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MODELLING OF AMMONIA HEAT PUMP DESUPERHEATERS

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ABSTRACT

This paper presents a study of modelling desuperheating in ammonia heat pumps. Focus is on the temperature profile of the superheated refrigerant. Typically, the surface area of a heat exchanger is estimated using the Log Mean Temperature Difference (LMTD) method. The assumption of this method is that the specific heat is constant throughout the temperature glide of the refrigerant in the heat exchanger. However, considering ammonia as refrigerant, the LMTD method does not give accurate results due to significant variations of the specific heat. By comparing the actual temperature profiles from a one-dimensional discretized model with the LMTD, it is found that the LMTD method provides a higher temperature difference than the discretised model, and would therefore lead to an underestimation of the needed condenser area.

The lower temperature difference in the discretized model can be compensated for in two ways. The area of the heat exchanger can be increased or the condensation temperature can be raised to achieve the same temperature difference for the discretized model as for the LMTD. This would affect the compressor work, hence the COP of the system. Furthermore, for higher condenser pressure, and thus higher pressure in the desuperheater, a larger deviation between the two temperature difference models is observed. Using the discretization model the number of discretizations to get accurate estimates is found to be 20.

NOMENCLATURE

Parameter	Explanation	Unit
c_p	Specific heat capacity with constant pressure	kJ/(kgK)
ΔT	Temperature difference	°C
Δh	Enthalpy difference	kJ/kg
U	Heat transfer coefficient	W/(m^{2}K)
A	Heat transfer area	m^2
$\dot{Q}$	Heat transfer rate	W
$\dot{m}$	Mass flow rate	kg/s
LMTD	Log Mean Temperature Difference	°C

1. INTRODUCTION

The focus of this study is mainly on the modelling of the desuperheater in an ammonia heat pump. The desuperheater is part of the condenser unit in a heat pump, where the energy from the superheated gas from the compressor is transferred to the secondary media in the heat exchanger. Typically, the desuperheater is a counter flow heat exchanger to achieve the highest outlet temperature on the secondary media. Figure 1 illustrates the process of a simple heat pump in a log(P),h-diagram, where the desuperheater part is marked.

Determining the heat transfer area in a heat exchanger for a specific heat pump is critical for obtaining the desired working conditions. From eqn. (1) the amount of energy transferred ($\dot{Q}$) is dependant of the heat transfer coefficient (U), the heat transfer area (A) and the temperature difference (ΔT).

A larger heat exchanger area than needed gives a higher production cost and therefore a weaker business case.

A smaller heat exchanger area than needed results in a higher condensing temperature of the heat pump (which gives a higher ΔT) to transfer the same amount of energy with unchanged conditions on the secondary media. Consequently, a smaller area also results in a weaker business case due to larger energy consumption of the heat pump. Furthermore, situations exist where a higher condensing temperature is not a possible solution. This would result in either a lower heat capacity of the heat pump or a higher flow and a lower outlet temperature on the secondary media to compensate for the smaller area.

The prediction of the UA-value is important to obtain the best possible business case and therefore it is essential to have correct information about the temperature differences in the heat exchanger. Often calculations are based on a lumped parameter approach. This results in that ΔT is described by the Log Mean Temperature Difference (LMTD)- which is a method that takes into account the fact that the energy transferred per unit area is not constant through the heat exchanger due to varying temperature differences. This method helps determining the exact needed area of a heat exchanger to transfer a certain amount of energy, assuming a constant specific heat at constant pressure, c_p, for both media. (Incropera *et al.*, 2005).

