LIFETIME COMMISSIONING AS A TOOL FOR IMPROVING HEAT RECOVERY USING 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. An improved estimation approach of heat pump performance was introduced. This approach was developed by integrating manufacturer and building energy management system data. Direct and indirect measurements were combined using the data fusion method. This heat pump estimation approach was tested on substation where two heat pumps were integrated to support building energy supply system. Two approaches for exhaust air heat recovery: 1) within air handling unit and 2) within the entire building using heat pumps, were assessed. The results showed that use of only direct measurements could result in fail conclusion regarding energy savings. The results showed that exhaust air heat recovery within the entire building using heat pump was favourable solution when district heating price was above 60% of electricity price.

1. INTRODUCTION

In order to increase energy efficiency in buildings and decrease building demand, different technologies for heat recovery has been highly recommended. Heat recovery in buildings can imply different strategies, e.g., moving heat from one zone to another, integrated solutions, and using exhaust ventilation air for heating. Implementation of these strategies could imply use of heat pumps. Regardless of good intentions in energy efficiency, in practice problems such as oversized systems, faults, and poor integration are common. 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 been only sporadically tested as best. To achieve 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. To perform good building operation and quality control of a given energy system, it is necessary to have information about building systems and assessment tools.

During eighties the solution with the exhaust air heat recovery heat pump for ventilation, space heating, and domestic hot water was pushed into marked in Sweden and Germany (Fehrm et al., 2002). The exhaust air heat pump recovers two to three times more energy than air to air heat recovering within ventilation units, as noted in (Fehrm, Reiners et al., 2002). However, based on the Norwegian legislation on energy use in building, there is requirement on heat recovery within air handling unit (AHU) with an annual efficiency of 70% (Lovdata, 2011). Therefore, one of the aims of this study was to compare these two solutions for heat recovery in an office building.

The models on heat pump performance measurements were developed based on available literature references. Temperature measurements were used to establish direct measurements on the heat pump performance. Since temperature measurements sometimes suffer from noise, outlier, and systematic errors, use of data fusion technique can help to estimate real performance data as shown in (Huang et al., 2009). Detailed method for the heat pump performance estimation is explained in (Djuric et al., 2011). Heat pump models and data fusion method were developed on the MATLAB platform (MATLAB, 2010). 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, a new research work in Annex 53, Total Energy Use in Buildings: Analysis and Evaluation Methods ((IEA), 2010), has specified as one of task development of measurement techniques for the purpose of estimating real energy use in buildings. Therefore, in this study benefit and need for proper estimation was studied too. One year of measurements were used to estimate heat pump performance.

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

2. METHODS

Use of LTC procedures provided detailed data on heating, ventilation and air-conditioning (HVAC) systems and access to building energy management system (BEMS) gave 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.

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

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

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

where $\dot{m}_{wc}$ [kg/s] is water mass flow rate on condenser, $T_{w,out}$, $T_{w,in}$ are water temperatures after and before the condenser respectively. In similar way as 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:

$$t = \dot{W} \big/ \dot{W}_{FL} \ . \qquad (2)$$

where $\dot{W}$ [kW] is compressor power and $\dot{W}_{FL}(T_{cd}, T_{ev})$ is compressor power under the full load, and it is possible to get it 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 non-dimensional relation as defined in (Lemort et al., 2009):

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

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

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

$$Q_{cd,f} = \lambda_{cd,1} \cdot Q_{cd,d} + \lambda_{cd,2} \cdot Q_{cd,id} \qquad (4)$$

where coefficients $\lambda_{cd,1}$ and $\lambda_{cd,2}$ are obtained based on the model uncertainties. The fused measurement of the evaporator load was obtained in the same way as the measurement for the condenser load in eq. (4), by using information on evaporator model uncertainties. The measurements outliers were removed using Moffat distance as explained in (Duta and Henry, 2005) and (Djuric et al., 2011).

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 cooling, free-cooling system, two heat pumps, and heating and ventilation systems, which were integrated to district heating and supported by heat pumps as shown in Figure 1. Using LTC procedures detailed data included data from design, construction, and operation phases were collected.

