PARAMETRIC STUDY OF ENERGY SAVING POTENTIAL FOR CAPACITY CONTROLLED HEAT PUMPS – OPTIMAL CONDENSATION TEMPERATURE

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ABSTRACT

New controls such as variable speed compressors, pumps and fans as well as electronic expansion valves offer new possibilities for increased energy efficiency of heat pumps. These possibilities are first analysed through a literature survey that is divided into sections which each relates to the before mentioned controls. Thereafter optimisation of the condensation temperature for a brine-to-water heat pump is analysed. This optimal condensation temperature minimises the total power used by the compressor and the heating water circulation pump. The decrease in COP when deviating from the optimal temperature is asymmetric and non-linear. The decrease in COP will be larger for negative than for positive deviations from the optimal condensation temperature.

1. INTRODUCTION

The use of variable-speed compressors, pumps and fans as well as electronically controlled expansion valves is increasing. These new controls give opportunities for increased energy efficiency of heat pumps if they are used in a suitable way. Several investigations have been made in this field but they mainly focus on one of the above mentioned components. In order to optimise the performance of heat pumps the entire cycle should be analysed and the impact of each component on the overall performance should be determined.

MacArthur and Grald (1988), Svensson (1994) and Jakobsen and Skovrup (2001) all present investigations where the overall heat pump efficiency is in focus. MacArthur and Grald (1988) propose a strategy where the Coefficient Of Performance (COP) is maximised while at the same time a comfort index is fulfilled. Svensson (1994) proposes a steady-state energy optimisation strategy for a water-to-water heat pump. Jakobsen and Skovrup (2001) take a somewhat different approach where a static analysis is used to determine the energy optimal set point for a cooling system containing a vapour compression cycle and a cooling tower.

However, these investigations do not discuss all of the above mentioned controls. This will be done in an ongoing project at SP. The objective of this project is to examine the energy saving potential of the new controls mentioned above. The project is further described in the next section. This paper presents the results from a literature survey regarding this particular field. It also presents an analysis of the optimal condensation temperature.

2. INTEGRATED REFRIGERATION MANAGEMENT

Experiences from laboratory tests and investigations from real-world heat pump systems show that it is possible to make the systems more efficient using existing components. Utilising the new electronic controls offered on the market today will further increase this possibility. Heat pump installations are also influenced by a number of operational and safety functions such as defrosting, thermostats and pressure switches. The electronic controls and the operation and safety equipment together constitute a source for information about the status of the process. If this information is collected and used in an intelligent manner it should be possible to optimise the entire process with respect to heating or cooling COP, heating or cooling capacity or some other optimising criterion. Operational and safety functions and information on the operational status of the heat pump may be integrated in this system (Integrated Refrigeration Management (Fahlén 1988) and (Fahlén 1996)).

An ongoing project at SP intends to formulate criteria and survey the existing knowledge regarding optimisation of the operation of refrigeration and heat pump systems. This will be accomplished through the following steps:

- Survey of existing knowledge
- Determination and specification of measuring devices needed
- Development of theoretical models for control and analysing purposes
- Construction and testing of one ground-source heat pump and one air-source heat pump

The following sections present the results from the literature survey and the simple model.

3. LITERATURE SURVEY

3.1 VARIABLE-SPEED COMPRESSORS

Several investigations show that variable-speed compressors lead to more efficient heat pumps than do fixed-speed compressors. The reasons for this can be divided into the following points (Miller 1987; Garstang 1990):

- Better performance at part load
- Fewer cycles on/off
- Reduced need for defrosting
- Reduced supplemental heat

Fixed-speed compressors provide more heating capacity than needed during most of their operating time. This means that the compressor is working with an unnecessarily large pressure difference and thus uses more electric power than necessary. When the compressor speed is adjusted to meet the actual heat load the pressure difference between the high-pressure side and the low-pressure side is reduced and thus the power consumption is reduced. When the rejected power and the heat load match the compressor does not need to be switched on and off as many times as if the compressor was operated with a fixed speed. This switching on and off creates energy losses and reducing them will lead to increased efficiency. How large this benefit will be depends on the frequency span within which the compressor can operate and the range of load variations. A large span means few on and off cycles and thus larger savings. The evaporating temperature increases when the compressor speed is reduced. For air-source heat pumps this will lead to less frost formation and thus a reduced number of defrosts. The opportunities for increasing the compressor speed create opportunities for reducing or completely avoiding the supplemental heat. This opportunity is, like the reduction of the cycling losses, very much dependent of the operating range of the compressor.

