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FLOW BOILING OF R723 (AMMONIA/DIMETHYLETHER) IN SMOOTH HORIZONTAL TUBES IN THE PRESENCE OF OIL

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ABSTRACT

Direct evaporation of the refrigerant inside tubes (flow boiling) allows for a significant minimization of the plant refrigerant charge. However, very small quantities of lubricant can have an important impact on the flow boiling, thus causing a decrease of the flow boiling heat transfer coefficient, depending on the miscibility of oil with the refrigerant. In a previous paper we presented experimental data for the evaporation of the natural refrigerant ammonia (R717) inside smooth horizontal tubes: experiments run with a frequently-used fully-synthetic oil (PAO with ISO-VG 68), which is immiscible with ammonia, with oil contents from 0.1 to 3% showed a very important decrease of the flow boiling heat transfer coefficient of ammonia.

The azeotropic mixture R723 composed of 60%w/w ammonia and 40%w/w dimethylether has a better miscibility with mineral oils and with synthetic oils which are immiscible with ammonia. Therefore, the decline of the heat transfer is attenuated [eurammon (2003)].

In the present paper, the flow boiling measurements performed with R723 and the same fully-synthetic oil as in the pure ammonia tests (i.e. PAO with ISO-VG 68) are analyzed. Tests were run in a 24 m serpentine of double-pipe type with a plain stainless steel tube of 14 mm inner diameter. The test conditions were: evaporation saturation temperatures from −10 to +10°C, mass fluxes ranging from 100 to 260 kg/(m^2 s) and average heat fluxes between 15 and 60 kW/m^2 . The average values of the flow boiling heat transfer coefficient with R723 for an oil-free operation (with vapor contents of 10% at the inlet and dry-out conditions at the outlet of the evaporator) range between 6000 and 14000 W/(m^2 K). Experiments performed with oil contents from 0.1% to 2% showed a noticeable decrease of the flow boiling heat transfer coefficient, however, the impact of oil on flow boiling is considerably weaker with R723 in comparison to pure ammonia.

1. INTRODUCTION

A significant advantage of the refrigeration systems using flow boiling (also referred to as in-tube evaporation or direct expansion) is the strongly-minimized refrigerant charge. However, in most cases the flow boiling of the refrigerant is affected by the presence of lubricating oil from the compressor. Gross (1994) indicated that particular problems arose due to the variation (or lack) of the miscibility (i.e. solubility) of oil with the refrigerant. Shen and Groll (2003) reviewed the influence of lubricants on the boiling of refrigerants, stressing the importance of the modification of the local flow pattern distribution due to the presence of oil.

The use of ammonia in direct expansion systems remained limited due to its immiscibility with conventional lubricating oils: immiscible oil forms a layer with a poor conductivity, thus causing a strong reduction of the heat transfer coefficient. A short review of the literature dealing with this problem can be found in our previous publications.

The significant decrease of the heat transfer coefficient of ammonia in the presence of immiscible oil led in the 90's to the development of new synthetic oils miscible with ammonia. The experiments by Boyman *et al.* (1997) on the in-tube evaporation of ammonia in the presence of a miscible poly-glycol oil (with viscosity grade VG68) with mass fluxes between 30 to 65 kg/(m^2 s) did not show any deterioration of the heat transfer coefficient for oil contents up to 3% (by weight). However, in order to operate with these oils all the components of the plant have to be totally exempt of impurities resulting from immiscible oils and the plant must be assembled with the highest care without any hygroscopic traces. For this reason the manufacturers of ammonia plants are still using immiscible oils for compressor lubrication.

Recently we presented results on the impact of oil on flow boiling of ammonia in plain horizontal tubes [Boyman *et al.* (2004), Aecherli and Boyman (2004)]. Experiments run with a frequently-used fully-synthetic oil on the basis of polyalphaolefin and alkyl-aromatics (herein after referred to as PAO with ISO-VG 68), which is immiscible with pure ammonia, showed a significant decrease of the flow boiling heat transfer coefficient due to the presence of oil.

