CAPACITY CONTROL AND OPTIMISED OPERATION OF DOMESTIC HEAT PUMP HEATING SYSTEMS

F. Karlsson, Ph.D., SP Technical Research Institute of Sweden, Box 857, 501 15 Borås, Sweden
P. Fahlén, Professor, Building Services Engineering, Chalmers University of Technology, 412 96 Göteborg, Sweden

Abstract: The efficiency of ground-source heat pump heating systems can be increased considerably by using variable-speed compressors and pumps instead of fixed-speed components. For the heat pump systems evaluated in this work the increase in efficiency is in the range of 15 – 27 %, if the efficiency of the variable-speed compressor can be increased to the same level as the single-speed compressor.

Key Words: ***heat pumps, variable-speed, intermittent, seasonal performance, efficiency, control, optimization***

1 INTRODUCTION

Heat pumps for domestic use are connected to continuously varying loads and thus it must be possible to adjust their capacity. Conventionally, this is done by intermittently switching the heat pump on and off. There are other methods of capacity control, with continuously variable-speed control being the most efficient, as described by Pereira and Parise (1993), Qureshi and Tassou (1996) and Wong and James (1988). However, Poort and Bullard (2006) and Ilic, Hrnjak et al. (2002) report promising result for compressor short cycling as capacity control technique. According to Miller (1988) and Garstang (1990) the variable-speed capacity control is more efficient than the intermittent control due to better performance at part load through heat exchanger unloading and lower supply temperature, higher compressor efficiencies, fewer on/off cycles, reduced need for supplementary heating and reduced need for defrosting. Investigations where variable-speed compressors have been compared to intermittent controlled compressors in hydronic heat pump heating applications have been reported by Poulsen (1998), Tassou, Marquand et al. (1983), Aprea, Mastrullo et al. (2006), Bergman (1985) and Forsén and Claesson (2002). Comparison of performance at single operating points show improvement in COP by 10-30 % when using variable-speed compressors. Investigations of the seasonal performance factor show increases in performance in the order of 1-25 %. The difference in results depend on type of heat pump (air source or ground source), compressor type, operating range (speed) and of course design of capacity to heat load.

The use of variable-speed compressors and pumps add degrees of freedom for the control system and possibilities for increased energy efficiency. It is possible for a specific operating condition to find an optimal combination of compressor, pump and fan speeds which maximise the efficiency of the heat pump system. Such investigations have previously been presented by Jakobsen, Rasmussen et al. (2000), Jakobsen and Skovrup (2002), Skovrup and Jakobsen (2001), Liptak (1983), Hydeman and Zhou (2007) and Granryd (2007).

This paper is a summary of the work presented in the doctoral thesis of Karlsson (2007), which concern capacity control of compressors and pumps and how to use these components for increased energy efficiency. The analysis is made by a combination of laboratory measurements and computer simulations.

2 RESULTS FROM LABORATORY TESTS

Tests according to SS-EN 255-2 (1997) were performed on two ground-source heat pumps in SP's accredited laboratory. One (called X, Table 1) had a variable-speed scroll compressor which can be operated in the range 30-120 Hz. The other one (Y, Table 2) was a state-of-the-art heat pump with a single-speed compressor. A third, fictitious heat pump (Z) is included in the analysis. It has the characteristics of X but its efficiency is scaled such that it is as efficient as Y at the nominal frequency, i.e. 80 Hz. The COP of X and Y at the nominal speed are given in Figure 1 below, showing that heat pump Y is more efficient than X. However, the efficiency of X increases at part loads as shown in Figure 2.

