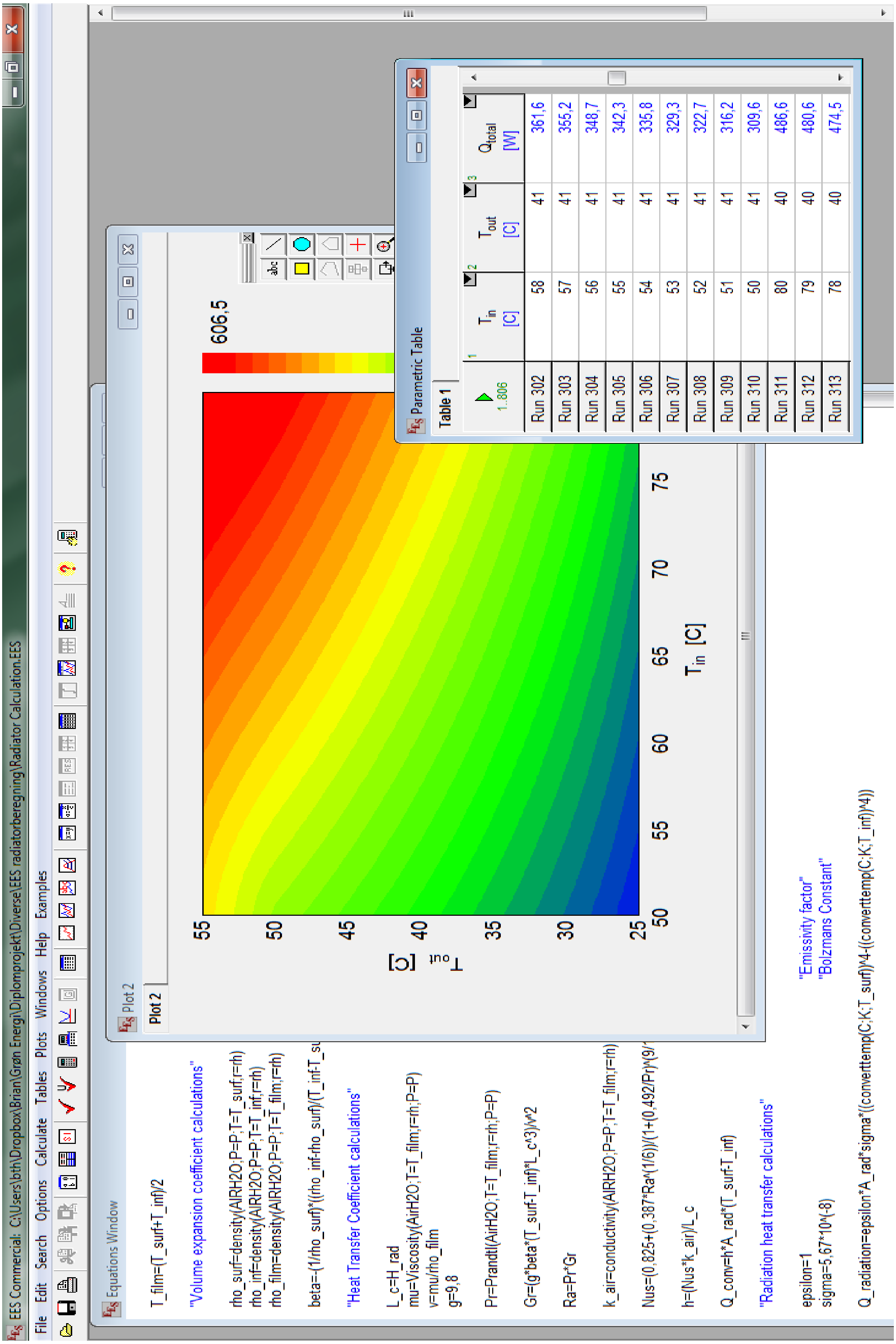
Radiation heat transfer calculations

$\varepsilon = 1$ *Emissivity factor*

$\sigma = 5{,}67 \cdot 10^{-8}$ *Bolzmans Constant*

$Q_{radiation} = \varepsilon \cdot A_{rad} \cdot \sigma \cdot \left[\mathbf{ConvertTemp}\left(C; K; T_{surf}\right)^{4} - \mathbf{ConvertTemp}\left(C; K; T_{inf}\right)^{4}\right]$

