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COMPARISON OF HEAT TRANSFER FOR R22 AND SOME ALTERNATIVES IN A 25 MW SHELL-AND-TUBE CONDENSER

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ABSTRACT

Calculations have been carried out for a 25 MW horizontal shell-and-tube condenser with shell-side condensation, part of a heat pump for district heating. Due to a coming ban on refilling R22, which is used in some district heating heat pumps, calculations of heat transfer were carried out for R22, R134a and four zeotropic refrigerant mixtures, to see how well they perform. It was found that some of the refrigerant mixtures are competitive to R134a as replacement for R22.

Two comparisons were made, one for fixed duty and one for a fixed heat pump system. In both comparisons some of the mixtures seem to have better performance than R134a when considering duty and heat transfer whilst maintaining lower flow rates.

NOMENCLATURE

A_i	-	inside surface area of tube [m^2/m]
A_o	-	outside surface area of tube [m^2/m]
A_{tot}	-	total heat transfer area [m^2]
c_p	-	heat capacity [J/mole]
h_{fg}	-	specific enthalpy of vaporization [J/kg]
$\dot{n}$	-	flux of comp. i towards interface [mole/s m^2]
$\dot{q}$	-	heat flux [W/m^2]
$\dot{q}_L$	-	sensible heat of condensate [J/kg K]
R_t	-	fouling resistance on tube side [m^2K/W]
T_b	-	bulk gas temperature [K]
T_c	-	coolant temperature [K]
T_i	-	interface temperature [K]
U	-	overall heat transfer coefficient [W/m^2K]
α_c	-	heat transfer coefficient on coolant side [W/m^2K]
α_g	-	heat transfer coefficient on gas side [W/m^2K]
$\alpha_g^\bullet$	-	α_g corrected for mass transfer [W/m^2K]
α_L	-	heat transfer coefficient from condensate interface to coolant [W/m^2K]
α_l	-	heat transfer coefficient in condensate film [W/m^2K]
δ_w	-	thickness of tube [m]
ε	-	correction factor for heat balace
λ_w	-	thermal conductivity in tube [W/mK]

1. INTRODUCTION

In Sweden there is a ban on refilling heat pump systems with R22 in the year 2002, which generates the problem of finding well-working substitutes. In high temperature applications, such as heat pump systems for district heating, where temperatures of up to 80 °C must be sustained, the only commercially available refrigerant today is R134a. This refrigerant can sometimes lead to a decrease in capacity for the heat pump of up to 35 %. Earlier work (Gabrielii and Vamling, 2000) has identified four zeotropic mixtures of refrigerants with possibly better performance that can be used instead of R134a. It is thus of interest to investigate the performance of heat transfer for these mixtures, compared to the performance of R22 and R134a. Since it is both difficult and expensive to make experimental studies a good alternative is to develop a calculation model to study the effects of a refrigerant change.

This paper presents calculations that have been carried out to compare heat transfer for the pure refrigerants and the refrigerant mixtures identified in a condenser. The condenser on which the calculations have been carried out is a 25 MW horizontal shell-and-tube condenser used in a heat pump for district heating. There are a total of 3000 finned tubes aligned in a staggered formation and the condensation takes place on the shell-side. There are 19 fins per inch and the outside/inside area ratio is 3.84. The shell is unbaffled and the tubes are one-pass on the waterside. Superheated vapour enters the condenser at the top and is transported vertically through the condenser while undergoing condensation and subcooled liquid leaves at the bottom.

A comparison of heat transfer for different refrigerants is made more difficult by the changing conditions due to different physical properties and the fact that the condenser is a component in a greater system that is in a kind of equilibrium. Many parameters change at the same time and comparisons can be made in many different ways. In this paper two different comparisons are presented, one where the duty is fixed and one where the heat pump system is fixed. The first case, with a fixed duty, usually means that costly modifications have do be done to the heat pump, for example due to increased volume flows. Therefore it is also interesting to make a comparison for a fixed system, where less or no modifications have to be done.

2. METHOD

The calculations of the condenser were carried out sequentially on one tube row at a time, following the refrigerant flow from the inlet at the top to the outlet at the bottom, and following the water from the inlet to the outlet along the tubes. Some simplifications were made:

- The conditions were assumed to be equal on all tubes in a tube row and therefore the calculations were carried out in only two dimensions, one along the tubes and one down the tube rows.
- Instead of integrating, every tube row was divided into a finite number of slices. Given the properties of the coolant and the refrigerant flowing into a slice, the heat and mass transfer equations were solved to get the condensing mass flux, duty and properties of the coolant and refrigerant flowing out of the slice.
- The condensate flow was assumed to be strictly vertical; there was no distribution along the tubes and the surroundings did not influence the condensate falling between two tubes.