3.2 EXPANSION VALVES

In order to prevent liquid hammer the refrigerant flow is controlled in such a way that superheated vapour enters the compressor suction line. A thermostatic expansion valve (TEV) or a capillary tube usually makes such an adjustment. The capillary tube has a very limited operating range. The TEV has a wider operating range and is the dominating flow control device in heat pumps today. From an energy optimisation point of view the superheat should be kept as small as possible. In practice the superheat cannot be adjusted down to zero due to a stability limit called the MSS-line (Minimum Stable Signal; Huelle 1967). This limit describes the interaction between the evaporator and the expansion valve, see figure 1. When the evaporator load is increased the superheat must also be increased in order to maintain a stable superheat control. If the superheat falls into the unstable region droplets of liquid refrigerant may enter the compressor, thus lowering the efficiency and increasing the risk for liquid hammer.

A TEV is set to keep the superheat in the stable region for the maximum evaporator load. If this is made in a proper way the superheat will be kept in the stable region also for lower loads due to the valve characteristic. Figure 1 shows that the TEV for part loads will keep the superheat unnecessarily large. The difference between the characteristic line of the valve and the MSS-line constitutes a potential for energy savings. This can be utilised if the superheat set point can be changed depending on the actual evaporator load. Electronic expansion valves (EEV) provide the means to accomplish this. The electronic control makes it possible to implement control strategies that keep the superheat just above the MSS-line. Jakobs (1989), Jolly et al. (2000) and Finn and Doyle (2000) describe such strategies.

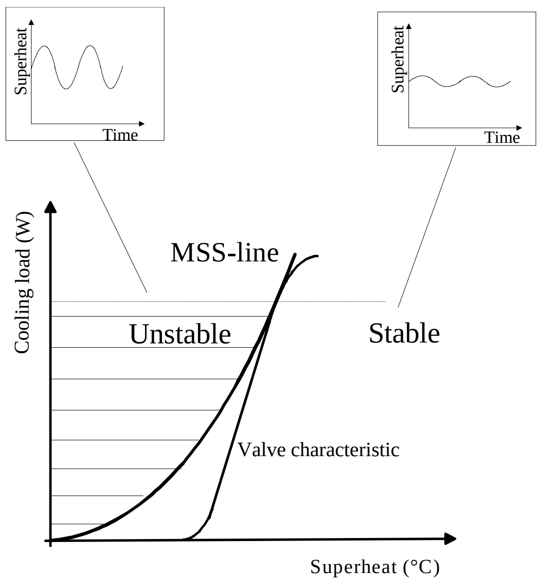

Figure 1. *To the left of the MSS-line the control of the degree of superheat is unstable. By reducing the refrigerant flow and thus increasing the degree of superheat the system falls to the right of the MSS-line and the control gets stable.*

3.3 VARIABLE-SPEED PUMPS AND FANS

The required heating capacity may be transferred to the heating system at different condensation temperatures. A decrease in condensation temperature decreases the power usage of the compressor. A decrease in temperature means that the heating water flow must increase and thus the power consumption of the auxiliary equipment is increased. This relationship has been described for a cold-storage room by Jakobsen and Skovrup (2001) and it is illustrated in figure 2. As can be seen from this figure there exists an optimal condensation temperature that minimises the total power absorbed by the compressor and the auxiliary equipment.

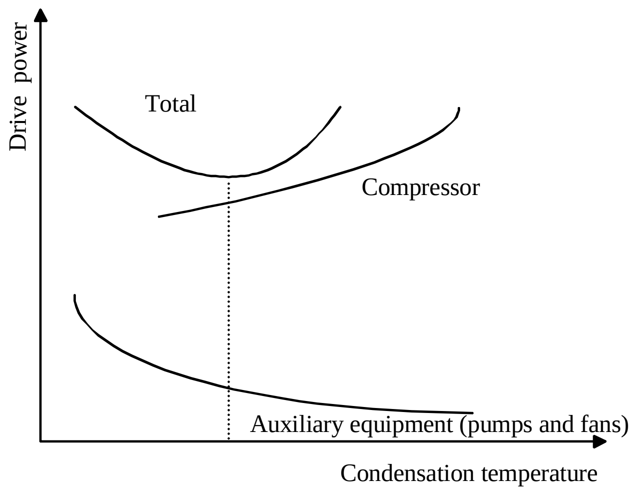

Figure 2. *The optimal condensation temperature minimises the total drive power to the compressor and the auxiliary equipment (Jakobsen and Skovrup 2001).*

