TOWARDS ENERGY-USE OPTIMISATION OF A DOMESTIC HEATING SYSTEM BASED ON A HEAT PUMP

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ABSTRACT

The overall efficiency of a heat pump system i.e. heat source, heat pump unit and distribution system, is strongly affected by the design temperature levels of the system. The temperature levels influence the performance, in every single process from the time that energy flows from the ambient to the heat source until the heat is released from the building envelope to the environment. The heat exchanging processes in every step affect the energy flow, the losses and generally the energy efficiency of the heating system.

The objective of this paper is to show how the different temperature levels influence the energy efficiency and the performance of a heat pump system for domestic applications. The heat pump unit and the heat distribution system are approached as an integrated system. Hence, calculations are performed in order to estimate possibilities for energy savings and for improving the overall efficiency of a heating system based on a heat pump.

The results show mainly the increasing call reducing the supply temperature of heat distribution systems. The influence of the refrigerant characteristics on optimal heat pump designs will be discussed.

INTRODUCTION

Nowadays, there is a growing call for energy efficient and environmentally friendly heating systems. Conceptual thinking on energy use is an aspect that increases its value and importance considering residential heating. When designing, installing and operating a heating system, the term energy savings is an added factor to the primary aspects of cost and comfort level that are traditionally taken into account.

Due to many reasons, but mainly because of the growing call for efficient energy use and increased comfort standards there is a great interest and a leading trend towards low-temperature heat distribution systems. Heating systems with low supply temperatures and large surface areas are suitable for buildings with low specific energy requirements (Keller, 1997). The key elements are thus good insulation, sufficient thermal storage and large heating surfaces.

Considering a heat pump, its performance depends strongly on the quality of the heat source and the heat sink. For a heating system comprising a heat pump unit, a low temperature heat distribution system coupled to the condenser is the basic prerequisite for a good performance and a high efficiency. However, optimizing the heat pump unit itself is not enough for optimizing the whole system. On the contrary, the heating system must be approached as an integrated system in order to investigate for energy efficiency and savings. In such a case, the demand for an overall approach is necessary.

Energy flow versus temperature

The main objective of this paper is to show how different temperature levels influence the several steps in a heating system for domestic applications based on a heat pump. Therefore, Fig. 1 is used for exemplifying the energy flow for a heating system based on a heat pump versus the temperature change in every process. The figure depicts all the steps from the time energy flows from the heat source to the evaporator, until heat gets lost to the building surroundings. Each temperature difference and every heat exchanging process induces losses to the entire heating system. The magnitude of every step affects the energy flow and generally the efficiency of the heating system.

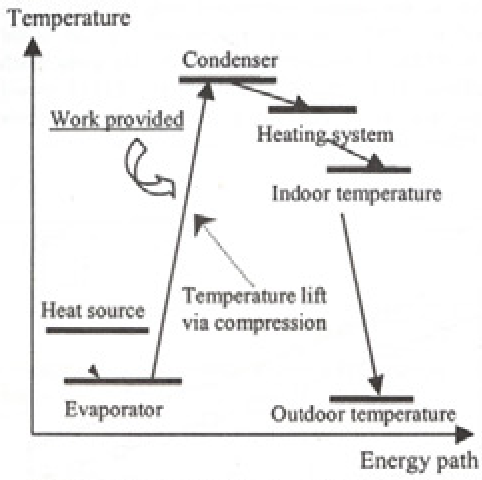

Figure 1. Energy flow in a heating system versus the temperature levels.

This is the main ground where the analysis is based on in order to find how different temperature levels influence the performance of a heat pump system for domestic applications. The rational model of fig. 1 is turned into a simulation tool so that it can be studied from a thermodynamic point of view.

The analysis necessitates exemplifying the whole idea with a system choice. After selecting the main system components calculations are done for one case. In this case, the condensation and the evaporation temperatures *T*□ and *T*□ are a result of the overall system design. The main objective is to maintain a low condensation temperature and so a high efficiency for the heat pump unit. The results show also that improvement of the performance at the heat distribution side requires a low condensation temperature and vice versa.

