Determination of energy quality for hydronic heating systems

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ABSTRACT

The quality of a hydronic energy system is influenced by supply and return temperatures. Instead of using the exergy term, a TEA (Temperature and Energy Accumulated) curve is introduced for evaluating an examplified planned 100 GWh district heating system outside Oslo.
Main focus is on normal running conditions off the design point. The following questions were raised when analyzing the energy system:

- how could we make a design that improves the use of low temperature energy?
- is a design with low flow rate, i.e. low pipeline costs, compatible with the use of low temperature energy?
- can we visualize the quality of the energy used through the distributed temperatures and the corresponding quantity of energy?

A spreadsheet analysis tool was developed and used for the heating plant design and optimization.

This paper presents the TEA curve and shows how evaluations may be done. Different design temperatures ranging from 60 °C/50 °C to 110 °C/40 °C are included in the discussion.

Focus on low return temperatures is often more important than avoiding high supply temperatures at design conditions. The main findings are:

- The TEA curve demonstrates that it is a useful tool in order to consider the energy quality of a hydronic heating system.
- Temperature mixing with shunt valves and recirculation of supply water into the return water pipeline has a negative effect on the possibility to use low temperature energy sources. High temperatures should be utilized when possible.
- A 75 °C/40 °C design improves the level of energy use compared to a 60 °C/50 °C design.

1. INTRODUCTION

A new suburban district is being established in Baerum close to Oslo. The plans have included a district heating and cooling plant, partly based on utilizing the Oslo Fjord as a basis for a combined heat pump (HP) and chiller. Peak power for heating is approximated to 50 MW and the energy demand estimated to 100 GWh per year. The corresponding cooling demand is 22 MW and 28 GWh. The combined heat pump covers 500,000 m^2 of office space and 700,000 m^2 of residential buildings.
This paper analyses the following:

- How to do a system design that improves the use of low temperature energy
- Investigation of how design with low flow rate, i.e. low pipeline costs, is compatible with the use of low temperature energy.
- Investigate if there is a correlation between the quality of the energy used through the distributed temperatures and the quantity of energy?

This paper will focus on how energy supply and return temperatures can impact on possible heat pump performances and how to utilize low temperature energy. As a basis for the evaluation a TEA curve analysis is presented.

For this project report a spread sheet analysis tool is developed, presenting the Temperature and Energy Accumulated (TEA) curves for different locations and system designs. An easy-to-use analysis tool can be based on a mathematical expression for the outdoor temperature duration curve. The expression derived uses the annual mean temperature as sole input to get values for duration. Here statistical Oslo annual temperature duration data is used.

The temperature impact on the district cooling plant and pipeline optimization is considerable, but is not included in this paper.

2. EXERGY – LOW TEMPERATURE ENERGY UTILIZATION

Utilizing low temperature energy has a positive impact on the environment. Norway, as a large producer of high quality energy through hydro power, has historically had low focus on this topic. In modern time an increasing desire to be flexible with regards to energy source has set interest on district heating and low temperature energy utilization.

The use of flow control instead of shunt valve control is emphasized and is demonstrated in this paper. An increased temperature difference between supply and return water eases system control. Increased temperature difference will also reduce necessary valve authority, i.e. reduce valve pressure drop.

In recent time natural refrigerants as CO_2 and NH_3 has been introduced as a useful alternative to CFCs for heat pumps. Of particular interest are systems working with large temperature differences. The yearly performance of a heat pump system influences on the possibility to deliver low-temperature energy into a district heating system. The yearly performance is how your system provides and utilizes the supply water temperature and gives a low return temperature.

The aim is to present a tool to analyze how different solutions can work - and do it without mentioning the word *exergy*. The hope is that some of the examples presented will have an eye-opening effect.

3. ANNUAL ENERGY DEMAND

Oslo has an outdoor design temperature of -20 °C, and about 4,600 degree-days below 17 °C. The annual duration curve of the outdoor temperature, together with the anticipated heating demands, are shown in Figure 1. The 2 MW part of the power demand curve indicates the hot tap water supply demand, which is directly coupled to the demand for a minimum 60 °C supply temperature from the heating plant.

The specification of the heating demand is as follows:

- A diversity factor of 80 % is used for the centralized heating plant. I.e. the theoretical local peak demand of 50 MW is only 40 MW at the heating plant.
- The heating demand is reduced proportional to the increasing outdoor temperature from a maximum of 40 MW at -20 °C to a minimum of 2 MW at 12 °C.
- For all outdoor temperatures above 12 °C, the heat demand is constant 2 MW.
- The annual duration curve is a typical curve for the Oslo/Norway area; i.e. with an annual average outdoor temperature of about +6 °C.

