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# HEAT TRANSFER OF R-22 AND ALTERNATIVES IN A PLATE-TYPE EVAPORATOR

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## ABSTRACT

This paper deals with the investigation of substitutes for R-22 and their heat transfer behaviour in a plate-type evaporator with a falling water film as heat source. Alternatives investigated were R-134a and two new refrigerant mixtures.

Measurements on a 3m x 0,5m plate showed a superior performance of R-134a compared to R-22 and mixtures at equal test conditions and a distinct difference between mixtures and pure fluids at very low water temperatures. Considering the decrease in duty when changing to R-134a in an existing plant, it was seen that with decreasing evaporator duty the U-value decreased too.

A dependency on conditions prior to start-up was observed, with U-values differing as much as 15%. Reasons discussed are activation of nucleation sites and different flow patterns in the plate.

## NOMENCLATURE

| | | |
|---|---|---|
| A | - | evaporator area, $m^2$ |
| LMTD | - | logarithmic mean temperature difference, K |
| U | - | overall heat transfer coefficient, $W/m^2K$ |
| Q | - | duty, W |
| $T_{w,in}$ | - | water temperature to the evaporator, K |
| $T_{w,out}$ | - | water temperature from the evaporator, K |
| $T_{R,bubb}$ | - | refrigerant bubble point temperature, K |
| $T_{R,out}$ | - | refrigerant temperature at the outlet, K |

# 1 INTRODUCTION

## 1.1 Background

In Sweden the ban on refilling existing large-scale heat pumps with R-22 in the year 2002 generates the problem of finding adequate substitutes. Especially high temperature applications, like heat pumps for district heating, demand working fluids with critical properties, which can sustain condensation temperatures of around 80°C. A possible substitute is R-134a, but its use can result in duties 30-36% lower than that of R-22. A number of blends offering a somewhat higher duty than R-134a have, however, been identified (Gabrielii and Vamling, 2000).

Several heat pumps for district heating in Stockholm, Sweden use plate-type evaporators with a falling film of sea water as heat source. This is mostly due to the fact that in the Baltic Sea during winter time very low water temperatures can occur; thus the risk of ice formation on the evaporator panels is evident. This ice formation does not cause any harm when plate-type heat exchangers are used. For the current work one heat exchanger panel was taken out of the large-scale evaporator, which consists of 1020 of those panels. This evaporator is of the self-circulation type, i.e. after the expansion valve there is a gas-liquid separator, where the liquid is led down to the inlet of the evaporator itself. The outlet is connected to the separator, from which the gas proceeds to the compressor inlet.

Sea water is employed as a falling film on the outside of the evaporator plates at flow rates of about 2500 kg/s. The sea water temperature is subject to a seasonal change, varying between 2°C in wintertime and up to 18°C in late summer. The total duty of the evaporator varies over the year between 15-20 MW.

## 1.2 Objectives

In a first step the possible substitutes for R-22 should be compared in terms of heat transfer at similar conditions, i.e. same evaporator duty, refrigerant flow and water inlet temperature. At low water temperatures (2°C) it should be determined what maximum duty could be achieved without the formation of ice on the panel.

R-134a requires a volumetric compressor displacement about 50% greater than that of a R-22 compressor of the same cooling capacity, due to lower pressure vs. temperature correlation at saturation (A. Cavallini, 1995). This means that the limited compressor volume flow in the existing heat pump involves a decrease in evaporator duty and therefore it was important to investigate the heat transfer also at lower duties.

# 2 EXPERIMENTAL

The test facility as seen in Figure 1 consists of a refrigerant and a water loop, of which the latter is an open system since the water is employed as a falling film on the outside of the evaporator plate. The water film provides the heat input, which can be calculated by measuring water flow and temperature difference at inlet and outlet. Water flow as well as water inlet temperature are the two parameters, which can be controlled on the water side. The use of tap water was reasonable, since the full-scale evaporator uses brackish water with a very low salt content.