4. OPTIMAL CONDENSATION TEMPERATURE

4.1 METHOD

In the preceding sections the possibility for better performance of heat pumps using variable-speed compressors, pumps, fans and electronically controlled expansion valves was indicated. In this study the optimal condensation temperature for a brine-to-water heat pump working with R407C was analysed (see figure 3). The heat pump was considered connected to a typical Swedish villa with a hydronic heating system. A calculation method described by Fehrm and Hallén (1981) was used to determine the heat load and the supply and return temperatures of the heating system for different outdoor air temperatures. The inputs to the method are the DOT (Design Outdoor Temperature), the supply and return heating water temperatures at DOT, the annual mean outdoor temperature and the heat loss factor. For this analysis a so called "55/45" heating system was considered. This implies that the supply temperature is 55˚C and the return temperature is 45˚C at the DOT.

For the analysis a simple steady-state model of the heat pump was implemented in EES (Klein and Alvarado 2001). Only the compressor and condenser with auxiliary equipment were considered. In order to determine the optimal condensation temperature the power used by the compressor and the auxiliary equipment must be determined. These are dependent on the heat transferred through the condenser, which in its turn is dependent both on the refrigerant flow and the water flow.

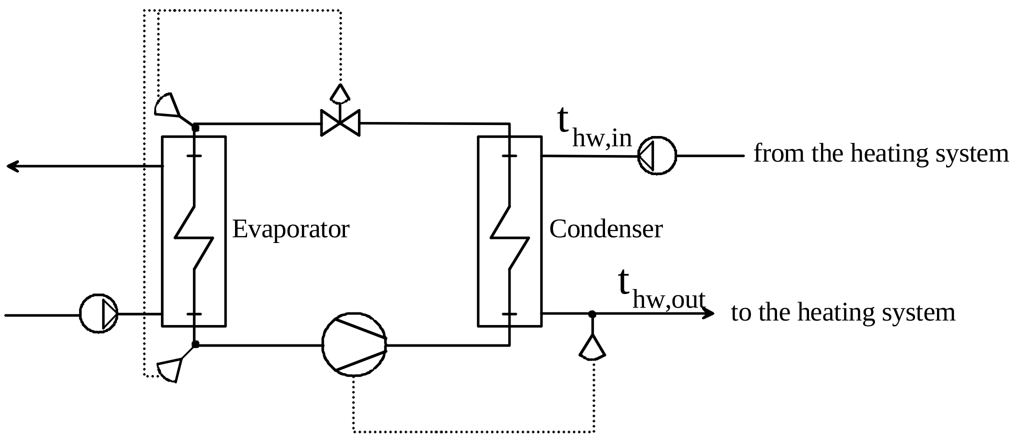

Figure 3. *The heat pump considered in this paper is a brine-to-water heat pump.*

The compressor inlet condition was considered constant; the evaporation temperature was set to -5 ˚C and the degree of superheat was set to 5 ˚C. The compressor power usage was determined from the following expression for a given heat load $\dot{Q}_c$:

$$\dot{Q}_c = \eta_C \cdot \frac{T_c}{T_c - T_e} \cdot \dot{W}_{comp} \tag{1}$$

The Carnot efficiency, η_C, was supposed to be independent within a limited range of variation of the pressure ratio and the rotational speed of the compressor.

A plate heat exchanger was considered as condenser since this is the most common heat exchanger used in brine-to-water heat pumps in Sweden. There are few investigations made public on the field of two-phase heat transfer in plate heat exchangers and no correlation for determining the overall heat transfer coefficient when operating with refrigerants was found. In order to simplify the analysis the heat transfer resistance on the refrigeration side as well as the heat transfer resistance through the plates were considered small and constant. The heat transfer coefficient on the water side was calculated by an equation in the form:

$$Nu = a \cdot \mathrm{Re}^b \cdot \mathrm{Pr}^c \tag{2}$$

The values of a, b and c depend on the specific heat exchanger. For this analysis the values specified by Yan et al. (1999) were used; a = 0.2121, b = 0.78 and c = 0.33. These values are also within the ranges specified by Wang et al. (2000).