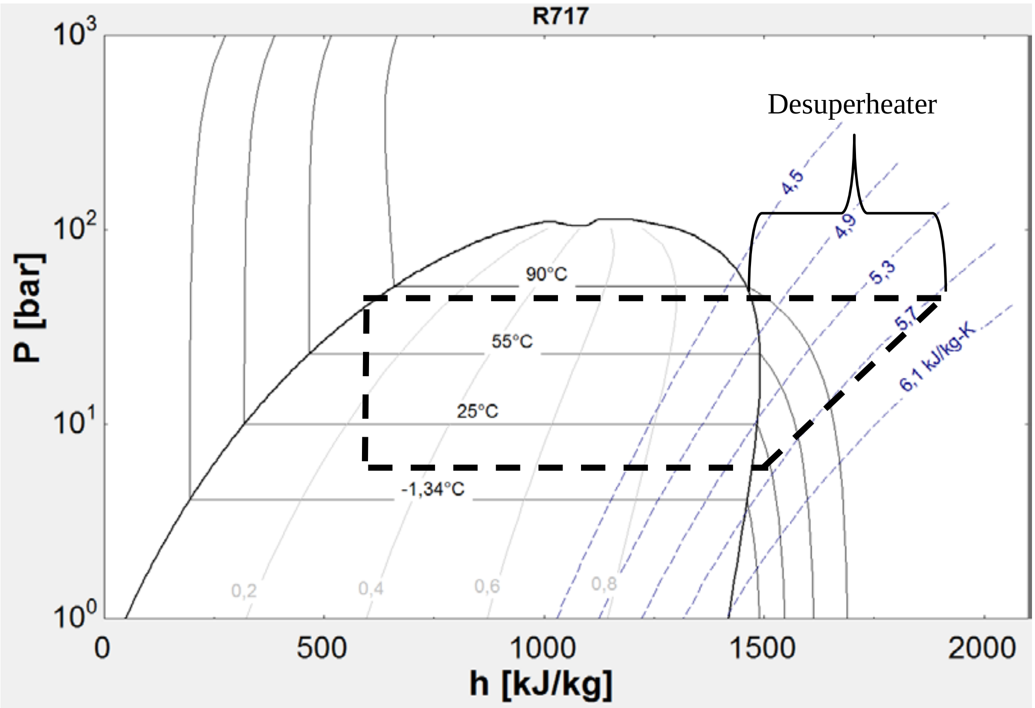

Figure 1. Log(P),h-diagram with a principal sketch of a simple heat pump process. The desuperheating is from the outlet of the compressor to the start of the two phase area as marked in the sketch.

Bell (Bell, 1972) cautioned that the use of LMTD could be invalid in some cases for desuperheating, which will have a direct influence on the prediction of heat exchanger area. The reason for the invalidation of LMTD is that the refrigerant may not have a constant c_p value in the superheated region. Instead of using the LMTD-method it is possible to discretize the desuperheater and use arithmetic mean temperature differences for every discretization to calculate a more accurate UA-value for each step. This will give a better profiling of the temperatures through the heat exchanger and therefore a more accurate estimate UA-value needed to transfer the desired amount of energy. This method is described further in section 2.METHODS.

Heat pumps are designed to have a relatively low condenser pinch point temperature difference to ensure a high COP. When temperature differences between the two media are relatively low the change in c_p for one of the media will have a relatively high influence on the overall temperature differences and therefore the deviation from LMTD to a discretized model will be significant.

LMTD is a form of average temperature difference between the media through the heat exchanger, whereas the discretized model gives exact temperature differences for every discretization. Therefore, it has been chosen to compare the UA-values of the desuperheater from the two different approaches to illustrate the differences of the two approaches.

A case study based on heat pump for application in a low temperature district heating net is applied for illustrating the potential of using a more accurate temperature difference in the heat transfer calculation. The heat demand for this specific case is 2792 kW, out of which 500 kW is transferred in the desuperheater.

The conditions set for this investigation is that the desuperheater is a counter flow heat exchanger with water and R-717 and the inlet temperature of the water is 5 °C below the condensing temperature of R-717. This is illustrated in Figure 2.

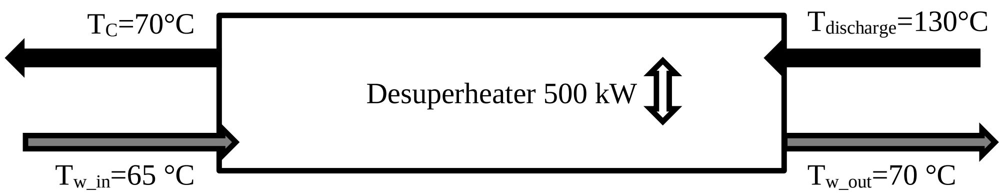

Figure 2. Conceptual sketch of desuperheater used for discretizing

The total UA-value calculated from eqn. (1) with the LMTD method is 22.59 kW/K whereas the total UA-value calculated with the discretized model is 24.94 kW/K. The two methods deviate from each other and this deviation is investigated further in detail for various conditions of the heat pump.

2. METHODS

The energy transfer rate in a heat exchanger is described as

$$\dot{Q} = UA \cdot \Delta T \tag{1}$$

where U is the heat transfer coefficient [W/(m^{2}K)], A is the heat transfer area [m^2] and ΔT is the driving temperature difference between the two media in the heat exchanger [°C], (Incropera *et al.*, 2005).