- After the mass and heat transfer equations were solved for all slices in a tube row the outlet flow of refrigerant vapour was redistributed so that the volume flow was equal all over the tubes on the next tube row. This was accomplished by shifting refrigerant from slices next to each other, in order to maintain differences in composition along a tube.
- Integral condensation was assumed, which means perfect mixing between new condensate and condensate drainage from the tube above.
- The flow field was not taken into account and the pressure was constant along the tubes in a tube row.

To solve the heat and mass transfer equations for a slice an iterative approach was used. For a given slice with known conditions on coolant and refrigerant entering, a condensing mass flux of each component was guessed. With a known concentration gradient in the gas film Krishna and Standart's method (1976) was used to solve Maxwell and Stefan's transport equations for a multi-component gas mixture. With a known condensing mass flux two equations for heat transfer were solved and compared; the heat transferred from the gas bulk to the condensate interface,

$$\dot{q}_1 = \dot{n}_{tot} \cdot h_{fg} + \dot{q}_L + \alpha_g^{\bullet} \cdot (T_b - T_i) \tag{1}$$

and the heat transferred from the interface to the coolant inside the tubes,

$$\dot{q}_2 = \alpha_L \cdot (T_i - T_c) \tag{2}$$

$\dot{n}_{tot}$ is the total condensing flux, h_{fg} is the specific enthalpy of vaporization, $\dot{q}_L$ is the sensible heat from the condensate falling from the tube above and $\alpha_g^{\bullet}$ is the gas-phase heat transfer coefficient corrected for mass transfer effects according to Ackerman (1934),

$$\alpha_g^{\bullet} = \alpha_g \frac{\theta}{1 - e^{-\theta}} \tag{3}$$

where α_g is the gas-phase heat transfer coefficient for a finned tube bundle in cross flow (VDI Wärmeatlas, 1998) and

$$\theta = \frac{\sum_i \dot{n}_i c_{P_i}}{\alpha_g} \tag{4}$$

where $\dot{n}_i$ is the condensing flux of component i and c_{P_i} is the partial molar heat capacity for component i. α_L in (2) is the heat transfer coefficient from the condensate interface to the coolant, and can be written

$$\frac{1}{\alpha_L dA_o} = \frac{1}{\alpha_l dA_o} + \frac{\delta_w}{\lambda_w dA_w} + \frac{R_t}{dA_i} + \frac{1}{\alpha_c dA_i} \tag{5}$$

where α_l is the heat transfer coefficient of the condensate film on a finned tube according to Beatty and Katz (1948), α_c is the heat transfer coefficient on the coolant side of the tube according to Dittus and Boelter (1930), R_t is thermal resistance due to fouling on the coolant side of the tube, and λ_w is the thermal conductivity of the tube wall.

The iterations were carried on until local interface equilibrium was established and $\dot{q}_1$ was equal to $\dot{q}_2$. Given the condensing flux and the heat transferred in the slice, the outlet conditions of coolant and refrigerant could be calculated. Next the procedure was repeated for the next slice downstream the same tube row. When the calculations of one tube row were finished, the refrigerant gas was redistributed over the next tube row and the calculations continued in the slice at the coolant inlet. This continued until all tubes were calculated.

3. VALIDATION OF THE CALCULATIONS

The calculations were compared to measurements on a condenser in a heat pump used for district heating in Stockholm, Sweden. Pressure, inlet and outlet coolant temperatures and the coolant flow were measured. The superheating of the refrigerant vapour entering the condenser and the refrigerant flow were not measured. Instead, they were given by system simulations (Gabrielii and Vamling, 2000). To reach the desired outlet conditions of the condenser, a correction factor ε was introduced in the heat transfer equation (2),

$$\dot{q}_2 = \varepsilon \cdot \alpha_L \cdot (T_i - T_c) \qquad (6)$$

The correction factor was adjusted until the calculations agreed with the measured data. This was done for four different conditions using R22, presented in table 1. Conditions 1, 3 and 4 aim at maximum duty, while condition 2 aims at a target output temperature, in this case 78 °C. A value of ε less then one means that the calculations overestimate the heat transfer and a value greater than one that the heat transfer is underestimated. The correction factors can be seen in table 1 for all conditions. As seen, the calculations predict the heat and mass transfer within ± 9 % under various conditions, with no obvious trend.