To determine the power used by the circulation pump the total pressure drop had to be determined. The total pressure drop consists of the pressure drop through the hydronic heating system and the pressure drop through the condenser. The pressure drop through the heating system was supposed to vary quadratically with the flow:

$$\Delta p_{rad} = k_1 \cdot \dot{V}^2 \tag{3}$$

The pressure drop through the condenser was assumed to vary as:

$$\Delta p_{cond} = k_2 \cdot \dot{V}^{1.8} \tag{4}$$

The power 1.8 was chosen from Wang et al. (2000).

Thus knowing the total pressure drop, the power absorbed by the circulation pump was determined using the following expression:

$$\dot{W}_{hwp} = \frac{\Delta p \cdot \dot{V}_{hw}}{\eta_{hwp}} \tag{5}$$

The efficiency, η_{hwp}, was considered constant and independent of the rotational speed of the circulation pump. The volume flow was determined by the following equations:

$$\dot{Q}_c = \alpha_{hw} \cdot A \cdot \theta \tag{6}$$

$$\theta = \frac{(t_c - t_{hw,in}) - (t_c - t_{hw,out})}{\ln\left(\frac{t_c - t_{hw,in}}{t_c - t_{hw,out}}\right)} \tag{7}$$

$$\dot{Q}_c = \rho_{hw} \cdot \dot{V}_{hw} \cdot c_{p_{hw}} \cdot (t_{hw,out} - t_{hw,in}) \tag{8}$$

Thus, by giving the mean temperature of the heating water in the condenser ($t_{hw,mean} = \frac{(t_{hw,out} + t_{hw,in})}{2}$) and the actual heat load as inputs, the total power absorbed by the compressor and the circulation pump could be determined as a function of the condensation temperature.

4.2 RESULTS

A rule of thumb says that an economically optimal heat pump has a relative heat pump size equal to 0.5, i.e. the heat pump delivers half of the heat demand at the DOT.

$$\text{Relative Heat Pump Size} = \frac{\dot{Q}_{c,DOT}}{\dot{Q}_{house,DOT}} = 0.5$$

A heat pump is normally connected to the heating system return line. With a relative size of 0.5 and operating at the DOT in a 55/45 system the heat pump raises the heating water temperature from 45 to 50 °C. Auxiliary heating provides the other half of the temperature rise. Thus the mean temperature of the heating water in the condenser was 47.5 °C and, for the house considered, the power to be delivered was 5.55 kW. The total power usage was determined for different condensation temperatures and an optimal condensation temperature was found, see figure 4. For this optimal condensation temperature the power usage is minimised and thus the

COP is maximised. In this example, where η_{hwp} = 0.3 and η_C = 0.55, the optimal condensation temperature was 50.6 °C.

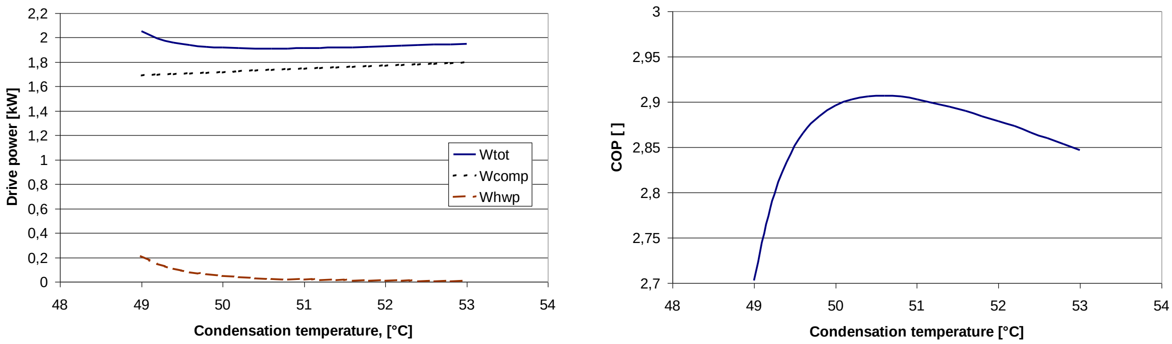

Figure 4. *The optimal condensation temperature minimises the total power usage (left) and thus maximises the COP (right). For this example* $\dot{Q}_c$ *= 5.55 kW,* η_{hwp} *= 0.3,* η_C *= 0.55 and the mean temperature of the heating water was 47.5 °C.*

Making the same analysis for the same heat pump and heating system but for the outdoor temperature +2 °C the optimal condensation temperature was 37.2 °C. The mean temperature of the heating water was 34.5 °C.