System choice - design criteria. The simulation program is based on assumptions that correspond to a realistic approach. The system in the following analysis concerns a ground coupled heat pump unit connected to a low temperature, hydronic distribution system. The system is selected so that it serves also for the preparation of hot sanitary water and for this reason the heat pump unit includes a hot gas cooler between the compressor and the condenser. The heating process is regulated from the hot sanitary water temperature and the indoor temperature. The best way to run the system is to operate the heat pump so that it only serves for the preparation of the hot tap water until a desired high temperature is reached. Then the heat pump continues heating the water for the heat distribution system (Granryd, 1997). In Sweden it is common to dimension the heat pump so that it covers a certain portion of the maximum energy demand. An auxiliary heating system supports the heat pump during the coldest period of the year when the maximum load occurs. The auxiliary heating system and its connection to the heating system varies depending on its characteristics and the type of application. In this case, the auxiliary heat is electricity and is supplied to the same distribution system.

For the sake of simplicity, climatic and ground conditions are taken for the region of Stockholm. For buildings in Stockholm the outdoor design temperature (*ODT*) is set to -17°C.

Beginning with the heat source, an indirect system is chosen. A secondary refrigerant (brine) circulates between the heat source and the evaporator. An aqueous solution of ethylene glycol with a concentration of 30,5 % by weight is selected. The main assumption considering the heat source is that the amount of energy that is extracted from the ground is equal to 30 W/m. This is typical for a single bore hole presupposing a sufficient installation in rock with proper back-filling (Granryd, 1997).

Continuing with the heat pump unit, the most important component to be selected is the refrigerant. The thermodynamic properties of the selected refrigerant for given operating pressures determine the maximum temperature, the heat output and generally the whole vapor compression cycle. In this discussion, R410A is taken as the working fluid. R410A has a negligible temperature glide, has a rather low molar mass ($\tilde{M} = 72{,}6$ *kg/kmol*) and allows for obtaining high temperatures at the compressor's outlet while maintaining a low condensation temperature. This is favorable for heating the sanitary water up to a sufficient high temperature. In this case, subcooling is not included since R410A does not benefit from subcooling.

The compressor has to be carefully selected in order to withstand the high pressures and be compatible with R410A. The empirical equations, used for the calculation of the isentropic, volumetric and motor electric efficiencies give satisfying results. These equations depend on the pressure ratio and the refrigerant's molar mass (Pierre, 1979).

High heat transfer with minimum charge and low leakage probability demand compact heat exchangers. Plate type heat exchangers (evaporator, condenser and gas cooler) with negligible heat losses are chosen. Hydronic-floor heating with an option of combining to some extent with convectors is used as the heat distribution system. The chosen floor heating design is an ordinary assembly consisting of a layer of concrete, an insulation layer, heating plates on which the pipes are fastened and an upper floor layer of parquet. The total thermal conductivity of the floor in the calculation tool changes depending on the thickness of the layer of insulation placed on the ground.

The house is assumed to be tight and well insulated. In the simulation model the main assumption is that the house is one-story dwelling with the floor placed directly on the ground. Comparisons are made between the heat transfer losses of the house for two cases. The first case assumes that the heating system consists of radiators. In the second case the heating system comprises a floor heating system with a possibility of supplementing it with convectors.

Description of the simulation program. In order to examine the effect of different temperature levels on the performance of the selected heating system a simulation program has been developed. The program gives also results for the influence of different temperature levels on dimensioning parameters as the borehole depth and the insulation thickness under the floor heating construction. The main set variables in the simulation tool is:

- Heat source: The inlet brine temperature is taken constant and equal to $T_{in\text{-}b} = -3{,}5°C$.
- Heat pump unit: The overall heat provided by the heat pump is constant and equal to *Q*□ = *5000 W*.
- There is no sub-cooling.
- There is a steady superheating of $SH = 5K$.

- The eqs. (1) and (2) allow for the calculation of the coefficient of performance for the heat pump. The compressor's overall efficiency (eq. (3)) depends on the isentropic efficiency, the electric motor's efficiency and the transmission efficiency.

$$COP_1 = COP_2 + 1 \qquad \text{(I)}$$

$$COP_2 = n_{all} \cdot \left(\frac{q_v}{e_v}\right) \qquad \text{(II)}$$

$$\eta_{all} = f(\eta_{is}, \eta_{el}, \eta_{tm}) \qquad \text{(III)}$$

- Heat distribution system: The overall heat transfer coefficient between the house and the surroundings (the floor is not taken into account) is set equal to $\Sigma(kA)w$ = *60 W/K*.
- The overall heat transfer coefficient between the floor without floor heating and the ground is taken $kAgrad$ = *25 W/K*.
- The overall heat transfer coefficient between the floor with floor heating and the ground depends on the thickness of the extra insulation layer underneath the floor heating construction (eq. 4).

