Improved measurements for better decision on heat recovery solutions with heat pumps

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Abstract

The aim of the study was to show the need for performance documentation, monitoring, and data integration during the lifetime of an energy system to achieve proper decision making. An improved measurement approach for heat pump performance was introduced. This approach was developed by integrating manufacturer and building energy management system data. Direct and indirect measurements were combined into fused measurements. This heat pump estimation approach was tested on a substation where integrated heat pumps supported a building energy supply system. Two approaches were assessed for exhaust air heat recovery: within the air handling unit and by using heat pumps. The results showed that improved measurements were cost-effective and highly reliable in the decision making. The exhaust air heat recovery heat pump was the favorable solution when district heating price was 60% above the electricity price and in the case when electricity was produced by renewable energy sources.

Keywords: heat pump performance, data fusion, direct measurement, indirect measurement, heat recovery, zero emission building (ZEB)

1. Introduction

In order to achieve ambitious targets for energy efficiency and zero energy/emission buildings (ZEB), technologies for heat recovery have been highly recommended. Heat recovery in buildings can involve different strategies, among others, moving heat from one zone to another, integrated solutions, and using exhaust ventilation air for heating. Implementation of these strategies could imply the use of heat pumps. In practice, problems such as oversized systems, faults, and poor integration are common, regardless of good intentions in energy efficiency. In (Moersfelder et al. 2010) it is shown that the comprehensive integration of energy-efficient designs and technologies with renewable energy technologies to achieve net-zero energy buildings has only been sporadically tested at best. To achieve the full potential of energy efficient solutions, it is necessary to perform quality control of the complete energy system. Lifetime commissioning (LTC) has been recognized as a quality control tool for building energy performance through the entire system lifetime (Visier 2004; Djuric and Novakovic 2009; Xiao and Wang 2009). To perform a good building operation and quality control of a given energy system, it is necessary to have information about the building systems and assessment tools.

During the eighties, a solution with an exhaust air heat recovery heat pump for ventilation, space heating and domestic hot water heating was introduced into markets in Sweden and Germany (Fehrm et al. 2002). The exhaust air heat pump (EAHP) uses exhausted ventilation air as heat source. The exhausted building ventilation air has a stable temperature during the year. The exhausted heat in the ventilation air is lifted to a higher temperature by using electricity in the EAHP. This way, the EAHP recovers two to three times more energy than air to air heat recovery within ventilation units, as noted in (Fehrm et al. 2002). However, based on the Norwegian legislation on energy use in buildings, there is a requirement on heat recovery within air handling units (AHU) with an annual efficiency of 70% (Lovdata 2011). The heat recovery within AHU implies heat recovery from the exhausted ventilation air to supply fresh air by using a rotary or a plate heat exchanger. These heat recovery strategies are known and widely implemented in buildings. Therefore, one of the aims of this study was to estimate these two solutions for heat recovery in an office building by using building energy management system (BEMS), rather to compare features of these two solutions.

The research work in Annex 47, Cost-effective Commissioning for Existing and Low Energy Buildings ((IEA) 2009), showed a big need for sensor deployment for the purpose of fault detection and diagnosis, improvement in operation, and performance optimization. Further, the research work in Annex 53, Total Energy Use in Buildings: Analysis and Evaluation Methods ((IEA) 2010), has specified as one of the tasks the development of measurement techniques for the purpose of estimating real energy use in buildings. However, practical experience shows that measurement and monitoring of the building energy performance can be challenging and expensive depending on the monitoring platform, the monitoring platform ownership, the age of the building, when the building monitoring platform was installed, etc. (Djuric et al. 2012). Therefore, in this study, benefit and need for proper estimation by using available BEMS data were studied. One year of detailed measurements were used to estimate the heat pump performance.

The models on heat pump performance measurements were developed based on available literature references. The direct models on condenser load were developed based on (Bourdouxhe et al. 1994; Vujic 2000; Winandy et al. 2002). The indirect models on condenser load were developed based on manufacturer data and relations established in (Lemort et al. 2009). Manufacturer data were found enough reliable for the purpose of the study because the use of them is also recommended by the standard EN 15450 (2007). Developed models were supplied with measurement data from BEMS to produce virtual measurements on heat pump performance. Temperature measurements were used to establish direct measurements on heat pump performance. Since temperature measurements sometimes suffer from noise, outliers, and systematic errors, the use of data fusion techniques can help to estimate real performance data as shown in (Huang et al. 2009). A detailed method for the heat pump performance estimation based on the data fusion method is explained in (Djuric et al. 2011). Heat pump models and the data fusion method were developed on the MATLAB platform (MATLAB 2010).

This article consists of four parts. The first part introduces method for detail documentation and three approaches for heat pump performance estimation. In the second part, a case study and two strategies for exhaust air heat recovery are introduced. The third part gives a comparison of the heat pump performance data obtained using different estimation methods. Finally, the performance data were used for cost-benefit and CO_2 emission analyses.

2. Methods

The use of the Norwegian LTC procedures provided detailed data on heating, ventilation, and air-conditioning (HVAC) systems, while access to the BEMS gave the possibility to monitor system performance data. The LTC procedures imply use of a generic framework on building performance, so that both follow-up and different manipulations of performance data are enabled (Djuric and Novakovic 2010). Data fusion implies the use of techniques that combine data from multiple sources and gather information in order to achieve inferences, which is more efficient and potentially more accurate than if they were achieved by means of a single source.

2.1. Norwegian LTC procedures

The Norwegian LTC procedures were developed based on international commissioning experience and national practical experience. The aim of the procedures is to create a good information system between all the participants during the building lifetime. The focus is on ensuring the owner's project requirements so that the performance quality control is enabled. Practically, the necessary information for fulfilment of the LTC procedures can be collected in different ways. In these procedures, a generic framework on building performance is suggested. This framework describes building performance as a data model (Djuric and Novakovic 2010). This means that a building element can be defined by a few performance data. A building element can consist of a few sub-elements, which can be defined by a few functions. A function of a building element is a building performance data. For example, a building element can be AHU, which consists of few sub-elements like fan, filter, heating and cooling coil. Further, fan functions can be: air flow rate, pressure difference, motor power effect, and the specific fan power (SFP). The function numbers of an element depends on which performance data are necessary for performance estimation. To follow-up desired functions during operation, it is necessary to define measurement of that function. Therefore, measurement of desired performance data should be defined as soon as an element is involved in a building project. This suggested framework on building performance enables generic definition of performance data and their requirements. In that way, different manipulation of performance data is enabled for different purposes (Djuric et al. 2011).

In our study, data from design and manufacturer were organized in the above explained generic framework, so that it was simple to extract necessary data for the heat pump analysis. Further, the design and manufacturer data were compared and combined with the operation data as explained in the next section.