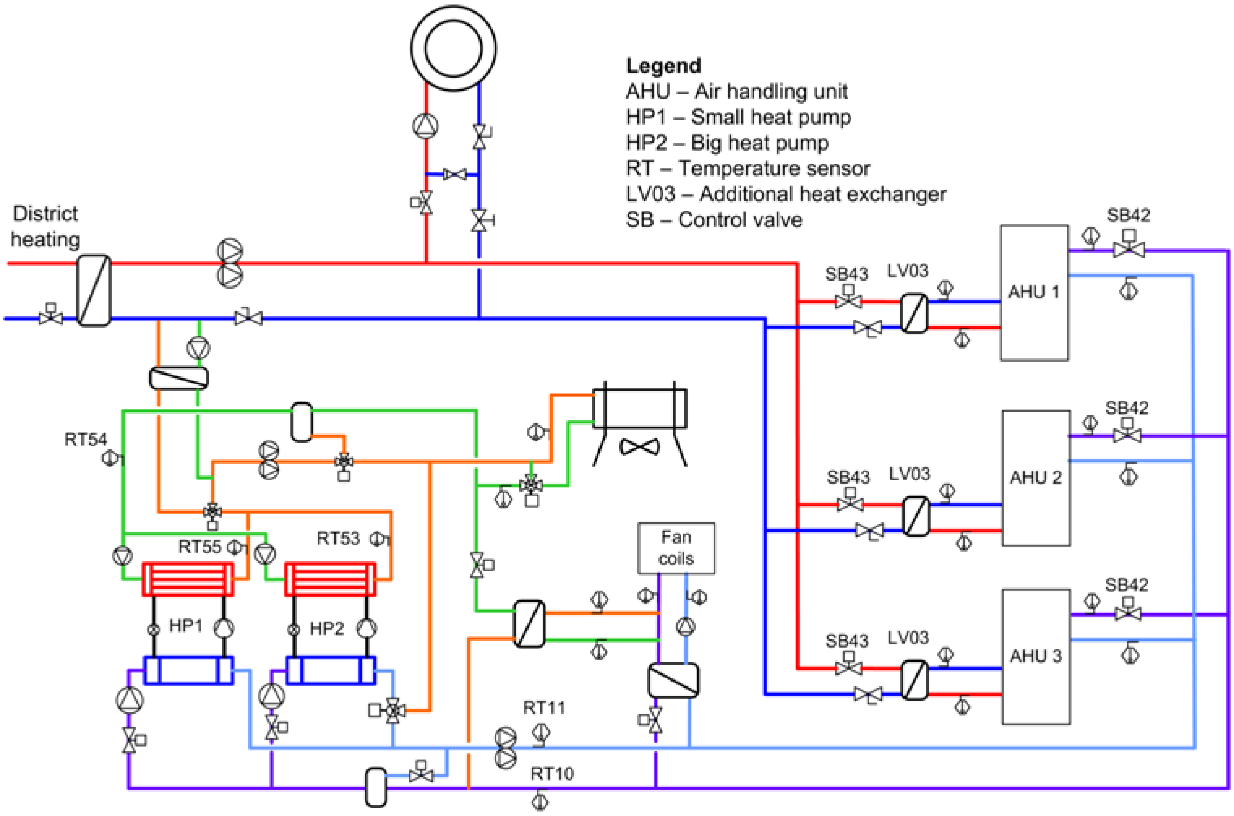

Figure 1. Scheme of the substation with two heat pumps and three AHUs

The heat pumps in Figure 1, HP1 and HP2 were frequency controlled. The installed cooling capacity of HP1 is 200 – 600 kW, while the cooling capacity of HP2 is 420 – 1200 kW. The working fluid in both heat pumps is R-134a. One year of measurements with two minutes step were logged from BEMS for the purpose of this study. Depending which type of heat recovery is used, AHU could have slightly different construction, as shown in Figure 2. These different approaches for heat recovery from exhaust air were named as following in this study: heat recovery within AHU shown in Figure 2.A, and total heat recovery from exhaust air within the entire building shown in Figure 2.B. Heat realized on the condensers is used to support direct heating in the building. Depending on approach for heat recovering, amount of condenser heat could be quite different. Based on national legislation for heat recovery within AHU (Lovdata, 2011), the analyzed building had used heat recovery only within 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 the control valve SB43. This approach implied that HP1 and HP2 were used primarily to produce cooling for fan-coils in IT-rooms, while condenser heat was secondary priority. In design phase HP1 and HP2 were designed to utilize completely heat from the exhaust air. However, due to lack in information between design and construction team, the idea to implement the total heat recovery was not realized in operation. Total heat recovery would imply that condenser heat could be enough for the building heating demand. Therefore, in this study these two approaches for heat recovery were analyzed by using introduced assessment methods.