$Q_{total} = Q_{conv} + Q_{radiation}$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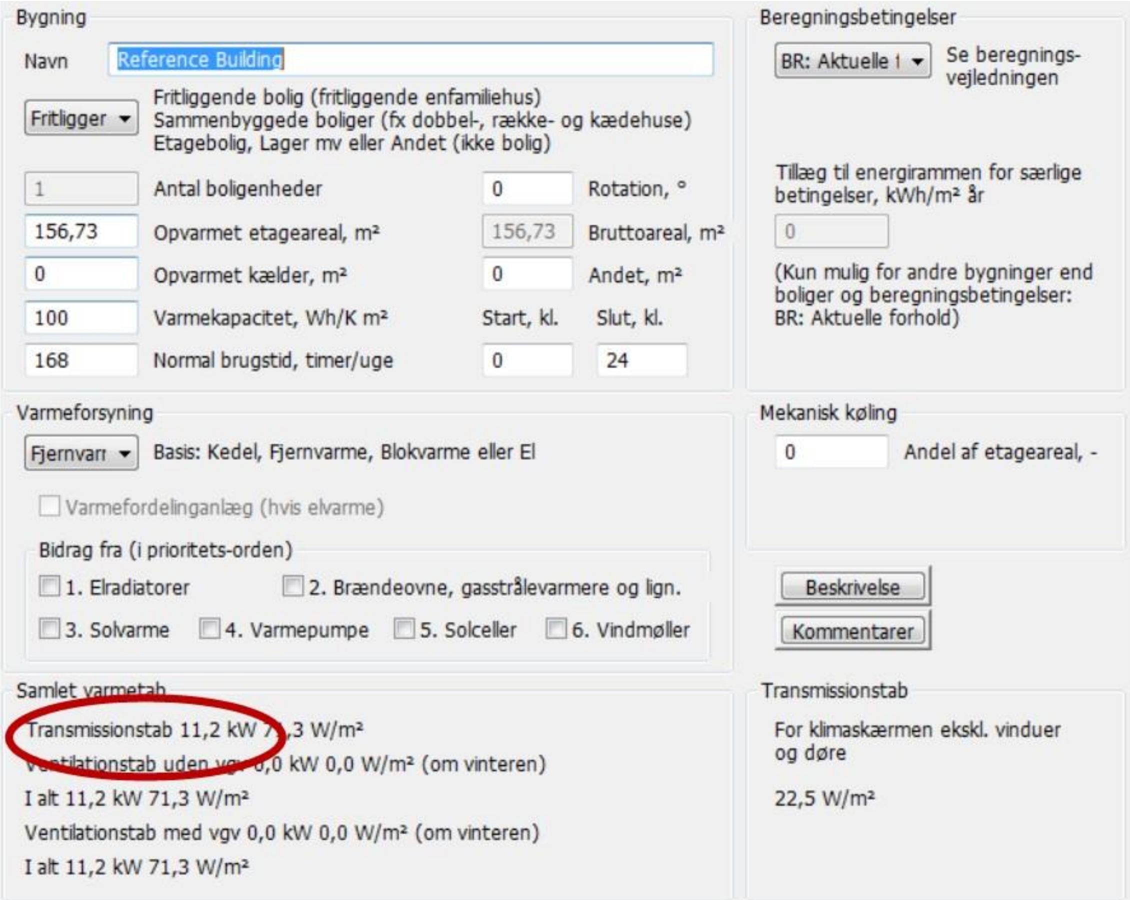
File Edit Search Options Calculate Tables Plots Windows Help Examples
Equations Window
T_film=(T_surf+T_inf)/2
"Volume expansion coefficient calculations"
rho_surf=density(AIRH2O;P=P;T=T_surf;r=rh)
rho_inf=density(AIRH2O;P=P;T=T_inf;r=rh)
rho_film=density(AIRH2O;P=P;T=T_film;r=rh)
beta=-(1/rho_surf)*((rho_inf-rho_surf)/(T_inf-T_su
"Heat Transfer Coefficient calculations"
L_c=H_rad
mu=Viscosity(AirH2O;T=T_film;r=rh;P=P)
v=mu/rho_film
g=9,8
Pr=Prandtl(AirH2O;T=T_film;r=rh;P=P)
Gr=(g*beta*(T_surf-T_inf)*L_c^3)/v^2
Ra=Pr*Gr
k_air=conductivity(AIRH2O;P=P;T=T_film;r=rh)
Nus=(0,825+(0,387*Ra^(1/6))/(1+(0,492/Pr)^(9/
h=(Nus*k_air)/L_c
Q_conv=h*A_rad*(T_surf-T_inf)
"Radiation heat transfer calculations"
epsilon=1 "Emissivity factor"
sigma=5,67*10^(-8) "Bolzmans Constant"
Q_radiation=epsilon*A_rad*sigma*((converttemp(C;K;T_surf))^4-((converttemp(C;K;T_inf))^4))
Plot 2
606,5
T_out [C]
55 50 45 40 35 30 25
T_in [C]
50 55 60 65 70 75
Parametric Table
Table 1
1..806
T_in [C] | T_out [C] | Q_total [W]
Run 302 58 41 361,6
Run 303 57 41 355,2
Run 304 56 41 348,7
Run 305 55 41 342,3
Run 306 54 41 335,8
Run 307 53 41 329,3
Run 308 52 41 322,7
Run 309 51 41 316,2
Run 310 50 41 309,6
Run 311 80 40 486,6
Run 312 79 40 480,6
Run 313 78 40 474,5