2.2. Data fusion estimation

Measurement and monitoring of the building energy performance can be difficult and challenging due to, for example, technological issues, lack of expertise and poor communication between players during the building lifetime, and building operation and maintenance economy. Technological issues in the building monitoring are caused by chosen monitoring platform and interoperability among equipment and the platform. Currently, different monitoring platforms for BEMS have limitations related to number of measurements and storage of measurement history. Even though many manufactures of equipment claim that monitoring of the equipment is simple and open because the control unit of the equipment is interoperable with the BEMS platform, the communication between the equipment and BEMS can be problematic. Unfortunately, heat pumps and cooling plants are often equipment with limited performance data available to monitor in BEMS. This means that a practical monitoring for operation and maintenance purposes of heat pumps and cooling plants by using BEMS is quite different than the experimental monitoring in manufacturer laboratories. On the other side, the building lifetime includes different players, such as, designers, managers, caretakers, users, owners, etc. Due to this diversity among the players and their expertise, there is a lack of communication between them. For example, it could appear that a designer team had one idea related to the building plant, but something else was installed. This topic will be further discussed in this article. Also, necessary documentation of the installed equipment is not always delivered complete. This lack in communication means that the implemented BEMS do not properly represent the building energy performance data (Djuric et al. 2011). BEMS with many measurements, a large history database, and a good user interface can be expensive. Depending on the building ownership relationships to a building, the interest in good building operation and maintenance can be different. Hence, the building ownership relationships, building owner and user expertise, and energy efficiency awareness can strongly influence the choice of monitoring platform. Regardless of all the above mentioned issues in performance monitoring, in order to properly maintain the building plants, it is necessary to develop tools that could encourage proper building operation and maintenance. Therefore, in this study a data fusion method was suggested to estimate energy performance of the heat pump plant. This method could be also used to compare performance in operation with the manufacturer data for the purpose of documentation and verification. In this study, the data fusion method utilized data from the design phase, manufacturer data, and BEMS data.

In this study, compressor power was measured directly from the BEMS. Condenser and evaporator load were estimated using direct and indirect virtual measurements. Finally, these two measurements were combined into a fused measurement. The indirect estimation utilized the manufacturer technical guide *and* BEMS data.

The direct measurement of the condenser load was obtained using the water temperature difference:

$$\dot{Q}_{cd,d} = \dot{m}_{wc} \cdot c_{pw} \cdot \left(T_{w,out} - T_{w,in}\right), \tag{1}$$

where $\dot{m}_{wc}$ [kg/s] is the water mass flow rate of the condenser; $T_{w,in}$ and $T_{w,out}$ are the inlet and outlet water temperatures of the condenser respectively. In a similar way as in Eq. (1), the direct measurement of the evaporator load was obtained.

Before the indirect model of the heat pump is introduced, the compressor part load is defined as:

$$t = \dot{W}/\dot{W}_{FL}\ . \tag{2}$$

where $\dot{W}$ [kW] is the compressor power and $\dot{W}_{FL}(T_{cd}, T_{ev})$ is the compressor power under full load. It is possible to get $\dot{W}_{FL}$ from manufacturer data based on condensation temperature T_{cd} and evaporation temperature T_{ev}. The indirect measurement of the condenser load can be calculated by using the non-dimensional relation defined in (Lemort et al. 2009):

$$\dot{Q}_{cd,id} = \left[1 - \exp(-\frac{UA_{cd,FL}}{\dot{m}_{wc} \cdot c_{pw}} \cdot t^m) \cdot \dot{Q}_{cd,FL}(T_{cd}, T_{ev})\right] \Big/ \left[1 - \exp(-\frac{UA_{cd,FL}}{\dot{m}_{wc} \cdot c_{pw}})\right], \tag{3}$$

where $\dot{Q}_{cd,FL}(T_{cd}, T_{ev})$ is the condenser load under full load, which is possible to get from manufacturer data. UA [W/m^{2}K] and m (Bourdouxhe et al.) are condenser parameters. The indirect measurement of the evaporator load was calculated in a similar way as in Eq. (3).

After the direct and indirect measurements of the heat pump performance were estimated, the fused measurements were calculated using the combined best estimate method as shown in (Duta and Henry 2005). This way, the fused measurement of the condenser load can be obtained as:

$$Q_{cd,f} = \lambda_{cd,d} \cdot Q_{cd,d} + \lambda_{cd,id} \cdot Q_{cd,id} \tag{4}$$

where coefficients $\lambda_{cd,d}$ and $\lambda_{cd,id}$ are calculated based on the model uncertainties. The fused measurement of the evaporator load was estimated in the same way as the measurement for the condenser load in Eq. (4), by using information on evaporator model uncertainties. The measurement outliers were removed using the Moffat distance as explained in (Duta and Henry 2005; Djuric et al. 2011). In this study, it was assumed that when an outlier was detected, it was replaced by an indirect measurement. This assumption was implemented because it was assumed that indirect measurement based on the electrical signal had higher accuracy than the temperature measurements that could sometimes suffer from noise, outliers, and systematic errors.

3. Case study

The case building is located in Stavanger, Norway, where design outdoor temperature is -9 ^{o}C, while the average annual outdoor temperature is 7.5 ^{o}C. This building has been in use since June 2008 and is rented as an office building. The heated area of the building is 19,623 m^2. The ventilation system consists of three variable air ventilation systems, where the maximal air volume is 90,000 m^3/h for two ventilation systems and 75,000 m^3/h for the third system. In total, both the inlet and exhaust maximal air volume are 255,000 m^3/h. The analyzed substations included a cooling system, free-cooling system, two heat pumps, and heating and ventilation systems, which were connected to district heating as shown in Figure 1. Using the LTC procedures, detailed data from design, construction, and operation phases were collected. In the substation in Figure 1, condenser heat was transferred to a heat exchanger to support the building heating. If there was no necessity for building heating, the condenser heat was either stored or utilized in the outdoor dry cooler. Depending on the weather conditions, the outdoor dry cooler was used to support free-cooling.