Table 1: Component list for heat pump X

Component	Designation
Compressor	Mitsubishi AEV60F (scroll)
Condenser	CP 415-50 plates
Evaporator	CP 415-60 plates
Thermostatic expansion valve	Danfoss TCAE, 068U4325
Frequency converter	Danfoss VLT
Refrigerant (charge)	R407C (2.5 kg)

Table 2: Component list for heat pump Y

Component	Designation
Compressor	Bristol H75A42QDBE (piston)
Condenser	Cetetherm (50 plates)
Evaporator	Cetetherm (50 plates)
Thermostatic expansion valve	Honeywell, FLICA TLEX-3,5
Refrigerant (charge)	R407C (2.4 kg)

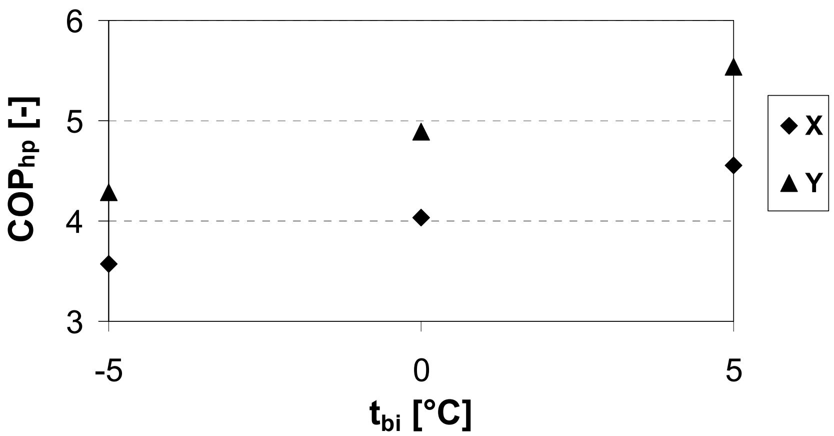

Figure 1: The COP for the two heat pumps, not considering the pumps. Supply temperature 35 °C

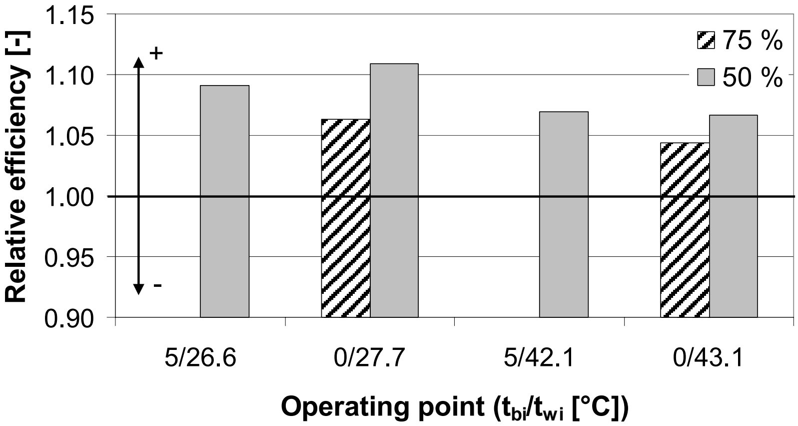

Figure 2: The relative efficiency is the compressor efficiency at part load divided by the compressor efficiency at full capacity (at 80 Hz)

3 SEASONAL PERFORMANCE CALCULATIONS

To compare the efficiency of variable-speed control to that of intermittent control the seasonal performance factor was calculated for the three heat pumps. Two different models was used, one steady-state model and one transient model. The models are described by Karlsson (2007).

3.1 Method

As the heat pumps originally were of different capacities they have been scaled to have the same capacity at DOT. Heat transfer flows and power used by pumps are scaled with the heat pump capacity such that all heat pumps with the same capacity at DOT have pumps of the same capacity. Furthermore, the circulation pumps were considered as single-speed pumps having a constant efficiency of 10 %. This may seem as a low value but is actually the reality for small fixed-speed pumps which may have efficiencies even lower than this.

The steady-state model assumes a constant return temperature from the heating system in each time step. This is the method most commonly used in seasonal performance calculation methods. In the transient model the thermal inertia of the heating system is added, which then considers the variation of return- and supply temperatures. An example of this is given in Figure 3. It was expected that the average supply temperature from the heat pump when the compressor was in operation would increase for fixed-speed heat pumps using the transient model compared to the steady-state model.