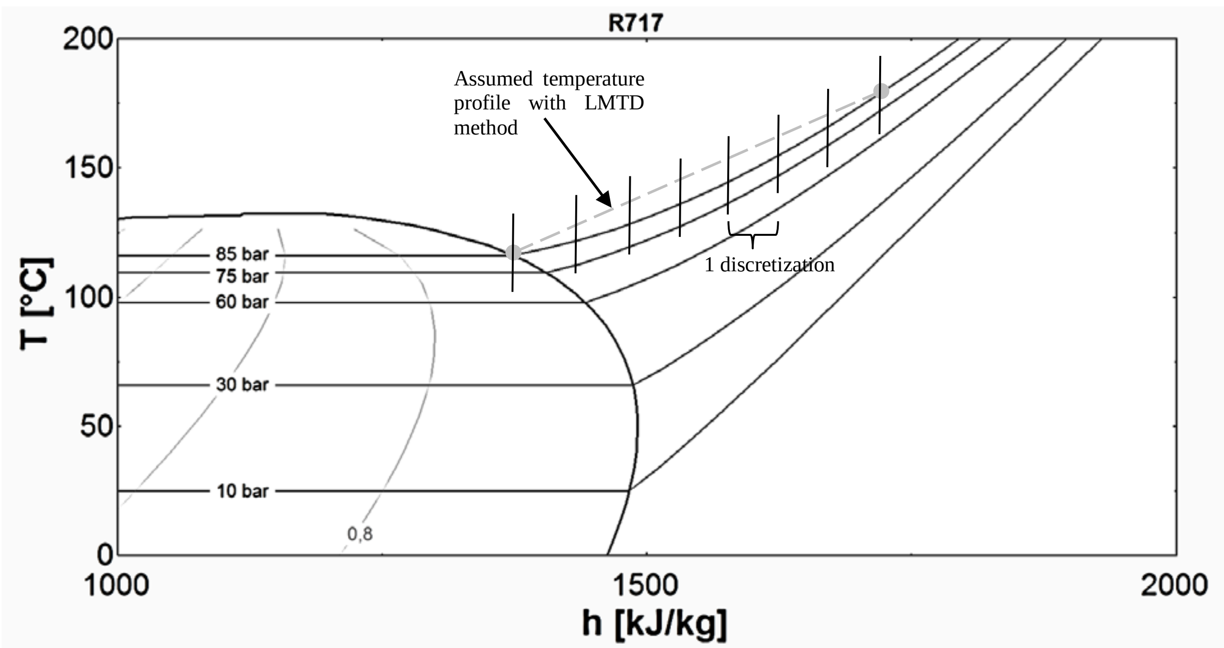

Figure 3. (h,T)-diagram showing nonlinear temperature curves in the gas area of R-717 indicating a varying c_p value in this area

Considering ammonia (R-717) as refrigerant, inaccuracies occur when using LMTD for calculating the temperature difference of the desuperheater. In order to keep the condensation temperature at the cycle design point, this results in a change of the area needed to transfer the heat. The reason for the inaccuracies is, as described in the introduction, that LMTD assumes constant c_p for both media through the heat exchanger. Recalling the definition of c_p:

$$c_p = \left(\frac{\partial h}{\partial T}\right)_p \tag{2}$$

It may be seen from Figure 3 that c_p is not constant in the superheated region for R-717, especially at high pressures.

The dashed grey line in Figure 3 indicates the assumption of the temperature profile using LMTD for a condensing pressure of 85 bar. Figure 3 shows that using the assumptions of LMTD gives a higher temperature difference between the two media in the heat exchanger than the actually available temperature difference, which is illustrated by the curved black line between the grey dots. Considering eq. (1) this results in an underestimation of the surface area needed for transferring a certain heat rate. Furthermore Figure 3 indicates the concept of discretizing the model by splitting the desuperheater into minor parts, in this case 7 parts split by vertical black lines. Even though a constant temperature profile is assumed in each discretization the temperature profile becomes much closer to the actual profile.

2.1 LMTD

LMTD depends on both inlet and outlet temperatures of the heat exchanger.

The inlet temperature of the refrigerant (the discharge temperature from the compressor) is given as T_1. The outlet temperature, T_2, of the refrigerant is the condensing temperature.