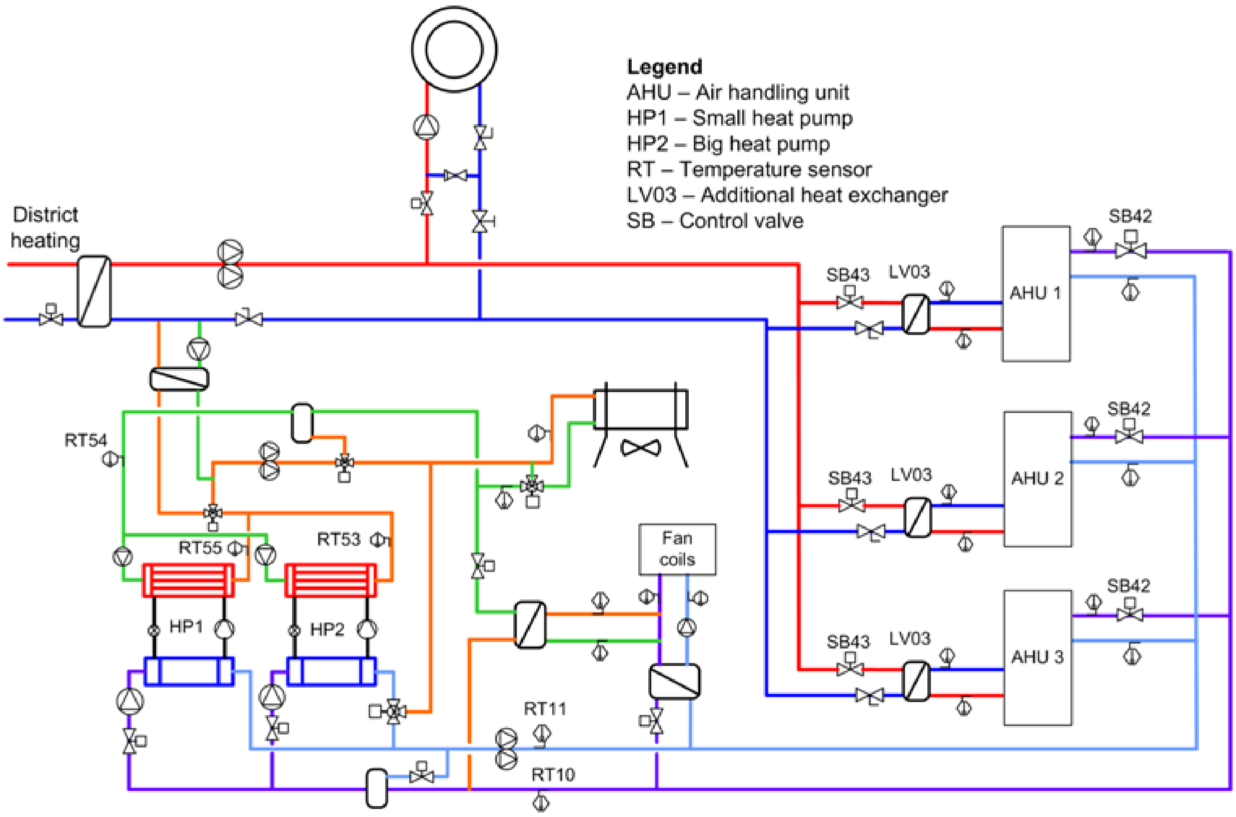

Figure 1. Schematic of the substation with two heat pumps and three AHUs

The heat pumps HP1 and HP2 in Figure 1 are frequency controlled. The installed cooling capacity of HP1 is 420 – 1200 kW, while the cooling capacity of HP2 is 200 – 600 kW. The working fluid in both heat pumps is R-134a. One year of measurements with two minute interval were logged from the BEMS for the purpose of this study. Depending on which type of heat recovery is used, the AHU could have slightly different design, as shown in Figure 2. These different approaches for heat recovery from the exhaust air were named as following: heat recovery within AHU shown in Figure 2.a, and exhaust air heat recovery heat pump in Figure 2.b.

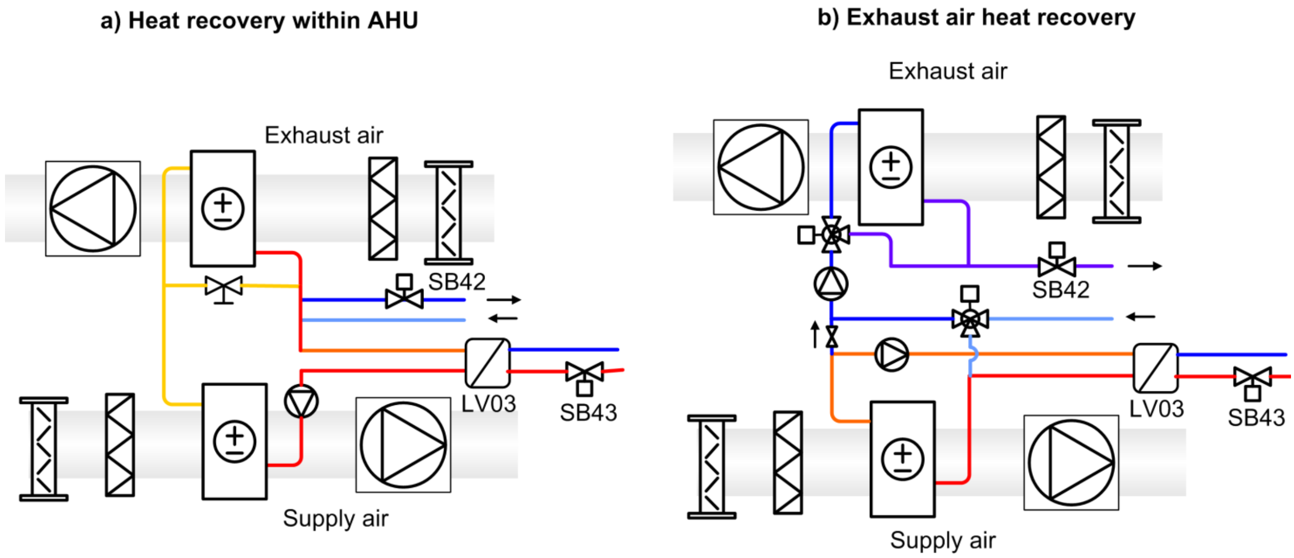

Figure 2. Approaches for heat recovery: a) heat recovery only within the AHU; b) exhaust air heat recovery heat pump

In the analyzed case building, heat recovery from exhaust air was implemented by using two coils; one coil was placed in the exhaust air stream, while the other was placed in the supply air stream. The difference between the analyzed approaches for the exhaust air heat recovery in Figure 2 was how heat from the exhaust air was utilized. In the case of heat recovery within the AHU, Figure 2.a, heat from the exhaust air was directly transferred to the supply air by using the water circuit between the two coils in the heating period. If necessary, additional heat was added via heat exchanger LV03. In the cooling period, heat from the exhaust air was not utilized and cooling was provided to the supply air coil via the control valve SB42 and the by-pass line. In the case of the exhaust air heat recovery heat pump strategy, Figure 2.b, heat from exhaust air stream was transferred to the heat pumps evaporators in Figure 1. In Figure 2.b, the water circuits of the two coils are separated by the three-way valve. In the heating period, supply air was provided by heat via the heat exchanger LV03. In the heating period, the three-ways valve was opened from right to left. In the cooling period, the three-ways valve was opened from right to down, so that cooling water could supply cooling for the supply air stream.