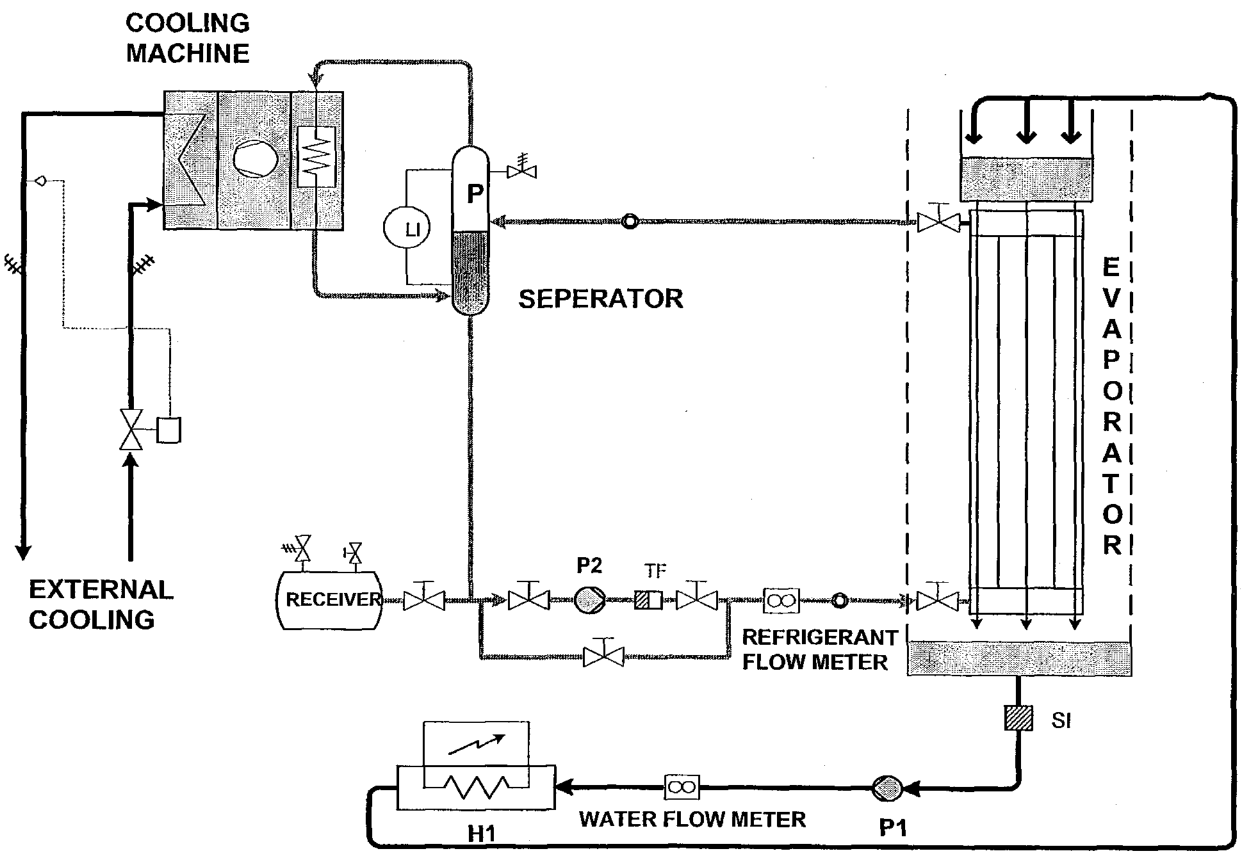


**Figure 1 Flow chart of the test facility**

The refrigerant loop consists of the evaporator plate, a separator and a condenser and can be operated using natural or forced circulation. During forced circulation a by-pass with a pump is utilised and in this way the refrigerant flow can be controlled. A cooling machine, where the compressor has a variable speed drive, controls the pressure in the separator and thereby the evaporation pressure.

## 2.1 The Evaporator

According to Haukås (1984), narrow vertical refrigerant flow channels assure high plate efficiency. Vertical flow channels in the investigated heat exchanger were created by attaching two sheets by means of point weldings. This of course allows exchange of fluid between the different channels.

The evaporator plate as seen in Figure 2 is made of stainless steal (Avesta 254 SMO) and is 3000 mm high, 500 mm wide and, at channel centre, 5 mm thick. The wall thickness is about 1 mm.