Table 1: Comparison of conditions and correction factors

Condition	1	2	3	4
Refrigerant	R22	R22	R22	R22
Duty [MW]	24.8	16.6	19.7	19.4
Shell-side pressure [MPa]	2.7	3.6	2.6	3.0
Coolant inlet temp. [°C]	45	63	52	60
Coolant outlet temp. [°C]	63	78	60	69
Refrigerant superheat [°C]	41	46	40	43
Correction factor ε	1.00	1.08	0.93	0.91

4. COMPARISON OF HEAT TRANSFER FOR CONSTANT DUTY

A comparison of heat transfer performance for constant duty was made for R22, R134a and four zeotropic refrigerant mixtures identified in earlier work (Gabrielii and Vamling, 2000), all with glides less than 4 °C. The compositions of the four mixtures are in the condenser:

Mix1:	R32/R134a	20/80 % by volume
Mix1 mod:	R32/R134a	25/75 % by volume
Mix2:	R125/R134a/R143a	25/45/30 % by volume
Mix3:	R125/R134a/R143a	30/15/55 % by volume

The comparison cannot be made under entirely equal conditions because of differences in physical properties. Instead, condition 1 in table 1 for R22 was chosen as reference, and the following parameters were fixed:

- The degree of superheating of the incoming vapour, 41 °C.
- The degree of subcooling of the leaving condensate, 1 °C.
- The duty, 24.8 MW.
- The coolant inlet and outlet temperatures, 45 °C and 63 °C.
- The correction factor, 1.00.

The inlet pressure on the refrigerant side was chosen to obtain a temperature driving force large enough to make use of all condenser area and to reach one degree of subcooling. An increase in pressure gives a higher condensation temperature and therefore a greater temperature driving force. The mass flow of refrigerant was calculated from the fixed duty and the inlet and outlet temperatures, defined by the dew point at the chosen pressure and the fixed superheating and subcooling.

To evaluate the results an averaged overall heat transfer coefficient was calculated,

$$\langle U \rangle = \frac{1}{A_{tot}} \sum_j \frac{\dot{q}_j \cdot A_j}{\Delta T_j} \tag{7}$$

where index j denotes the slice, ΔT is given by

$$\Delta T = T_b - T_c \tag{8}$$

and the area is based on the fin-root diameter of the tube. <U> was calculated for the entire condenser, for the desuperheating part where no condensation takes place and for the condensing part of the condenser. The results can be seen in table 2. There are differences in the heat transfer coefficient, but the heat transfer qualities are good for all refrigerants when compared. An important factor to consider in the context is the volume flow, which increases for R134a, Mix1 and Mix1 mod.

Table 2: Comparison of heat transfer coefficient for R22, R134a and four zeotropic mixtures.

	R22	R134a	Mix1	Mix1_mod	Mix2	Mix3
Pressure [MPa]	2.7	1.9	2.2	2.3	2.5	2.9
<U> [W/m^{2}K]	1230	1330	1260	1230	1370	1300
Change in <U> comp. to R22 [%]	-	+8	+2	±0	+11	+6
<U> desuperheating [W/m^{2}K]	480	610	630	630	690	750
<U> condensation [W/m^{2}K]	1310	1440	1350	1310	1500	1450
Desuperheating area [%]	10	13	12	12	16	20
Inlet volume flow [m^3/s]	1.51	1.81	1.77	1.74	1.52	1.43
Change in flow comp. to R22 [%]	-	+20	+17	+15	0	-5

5. COMPARISON OF HEAT TRANSFER FOR A GIVEN SYSTEM

Fulfilling the assumption of constant duty used above demands changes in the heat pump like a new or modified compressor and changed piping to cope with greater flows, and in many cases even changes in the evaporator is needed. Instead of making these changes an alternative is to keep the equipment as it is and find the optimal conditions for the new refrigerant. This will imply that pressure, mass flow and inlet superheating will differ a lot for the different refrigerants. One question that will arise is how the heat transfer performance is influenced by the different conditions.

System simulations have been carried out (Gabrielii and Vamling, 2000) for R134a and the four mixtures, based on the four different conditions presented in table 1. One assumption in these simulations was a constant overall heat transfer coefficient for the condenser. It is thus of interest to see if this assumption is acceptable. Therefore input data from the system simulations was used for R134a and the mixtures, and the correction factor described above was adjusted to reach the same outlet conditions as in the system simulations. One example of input data for condition 1 is given in table 3. The trends for the three other conditions are similar and therefore not presented here. The correction factors that give the desired outlet conditions are presented in table 4. Correction factors for Mix2 are missing for conditions 2 and 4 because of a lack in physical properties at high temperatures and factors for Mix3 are missing for conditions 2 and 4 due to supercritical conditions.