A circulation pump with higher efficiency changes the optimal condensation temperature towards lower values as shown in figure 5. The lower condensation temperature leads to a higher COP. Combining these effects in one diagram gives a diagram showing the COP as a function of the circulation pump efficiency, figure 5.

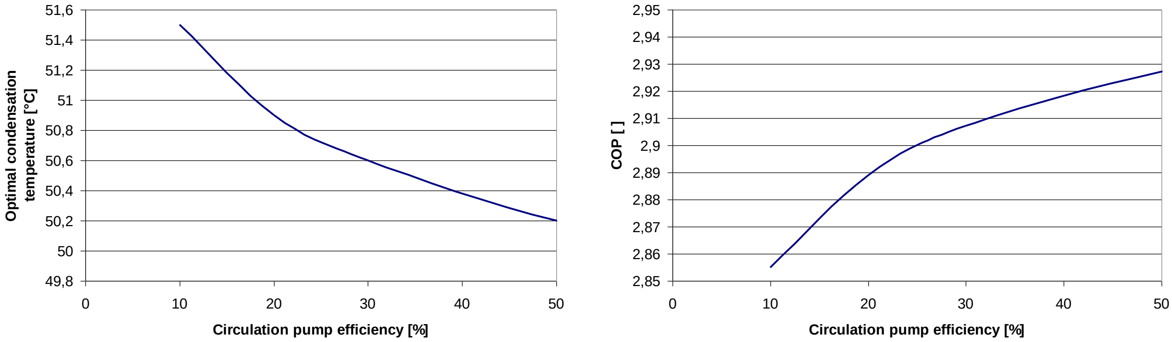

Figure 5. *The diagram to the left shows how the efficiency of the circulation pump affects the optimal condensation temperature. The higher circulation pump efficiency also increases the COP as shown in the diagram to the right.*

As can be seen from figure 5 the change in COP with circulation pump efficiency is quite weak. An increase in η_{hwp} from 20 % to 30 % will increase the COP with less than 1 %. Figure 5 also shows that the same increase in circulation pump efficiency will change the optimal condensation temperature with 0.3 °C. The influence on COP from changes in the brine pump efficiency will, however, be more significant.

Figure 6 below shows how much the COP will decrease when the condensation temperature deviates from the optimal. The figure also shows that the decrease is larger if the condensation temperature is kept below the optimum than if it is kept above. If the condensation temperature is 1 °C below the optimum the COP will decrease with 1.4 %, while if 1 °C above the optimum the COP will decrease only 0.6 %.