Heat realized from the condensers in Figure 1 is used to support space and ventilation heating in the building. Depending on the approach for heat recovering, amount of condenser heat could be quite different. Based on national legislation for heat recovery within the AHU (Lovdata 2011), the analyzed building has used heat recovery only within the AHU as in Figure 2.a. This implied that supply air was firstly heated up by using exhaust air heat and then, if necessary, additional heat was added via heat exchanger LV03 and controlled by control valve SB43. The installed capacities of the additional heat exchangers were about 922 kW for LV03s supplying the AHU1 and AHU 3 and 765 kW for LV03 supplying the AHU2 in Figure 1. This approach implied that HP1 and HP2 were primarily used to produce cooling for fan-coils in the IT-rooms, while condenser heat was a secondary priority. The heat pumps in Figure 1 were controlled based on the outdoor air temperature. Such control strategy implied that the heat pump HP2 was working all the time, while the heat pump HP1 started when the outdoor temperature was higher than 26^{o}C.

In the design phase, HP1 and HP2 were designed to exclusively utilize heat from the exhaust air. However, due to lack of communication between the design and construction teams, the idea to implement the exhaust air heat recovery heat pump strategy was not realized in operation. The exhaust air heat recovery heat pump strategy would imply that condenser heat could be enough for the building heating demand. The control strategy for the exhaust air heat recovery would imply the control based on the evaporator load. The evaporator load was defined by the amount of heat in the ventilation exhaust air and cooling demand for the IT-rooms. This amount of heat would influence temperature difference on the evaporators of the HP1 and HP2. If the amount of evaporator load was increased, the input signal to the HP2 was increased. After the HP2 would reach the performance maximum, the HP1 would be started. In this study, the above two mentioned approaches for heat recovery were analyzed by using introduced assessment methods.

For the substation in Figure 1, it was possible to perform measurement by using a web-based BEMS. The following performance data related to the heat pumps were logged: the heat pump compressor power, the outlet condenser temperature RT53 and RT55, the inlet condenser temperature RT54, the outlet evaporator temperature RT11 and the inlet evaporator temperature RT10, and outdoor air temperature. The following performance data related to the AHUs were logged: supply and exhaust air amount, the supply air temperature, the exhaust air temperature, and the outdoor air temperature. The outdoor temperature was a common performance datum for both the heat pumps and the AHUs. Since the web-based system was using for data logging and only assess was allowed, it was not possible to make any changes in sensor positions. During our research, the operation and maintenance personal in the analyzed building was helpful, but change in the plant was not possible.

4. Results

The results showed a difference in operation and heat pump performance data when the two analyzed approaches for heat recovery were used. In addition, influence of different measurements on heat pump performance is presented.

4.1. Operation analysis

Before the performance estimation is introduced, differences in operation of the two analyzed strategies for exhaust air heat recovery are presented. The idea was to show briefly the main operation features of the exhaust air heat recovery strategies. In addition, operation issues due to that the substation in Figure 1 was designed for exhaust air heat recovery heat pump rather than for heat recovery within AHU are briefly discussed her. The main differences between the two analyzed exhaust air heat recovery strategies can be explained as: difference in ventilation load and difference in heat pump mode.

Ventilation load is heating or cooling energy necessary to condition outdoor air to the desired supply air temperature. In this case, the ventilation load was transferred from water to the supply air via coil in AHU, as shown in Figure 2 with the coils in the supply air stream. These coils were energy supplied via the additional heat exchangers LV03 as shown in Figure 1. For the purpose of this study and to show differences in load, the total ventilation load for the three coils operating in the three AHUs is shown in Figure 3.

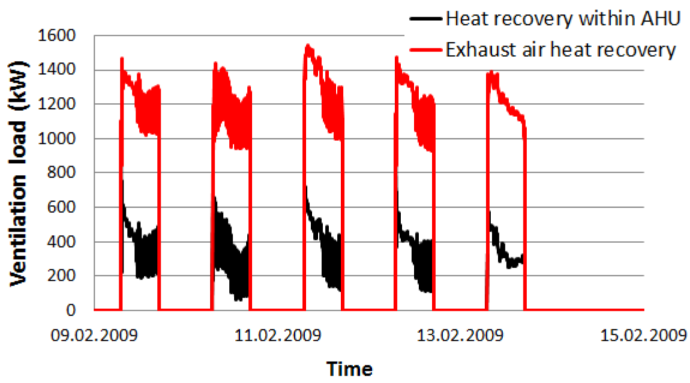

Figure 3. Ventilation load for different heat recovery strategies

In Figure 3 it is shown that in the case of the heat recovery within the AHUs, a smaller amount of additional heat (black line in Figure 3) was necessary for ventilation compared to the exhaust air heat recovery heat pump strategy (red line in Figure 3). Compared to the installed capacity of the additional heat exchangers LV03, in total 2609 kW, it could be concluded that the additional heat exchangers LV03 in Figure 1 were poorly utilized for the purpose of exhaust heat recovery within

AHU. A summary of the operation time of control valves SB43 (see Figure 1) in February and June is given in Table 1. Since the ventilation heat load was low in the case of heat recovery within AHUs as shown in Figure 3, the average valve position was quite low. The low valve positions and short operation period indicated that both the valves SB43, and the additional heat exchangers LV03, were oversized. This indicated that the substation in Figure 1 was designed for the exhaust air heat recovery heat pump, rather than for heat recovery within the AHUs.

Table 1. Operation time and valve position of ventilation control valves

Valve	Second week in February 2009			Second week in Jun 2009		
	Position (%)	Operation time (hour)	Operation time with more than 95 % (%)	Position (%)	Operation time (hour)	Operation time with more than 95 % (%)
AHU1_SB43	12.97	45.5	20	0.01	5.5	0
AHU2_SB43	13.07	45.5	14	0.03	0.5	0
AHU3_SB43	14.06	45.5	22	0	0	0

Related to the operation of the heat pumps HP1 and HP2 in Figure 1, in the case of the heat recovery within the AHUs, the evaporators were primary utilized to produce cooling for fan-coils in the IT-rooms. Therefore, the evaporator load was only dependent on cooling demand. In the case of the exhaust air heat recovery heat pump strategy, heat in 255,000 m^3/h of exhaust air would be utilized as evaporator load. Since the heat recovery within the AHUs strategy was currently utilized and evaporator load was lower, most of the year HP2 was working and HP1 was shut down. For example, in the case of the heat recovery within the AHUs, both heat pumps were in use only 116 hours during the year. This situation could indicate that the heat pumps were oversized for this approach of heat recovery. In the case of the exhaust air heat recovery heat pump when the heat in 255,000 m^3/h of exhaust air was utilized, the installed heat pumps would be better utilized. The heat pumps mode for these two heat recovery approaches is given in Figure 4. In Figure 4, 1 means that only HP2 is in use, 2 means that only HP1 is in use, and 3 means that both HP1 and HP2 are in use.