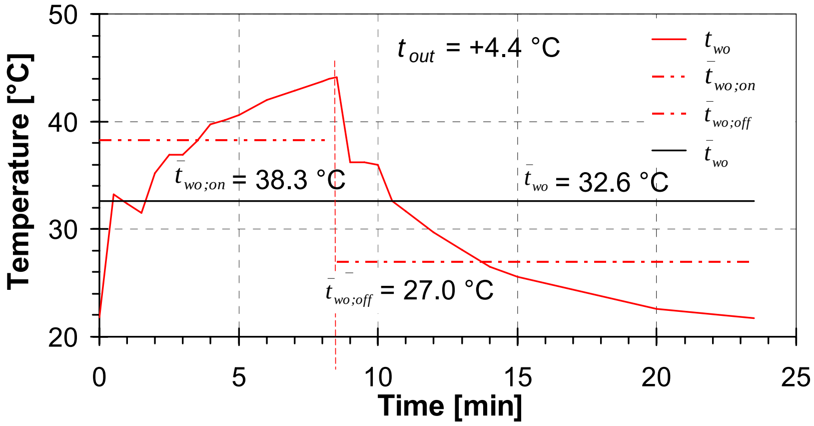

Figure 3: The figure shows an example of how the supply temperature to the heating system varies with time when the heat pump is running in intermittent operation (Fahlén 2004).

The heat pumps are connected in parallel to the heating system, as shown in Figure 4. The following conditions apply for the analyses in this chapter:

- A space heating demand is present until the outdoor air temperature exceeds +11 °C.
- The calculations were made in steps of 24 hours under which the outdoor climatic conditions were considered constant
- The building has an annual space heating demand of 25 MWh per year and the average annual outdoor air temperature is +6 °C.
- The heating system, which in this chapter will be a radiator system, supply and return temperatures were 55/45 at the design outdoor air temperature (DOT)
- The radiator system is designed according to manufacturer data (Thermopanel 2006)
- The pumps of the heat pump system, i.e. the pump for the bore hole and the condenser water pump, are running simultaneously with the compressor. The pump for the space heating system is in continuous operation throughout the heating season
- The temperature of the bore hole was calculated using a relation between outdoor air temperature and bore hole temperature measured by (Fahlén 2002)
- When the heat pump can not match the load, the lack in capacity is supplied by an electric resistance heater with an efficiency of 95 %.
- The controller dead band, i.e. the temperature difference between compressor start and compressor stop is 5 K
- The thermal inertia of the condenser is neglected

The Seasonal Performance Factor for the heat supply systems is defined as:

$$SPF_{hs} = \frac{Q_{hp} + \eta_{sh} \cdot W_{sh}}{W_{comp} + W_{bp} + W_{wp} + W_{radp} + W_{sh}} \qquad \text{(Eq. 1)}$$

where:
Q_{hp}: Heat supplied by the heat pump [kWh]
η_{sh}: Efficiency of supplementary heater [-]
W_{sh}: Total drive energy to the supplementary heater [kWh]
W_{comp}: Drive energy to the compressor [kWh]
W_{bp}: Drive energy to the brine pump [kWh]
W_{wp}: Drive energy to the condenser water pump [kWh]
W_{radp}: Drive energy to the radiator pump [kWh]

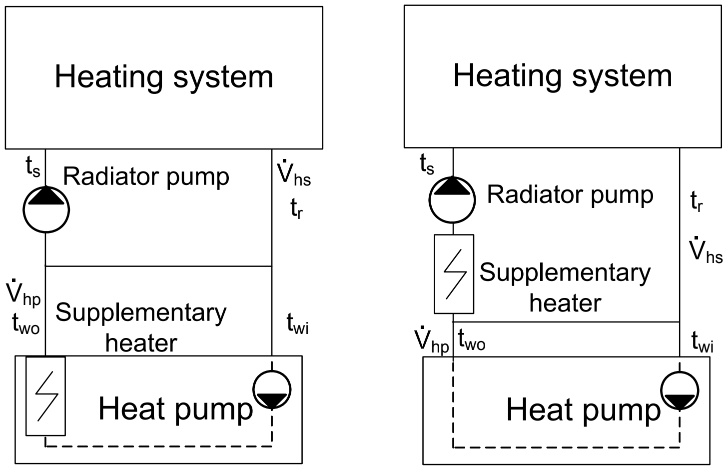

Figure 4: The heat pump is connected in parallel with the heating system; either with a supplementary heater included in the heat pump unit (left) or with an external supplementary heater (right).