12 Annex 4

Printout from Be10 model of the reference house:

Reference scenario without any improved insulation:



Be10 model: Reference Building Dato 4.01.2016 18.59

Reference Building	
Bygningen	
Bygningstype	Fritliggende bolig
Rotation	0,0 deg
Opvarmet bruttoareal	156,7 m²
Areal opvarmet kælder	0,0 m²
Areal eksisterende / anden anvendelse	0,0 m²
Opvarmet bruttoareal inkl. kælderandel	156,7 m²
Varmekapacitet	100,0 Wh/K m²
Normal brugstid	168 timer/uge
Brugstid, start - slut, kl	0 - 24
Beregningsbetingelser	
Beregningsbetingelser	BR: Aktuelle forhold
Tillæg til energirammen	0,0 kWh/m² år
Varmeforsyning og køling	
Grundvarmeforsyning	Fjernvarme
Elradiatorer	Nej
Brændeovne, gasstrålevarmere etc.	Nej
Solvarmeanlæg	Nej
Varmepumper	Nej
Solceller	Nej
Vindmøller	Nej
Mekanisk køling	Nej

Rumtemperaturer, setpunkter	
Opvarmning	20,0 °C
Ønsket	23,0 °C
Naturlig ventilation	24,0 °C
Mekanisk køling	25,0 °C
Opvarmning lager	15,0 °C
Dimensionerende temperaturer	

Rumtemp.	20,0 °C
Udetemp.	-12,0 °C
Rumtemp. lager	15,0 °C

Ydervægge, tage og gulve

Bygningsdel	Areal (m²)	U (W/m²K)	b	Dim.Inde (C)	Dim.Ude (C)
Wall - North	37,7	1,00	1,000	20	-12
Wall - South	34,2	1,00	1,000	20	-12
Wall - East	19,1	1,00	1,000	20	-12
Wall - West	19,9	1,00	1,000	20	-12
Floor	140,0	0,45	0,700		
Ceiling	156,7	0,60	0,700		
Ialt	407,7	-	-	-	-

Fundamenter mv.