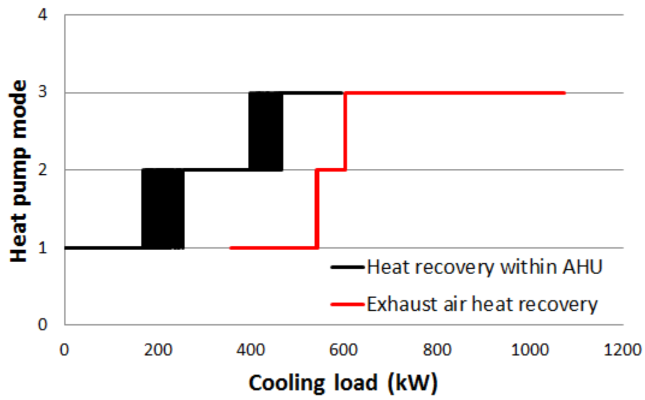

Figure 4. Heat pump mode

In the developed model for the exhaust air heat recovery heat pump, the heat pumps were controlled based on the evaporator load. Currently, in the analyzed building, the heat pumps HP1 and HP2 were controlled based on the outdoor temperature, as explained before. The heat pump control based on the outdoor temperature implied that the heat pumps started even when they do not have enough load as shown in Figure 4 (the black line). In the case of the exhaust air heat recovery heat pump strategy and heat pump control based on the evaporator load, the heat pumps were better utilized as shown in Figure 4 (the red line). In the case of the exhaust air heat recovery heat pump, both heat pumps would work about 4400 hours per year. All this indicated that the substation in Figure 1 was designed for the exhaust air heat recovery heat pump strategy rather than for heat recovery only within the AHUs.

4.2. Performance estimation

Two performance estimation approaches were suggested in this study: direct and fused measurements. Features of these two estimation approaches are presented in this section. To estimate the condenser load by using the direct measurement, the water temperature difference on the condenser was used as shown in Eq. (1). In the case of the substation in Figure 1, the relevant water temperatures of the condenser are marked as RT53 and RT54 for the outlet and inlet water temperatures, respectively. Daily variations of the condenser water temperatures are shown in Figure 5.

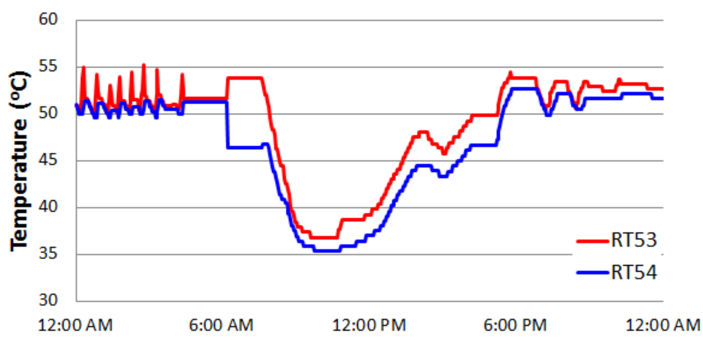

Figure 5. Condenser inlet and outlet water temperatures

The results in Figure 5 show a typical daily profile of the condenser temperatures. The data were logged March 2nd, 2009. In Figure 5, it is possible to notice that the condenser temperature difference was quite low in the morning and in the afternoon. In the morning, high oscillations often appeared in the outlet condenser temperature. During working hours from 6 AM until 6 PM, the condenser temperature difference was stable and had values between 2 to 7 K. Finally, if the condenser water temperatures, as shown in Figure 5, were used for the direct estimation of the condenser load, the results would be as displayed in the upper part of Figure 6 (the blue line). To understand the reasons for the high oscillations of the condenser temperature outside working hours, the compressor power was analyzed. For comparison, the compressor power is displayed in the upper part of Figure 6 (the purple line). To recall, the compressor power was directly measured from BEMS. Further, in the upper part of Figure 6, the indirect and fused measurements are also displayed. To analyze the quality of each estimation method, uncertainty associated with the condenser measurements were calculated and is displayed in the lower part of Figure 6.

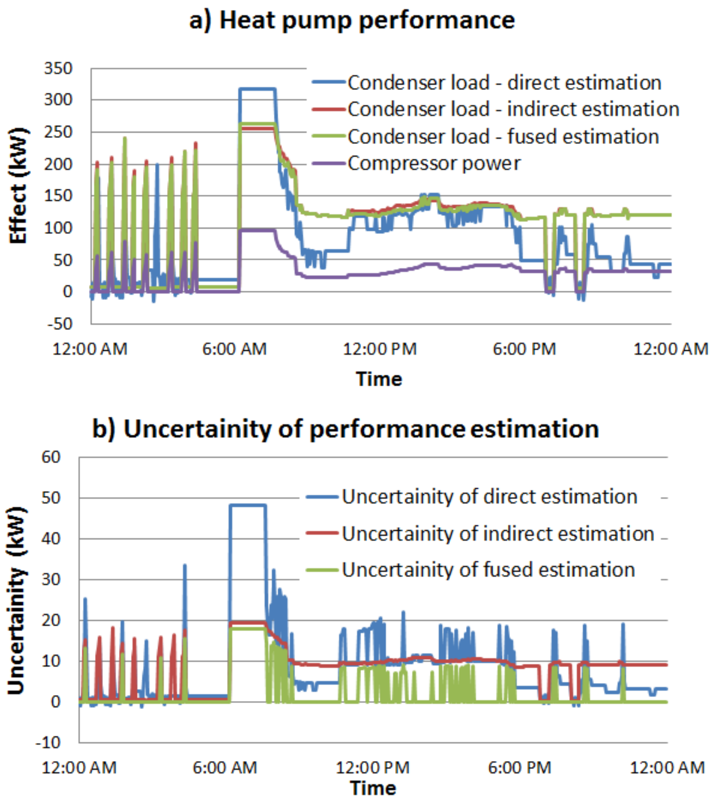

Figure 6. Heat pump performance estimation. a) Performance estimation based on different estimation approaches. b) Associated uncertainty of the heat pump performance estimation

In the upper part of Figure 6 it is shown that the heat pump compressor was frequently switching between ON and OFF in the morning. For example, the compressor was OFF for 10 to 30 minutes and then ON for 5 minutes. In the same period, the condenser outlet temperature was oscillating as shown in Figure 5. By analyzing results for the condenser temperatures in Figure 5 and compressor power in the upper part of Figure 6, it was possible to note the following:

- when the compressor was switched ON, the outlet water temperature was increased and was higher than the inlet condenser temperature, as expected;

- when the compressor was switched OFF, the outlet water temperature was slowly decreasing and reaching the value of the inlet water temperature.