3.2 Results

Figure 5 show how the SPF varies with the capacity coverage of the heat pump ($\dot{Q}_{hp} / \dot{Q}_{bldg}$). The capacity coverage is related to the heat pump's capacity at the nominal speed which for the variable-speed heat pump is 80 Hz. Due to the possibility of operating the compressor up to 120 Hz, the variable-speed heat pumps X and Z cover the full load at the capacity coverage of 70 %. The intermittently operated heat pump results in a higher SPF_{hs} than both the real variable-speed heat pump, X, and the fictitious variable-speed heat pump, Z. The reason for X being worse is the lower performance at the nominal capacity as pointed out in Figure 1. The reason for Z being worse is the drive power used by the pumps as will later be discussed, and the fact that the steady-state model overestimates the performance of heat pump Y.

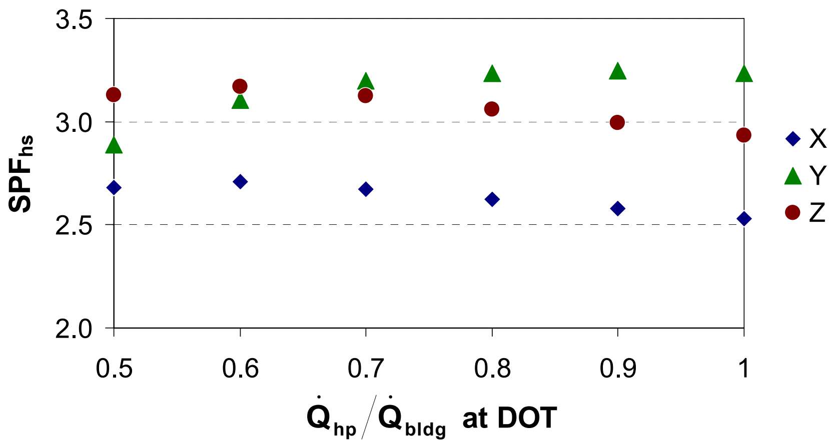

Figure 5: The diagram show the Seasonal Performance Factor for the heat supply system as a function of the relative capacity at design conditions for the three heat pumps. The results are based on the steady-state model.

Figure 6 show a comparison between the steady-state model and the transient model. The circumstances are the same as for Figure 5. From the figure it shows that the SPF_{hs} is practically unchanged for the two variable-speed heat pumps, but for the intermittently controlled heat pump, Y, the SPF_{hs} is reduced. For the full covering heat pump (capacity coverage of 100 %) the SPF_{hs} is reduced by 7 % when using the transient model compared to the steady-state model. The reason for the lower efficiency for heat pump Y when using the transient model is illustrated in Figure 7. It shows that the average supply temperature during compressor operation, $\bar{t}_{wo;on}$, is higher for the transient model than for the steady-state model, as expected. The increased supply temperature results in a higher condensation temperature and thus a lower COP and lower seasonal performance factor.

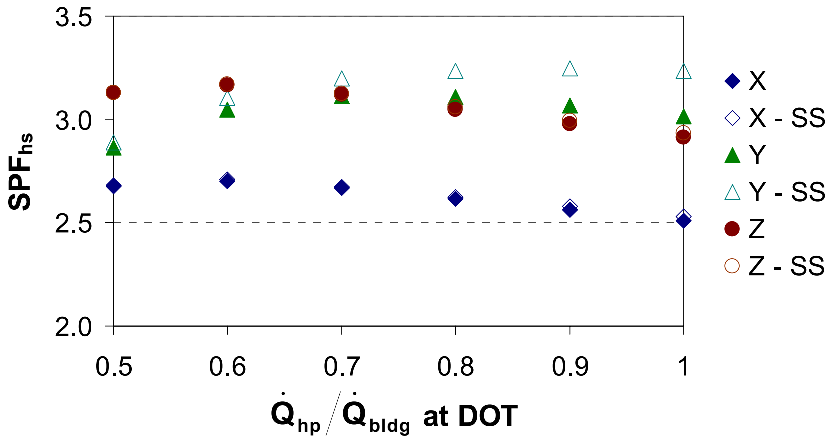

Figure 6: The diagram show a comparison between the results using the steady-state model and the transient model.