In order to improve the solubility of lubricant with the refrigerant, researchers at ILK Dresden added dimethylether to ammonia. Their work has focused on the azeotropic mixture with 60% w/w ammonia and 40% w/w dimethylether with the refrigerant denomination R723. Lippold and Heide (1997), Lippold (2001), eurammon (2003), Krauss and Schenk (2005) reported, besides improved lubricant solubility, a lower compressor discharge temperature as a result of the higher molar mass of R723 in comparison to R717, and also a higher heat transfer both in evaporator and condenser.

The present work has the main goal of reporting on the experimental results obtained with R723 and on the impact of oil, which is the same as in the former ammonia tests performed by our group. The preliminary tests were carried out by Camenzind (2005).

A comparison of the results for flow boiling heat transfer with both refrigerants (R717 and R723) under similar operating conditions with small quantities of lubricating oil will confirm the improvement mentioned in the literature, but will also show the limitations observed.

2. EXPERIMENTAL TEST FACILITY AND THE TEST CONDITIONS

A detailed description of the experimental test facility with the flow circuit diagram can be found in Boyman *et al.* (2004). The test facility has been operated as refrigeration plant based on the single-stage vapor compression principle. The experiments presented in this paper have been carried out with a manually-controlled expansion valve at the inlet of the evaporator in order to run under constant operating conditions.

The schematic arrangement of the measurement points on the evaporator is summarized in Figure 1 and a picture of the evaporator prior to insulating is given in Figure 2.

The evaporator of the test facility consists of two sections (***Sections 1*** and ***2*** in Fig.1) each 12 m long. Each section is a serpentine coil with horizontal straight tube sections of 2 m (plain stainless steel tube with 14 mm inner diameter) connected by U-bends with the same inner diameter. Water-glycol is fed in the concentric outer tube in counter-current flow direction. Both ***Sections*** can be operated in parallel or in series, thus allowing the operation in a wide range of mass fluxes. The test facility is equipped with special tubular sight glasses with the same inner diameter as the evaporator tubes for the observation of the flow patterns at different intermediate stations (see Fig.2). The temperatures were measured in the refrigerant loop and in the water-glycol loop with thermocouples calibrated with an accuracy of ±0.1 K.

The tests for the present work were performed with PAO with viscosity grade ISO-VG 68. The same oil was used for the ammonia tests. The oil content in the refrigerant loop has been adjusted through a special circuit and a pump from 0 up to 2% (by weight). The oil injected prior to the Coriolis flow meter has been collected in the liquid separation vessel at the outlet of the evaporator and led back into the oil reservoir at the end of a test run. The oil flow rate has been determined by means of level measurement in the oil reservoir with accuracies better than ±0.05%. The flow rate, the density and the temperature of the mixture "refrigerant – oil" are measured after the condenser online by means of a Coriolis type instrument.