The pinch point temperature in the condenser is defined to be 5 °C which means that the inlet temperature of the water to the desuperheater is $T_{W_1} = T_c - 5°C$. The outlet temperature of the water, T_{W_2}, is equal to the condensing temperature of the refrigerant.

The LMTD is calculated as follows:

$$LMTD = \frac{\Delta T_{in} - \Delta T_{out}}{\ln(\frac{\Delta T_{in}}{\Delta T_{out}})} \tag{3}$$

Where $\Delta T_{in} = T_1 - T_{W_2}$ and $\Delta T_{out} = T_2 - T_{W_1}$.

For a discharge temperature from the compressor of 130 °C and the condensing temperature of 70 °C the LMTD would be:

$$LMTD = \frac{60°C - 5°C}{\ln(\frac{60°C}{5°C})} = 22.13°C \tag{4}$$

2.2 One-dimensional Discretization

The lumped parameter approach may be compared to a model based on one-dimensional discretization of the heat transfer calculation. When discretizing, the heat exchanger it is split into n control volumes which are equidistant in heat transfer rate between the two media. This also means that the discretization is equidistant in enthalpy difference of each fluid.

$$\dot{Q}_\iota = \dot{m} \cdot \Delta h_i, \qquad \text{for } i = 1, n \tag{5}$$

Where Δh is the difference in enthalpy of each control volume.

The arithmetic mean temperature difference is used to estimate the driving temperature difference for each discretization in the counter flow heat exchanger.

$$\Delta T_i = \frac{T_{ref_{in,i}} - T_{ref_{out,i}}}{2} - \frac{T_{w_{out,i}} - T_{w_{in,i}}}{2} \tag{6}$$

Where ΔT_i is the arithmetic mean temperature difference corresponding to the ith discretization, $T_{ref_{in}}$is the temperature of the refrigerant into the control volume, $T_{ref_{out}}$is the temperature of the refrigerant out of the control volume, $T_{w_{in}}$ is the temperature of the water into the control volume, $T_{w_{out}}$ is the temperature of the water out of the control volume.

From ΔT_i it is possible to calculate the corresponding UA-value to transfer the desired amount of energy from the following equation.

$$\frac{\dot{Q}}{n} = UA_i \cdot \Delta T_i \tag{7}$$

$$UA_{total} = \sum_{i=1}^{n} UA_i \tag{8}$$

Pressure losses are neglected. It is expected that this will not have a significant influence due to the relatively small pressure losses found in common desuperheater designs.

2.3 Model implementation

A model of a desuperheater was implemented in EES (Engineering Equation Solver, see references) for both the lumped parameter approach and the discretized formulation.

3. RESULTS

From eqn. (1), the needed UA-value for the desuperheater to transfer 500 kW of energy by using LMTD and the same boundary conditions as in section 2. METHODS is

$$UA = \frac{500\ \mathrm{kW}}{22.13\ °C} = 22.59\ \mathrm{kW/K} \tag{9}$$

The total UA-value with the discretized model for *n*=20 is 24.94 kW/K which is 10.4% higher than for the LMTD method.

3.1 Number of discretizations

A low number of discretizations will not be very accurate, but when letting the number of discretizations approach infinity the temperature profile will go towards the actual temperature profile and thus the calculated surface area will tend towards the actual needed surface area to transfer the heat.
Using an infinite amount of discretizations requires an infinite amount of calculations. To save calculation time it is investigated how much the total UA-value varies when using different discretizations for different desuperheater conditions.

The number of control volumes is varied for condensing temperatures between 60 °C and 130 °C and discharge temperatures from the compressor between 90 °C and 210 °C. The change of temperature difference is expressed by the UA-value as shown in Figure 4, which shows the total UA-value as a function of the number of control volumes. It is found from analyzing condensing temperature in the interval [60 °C; 130 °C] and discharge temperatures in the interval [90 °C; 210 °C] that the calculated UA-value converges to an accuracy of minimum 99.5% at 20 control volumes for all calculations. Hence, it seems to be sufficient to use 20 discretizations to investigate the deviations from calculation of the UA-value by the LMTD approach.