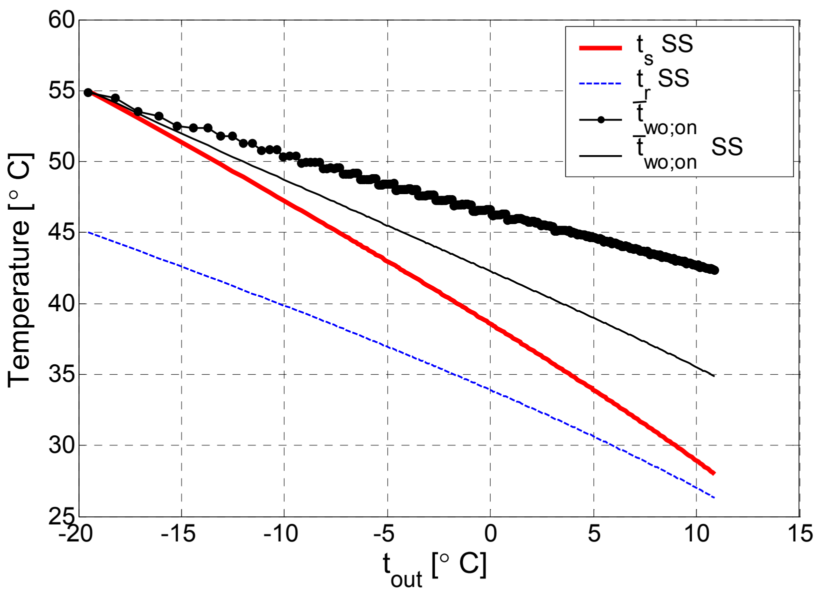

Figure 7: The figure show the difference in the supply temperature from the heat pump depending on if the steady-state model or the transient model is used.

Effects such as different thermal inertias, lag time, controller dead band, type of heat emitter (i.e. floor heating, radiator or fan-coil) and temperature of the heating system have been studied (Karlsson 2007). The parameter having the largest impact on the seasonal performance factor is the temperature in the heating system, which is well known. Changing the temperature at design conditions from 55 °C to 35 °C increased the SPF_{hs} by 30-35 %. Beside the temperature of the heating system the inclusion of thermal inertia in the calculation model is the parameter having the greatest impact on the SPF_{hs}, the effect of different inertias is smaller. The other parameter having an effect on the seasonal performance to an extent greater than a few percent is the lag time. The lag time is the time it takes for the water in the heating system to travel once through the heating system. The lag time reduces the average temperature during compressor operation and thus increases the SPF_{hs} of an intermittently operated heat pump. For a lag time of 10 minutes the SPF_{hs} increased by approximately 5 % for heat pump Y.

4 OPTIMIZED OPERATION OF VARIABLE-SPEED PUMPS

As implied in the previous chapter the pumps used for transporting the heat transfer media to and from the heat pump use a considerable amount of the drive energy for the heat pump system, especially for variable-speed heat pump systems. As the pumps are operated simultaneous with the compressor, the pumps will have longer running times when used in a variable-speed operated heat pump system. Thus, if the capacity of the pumps is not controlled they will constitute an increasing part of the total energy use when the capacity of the compressor is reduced. Figure 8 show how the drive energy is distributed among the components of heat pump system X. At the capacity coverage of 100 %, the pumps use 23 % of the total drive energy. For the intermittently operated heat pump system, Y, the corresponding value is 15 % (Karlsson and Fahlén 2007). Thus the SPF_{hs} decrease by 8 % because of the longer running hours with non-controlled pumps for heat pump X and Z. This chapter will therefore deal with the issue of which gain there may be of using variable-speed pumps and how these can be controlled.