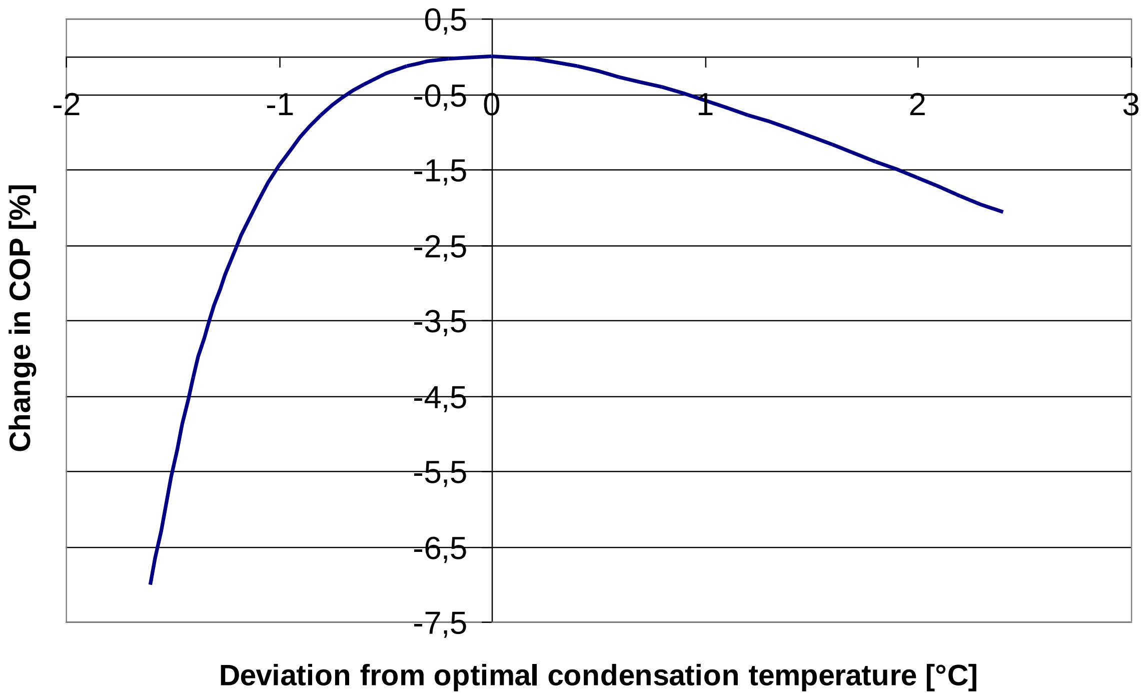

Figure 6. *When deviating from the optimal condensation temperature the COP decreases. The decrease is larger for negatives than for positive deviations.*

5. DISCUSSION

This paper presents the results from an analysis of the optimal condensation temperature. A simple model of a brine-to-water heat pump is used for this purpose. The optimal condensation temperature minimises the drive power for a given heat load and thus maximises the COP. A deviation from the optimal condensation temperature causes a decrease in COP. This relationship is not symmetric; a temperature lower than the optimal temperature will decrease the COP more than the same deviation above the optimal temperature. This is in agreement with the findings of Jakobsen and Skovrup (2001) who analysed a large refrigeration system with a condenser connected to a cooling tower. This relationship will be more pronounced if the circulation pump efficiency is lower than the 30 % used in this example. This will in mostly be the case since small circulation pumps often have poor efficiencies (less than 10 %). The results also indicate that the circulation pump efficiency has a quite low impact on the optimal condensation temperature. This implies that for controlling purposes the circulation pump efficiency does not need to be known with a high accuracy in order to determine the optimal set point. The heat transfer resistance for the condensing refrigerant and the heat conduction in the plates have been neglected in this analysis and in the further work this will be included in order to determine its impact on this analysis.

For the heat transfer correlation in equation 8 the constants have been chosen within the ranges specified by Wang et al. (2000). They investigate the heat transfer for steam condensation in plate heat exchangers and the constants were fitted to the data from these tests. Yan et al. (1999) also investigate the condensation heat transfer for R134a and the heat transfer on the water side was fitted to the test data. Equation 2 is used and the constants fall into the ranges specified by Wang et al. (2000). Thus it is considered that for a typical plate heat exchanger the one-phase heat transfer can be expressed as in equation 2 with constants specified by Yan et al. (1999). For more precise calculations these coefficients must be validated for the specific heat exchanger used.

6. CONCLUSIONS

From the quite simple model of a brine-to-water heat pump, with a plate heat exchanger as condenser, it is concluded that the decrease in COP when deviating from the optimal condensation temperature is asymmetric. A condensation temperature lower than the optimal will decrease the COP more than the same deviation towards higher temperatures. In the analysis the heat transfer resistance on the refrigeration side and through the plates have been neglected. The analysis has been made for typical components and is not valid for a specific heat pump. From the literature survey it is concluded that the use of electronic expansion valves and variable-speed compressors, pumps and fans will increase the energy efficiency of heat pumps if used properly.

7. ACKNOWLEDGEMENTS

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8. NOMENCLATURE

COP	Coefficient Of Performance
DOT	Design Outdoor Temperature
EES	Engineering Equation Solver (software)
EEV	Electronic Expansion Valve
MSS	Minimum Stable Signal
TEV	Thermostatic Expansion Valve
A	area
a	constant
b	constant
c	constant
c_p	specific heat capacity
g	acceleration due to gravity
Nu	Nusselt number
Pr	Prandtl number
Δp	Pressure difference
Re	Reynolds number
$\dot{Q}$	heat power
T	temperature, absolute (K)
t	temperature, centigrades (˚C)
$\dot{V}$	volume flow rate
$\dot{W}$	electric power

Greek letters

α	heat transfer coefficient
η	efficiency
θ	logarithmic mean temperature difference
ρ	density

Subscripts

C	Carnot
c	condensation, condenser
comp	compressor
e	evaporator, evaporation
hw	heating water
hwp	heating water circulation pump
in	inlet
out	outlet
mean	mean, average

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