3.2 Comparison of LMTD and discretized model

Twenty control volumes and simple arithmetic mean temperature differences for every control volume are used to predict the needed UA-value for a desuperheater to transfer 500 kW from R-717 to water with an inlet temperature of water at 5 °C below the condensing temperature of R-717 and an outlet temperature of the water equal to the condensing temperature of R-717. Due to the temperature profile of R-717 shown in Figure 3, the calculated UA-value will be higher with the discretized model than when using the LMTD approach. Figure 5 shows the deviation in calculated UA-value for various heat pump conditions.

$$\%\ deviation\ in\ UA\ value = \frac{UA_{discretization} - UA_{LM}}{UA_{LM}} \tag{10}$$

Where $UA_{discretization}$ is the UA value calculated from 20 discretizations and UA_{LM} is the UA-value calculated from the LMTD method for the same heat pump conditions.
The discharge temperature for the compressor is again varied from 70 °C to 210 °C and the condensing temperature is varied from 60 °C to 130 °C.

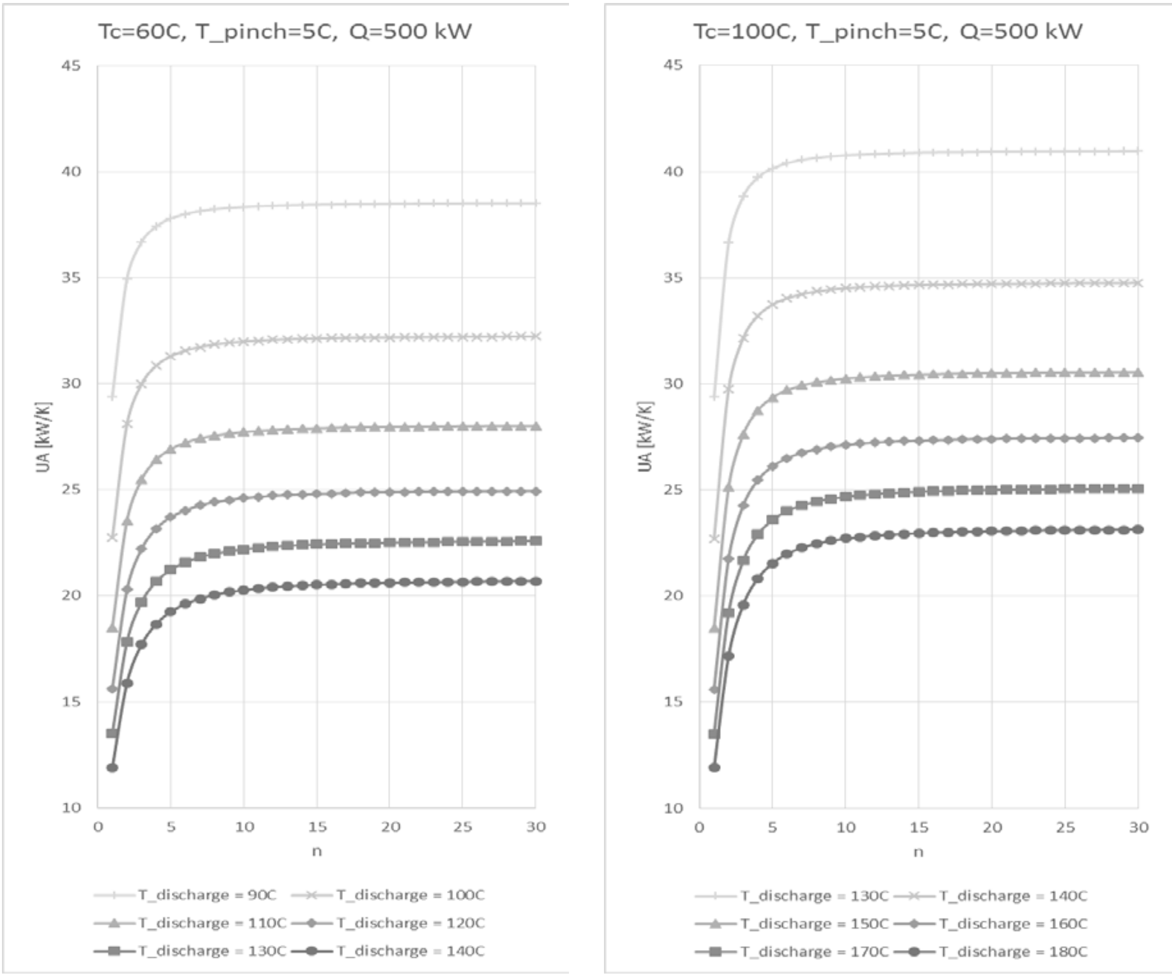

Figure 4. UA-value as a function of discretization for constant desuperheater capacity of $\dot{Q}$=500 kW

The results show that the deviation in calculated UA-value becomes significantly higher for higher condensing temperature/pressure. The typical discharge temperature for heat pump applications does usually not exceed 130 °C due to design limitations for valves and oil. Figure 5 shows the deviation in calculated UA-values, and illustrates that the required UA value is 8-12% higher based on the discretization method than with the LMTD method for discharge temperatures of around 130 °C. For higher discharge temperatures, significantly higher differences are found.