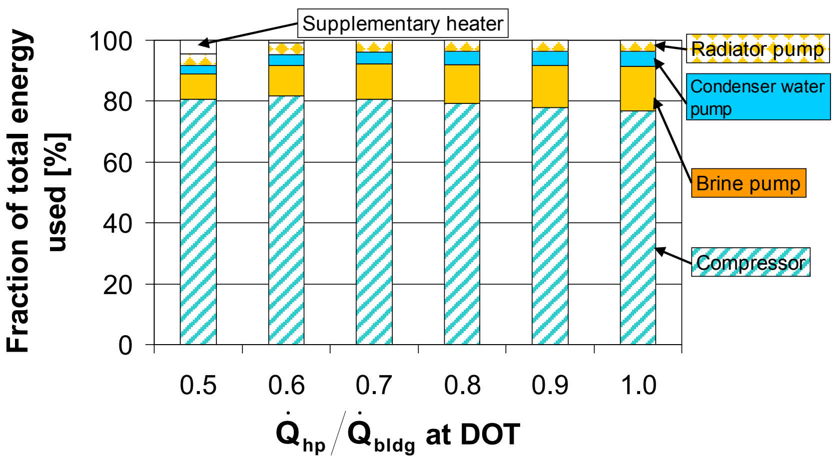

Figure 8: The diagram show the annual drive energy to the heat pump system divided upon the different components. The capacity coverage is given for the nominal compressor frequency.

4.1 Method

An increased heat transfer medium flow through a heat exchanger increases the heat transfer coefficient and reduces the required mean temperature difference. With the same inlet temperature and an increased flow rate the required mean temperature difference can be maintained with a lower condensation temperature. For the condenser in a heat pump this will mean a lower average temperature level and thus a lower condensing pressure and reduced compressor work. For the evaporator an increased flow will result in a higher evaporation pressure and consequently a reduced compressor work. The penalty for the increased flows is increased pump work. For each condition there will exist an optimal combination of pump and compressor speeds.

This issue was analysed using computer simulations and input data from measurements (Karlsson, Fahlén et al. 2005; Karlsson and Fahlén 2007), and from manufacturer soft wares. The heat transfer capacities was calculated using (SWEP 2006), and pump efficiencies and capacities was calculated from (Wilo 2005) and (Grundfos 2007). Refrigerant data was derived using Refprop (Lemmon, McLinden et al. 2002). The sink and source sides were analysed separately and the system boundaries are given in Figure 9. When optimising the sink system, the input data are the return temperature from the heating system, the evaporation temperature (-4.5 °C, obtained from measurements), the evaporator superheat (4 K, used in laboratory tests), the required capacity and the temperature difference t_5-t_{wi} = 0.5 K (derived from measurements). If nothing else is stated the heat pump is considered to be connected in parallel to the heating system and thus the circulator in the heat pump only services the condenser and the by-pass line. Similarly, when optimising the heat source system the input data are the return temperature from the bore hole, the temperature of the sub-cooled refrigerant leaving the condenser (t_{wi}-0.5), the condensing pressure and the evaporator superheat (4 K). Heating system return temperature and load, see Figure 10, are taken from the analysis in the previous chapter, for heat pump X. The description of the model can be found in Karlsson (2007). The objective for the optimisation routine was to maximise COP, while supplying the requested capacity and not violating the constraints imposed by limits in compressor and pump speeds. The COP is defined as:

$$\mathrm{COP} = \frac{\dot{Q}_{hp}}{\dot{W}_{comp} + \dot{W}_{p}} \quad \text{(Eq. 2)}$$

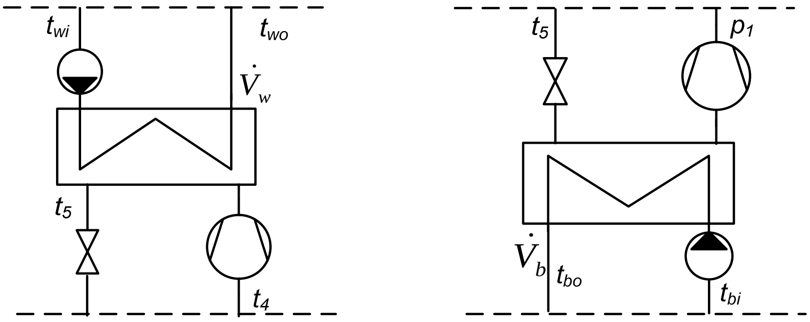

Figure 9: System boundaries for the optimisation of the heat sink system (left) and source system (right)