Due to the ON/OFF switching and the dynamic of the heat transfer in the condenser, it could happen that the outlet water temperature was lower than the inlet water temperature. When such oscillating temperature measurements were used for the direct condenser load estimation, it could happen that the condenser load had certain higher value, while the compressor was OFF. This can be noted at about 3 AM in the upper part of Figure 6. Further, in the case when the inlet water temperature was higher than the outlet temperature, the direct measurement of the condenser load appeared negative. In the period from 6 AM until 6 PM, when the compressor worked continuously, the condenser temperature difference was quite stable. Hence, all the analyzed measurements, the direct, indirect, and fused, had similar values, as shown in the upper part of Figure 6. Uncertainty associated with the different measurements of the condenser load showed that the direct estimation had the highest uncertainty during transients, like before switching OFF or ON, as displayed in the lower part of Figure 6. Finally, comparing results in Figures 5 and 6, it can be concluded that dynamics in the heat pump operation caused a high uncertainty in the direct measurement of the condenser load. If the heat pump would work more stable and seldom switching between ON and OFF, the direct and indirect measurements would have similar values. Finally, the fused measurement would equally combine these two measurements. The fused measurements were obtained by combining the direct and indirect measurements as explained in Section 2.2. Depending on the uncertainty associated with the measurements, the contributions of the direct and indirect measurements to the fused measurement could be different. Actually, the data fusion coefficients, $\lambda_{cd,d}$ for the direct measurement and $\lambda_{cd,id}$ for the indirect measurement could have different values. The data fusion coefficients for the condenser load are shown in Figure 7. The values in Figure 7 are related to the same period as the data in Figures 5 and 6. These coefficients are displayed as the summarized area in Figure 7, to easily identify weighting of the direct and indirect measurements in the fused measurement.

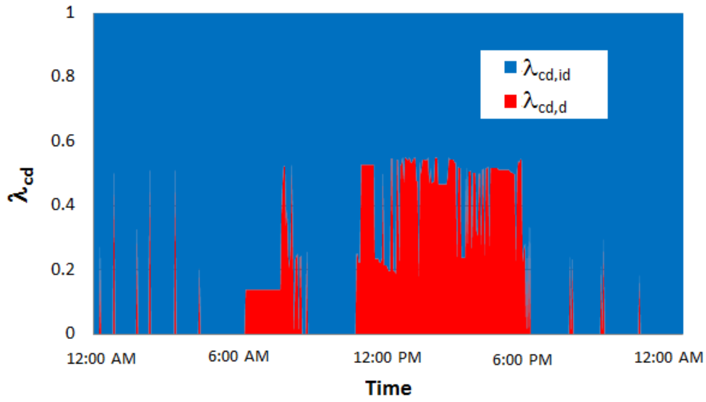

Figure 7. Data fusion coefficients for condenser load estimation

The results in Figure 7 show that when the direct and indirect measurements had similar values, the data fusion coefficients had approximately similar contributions to the fused measurement. This mostly occurred during stable operation of the heat pump. In the period when the heat pump was switching ON/OFF, the indirect measurement would mostly contribute to the fused measurement. As mentioned before, in the dynamic operation of the heat pump, the direct measurement had high uncertainty and many outliers as can be noticed in the upper part of Figure 6. Due to the procedure for removing the measurement outliers explained in (Duta and Henry 2005) and (Djuric et al. 2011), when an outlier was detected, it was replaced by an indirect measurement. Therefore, in the case of high uncertainty of the direct measurement, the indirect measurement was mostly used to calculate the fused measurement.

4.3. Energy use

The direct estimation of evaporator load was obtained based on the temperature difference measured by sensors RT10 and RT11 in Figure 1. The direct estimation of condenser load was obtained based on the temperature difference measured by sensors RT53, RT54, and RT55 in Figure 1. Data on the water flow through the condenser and evaporator were obtained from the LTC procedures. As mentioned before, the compressor electricity consumption was directly measured from the BEMS. The direct estimations of the heat pump performance over a year are given in Figure 8. The fused estimations of heat pump performance over a year are given in Figure 9. In the case of the exhaust air heat recovery heat pump, performance data over a year were obtained based on the evaporator load from the exhaust air and manufacturer data, while estimations of the compressor consumption and condenser load were obtained using indirect measurements. Results on the exhaust air heat recovery heat pump strategy are given in Figure 10.

In Figure 8 it is possible to notice that the evaporator load was six or more times higher than the compressor consumption, which is not correct based on thermodynamic fundamentals. This fault could occur due to faults in sensors, a wrong sensor position, or oscillation in the heat pump operation as explained in Section 4.2. Even though the sensors RT10 and RT11 are placed correctly in Figure 1, it could happen that in practice they are placed to include free-cooling. Therefore, this direct estimation was treated as faulty. Further, in Figure 8 it is possible to notice that in some months condenser load was even negative. This occurred because the temperature difference between RT53 and RT54 was small and even negative sometimes, as explained in Section 4.2.