$$kA_g = f(\Delta\delta) \qquad \text{(IV)}$$

- The extra transmission losses to the floor due to the floor heating application depend on the heat flux rate and the thickness of the extra insulation (eq. 5).

$$Q_{gloss} = f(\dot{q}, \Delta\delta) \qquad \text{(V)}$$

- The room temperature is set to Tr = *20°C*.
- The outdoor temperature is set to Tut = *-10°C*.
- The ground temperature is set to Tgr = *6°C*.
- Calculations for the domestic hot water are not included in this paper.
- The eqs. (6) and (7) provide the transmission losses for the house when a radiation system is applied and when a floor heating system is applied respectively.
- The factor z in eq. (7) determines the portion of losses that correspond to the floor heating system when it is supplemented with convectors.

$$Q_{grad} = \sum (kA)_w \cdot (T_r - T_{ut}) + kA_{grad} \cdot (T_r - T_{gr}) \qquad \text{(VI)}$$

$$Q_g = \sum (kA)_w \cdot (T_r - T_{ut}) + kA_g \cdot (T_r - T_{gr}) + Q_{gloss} \cdot z \qquad \text{(VII)}$$

Results. The selection of the refrigerant plays an important role on the operation and the design of the heat pump system. The simulation model has been developed taking into account that R410A is the chosen refrigerant. Fig. 2 shows the enhancement in *COP*□ when R410A is used in the model compared to R134a. A refrigerant such as R134a gives much lower discharge temperatures. A design for R134a would probably utilize a subcooler. Consequently, the design criteria should be different if another refrigerant is to be used.

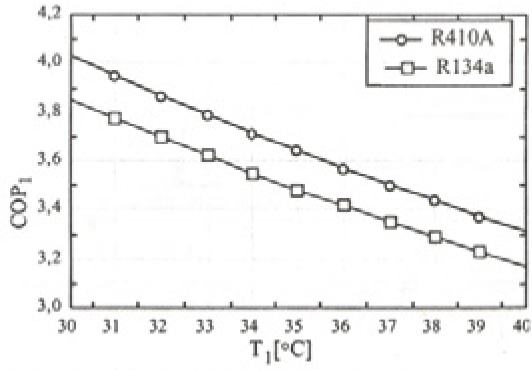

Figure 2. *COP*□ for the heat pump versus condensation temperature *T*□ for two different refrigerants.

Figs. 2 and 3 show the benefit of lower condensation temperatures. Moreover, Fig. 3 indicates that a better *COP*□ requires a lower logarithmic mean temperature in the evaporator, in other words, more efficient heat exchanger.

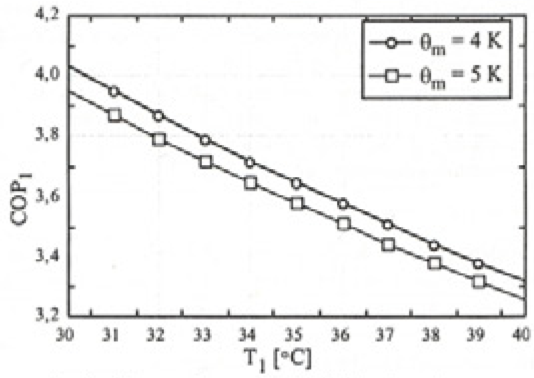

Figure 3. *COP*□ for the heat pump versus condensation temperature *T*□ for two different logarithmic mean temperature differences in the evaporator.

Fig. 4 shows the isentropic efficiency, the electric efficiency and the compressor's overall efficiency versus the condensation temperature *T*□. All efficiencies for the compressor increase for this particular study. However, the increase is rather small.

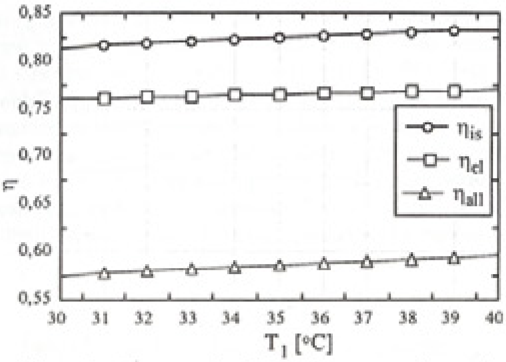

Figure 4. Compressor's efficiencies versus the condensation temperature *T*□.