A correction factor less than one implies that the heat transfer according to the calculations is better than in the system simulations. If the system simulations underestimate the heat transfer, the assumption of constant heat transfer coefficient is not valid. Instead it implies a higher value of the heat transfer coefficient. The opposite is valid for a correction factor greater than one.

The correction factor is fluctuating somewhat for the different conditions, but a trend that can be seen is a decrease in the value of the factor for Mix1, Mix1 mod and Mix2 compared to both R22 and R134a. That means a better heat transfer coefficient than the assumed constant value. Mix3, on the other hand, seems to have a somewhat worse heat transfer coefficient than the other mixtures.

Table 3: Measured data for R22 together with results from system simulations for condition 1, used in the condenser calculations.

	Duty [MW]	Pressure [MPa]	Mass flow [kg/s]	Superheat [°C]
R22	24.8	2.7	141	41
R134a	15.5	1.9	103	16
Mix1	17.8	2.2	105	26
Mix1 mod	18.4	2.3	106	29
Mix2	19.4	2.6	162	17
Mix3	21.2	3.1	216	15

Table 4: Correction factors for the different conditions.

Condition	1	2	3	4	Average
R22	1.00	1.08	0.93	0.91	0.98
R134a	0.97	1.10	0.71	0.75	0.88
Mix1	0.92	0.97	0.69	0.74	0.83
Mix1 mod	0.91	0.98	0.69	0.74	0.83
Mix2	0.93	-	0.72	-	0.83
Mix3	1.25	-	0.87	-	1.06

6. DISCUSSION

When comparing results for different refrigerants under different conditions, many factors must be taken under consideration. What is the objective with the operation? Is it maximum output, maximum temperature or not to make any design changes? Depending on the answer to this question, the results must be interpreted differently.

When looking at a constant duty situation, R134a, Mix2 and Mix3 seem to have the best average heat transfer coefficient, according to table 2, but when comparing the volume flows it turns out that R134a has more than 20 % higher volume flow than the other two. This means higher velocities in the condenser, and therefore a somewhat improved heat transfer. At the same time the pressure drop will increase and the compressor has to be modified to cope with the greater volume flows. Instead, Mix2 and Mix3 are the most interesting, since there is no increase in volume flow at all. However, the mass flow will increase 8 % for Mix2 and 16 % for Mix3. The comparison is made under constant superheating, which in reality may not be true. Usually the compressor discharge temperature is higher for R22 than for the other substances, due to differences in the enthalpy-temperature relationship, but for simplicity the superheating was here fixed.

When comparing different refrigerants in a fixed system, one of the most important factors to study is the duty. The typical trend is shown in table 3, where R22 has the highest duty and R134a has the lowest. The mixtures are somewhere in between. Considering the heat transfer coefficient, R134a, Mix1, Mix1 mod and Mix2 all seem to have a higher coefficient than R22, while Mix3 seem to have a lower coefficient. Therefore the assumption of constant heat transfer coefficient is not entirely valid and has to be taken under consideration. For rigorous simulations a more elaborate method might be needed. When making a decision about which alternative to use, these values are interesting in addition to the system calculations made for a constant heat transfer, since they indicate that no unpleasant surprises are to be expected when changing to R134a, Mix1, Mix1 mod or Mix2, at least not concerning the condenser.

7. CONCLUSIONS

Calculations were carried out for a 25 MW shell-and-tube condenser. The performance of R22, R134a and four zeotropic, low glide mixtures, were compared. Conclusions that can be drawn from the calculations are:

- The calculation model used predicts heat and mass transfer within ± 9 % under various conditions with no obvious trend.
- For a constant duty situation the heat transfer for the mixtures studied are all equal to or better than the heat transfer for R22. R134a also seem to have good heat transfer, but at the same time it has the highest volume flow. Two of the mixtures have a combination of good heat transfer and low volume flow.
- For a fixed heat pump system three of the mixtures seem to perform better than R134a, when considering heat transfer and duty.
- The assumption of constant heat transfer coefficient for a system when changing refrigerant must be taken under consideration when performing rigorous calculations.

- This kind of calculation provides a cost-efficient and time saving way of investigating the performance of different refrigerants under various conditions. It can be difficult and costly to do this experimentally, especially for equipment of the size dealt with in this paper.

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