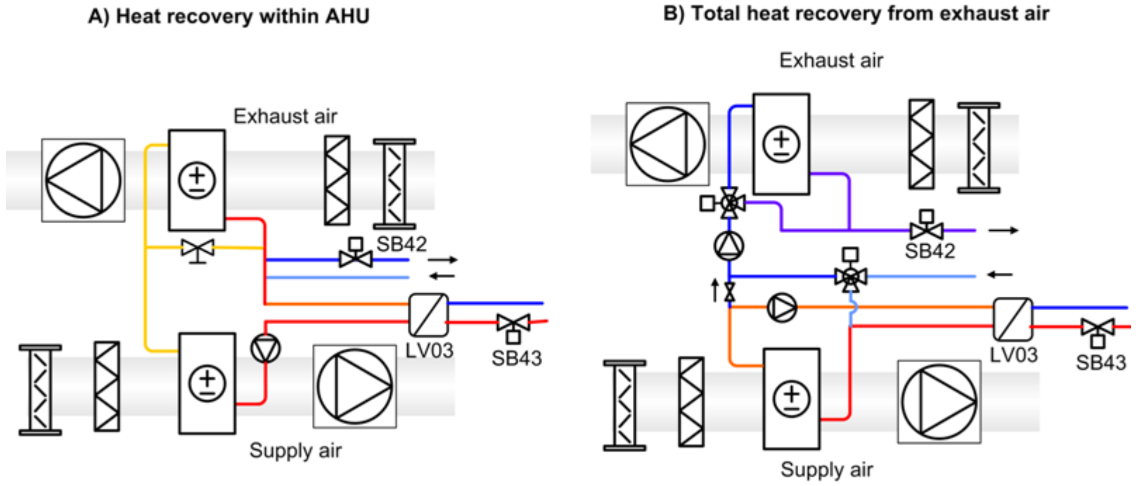

Figure 2. Approaches for heat recovery: A) heat recovery only within the AHU; B) total heat recovery from exhaust air within the building

4. RESULTS

The results show 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 was presented.

4.1. Improvements in Operation

Improvement in operation by using the total heat recovery within the entire building could be explained as avoiding unnecessary wear of control valves in AHUs and better utilization of installed heat pumps. The total heat recovery would imply higher ventilation load as shown in Figure 3.

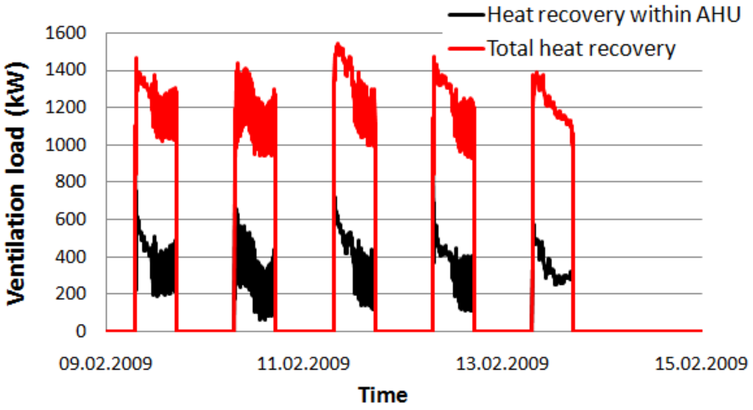

Figure 3. Ventilation load for different heat recovery strategies

In the case of heat recovery within AHUs, a small amount of additional heat was necessary for ventilation. Therefore, the additional heat exchangers LV03 in Figure 1 were poorly utilized. A brief summarization of operation time of control valves SB43 (see Figure 1) in February and June is given in Table 1. Since in the case of heat recovery within AHUs ventilation heat load was low as shown in Figure 3, the average valve position was quite low. These low valve positions and short operation hours indicate 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 total heat recovery, rather than for the heat recovery within the AHUs.