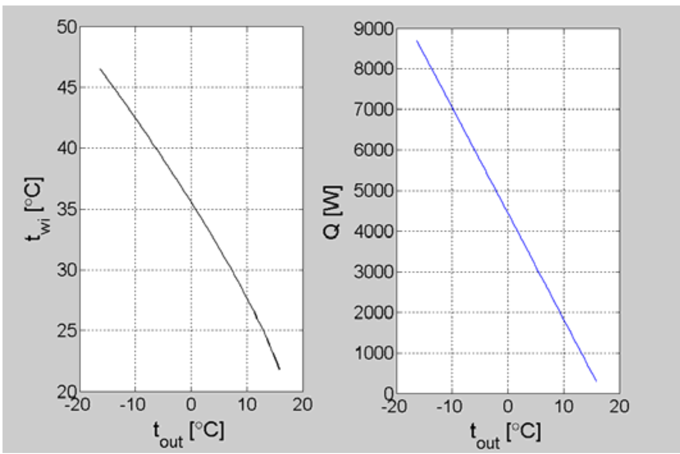

Figure 10: Actual return temperature from the heating system (left), and requested capacity (right). Inputs to the annual optimisation procedure.

The optimisation was made for one year, in steps of one week, and the seasonal performance factor was calculated. The seasonal performance factor is calculated differently in this chapter than in the previous one, and is defined in equation 3 below.

$$SPF = \frac{Q_{hp}}{W_{comp} + W_p} \qquad \text{(Eq. 3)}$$

The optimisation of pump speed can also be made for fixed speed heat pumps and the results for both fixed speed and variable-speed systems are given below.

4.2 Results

The optimization give that the optimum is relatively flat, i.e. it is not very important to find the exact optimum. This can be seen also from Table 3 and Table 4 below. For the sink system the optimization only improve the SPF by 1 % and for the source system 5 %, compared to the variable-speed pump run at fixed speed. However, changing from a standard pump to an efficient variable-speed pump with optimized control improve the SPF by 19 % (1-0.97*0.84) to 24 % (1-0.93·0.82) compared to using standard single speed pumps.

Table 3: Seasonal performance for different pumps in the heat sink system. Pumps 1-4 have PM motors. η_{mean} designates the average efficiency for the current conditions.

	Pump	SPF [-]	W_{wp} [kWh]	W_{comp} [kWh]	W_{tot} [kWh]	$SPF/SPF_{no.1}$ [-]	$\eta_{fwp=fwp;max}$ [%]	η_{mean} [%]
1	Stratos Eco 30/1-5	3.3	190	7520	7710	1.00	32	27
2	Magna 25/60	3.3	220	7510	7720	1.00	32	25
3	Stratos 30/1-6	3.3	200	7500	7700	1.00	35	30
4	Stratos Eco fixed speed	3.2	280	7490	7770	0.99	32	32
5	STAR RS 25/4	3.2	280	7630	7910	0.97	10 (step2 of 3)	10
6	TOP-S 25/7	3.0	890	7410	8300	0.93	19	19
7	TOP-S 25/5	3.1	650	7480	8100	0.95	15	15

Table 4: SPF for the heat source system. Pumps I to III have PM motors. η_{mean} designates the average efficiency for the current conditions.

	Pump	SPF [-]	W_{bp} [kWh]	W_{comp} [kWh]	W_{tot} [kWh]	$SPF/SPF_{no.I}$ [-]	$\eta_{fbp=fbp;max}$ [%]	η_{mean} [%]
I	Stratos 30/1-6	3.7	190	6560	6750	1.00	33	28
II	Stratos 30/1-12	3.7	260	6570	6830	0.99	32	25
III	Stratos 30/1-12 fixed speed	3.5	740	6370	7110	0.95	33	31
IV	TOP-S 30/10	3.0	1930	6320	8250	0.82	19	19
V	TOP-S 25/10	3.1	1730	6330	8060	0.84	24	24

The model used for the optimization is probably not feasible for implementation in a controller for a heat pump. However, considering the curve in Figure 11, for a practical implementation the pump speed could be set as a linear function of the compressor speed. Considering the flat optimum this will only have a small impact on the SPF.