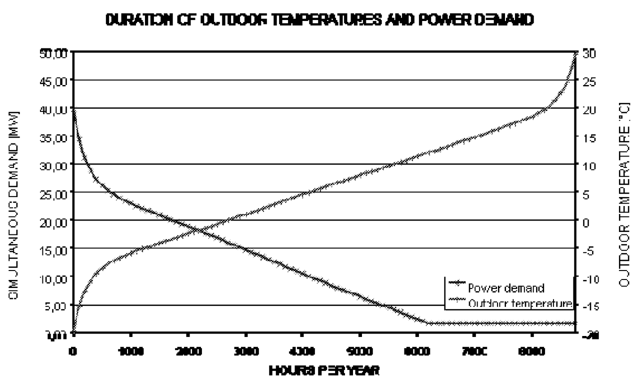

Figure 1. Duration of outdoor temperatures and simultaneous heating power demand.

The statistical weather data for Oslo has been used for the annual temperature duration curve.

4. TEMPERATURE AND ENERGY ACCUMULATED - TEA

The effect of supply and return temperatures can be analyzed by simulating different control strategies as indicated in the inner box of Figure 2. From the supply/return temperature diagram we can read interesting details. For instance, given the outdoor temperature of -10 °C, the supply temperature will be 65 °C and the return 35 °C. This means that: no energy will be delivered below 35 °C, 33 % will be delivered at temperatures below 45 °C, 50 % below 50 °C, and all the energy below 65 °C. The leading question is, can this information about temperatures and delivered energy be summarized and presented over a year? This is solved by introducing the TEA curve, i.e. supply temperature versus distributed energy presented as an accumulated curve. As an example the denoted system curve "75/40(22)" is presented in Figure 3. It reads the following:

Supply and return temperatures:

- The design supply/return temperature at -20 °C, 75/40 °C, has an outdoor temperature compensated supply temperature of 60 °C at -6 °C. Above this the supply temperature is kept constant due to the demand of hot tap water.
- The return temperature is presumed to be decreasing with increasing outdoor temperatures. Above +12 °C outdoors, the return temperature keeps around 22 °C.

Energy deliverance:

- 50 % of the energy is supplied below 45 °C which gives a high COP for a normal heat pump (NH_3).
- About 70 % of the energy is supplied at temperatures below 52 °C which is a maximum limit for NH_3 based heat pumps designed for 25 bars pressure. A NH_3 heat pump will have a COP close to twice as high as a standard, off the shelf R134a heat pump at this temperature level.
- 95 % of the energy will be delivered at temperatures below 60 °C which approximately is the limit for a 25 bar propane heat pump.

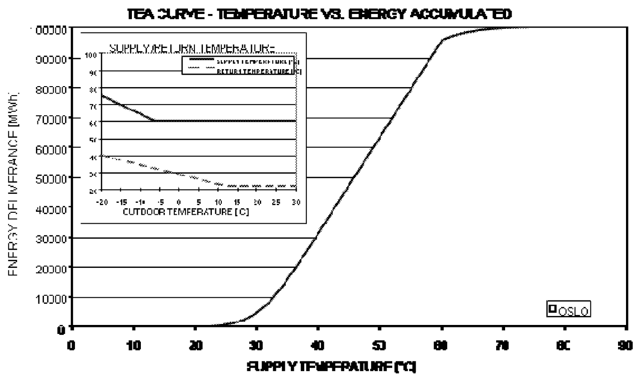

Figure 2. TEA curve: accumulated energy vs. supply temperature for a 75/40 (22) system.

The impact of different water temperatures on the possible use of low temperature energy is demonstrated by the following temperature presumptions:

110/40, i.e.: Supply 110 °C at -20 °C reduced to 60 °C at +10 °C, and the supply temperature is kept constant above 10 °C. Return temperature is assumed running at an average of 40 °C at all temperatures.

75/40, i.e.: Supply 75 °C at -20 °C reduced to 60 °C at -6 °C, and the supply temperature is kept constant above -6 °C. Return temperature is assumed constant at 40 °C at all temperatures.

60/50, i.e.: Supply and return temperatures are kept constant at all outdoor temperatures. Supply temperature is 60 °C and return temperature is 50 °C.

75/40(22), i.e.: Supply 75 °C at -20 °C reduced to 60°C at +7 °C, and the supply temperature is kept constant above +7 °C. Return temperature reduced to 22 °C at 12 °C and kept constant above 12 °C.

75/40 - shunt, i.e: A slightly oversized system (26 %) where the use of shunt control keeps a constant flow and an increased return temperature to as high as 49 °C at all temperatures.