Ommen *et al*. (2014) is considering discharge temperatures up to a temperature of 180 °C. In this case it can be seen from Figure 5 that the deviation from the LMTD method results in an increase of the UA value of 10% to 60% to compensate for the lower temperature difference.
Results in Figure 5 are derived from a pinch point temperature of 5 °C. If the pinch point temperature becomes smaller, as is the case for many heat pumps, the deviation of the two models becomes larger as the relative difference in the temperature profiles for the two media becomes more significant for R-717.
It should be noted that the deviation increases with pressure. This indicates that the pressure loss through the desuperheater will make the deviation a little less significant. Pressure loss is usually relatively low in a desuperheater and therefore the minor influence of this is neglected in this analysis.

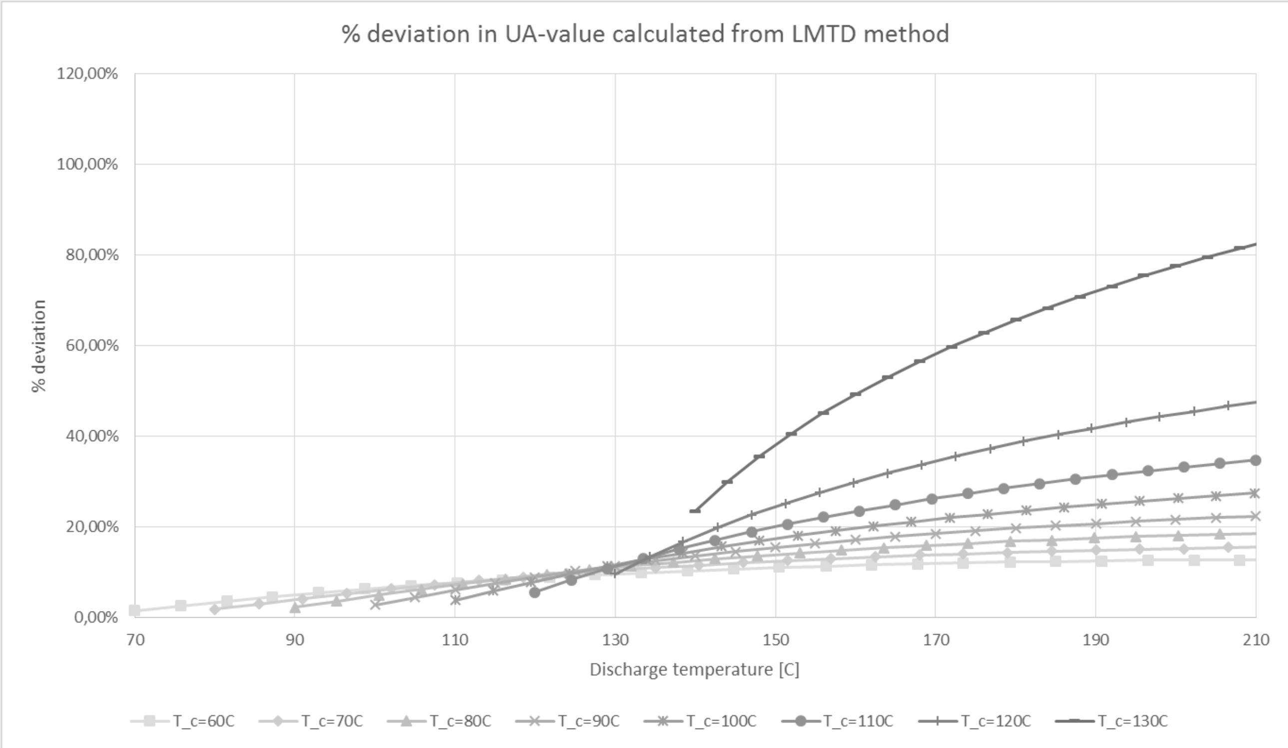

Figure 5. Deviation of UA-values compared to the LMTD method for various conditions of the desuperheater. The different lines in the graph correspond to different pressures/condensing temperatures of R-717 (T_c)

3.3 Other refrigerants

The investigation of ammonia shows that using the LMTD method results in an inaccuracy. Considering the results of the LMTD and the discretization models at constant condensation pressure an increase in UA-value is found due to an increase in c_p when gas is superheated. Furthermore, the results show that the deviation in c_p is higher for high pressure.