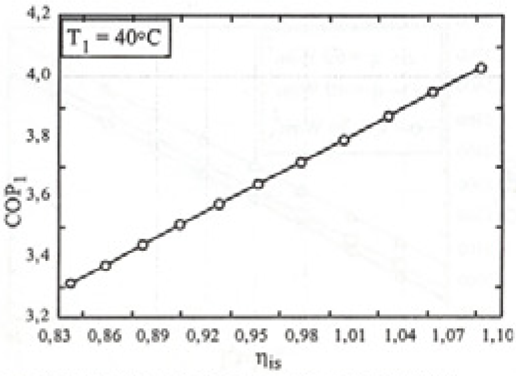

Figure 6. *COP*□ versus the isentropic efficiency *ηis*.

The condensation temperature *T*□ has a counter influence on the borehole depth *x*. For lower condensation temperatures *T*□ the *COP*□ becomes substantially higher (as shown in figs. 2 and 3). Hence, more heat has to be extracted from the borehole, which consequently has to be deeper. For this reason, when the condensation temperature decreases the depth of the borehole increases. Fig. 5 illustrates this effect. If this effect is neglected by assuming a constant borehole depth, then a lower condensation temperature may result in a system with a lower *COP*□ since the evaporation temperature will stabilize on a lower value and this consequently implies a higher temperature lift for the heat pump unit.

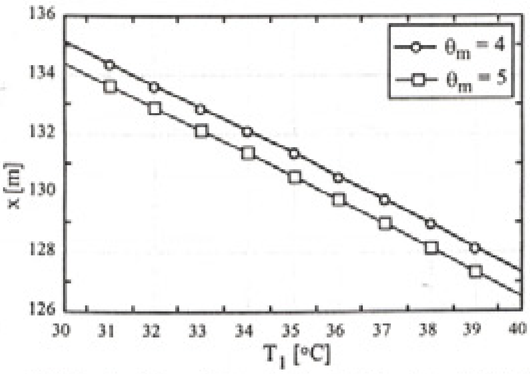

Figure 5. Bore hole depth versus the condensation temperature *T*□.

Fig. 6 is the result of the attempt to investigate the possibility of gaining the same enhancement in *COP*□ by changing another parameter than the condensation temperature. For this reason the condensation temperature is constant and the values of *COP*□ in the y-axis correspond to those of fig. 3 (*θm = 4 K*). Fig. 6 shows that the isentropic efficiency must be higher than one in order to retain the same *COP*□ as when *T*□ = *30°C*.

Considering the heat distribution side, fig. 7 gives the dependence of the heat transfer coefficient for the amount of energy that transfers from a heated floor to the ground on the water supply temperature *Ts*. Fig. 7 indicates the need for lower water supply temperatures consequently lower condensation temperatures. Moreover, the insulation thickness is an important factor to be considered.

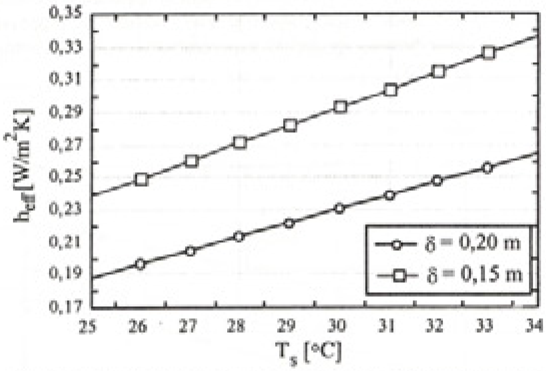

Figure 7. Heat transfer coefficient for transmission by a heated floor to the ground versus the water supply temperature *Ts*.

A low-temperature floor heating system is favorable because it lowers the condensation temperature. It is also favorable because it lowers the transmission losses. The indoor temperature is usually taken as constant when a heating system is designed. However, a lower indoor temperature is advantageous too. It decreases the overall transmission losses and at the same time increases the possibility for using lower supply water temperatures. Fig. 8 illustrates the transmission losses of a heated floor to the ground for different indoor temperatures *Tr*. A lower heat flux is also beneficial.