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

Valve	Second week in February 2009			Second week in Jun 2009		
	Position (%)	Operation time	Operation time with more than 95 % (%)	Position (%)	Operation time	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

In the case of the heat recovery within the AHUs, the evaporators were not well utilized. Most of the year HP1 was working and HP2 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 such approach of heat recovery. In the case of the total heat recovery 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 HP1 is in use, 2 means that only HP2 is in use, and 3 means that both HP1 and HP2 are in use.

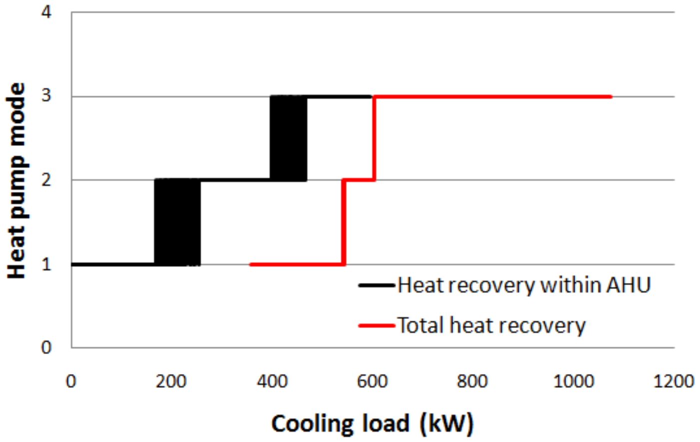

Figure 4. Heat pump mode

In the developed model for the total heat recovery, the heat pumps were controlled based on the cooling load. Currently, in the analyzed building, the heat pumps, HP1 and HP2, were controlled based on the outdoor temperature. 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 total heat recovery and heat pump control based on cooling load, the heat pumps were better utilized as shown in Figure 4, the red line. In the case of the total heat recovery, both heat pumps would work about 4402 hours per year. All these indicated that the substation in Figure 1 was designed rather for the total heat recovery than for heat recovery only within AHUs.

4.2. Energy Consumption

The direct estimation of evaporator load was obtained based on temperature difference measured by sensors RT10 and RT11 in Figure 1. The direct estimation of condenser load was obtained based on temperature difference measured by sensors RT53, RT54, and RT55 in Figure 1. Data on water flow through condenser and evaporator were obtained from LTC procedures. The compressor electricity consumption was directly measured from BEMS. The direct estimations of heat pump performance over a year are given in Figure 5. The fused estimations of heat pump performance over a year are given in Figure 6. Based on cooling load from exhaust air and manufacturer data, estimations of compressor consumption and condenser load using indirect measurements were obtained for the case of the total heat recovery. Results on total heat recovery are given in Figure 7.

In Figure 5 it is possible to notice that evaporator load was six or more times higher than compressor consumption, which is not correct based on the thermodynamic fundamentals. This fault could occur due to faults in sensors or fail sensors position. Even though in Figure 1 sensors RT10 and RT11 are placed correctly, it could happen that in practice they are placed in such way to include free-cooling. Therefore, this direct estimation was fail. Further in Figure 5 it is possible to notice that in some months condenser load was even negative. This occurred because temperature difference between RT53 and RT54 was small and even negative sometimes.