Bygningsdel	l (m)	Tab (W/mK)	b	Dim.Inde (C)	Dim.Ude (C)
Foundation	53,6	0,60	1,000	20	-12
Ialt	53,6	-	-	-	-

Vinduer og yderdøre

Bygningsdel	Antal	Orient	Hældn.	Areal (m²)	U (W/m²K)	b	Ff (-)	g (-)	Skygger	Fc (-)	Dim.Inde (C)	Dim.Ude (C)	Ot
Windows - 1,4x1,2 - north	3	0	90,0	1,7	2,90	1,000	0,00	0,63		1,00	20	-12-	0
Windows - 1,4x1,2 - south	4	180	90,0	1,7	2,90	1,000	0,00	0,63		1,00	20	-12	0
Window - 0,6x0,6 - east	1	90	90,0	0,4	2,90	1,000	0,00	0,63		1,00	20	-12	0
Window - 1,4x1,2 - west	1	270	90,0	1,7	2,90	1,000	0,00	0,63		1,00	20	-12	0
Glass Door - 2,0x2,2 - south	1	180	90,0	4,4	2,90	1,000	0,00	0,63		1,00	20	-12	0
Door - 1,2x2,2 - north	1	0	90,0	2,6	2,00	1,000	0,00	0,63		1,00	20	-12	0
Door - 0,95x2,2	1	90	90,0	2,1	2,00	1,000	0,00	0,63		1,00	20	-12	0
Ialt	12	-	-	22,9	-	-	-	-	-	-	-	-	

Skygger

Beskrivelse	Horisont (°)	Udhæng (°)	Venstre (°)	Højre (°)	Vindueshul (%)
Default	15	0	0	0	10

Sommerkomfort	
Gulvareal	0,0 m²
Ventilation, vinter	0,3 l/s m²
Ventilation, sommer, 9-16	0,9 l/s m²
Ventlation, sommer, 17-24	0,9 l/s m²
Ventilation, sommer, 0-8	0,6 l/s m²

Ventilation

Zone	Areal (m²)	Fo, -	qm (l/s m²), Vinter	n vgv (-)	ti (°C)	El-VF	qn (l/s m²), Vinter	qi,n (l/s m²), Vinter	SEL (kJ/m³)	qm,s (l/s m²), Sommer	qn,s (l/s m²), Sommer	qm,n (l/s m²), Nat	qn,n (l/s m²), Nat

Internt varmetilskud

Zone	Areal (m²)	Personer (W/m²)	App. (W/m²)	App,nat (W/m²)
Persons and apps.	157	1,5	3,5	0,0

Belysning

Zone	Areal (m²)	Almen (W/m²)	Almen (W/m²)	Belys. (lux)	DF (%)	Styring (U, M, A, K)	Fo (-)	Arb. (W/m²)	Andet (W/m²)	Stand-by (W/m²)	Nat (W/m²)

Andet elforbrug	
Udebelysning	0,0 W
Særligt apperatur, brugstid	0,0 W
Særligt apperatur, altid i brug	0,0 W

Parkeringskældre mv.

Zone	Areal (m²)	Almen (W/m²)	Almen (W/m²)	Belys. (lux)	DF (%)	Styring (U, M, A, K)	Fo (-)	Arb. (W/m²)	Andet (W/m²)	Stand-by (W/m²)	Nat (W/m²)

Mekanisk køling	
Beskrivelse	Mekanisk køling
Andel af etageareal	0
El-behov	0,00 kWh-el/kWh-køl
Varme-behov	0,00 kWh-varme/kWh-køl

Belastningsfaktor	1,2
Varmekap. faseskift (køling)	0 Wh/m²
Forøgelsesfaktor	1,50
Dokumentation	

Varmefordelingsanlæg					
Opbygning og temperaturer					
Fremløbstemperatur	70,0 °C				
Returløbstemperatur	40,0 °C				
Anlægstype	2-streng		Anlægstype		
Pumper					
Pumpetype	Beskrivelse	Antal	Pnom	Fp	
Varmerør					
Rørstrækninger i fremløb og returløb	l (m)	Tab (W/mK)	b	Udekomp (J/N)	Afb. sommer (J/N)