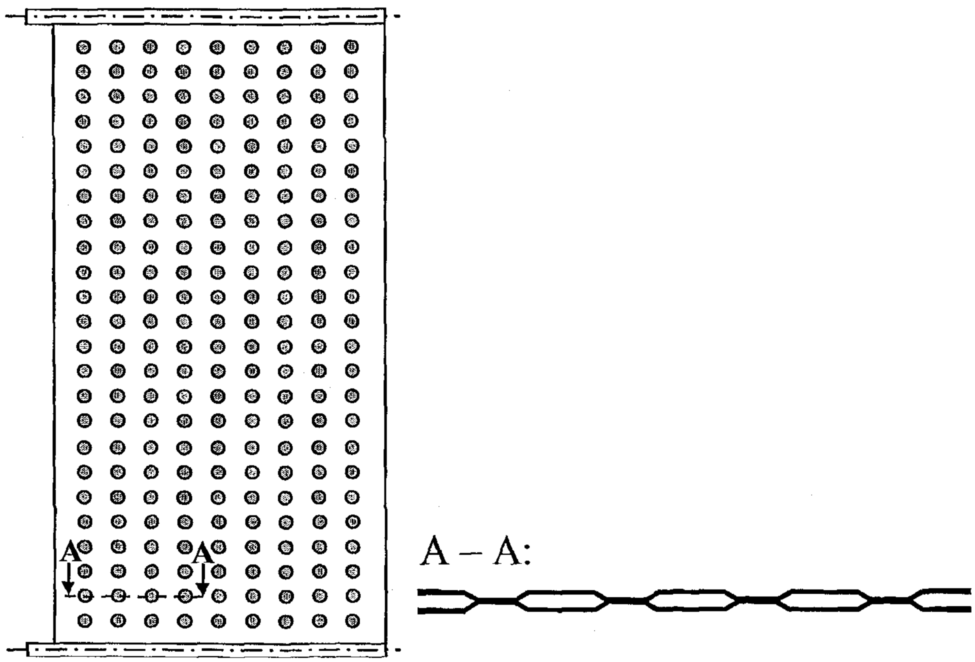


**Figure 2 The plate-type evaporator (not to scale)**

## 2.2 Conditions and media

Test conditions were determined by those conditions encountered at the full-scale evaporator, which are subject to seasonal changes. This lead to a water flow rates of 1,9 kg/s and 2,45 kg/s and water temperatures typically between 2°C and 12°C. Evaporation pressure was adjusted in a way that duties between 12-20 kW were attained.

Refrigerant flow was controlled by the pump and when natural circulation was applied the flow rate could be regulated by a valve placed before the evaporator inlet.

Tested refrigerants were:

1) R-22 (as reference)
2) R-134a
3) Mix1: 96,4% R-134a+ 3,6% R-32 (given in mass %)
4) Mix2: 67,2% R-134a+ 17% R-125 + 15,8% R-143a (given in mass %)

# 3 RESULTS

When the different working fluids are compared it is done by means of the overall heat transfer coefficient U, which was calculated by

$$U = Q/\left(A \cdot \mathrm{LMTD}\right), \qquad (1)$$

where the logarithmic mean temperature difference (LMTD) is defined on the basis of the water temperatures in and out, refrigerant bubble point and refrigerant saturation temperature at the evaporator outlet

$$\mathrm{LMTD} = \frac{(T_{W,in} - T_{R,out}) - (T_{W,out} - T_{R,bubb})}{\ln \dfrac{(T_{W,in} - T_{R,out})}{(T_{W,out} - T_{R,bubb})}} \quad (2)$$

In Figure 3 the results for measurements at similar conditions (duty and water inlet temperatures) are represented. Relatively to R-22 it was R-134a and Mix2, which performed better and Mix1 that performed worse. A decrease of the U-value with decreasing duty was noticeable.