These presumptions will of course impact on the system design parameters, and definitely the demands on systems control. The case with 22 °C return temperature is based on effective heating of tap water in the summer season.

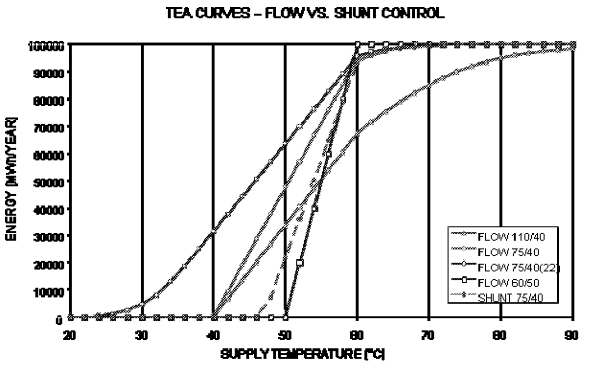

Figure 3. TEA curves for different control temperatures.

The TEA curves in Figure 3 show different aspects. Of particular interest we point out the comparison of 75/40 and 60/50. 75/40 is representative for design temperatures recommended by Borresen (1994). 60/50 is representative for many engineers' ideas of ideal design temperatures with respect to heat distribution in the buildings. Both 75/40 and 60/50 have the same logarithmic mean temperature difference of about 34.7 °C relative to a room temperature of 20 °C. Thus, they require the same area of heat exchanger surface. Of course the 75/40 only needs a 29 % water flow rate compared to the 60/50.

The collection of TEA curves clearly indicates the differences between the two designs. 75/40 has only 4 % delivered energy above 60 °C, while it delivers about 50 % below 50 °C. As indicated by the TEA curves a utilization of the "high" temperature (75°C) to handle peak loads do not only reduce pipeline costs, but a high supply temperature will also reduce the return temperature significantly.

By utilizing the ability to reduce the return temperatures the 75/40(22) can deliver 70 % of the energy below 52 °C. The 110/40 system will need 62 °C to reach the same energy amount.

A shunted system as indicated, will have severe problems supplying energy below 50 °C. This highly confirms that shunting should be avoided, if possible, when aiming at a utilization of low-temperature energy. The rule-of-thumb is that *"all mixing of temperatures will introduce energy quality losses",* and thereby reduce possibilities of utilizing low temperature energy. Shunting is temperature mixing.

5. WATER FLOW

The water flow versus the outdoor temperature for the different systems are shown in Figure 4. The figure clearly indicates that there are many advantages using high design supply temperatures to handle peak loads. One of the advantages are reducing the design water flow rate, and thereby reducing pipeline costs.

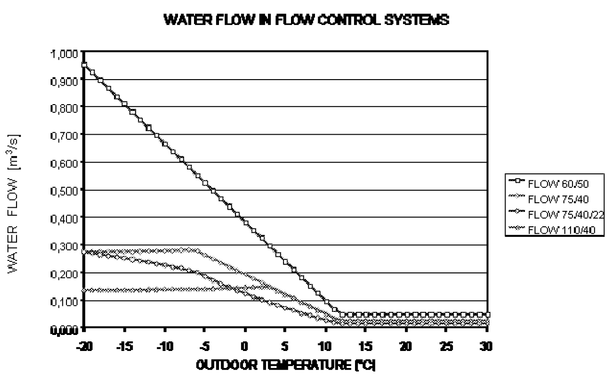

Figure 4. Pipe flow at different outdoor temperatures.

The dominating temperatures for the heating season is between -10 °C and +5 °C. Comparing the 75/40(22) system with a 110/40 system within this temperature range, there are no gains connected to yearly pumping costs when increasing the design temperature difference. Since the 75/40 (22) design reduces the actual-to-design flow ratio throughout the season far more than the other designs, the yearly pumping costs will be significantly less.

The 60/50 system will need significantly larger pipeline dimensions, because of the large design flow rate.

We here assume the same design criteria for all system designs when sizing the pipelines.

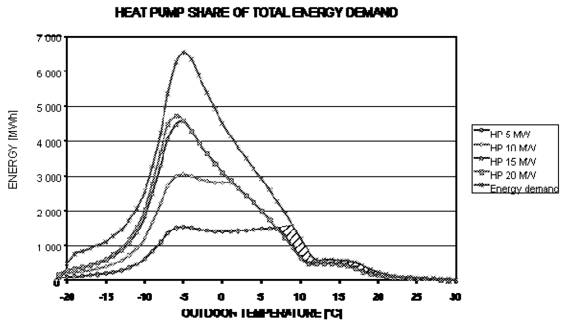

Figure 5. Energy supply at different outdoor temperatures for a 75/40 (22) system. Different sizes for a NH_3 heat pump are shown, reaching from 5 MW to 20 MW (maximum supply temperature 52 °C). The shaded area indicates the possible added energy deliverance to be obtained if the 5 MW energy source increases its supply temperature from 52 °C to 60 °C.