Varmt brugsvand	
Beskrivelse	Varmt brugsvand
Varmtvandsforbrug, gennemsnit for bygningen	0,0 liter/år pr. m²-etageareal
Varmt brugsvand temperatur	55,0 °C

Vandvarmere	
Elvandvarmer	
Beskrivelse	Elvandvarmer
Andel af VBV i separate el-vandvarmere	0,0
Varmetab fra varmtvandsbeholder	0,0 W/K
Temperaturfaktor for opstillingsrum	1,00
Gasvandvarmer	
Beskrivelse	Gasvandvarmer
Andel af VBV i separate gasvandvarmere	0,0
Varmetab fra varmtvandsbeholder	0,0 W/K
Virkningsgrad	0,5

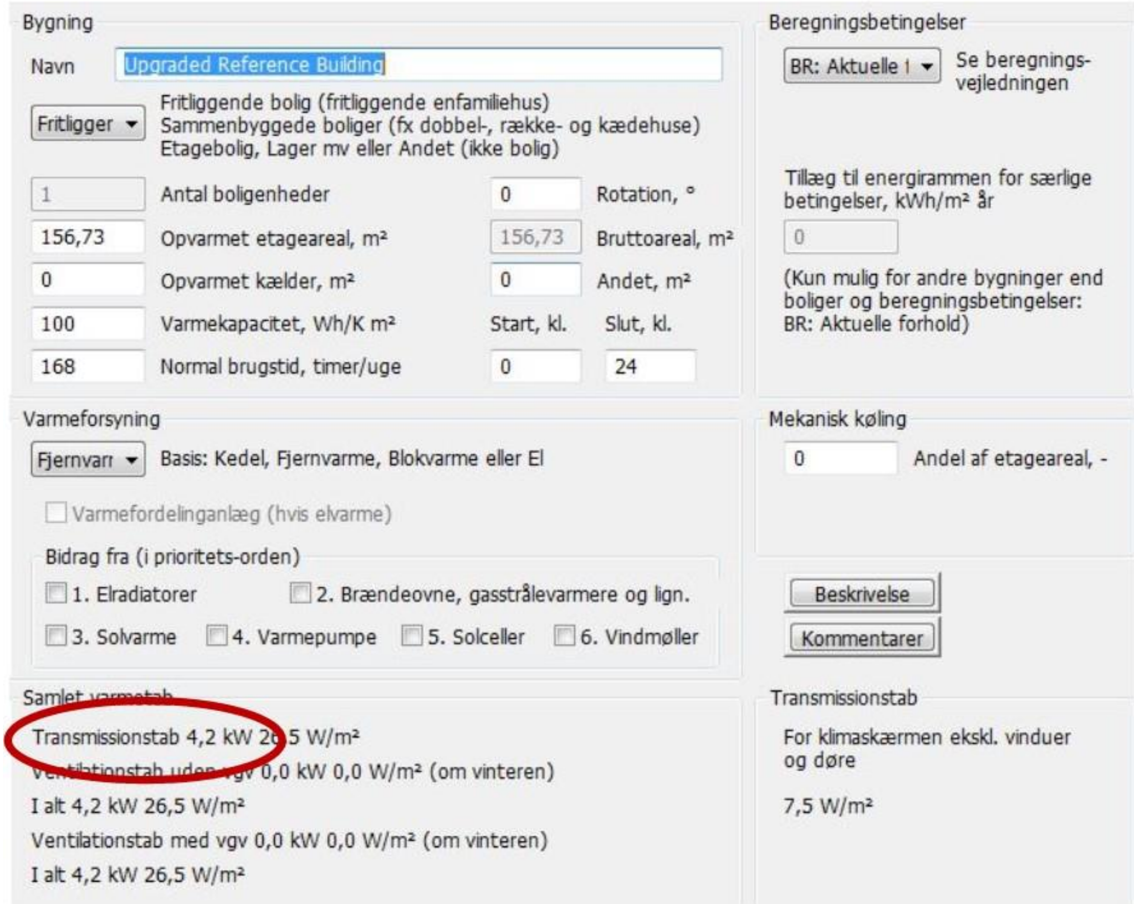
Pilotflamme	50,0 W
Temperaturfaktor for opstillingsrum	1,00