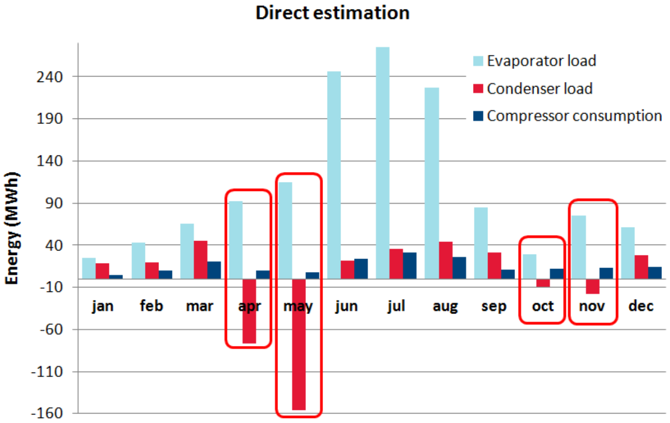

Figure 5. Direct estimation of the heat pump performance

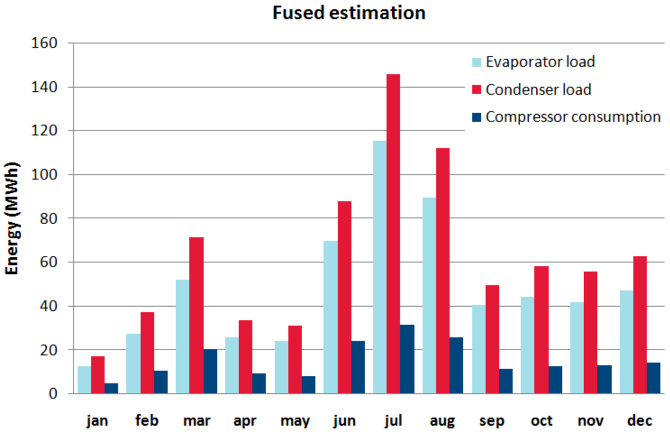

Figure 6. Fused estimation of the heat pump performance

In Figure 6 it is possible to notice that heat pump performance data fit well to the thermodynamic fundamentals, where COP is varying from 2.7 to 4.2 during the year. Since fused estimations of heat pump performance fitted better to the thermodynamic fundamentals, they should be treated as more reliable for further decision making and cost-benefit analysis.

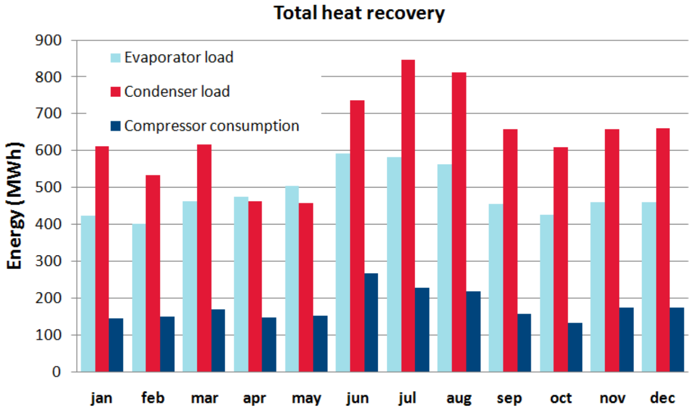

Figure 7. Heat pump performance for the total heat recovery of exhaust air

Comparison of results in Figure 7 and the total energy consumption of office building showed that condenser load in the case of total heat recovery 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 total heat recovery from exhaust air would result in higher leaving water temperature after the condenser. In the case when only the heat recovery within AHUs was used, leaving water temperature was lower. Comparison of leaving water temperatures is given in Figure 8.

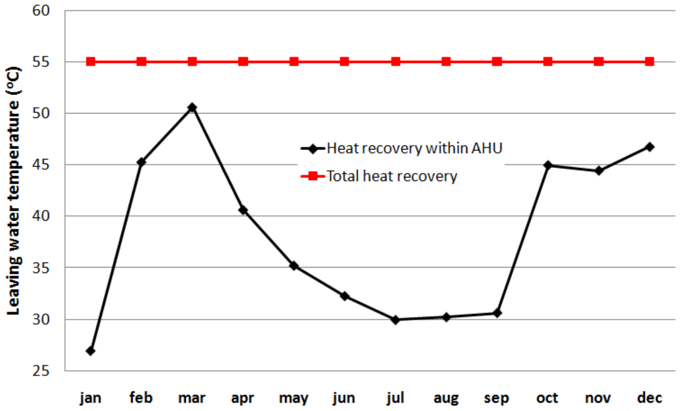

Figure 8. Leaving water temperature after the condenser

The condenser load when leaving 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 leaving water temperature is lower than 55°C.