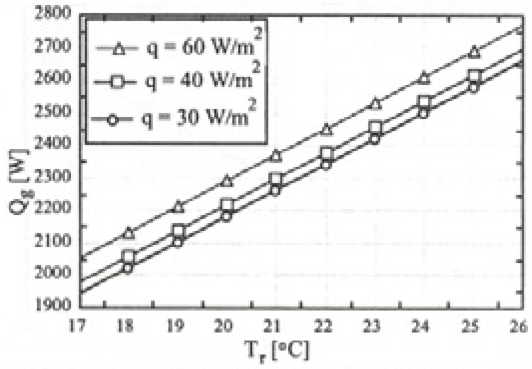

Figure 8. The transmission losses of a heated floor to the ground versus the indoor temperature *Tr*.

In fig. 9 the relation between the overall transmission losses of a house with radiators and the transmission losses of a house with floor heating system is plotted versus the thickness of the extra insulation layer $\Delta\delta$ required on the ground when floor heating is applied. When the relation is one, the losses are equal. When it is lower than one, the losses from applying a floor heating system are bigger. When $z = 1$ then all the energy demand in case that a floor heating system is applied, is covered totally by the floor heating system. The relation is improved when the floor heating system is supplemented with convectors ($z = 0,5$, $z = 0,8$).

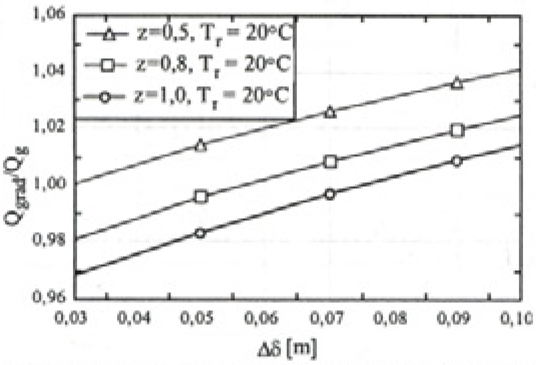

Figure 9. The relation between the transmission losses of a house with radiation system and with floor heating system versus the thickness of the extra insulation layer $\Delta\delta$ on the ground in the case of floor heating.

Conclusions. The results show clearly that lower temperature levels and smaller steps in the heat exchanging processes favor the heat pump and the heat distribution system. Since we deal here with the heating system and not the heat pump only the dependence of the condensation temperature on the water supply temperature of the heat distribution system and vice versa are taken into account. Correspondingly, the water supply temperature depends on the specific energy requirements of the house and the energy requirements are affected by the tightness of the house. In other words, the design parameters in the heating system depend on each other. It is not possible to isolate one process and to work intensively on improving its performance since this process is in a chain reaction with another part of the system. Hence, changing one parameter in one end brings results and effects on the other end. Approaching the heating system as a whole gives the possibility to see how the design temperature levels of the system affect it and every single process in it. Taking this approach into account it is even possible to find the energy savings for different sizes and different designs of the heating system (Forsén, Granryd, 1999).

NOMENCLATURE

Coefficient of performance for the heat pump	COP□	—
Coefficient of performance for the refrigeration process	COP□	—
Compressor's efficiency	η	—
overall	η_{all}	—
electric	η_{el}	—
isentropic	η_{is}	—
transmission	η_{tm}	—
Factor of heat demand covered by floor heating	z	—
Heat flux	$\dot{q}$	W/m²
Heat pump's overall energy output	Q□	W
Heat transfer coefficient for transmission to the ground for a heated floor	$heff$	W/m²K
Logarithmic mean temperature difference in the evaporator	θ_m	K
Molar mass	$\overline{M}$	Kg/kmol
Superheating	SH	K
Temperature	T	°C
brine inlet	$T_{in\text{-}b}$	°C
condensation	T□	°C
evaporation	T□	°C
ground	T_{gr}	°C
indoor	T_r	°C
outdoor	T_{ut}	°C
Outdoor Design	ODT	°C
water supply	T_s	°C
Overall heat transfer coefficient		W/K
between the floor and the ground	kA_{grad}	W/K
between the heated floor and the ground	kA_g	W/K
between the house and the surroundings	$\sum(kA)_w$	W/K
Thickness of extra insulation layer	$\Delta\delta$	m
Thickness of insulation	δ	m
Transmission losses		W
of the house with floor heating system	Q_g	W
of the house with radiation system	$Q_{g\,rad}$	W
to the ground due to floor heating	$Q_{g\,loss}$	W
Volumetric compression work	e_v	J/m³
Volumetric refrigerating effect	q_v	J/m³

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