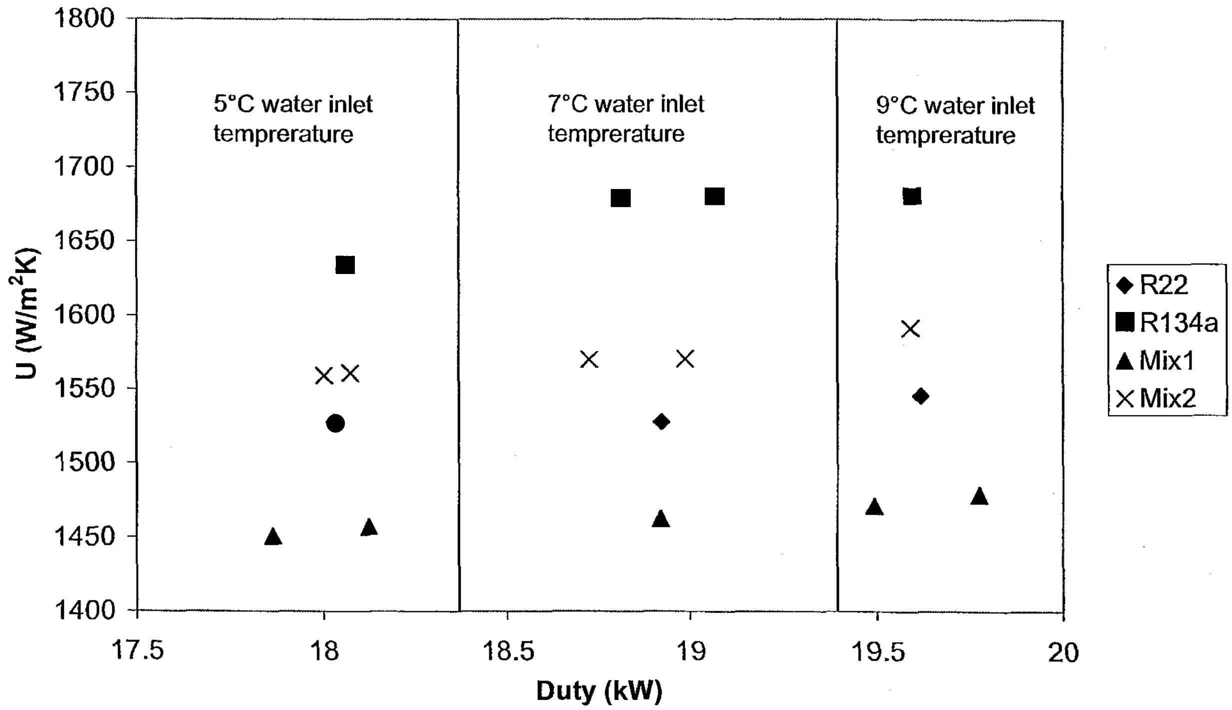


**Figure 3 Overall heat transfer coefficient vs. duty at different water inlet temperatures and water flow rate of 1,9 kg/s. (Pump circulation)**

At an inlet water temperature of 2°C, R-22 and R-134a showed equal performance and first ice formation on the panel occurred at equal duty and driving force (LMTD). Both were higher than for mixtures. Mix1 and Mix2 showed lower U-values at this low water temperature. When comparing the two mixtures it was observed that the heat transfer coefficient was worse for Mix2 but the duty could be slightly higher before ice formation arose.

As mentioned in chapter 3.2, it was of interest, if - at typical water temperatures - heat transfer would change with lower duties. Results for R-134a are shown in Figure 4.