5. DISCUSSION

Based on developed measurement approaches, cost-benefit analysis was performed. Energy savings in the analyzed case study was calculated as:

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

where c_{el}, c_{dh}, and c_{dc} are prices for electricity, district heating, and 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, energy producer is providing district cooling. In this analysis, energy price was not analyzed; rather the influence of relationship between district heating/cooling and electricity price was analyzed. The relationships c_{dh}/c_{el} and c_{dc}/c_{el} were assumed to be the same. In 2010 electricity price was about 1 NOK/kWh (1 EUR = 7.83 Norwegian krone (NOK)) (Statistics Noway, 2011). Depending on energy producers in different towns, district heating price was about 0.5 - 0.95 NOK/kWh. The results of cost-benefit analysis are given in Figure 9.

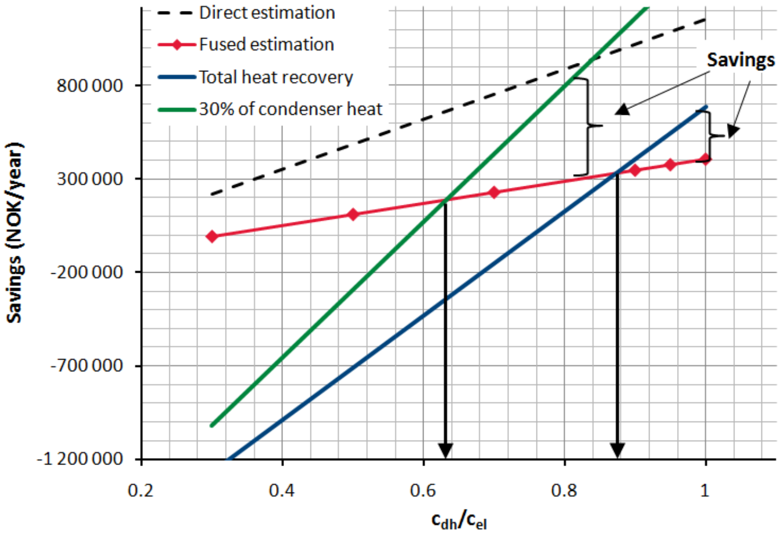

Figure 9. Energy savings for different estimation methods and heat recovery strategies

Condenser heat obtained using the heat recovery within the AHUs had low leaving water temperature as shown in Figure 8; therefore most of this condenser load was not treated in the cost-benefit analysis. The approach where heat recovery within the AHUs is used could provide only cooling load. In Figure 9 it appeared that the savings of using the heat recovery within AHUs are highest, the black dashed line. This fail occurred because cooling load estimation is quite high compare to compressor power, as discussed related to Figure 5. This fault in measurement could lead to fail decision to tread less efficient solution as better. Therefore, fused estimation, red line with diamonds in Figure 9, was treated as correct estimation of the energy savings using the heat recovery within the AHUs. Currently, the case building needed 22% of total heating energy obtained on condenser by implanting the total heat recovery. In that case, this solution with total heat recovery was favourable when district heating and cooling have a price around 88% of electricity price. If 30% of condenser heat would be utilized within the building, the green line in Figure 9, the total heat recovery solution could be favourable when district heating and cooling have a price above 60% of electricity price. This means if the building could use more of condenser heat or even could export heat to the district heating network, the solution with the total heat recovery from exhaust air would be quite preferable than only the heat recovery within AHUs.

6. CONCLUSIONS

Substation in the office building was analyzed. One year of detailed measurements were used to perform analysis. These measurements were combined into three virtual measurements: direct, indirect, and fused. The virtual measurements were obtained by integrating data from design, construction, and operation phase. Two approaches for heat recovery were compared: the heat recovery within the AHUs and the total heat recovery within the entire building. The results showed a huge need for detail documentation and monitoring of energy-efficient designs and technologies for the purpose to fulfill their aim. Further, fail measurements could lead into fail 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. The results showed that the approach with the total heat recovery was favourable when district heating and cooling have a price above 60% of electricity price. Future work should deal with improvement in control of the total heat recovery within the building.

7. REFERENCES

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