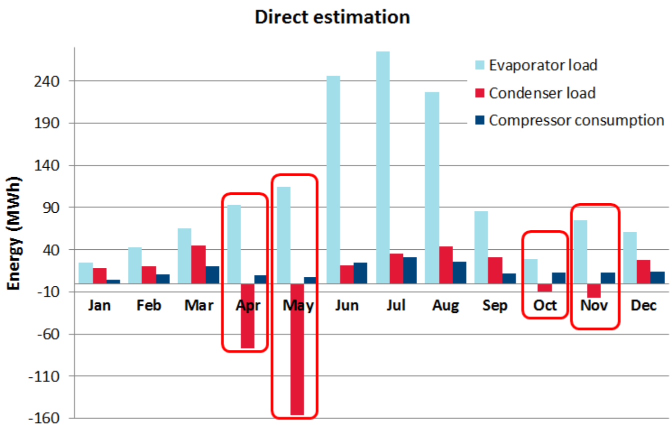

Figure 8. Direct estimation of heat pump performance

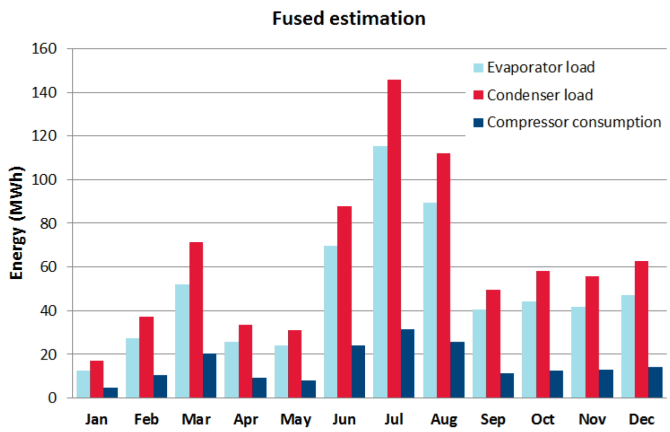

Figure 9. Fused estimation of heat pump performance

In Figure 9 it is shown that heat pump performance data fit well to thermodynamic fundamentals, where the COP is varying from 2.7 to 4.2 during the year. Since fused estimations of the heat pump performance fitted better to thermodynamic fundamentals, they should be treated as more reliable for further decision making and cost-benefit analysis.

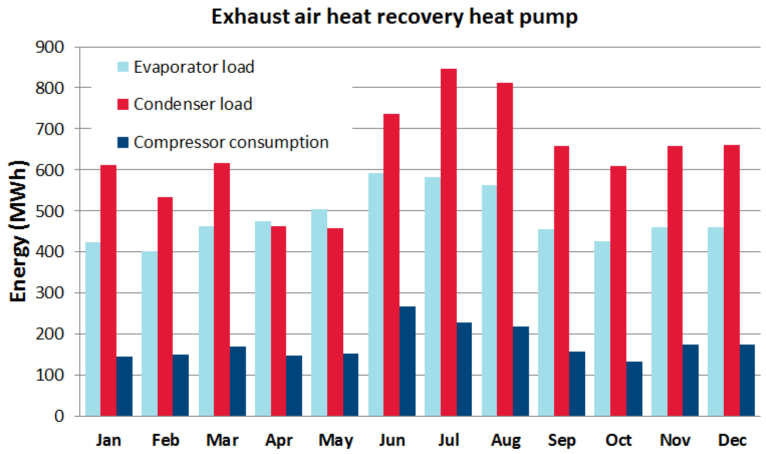

Figure 10. Heat pump performance for the exhaust air heat recovery heat pump

Comparison of the results in Figure 10 and the total energy consumption of the office building (Djuric 2011) showed that condenser load in the case of the exhaust air heat recovery heat pump could be several times higher than the building heating demand. This means that the installed HP1 and HP2 could completely cover the building heating demand. The exhaust air heat recovery heat pump strategy would result in higher outlet water temperature of the condenser. In the case when only the heat recovery within the AHUs was used, the outlet water temperature was lower. A comparison of the outlet water temperatures is given in Figure 11.

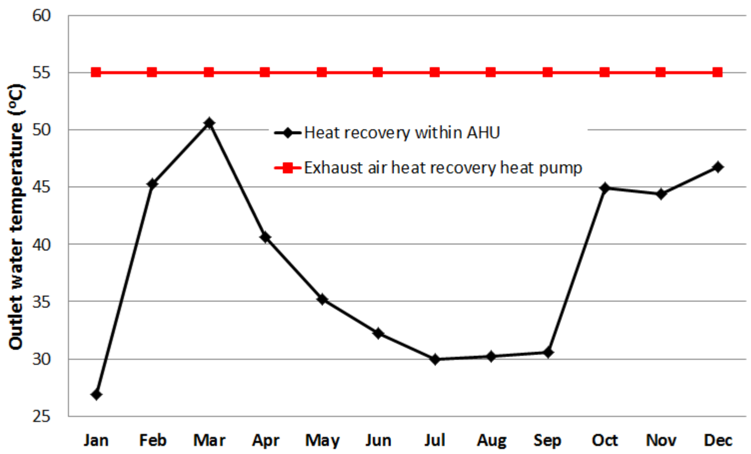

Figure 11. Outlet water temperature after the condenser

The condenser heat when the outlet water temperature was low could not be used for direct space heating. Therefore, in the cost-benefit analysis, the condenser heat was not treated when the outlet water temperature was lower than 55°C.

5. Discussion

Based on the developed measurement approaches, cost-benefit and CO_2 emission analyses were performed to compare the heat recovery solutions and estimation methods. Detailed monitoring data on the total heating and electricity use of the analyzed building in 2009 were obtained from the BEMS (Djuric 2011). The analyzed building that was supplied by the substation in Figure 1 had a heating energy use of about 480 MWh and a total electricity use of about 2,600 MWh in 2009.

Energy savings in the analyzed case study was calculated as:

$$Energy\ savings = -c_{el} \cdot W + c_{dh} \cdot Q_{cd} + c_{dc} \cdot Q_{ev}, \qquad (5)$$

where c_{el}, c_{dh}, and c_{dc} are the price for electricity, district heating, and district cooling respectively. W [kWh], Q_{cd} [kWh], and Q_{ev} [kWh] are compressor consumption, heating energy provided by condenser, and cooling energy provided by evaporator respectively. In Stavanger, the energy supplier is providing district cooling together with district heating. The cooling energy was also counted in the energy savings calculation, because the case building produced its own cooling and with that saved to purchase the district cooling. In this analysis, energy prices were not analyzed; rather the influence of the relationship between district heating/cooling and electricity price was analyzed. The relations c_{dh}/c_{el} and c_{dc}/c_{el} were assumed to be the same. In 2010, the electricity price was about 1 NOK/kWh[I] (2011). Depending on the energy suppliers in the different towns, district heating price was about 0.5 - 0.95 NOK/kWh. The results of the cost-benefit analysis are given in Figure 12.