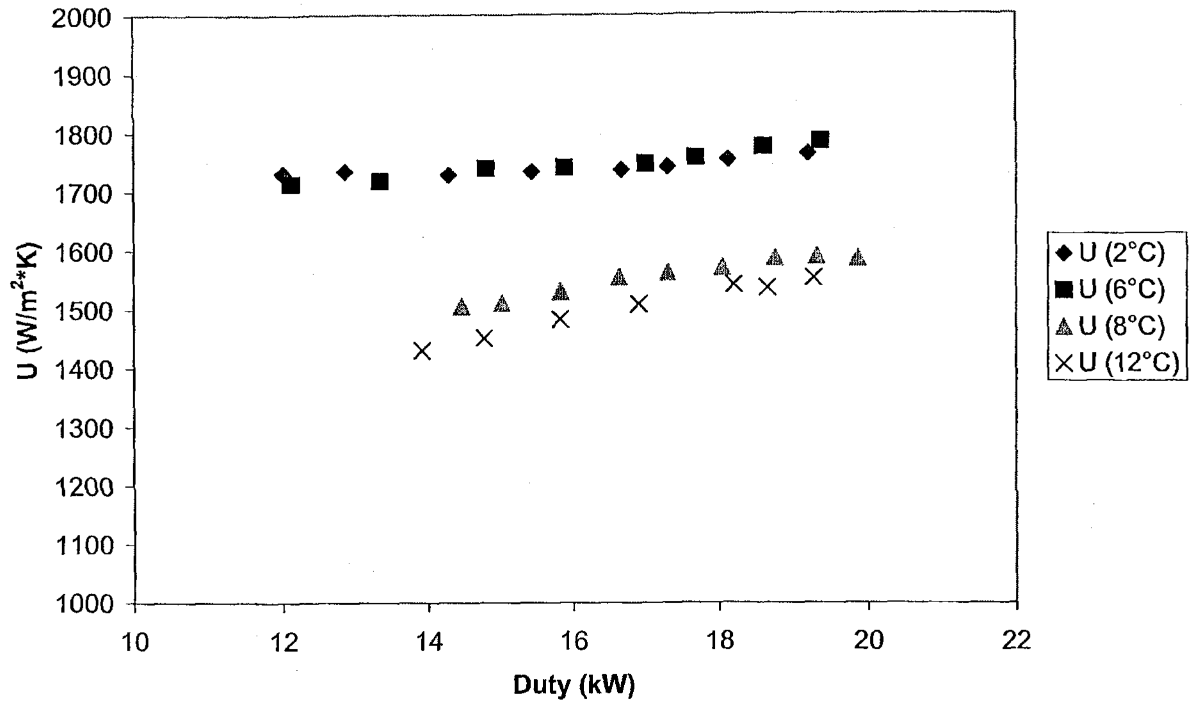


**Figure 4 U-values as a function of the duty at typical water temperatures. R-134a, natural circulation, water flow rate 2,45 kg/s.**

It can be clearly seen that the U-value is decreasing towards lower duties. Apparently there is a water temperature dependency, since the U-value falls most for 12°C water temperature (8%) and least for 2°C water temperature (1%) considering the range from 19 kW to 14 kW.

Reasons for the unexpected lower U-values at 8°C and 12°C water temperature are discussed in the following chapter.

# 4 DISCUSSION

In Figure 3 the large difference between R-134a and Mix1 should be pointed out, since those fluids differ from each other by just 3,6 % R-32. If this deviation is due to the phenomena discussed further below or to characteristics inherent to the mixture is difficult to anticipate.

When considering the results for ice formation it is important to focus on the main difference between pure fluids and mixtures: the temperature glide during evaporation. The maximum load will be achieved at operating conditions giving the most uniform outside plate temperature (Haukås, 1984). Pure fluids provide a larger temperature drop over the panel, which leads to the the above mentioned condition and thus to a higher possible load than in comparison to mixtures. In addition when using mixtures, the evaporator inlet temperature has to be lower in order to reach the same load as for pure substances. This fact could introduce a higher risk of ice formation at the very low part of the panel.

It should be mentioned that ice formation can occur very rapidly and therefore the actual start of ice formation is very difficult to determine. During laboratory tests, conditions can be changed very slowly and carefully and therefore high duties during low water temperatures can be achieved. During real conditions the evaporator is subject to more disturbance and thus ice formation might happen earlier.

The increase of the U-value towards higher duties in Figure 4 is probably due to augmenting vapour fraction at the evaporator outlet. This leads to higher flow velocities and reduced film thickness, which is accompanied by a more efficient convective heat transport. An additional effect is increased nucleate boiling in the lower part of the evaporator. The degree of the mentioned effects is apparently dependent on the water temperature on the outside of the panel.