Fjernvarmeveksler	
Beskrivelse	Ny fjernvarmeveksler
Nominel effekt	0,0 kW
Varmetab	0,0 W/K
VBV opvarmning gennem veksler	Nej
Vekslertemperatur, min	60,0 °C
Temperaturfaktor for opstillingsrum	1,00
Automatik, stand-by	5,0 W

13 Annex 5

Printout from Be10 model of the reference house:

Reference scenario with improved insulation:



Be10 model: Annex 5 - Upgraded Reference Building Dato 7.01.2016 11.17

Upgraded Reference Building	
Bygningen	
Bygningstype	Fritliggende bolig
Rotation	0,0 deg
Opvarmet bruttoareal	156,7 m²
Areal opvarmet kælder	0,0 m²
Areal eksisterende / anden anvendelse	0,0 m²
Opvarmet bruttoareal inkl. kælderandel	156,7 m²
Varmekapacitet	100,0 Wh/K m²
Normal brugstid	168 timer/uge
Brugstid, start - slut, kl	0 - 24
Beregningsbetingelser	
Beregningsbetingelser	BR: Aktuelle forhold
Tillæg til energirammen	0,0 kWh/m² år
Varmeforsyning og køling	
Grundvarmeforsyning	Fjernvarme
Elradiatorer	Nej
Brændeovne, gasstrålevarmere etc.	Nej
Solvarmeanlæg	Nej
Varmepumper	Nej
Solceller	Nej
Vindmøller	Nej
Mekanisk køling	Nej

Rumtemperaturer, setpunkter	
Opvarmning	20,0 °C
Ønsket	23,0 °C
Naturlig ventilation	24,0 °C
Mekanisk køling	25,0 °C
Opvarmning lager	15,0 °C
Dimensionerende temperaturer	

Rumtemp.	20,0 °C
Udetemp.	-12,0 °C
Rumtemp. lager	15,0 °C

Ydervægge, tage og gulve					
Bygningsdel	Areal (m²)	U (W/m²K)	b	Dim.Inde (C)	Dim.Ude (C)
Wall - North	37,7	0,33	1,000	20	-12
Wall - South	34,2	0,33	1,000	20	-12
Wall - East	19,2	0,33	1,000	20	-12
Wall - West	19,9	0,33	1,000	20	-12
Floor	140,0	0,17	0,700		
Ceiling	156,7	0,09	0,700		
Ialt	407,7	-	-	-	-

Fundamenter mv.					
Bygningsdel	l (m)	Tab (W/mK)	b	Dim.Inde (C)	Dim.Ude (C)
Foundation	53,6	0,40	1,000	20	-12
Ialt	53,6	-	-	-	-

Vinduer og yderdøre													
Bygningsdel	Antal	Orient	Hældn.	Areal (m²)	U (W/m²K)	b	Ff (-)	g (-)	Skygger	Fc (-)	Dim.Inde (C)	Dim.Ude (C)	Ot
Windows - 1,4x1,2 - north	3	0	90,0	1,7	1,50	1,000	0,00	0,63		1,00	20	-12-	0
Windows - 1,4x1,2 - south	4	180	90,0	1,7	1,50	1,000	0,00	0,63		1,00	20	-12	0
Window - 0,6x0,6 - east	1	90	90,0	0,4	1,50	1,000	0,00	0,63		1,00	20	-12	0
Window - 1,4x1,2 - west	1	270	90,0	1,7	1,50	1,000	0,00	0,63		1,00	20	-12	0
Glass Door - 2,0x2,2 - south	1	180	90,0	4,4	1,50	1,000	0,00	0,63		1,00	20	-12	0
Door - 1,2x2,2 - north	1	0	90,0	2,6	1,40	1,000	0,00	0,63		1,00	20	-12	0
Door - 0,95x2,2	1	90	90,0	2,1	1,40	1,000	0,00	0,63		1,00	20	-12	0
Ialt	12	-	-	22,9	-	-	-	-	-	-	-	-	