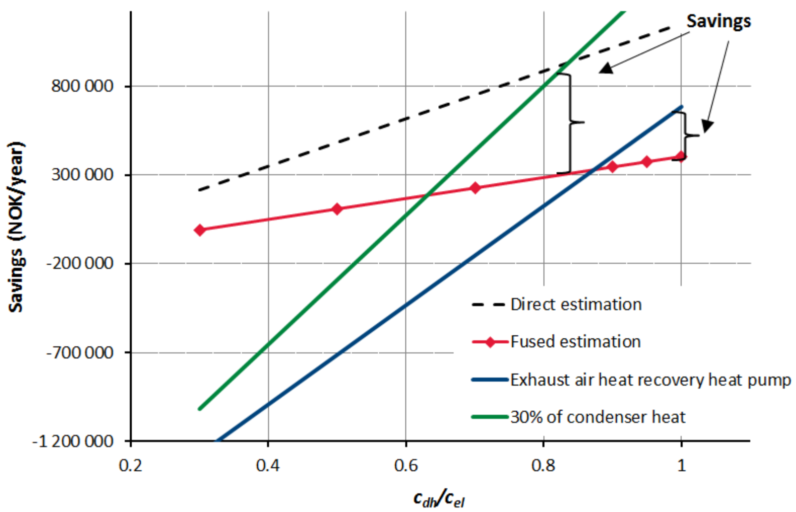

Figure 12. Energy savings for different estimation methods and heat recovery strategies

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[I] 1 EUR = 7.83 Norwegian krone (NOK) at date

The condenser heat obtained using heat recovery within the AHUs had low outlet water temperature, as shown in Figure 11; therefore most of this condenser load was not treated in the cost-benefit analysis. The approach where heat recovery within the AHUs was used could only provide cooling load. In Figure 12, it appears that the savings of using heat recovery within the AHUs, which were estimated using the direct method, are highest (the black dashed line). This fault occurred because cooling load estimation was quite high compared to the compressor power, as discussed related to Figure 8. The difference between the red and the black dashed lines in Figure 12 presents fault in the savings estimation caused by the heat pump performance estimation fault. The cost of this fault could be from about NOK 220,000 to 700,000 per year for the analyzed energy prices. Such a big fault in the energy savings estimation clearly indicated that implementation of LTC procedures and introduction of virtual measurements were cost effective. Finally, this faulty in measurement could lead to a wrong decision and treat the less efficient solution as better. Therefore, fused estimation (red line with diamonds in Figure 12) was treated as the correct estimation of the energy savings using heat recovery within the AHUs. Currently, the case building needed 22 % of the total heating energy obtained by the condenser by implementing the exhaust air heat recovery heat pump strategy. In this case, the solution with the exhaust air heat recovery heat pump strategy was favorable when district heating and cooling had a price above 80 % of the electricity price. If about 30 % of condenser heat would be utilized within the building (the green line in Figure 12), the exhaust air heat recovery heat pump strategy could be favorable when district heating and cooling have a price above 60 % of the electricity price. This means, that if the building could use more of condenser heat or even could export heat to the district heating grid, the solution with exhaust air heat recovery heat pump strategy would be preferable to only heat recovery within the AHUs.

Since the heating energy supply system was analyzed in this study, CO_2 emissions of the heating energy use were compared in Figure 13. The aim of this comparison was to find for which energy sources the analyzed heat recovery solutions could be preferable. In addition, the aim was to show that correct measurements are necessary for a proper estimation of CO_2 emissions. Different combinations for electricity and heat production were considered. CO_2 emissions were calculated for both heat recovery approaches and both estimation methods by using factors given in the standard EN15603 (2008). In the case of the exhaust air heat recovery heat pump strategy, CO_2 emissions were induced only by compressor electricity use. Results in Figure 10 for compressor consumption of the exhaust air heat recovery heat pump strategy were used to estimate the CO_2

emissions. In the case of heat recovery within the AHUs, CO_2 emissions were induced by compressor electricity use and district heating. For this case CO_2 emissions were estimated by using both direct and fused measurements methods. Results in Figure 9 were used to estimate the CO_2 emissions.

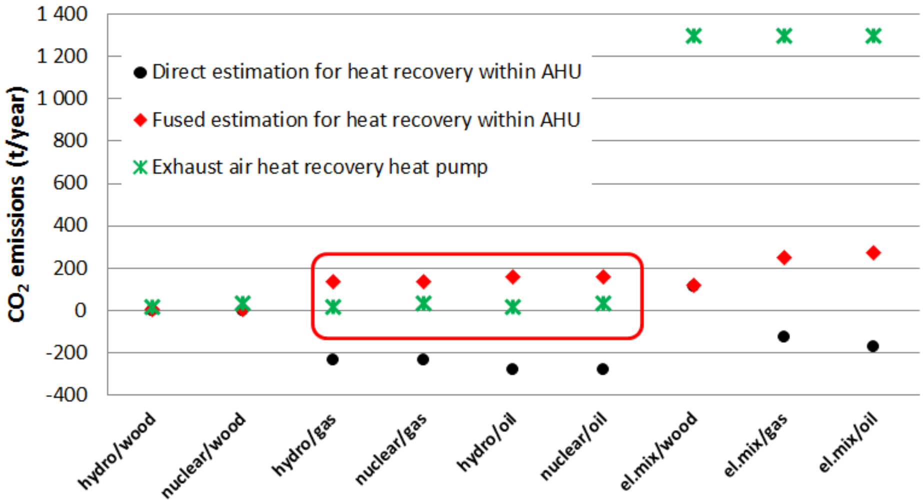

Figure 13. CO_2 emission of heating energy use

Results in Figure 13 obtained by using direct measurements (black points) indicate that heat recovery within the AHUs could induce even negative CO_2 emissions. Since the analyzed substation in Figure 1 was using energy, such conclusion cannot be accepted. This conclusion clearly indicates the importance of reliable measurements for technology decision making for relevant ZEB solutions. Finally, this result show that to prove ZEB definitions it is necessary to have reliable measurements, otherwise the conclusion may be misleading. Since the direct measurements were discarded, only the fused and indirect measurements were treated for further discussion. Results in Figure 13 show that the solution with the exhaust air heat recovery heat pump strategy could be the preferred energy supply solution in the case when electricity was produced from low CO_2 emission sources, like hydro power. This solution could be advantageous when electricity was produced by renewable energy sources. When electricity was provided from an electricity mix, this solution could not be advantageous.

6. Conclusions

A substation in an office building was analyzed. One year of detailed measurements were used to perform the analysis. These measurements were combined into the three virtual measurements: direct, indirect, and fused. The virtual measurements were obtained by integrating data from design, construction, and operation phases. Two approaches for heat recovery were compared: heat recovery within the AHUs and the exhaust air heat recovery heat pump strategy. The results showed a need for detailed documentation and monitoring of energy-efficient designs and technologies for the purpose of fulfilling their aim and for correct performance estimation. Further, fault measurements could lead into the wrong decision regarding the choice of technologies. The less efficient solution appeared to give higher savings. Therefore reliable measurements are necessary to better estimate cost-benefit of implemented technologies. Related to ZEB solutions, the improved measurements seemed to be very important for technology decision making and proof of the ZEB concept. The results showed that the approach with the exhaust air heat recovery heat pump strategy was favorable when district heating and cooling have a price above 60% of electricity price. Further, the results indicated that this solution could be advantageous in the case of renewable electricity production. Future work should deal with improvements in control of the exhaust air heat recovery heat pump strategy.

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