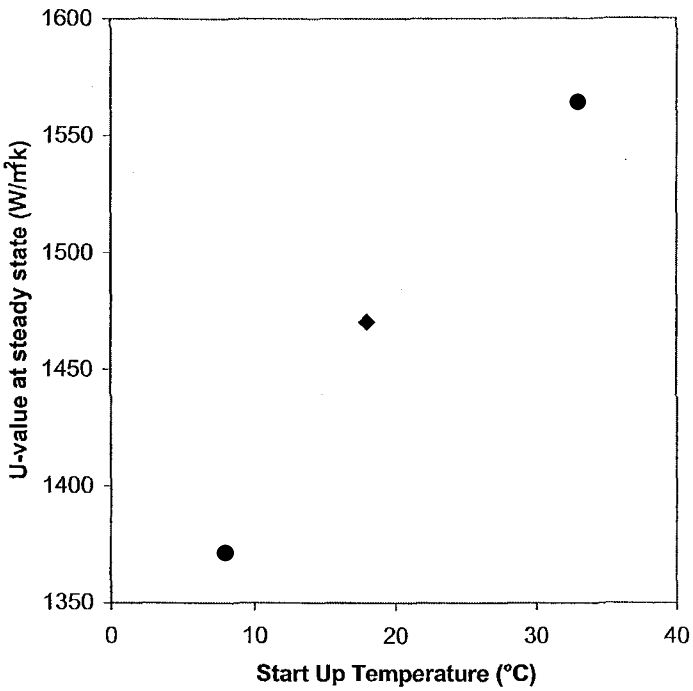


**Figure 5 Effect of different start up temperatures on steady state U-value at 8°C water inlet temperature. R-134a.**

A typical effect experienced during measurements is illustrated by the gap between curves in Figure 4. Tests with R-134a as well as those with mixtures showed a dependency on conditions prior to start-up and resulting U-values could differ as much as 15%. When starting the test facility, in most cases the water at start-up has a higher temperature than the desired water inlet temperature. Thus, during start-up the water needs to be cooled down, by means of the evaporator plate, until steady state is reached. An example how different start-up water temperatures can influence the resulting steady state U-value is shown in Figure 5. One possible explanation could be activation of nucleation sites. Such boiling hysteresis phenomena are usually more distinct at enhanced surfaces than at smooth surfaces (Hsieh and Wenig, 1997). However, point weldings could provide sufficient nucleation sites and therefore the effect of hysteresis might be more distinct. It does not seem very probable though that the effect illustrated in Figure 5 alone can explain the large deviation of U-values in Figure 4.

This leads to the discussion of the special design of the investigated evaporator plate. Its point weldings make the exchange between different flow channel possible and therefore distribution amongst them rather unpredictable. Caused by different start-up conditions the flow pattern in the panel could change from run to run and in that way influence the resulting performance of the tested fluids. An existing flow pattern inside the panel could be changed by a rapid alteration of conditions, like refrigerant flow or water temperature, leading to a new flow pattern. This could be followed by a different quality of heat transfer and may be described as flow hysteresis.

## 5 CONCLUSIONS

Heat transfer in a plate type evaporator during varying conditions for R-22 and alternatives was studied. The results and discussion lead to the following conclusions:

(1) The comparisons of the different fluids at similar conditions show in all cases the same ranking. However, results have to be treated with caution, since they can be dependent on previous conditions.

(2) At low temperatures, ice formation on the panel occurred at lower duties for mixtures than for pure substances, which is probably due to the mixtures' glide and thus a less uniform outside plate temperature.

(3) For R-134a, a variation of duty at different water temperatures showed a decrease of U-value towards lower duties. Hence, when comparing R-22 and alternatives, the limited compressor volume flow in the existing heat pump has to be considered if an erroneous basis of comparison should be avoided. The magnitude of decrease showed a dependency on water temperatures.

(4) A dependency on conditions prior to start-up was observed for all fluids and it is believed that the special design of the plate could involve the possibility of boiling and/or flow hysteresis.

## ACKNOWLADGEMENTS

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## REFERENCES

1. **Cavallini A.**, Working Fluids for Mechanical Refrigeration, *Proceedings – 19th International Congress of Refrigeration 1995*, Vol. IVa, pp. 25-42.
2. **Gabrielii C. and Vamling L.**, Replacement of R22 in heat pumps used for district heating (R19), submitted paper, *International Refrigeration Conference at Purdue, USA 2000.*
3. **Haukås H.T.**, Design of a plate-type evaporator for heat pumps, *International Journal of Refrigeration*, Volume 7, Number 1, January 1984, pp. 59-63.
4. **Hsieh S.-S. and Wenig C.-J.**, Nucleate pool boiling from coated surfaces in saturated R-134a and R-407c, *International Journal of Heat and Mass Transfer*, Volume 40, Number 3, 1997, pp. 519-532.