Skygger

Beskrivelse	Horisont (°)	Udhæng (°)	Venstre (°)	Højre (°)	Vindueshul (%)
Default	15	0	0	0	10

Sommerkomfort	
Gulvareal	0,0 m²
Ventilation, vinter	0,3 l/s m²
Ventilation, sommer, 9-16	0,9 l/s m²
Ventlation, sommer, 17-24	0,9 l/s m²
Ventilation, sommer, 0-8	0,6 l/s m²

Ventilation

Zone	Areal (m²)	Fo, -	qm (l/s m²), Vinter	n vgv (-)	ti (°C)	El-VF	qn (l/s m²), Vinter	qi,n (l/s m²), Vinter	SEL (kJ/m³)	qm,s (l/s m²), Sommer	qn,s (l/s m²), Sommer	qm,n (l/s m²), Nat	qn,n (l/s m²), Nat

Internt varmetilskud

Zone	Areal (m²)	Personer (W/m²)	App. (W/m²)	App,nat (W/m²)
Persons and apps.	157	1,5	3,5	0,0

Belysning

Zone	Areal (m²)	Almen (W/m²)	Almen (W/m²)	Belys. (lux)	DF (%)	Styring (U, M, A, K)	Fo (-)	Arb. (W/m²)	Andet (W/m²)	Stand-by (W/m²)	Nat (W/m²)

Andet elforbrug	
Udebelysning	0,0 W
Særligt apperatur, brugstid	0,0 W
Særligt apperatur, altid i brug	0,0 W

Parkeringskældre mv.

Zone	Areal (m²)	Almen (W/m²)	Almen (W/m²)	Belys. (lux)	DF (%)	Styring (U, M, A, K)	Fo (-)	Arb. (W/m²)	Andet (W/m²)	Stand-by (W/m²)	Nat (W/m²)

Mekanisk køling	
Beskrivelse	Mekanisk køling
Andel af etageareal	0
El-behov	0,00 kWh-el/kWh-køl
Varme-behov	0,00 kWh-varme/kWh-køl

Belastningsfaktor	1,2
Varmekap. faseskift (køling)	0 Wh/m²
Forøgelsesfaktor	1,50
Dokumentation	

Varmefordelingsanlæg					
Opbygning og temperaturer					
Fremløbstemperatur	70,0 °C				
Returløbstemperatur	40,0 °C				
Anlægstype	2-streng		Anlægstype		
Pumper					
Pumpetype	Beskrivelse	Antal	Pnom	Fp	
Varmerør					
Rørstrækninger i fremløb og returløb	l (m)	Tab (W/mK)	b	Udekomp (J/N)	Afb. sommer (J/N)

Varmt brugsvand	
Beskrivelse	Varmt brugsvand
Varmtvandsforbrug, gennemsnit for bygningen	0,0 liter/år pr. m²-etageareal
Varmt brugsvand temperatur	55,0 °C

Vandvarmere	
Elvandvarmer	
Beskrivelse	Elvandvarmer
Andel af VBV i separate el-vandvarmere	0,0
Varmetab fra varmtvandsbeholder	0,0 W/K
Temperaturfaktor for opstillingsrum	1,00
Gasvandvarmer	
Beskrivelse	Gasvandvarmer
Andel af VBV i separate gasvandvarmere	0,0
Varmetab fra varmtvandsbeholder	0,0 W/K
Virkningsgrad	0,5

Pilotflamme	50,0 W
Temperaturfaktor for opstillingsrum	1,00

Fjernvarmeveksler	
Beskrivelse	Ny fjernvarmeveksler
Nominel effekt	0,0 kW
Varmetab	0,0 W/K
VBV opvarmning gennem veksler	Nej
Vekslertemperatur, min	60,0 °C
Temperaturfaktor for opstillingsrum	1,00
Automatik, stand-by	5,0 W

14 List of Figures

15 List of Tables

16 Bibliografi

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