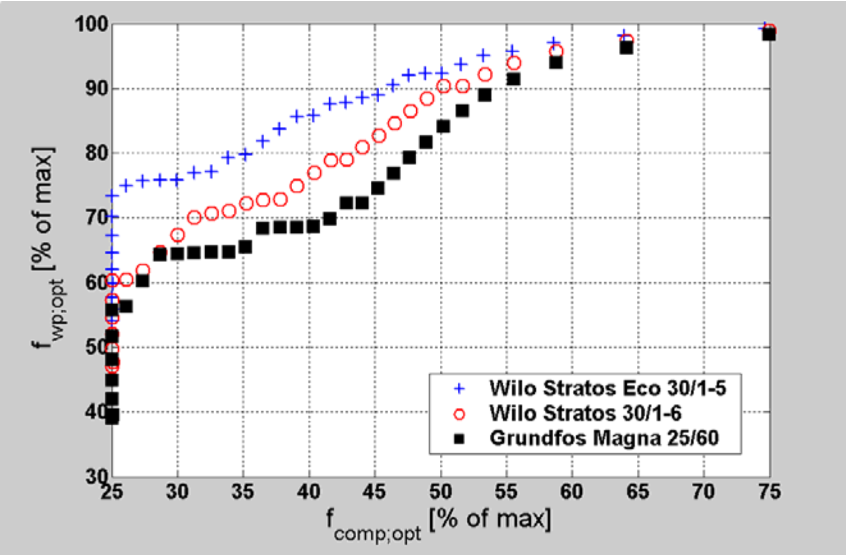

Figure 11: Optimal pump speed as a function of the optimal compressor speed for the sink system

5 DISCUSSION

Aggregating the outcomes of the above investigations give the result shown in Figure 12. When adding first the effect of using the transient model and then the optimisation of the pump speed the two variable-speed heat pumps X and Z becomes more efficient than the fixed-speed heat pump Y. If heat pump Z has both variable-speed pumps and compressor the SPF_{hs} will be 27 % higher than that of a standard fixed speed heat pump. If adding optimised pump operation to the fixed-speed heat pump the SPF_{hs} of heat pump Z is 15 % higher then that of heat pump Y.

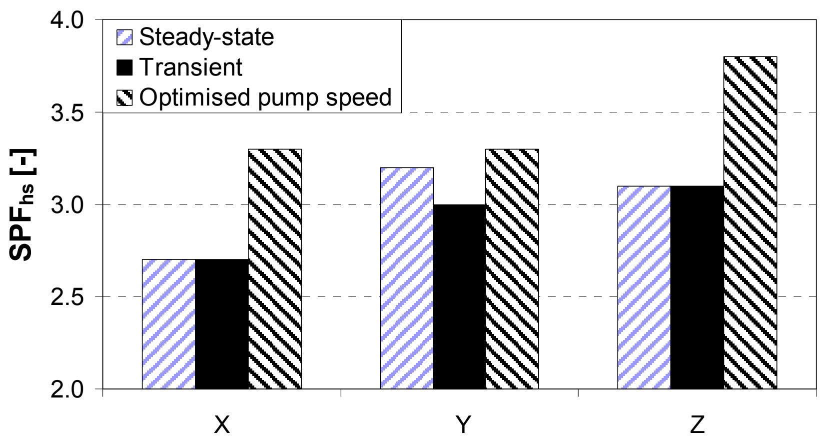

Figure 12: Aggregated results, starting with the steady-state model. Then adding the effect of the transient model and the optimisation of pump speed.

6 CONCLUSIONS

Variable-speed heat pumps can be more efficient than fixed-speed heat pumps but then it is important that the efficiency of compressor and pumps is high and that the capacity of especially the source pump is controlled. Furthermore, it is very important to include the inertia and dynamics of the heating system in order to get a fair comparison between fixed-speed and variable-speed heat pumps.

7 ACKNOWLEDGEMENTS

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