6. HEAT PUMP PEAK LOAD AND TEMPERATURE

How the peak load limits the heat pump coverage is shown in Figure 5. The figure is based on a 75/40 (22) system.

The curves in the figure are based on low temperature heat sources, with a maximum supply temperature of 52 °C. Typically the heat sources can be a NH_3 heat pump. Different heat pump sizes from 5 MW to 20 MW are shown. For the 5 MW heat pump the curves clearly indicate that for an outdoor temperature above 7 °C the upper limit of 52 °C will restrict its energy deliverance. The shaded area indicates the possible increase in deliverance if the source could provide 60 °C. We also notice that 92 % is delivered above outdoor temperatures of -10 °C, and 86 % is delivered between -10 °C and +12 °C when the heat pump is larger than 15 MW. It can be considered to put the heat pump out of operation outside these temperature ranges. This also indicates that the use of the heat pump as a combined heat pump and chiller above +12°C can be questioned.

It can be concluded from the figure that increasing the heat pump size from 15 MW to 20 MW has virtually no effect on the delivered HP energy. This is due to the maximum temperature of the heat source.

The following design statements can be concluded from Figure 5:

- For less than 15 % of the peak load for small heat pumps, there are negligible energy gains by increasing maximum temperature level for the heat pump.
- For more than 25 % of the peak load of larger heat pumps, you have to increase the maximum temperature level to utilize an increase in heat pump size.

7. CONNECTING TO DISTRICT HEATING

The example under describe one alternative solution how to connect the a building system to district heating.

We observe that we also use the return flow. This can be possibilities if there is an existing district heating system which is designed for higher temperatures. The solution shown is tested out in a 40 000 m^2 hospital in Norway.

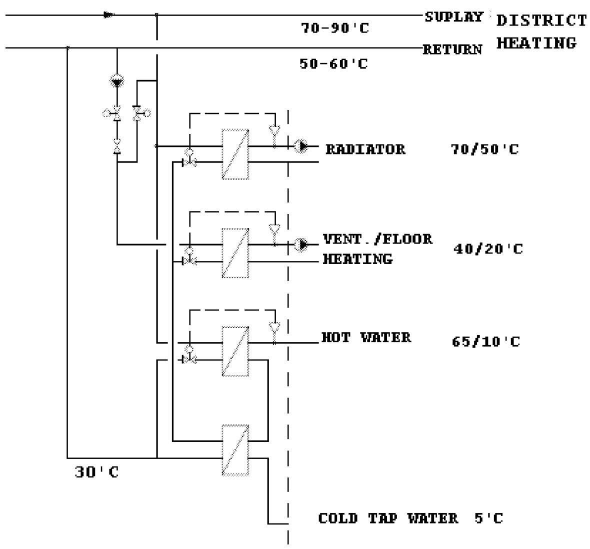

Figure 6. System coupling utilising and reducing the district heating return flow temperature.

8 Experience

Experiences from the heat pump driven district heating plants in Baerum county outside Oslo show that return temperatures seldom are lower than 50 °C. This indicates that shunting and recirculation of supply water into the return water pipeline is still widely used. (The circulation can be deliberately or non deliberately). This has a significant negative effect on the possibility to use low temperature energy sources. An alternative to reduce the return temperature is to use the return flow to heat low temperature systems.

9. CONCLUSIONS

The main findings are:

- The TEA curve demonstrates that it is a useful tool in order to consider the energy quality of a hydronic heating system.
- Temperature mixing with shunt valves and recirculation of supply water into the return water pipeline has a negative effect on the possibility to use low temperature energy sources. High temperatures should be utilized when possible.
- A 75 °C/40 °C design improves the level of low temperature energy use significantly compared to a 60 °C/50 °C design.

The introduction of the TEA curve gives a simple way of indicating the use of energy quality for a system, i.e. the temperature of the energy used. This will be an important tool for future evaluation of heat pumps or energy from other low temperature heat sources. The impact of different design and control strategies can be clearly demonstrated: An increased supply design temperature, for instance 75 °C/40 °C, may significantly improve the use of low temperature energy, i.e. for instance compared to a 60 °C/50 °C system.

The use of temperature mixing devices, for instance shunt valves, will decrease system performance and thereby reduce the possibility of utilizing low temperature energy.

The use of the return flow to heat low temperature system can reduce the main return flow temperature substantially and reduce the total flow.

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