The change in c_p is investigated for different refrigerants to evaluate the impact of using the lumped parameter approach.

Table 1 gives an overview of the change in c_p from in- to outlet of the desuperheater. The inlet temperature in the desuperheater is 180 °C (Ommen *et al.*,2014) and the outlet temperature is defined to be the condensing temperature.

In Table 1 the percentage change in c_p is calculated by

$$\%\ change\ in\ c_p = \frac{c_{p_{out}} - c_{p_{in}}}{c_{p_{out}}} \tag{11}$$

Where $c_{p_{out}}$ is the c_p value at the condensing point and $c_{p_{in}}$ is the c_p value at 180 °C at the pressure set in the desuperheater. The value of 180 °C has been chosen only to illustrate the difference in c_p values for various refrigerants and should not necessarily be seen as the realistic input temperature for a desuperheater.

T_C\media	R-717	R-134a	R-600a	R-290
60	0,37	0,17	-0,13	0,13
70	0,41	0,27	-0,07	0,25
80	0,46	0,41	-0,01	0,43
90	0,52	0,61	0,06	0,70
100	0,59	0,96	0,14	
110	0,67		0,26	
120	0,78		0,43	

Table 1. percentage change in c_p value ((c_p _out- c_p _in)/ c_p _out) in the desuperheater.

Table 1 indicates that the assumption of constant c_p in the superheated region is not completely valid for any of the refrigerants in the superheated area. The investigation also shows the tendency that with increasing pressure the deviation between the real c_p and the assumption of constant c_p increases. It is also found that the c_p -value decreases when gas is superheated for relatively low condensing pressures of R-600a. This has the consequence that by using LMTD the heat exchanger will be slightly over-dimensioned.

4. DISCUSSION

The desuperheater only transfers a part of the heat in the condenser of a heat pump, but the heat transfer coefficient is low in the gas phase and thus the desuperheater has a relatively high share of the required heat transfer area.
If the LMTD calculation is used to dimension the desuperheater, the heat pump may still function, but the performance will be less than expected as the condenser pressure will increase to compensate for the missing heat transfer area.
To calculate the correct surface area needed to exchange energy it is important to fully understand the variation of the U-value through the desuperheater. The U-value can be obtained from correlations, e.g., for plate heat exchangers by (Martin, 2010) but it is important to also discretize this value to ensure that correct results are obtained together with the correct temperature profiles. In this respect, the accuracy of the heat transfer correlation may have significant impact. Also condensation on the heat transfer surface in the superheated region becomes important to predict the needed surface area (Kondou C. and Hrnjak P., 2012).
The same phenomenon as discussed in this for desuperheaters of subcritical condensers is seen in gas coolers with CO_2 where a discretized model is usually needed for analysing the heat transfer.
Similar observations may be found for other applications, e.g., in steam power plants where the water vapour is superheated by combustion products. In superheater sections a varying specific heat capacity will result in inaccuracies if the LMTD method is used.

5. CONCLUSIONS

The temperature profile of ammonia in a desuperheater has been investigated to illustrate the importance of using a discretized model in dimensioning instead of the lumped parameter LMTD method. It has been found that the assumption of constant c_p in the superheated region is not valid for ammonia as well as other refrigerants. The variation of c_p has a direct influence on the estimated heat exchanger area, hence the discretized model is important for design purposes.
Twenty discretization steps are sufficient to get a more accurate calculation of the UA-value needed to transfer a certain amount of energy in the desuperheater of an ammonia heat pump.
The deviation from using the common LMTD method is approximately 10 % in UA-value for common conditions for a heat pump with a discharge temperature from the compressor around 130 °C. The deviation becomes larger with rising pressure and the deviation in the UA-value for very high condensing temperatures may reach 50 %.
It is found that c_p is varying with up to almost 100% in the superheated region for other common refrigerants than ammonia and hence discretization is recommended to ensure accurate calculations for all refrigerants.

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