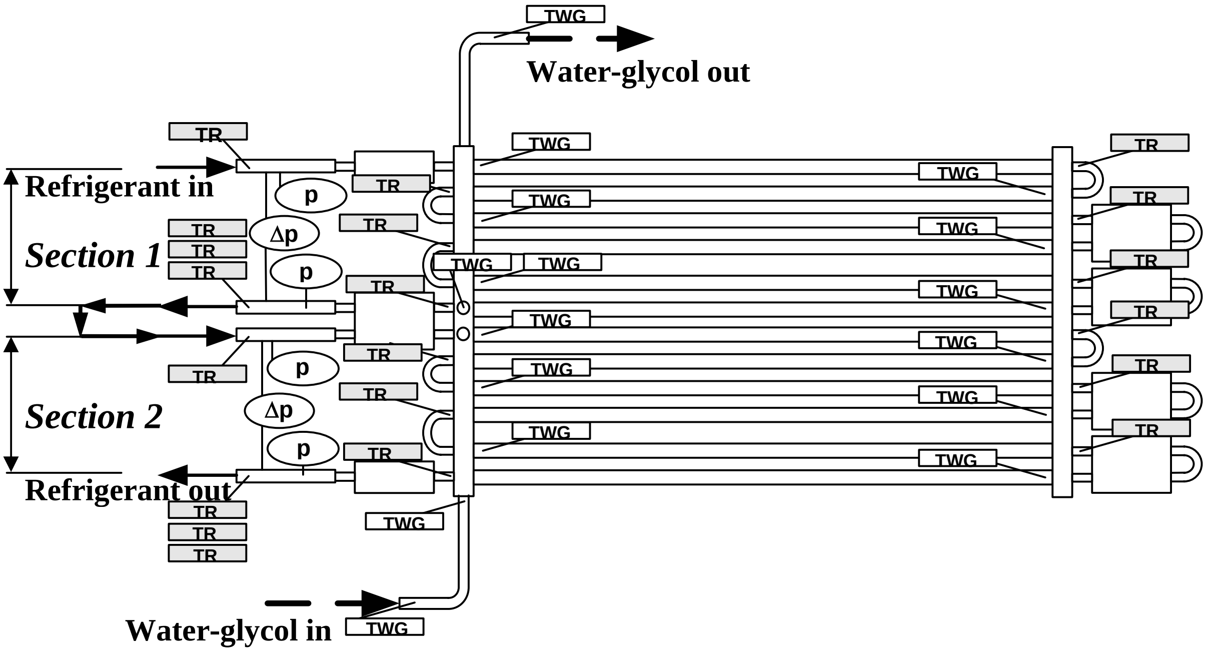

Figure 1: schematic arrangement of the measurement points on the evaporator
TR: Refrigerant temperature
TWG: Water-Glycol temperature
p and **Δp**: Pressure resp. Pressure difference

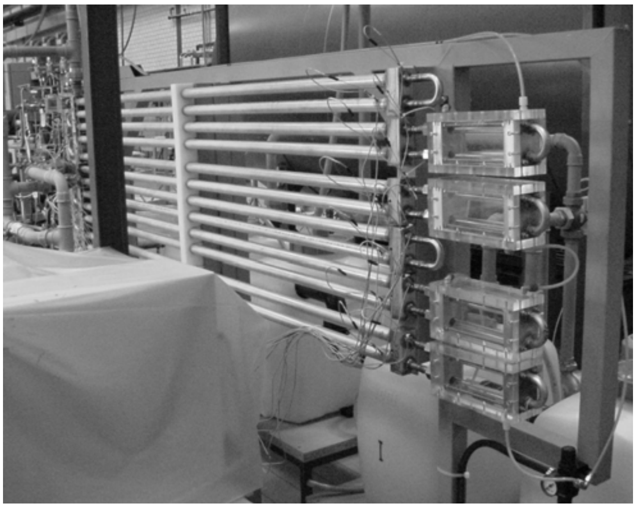

Figure 2: evaporator with sight glasses for observation of the flow regime

The experimental conditions for this study are:

- Evaporation temperature (at the evaporator inlet): $t_{sat\ evap\_in} = -10$ to $+10°C$
- Mass flux of R723 (during tests without oil injection): G= 100 to 260 kg/(m^2 s)
- Refrigeration capacity: $Q_{dot\ evap}$ = 15 to 30 kW
- Average heat fluxes at evaporator: $q_{dot\ evap}$ = 15 to 60 kW/m^2
- Vapor void fraction (=vapor quality) (at the evaporator inlet): x_{evap_in} = approx. 0.10
- Oil content in the R723-oil mixture in the evaporator: ξ_{oil} = 0 to 2% (by weight)
- Superheat at the evaporator exit: $\Delta T_{superheat}$ = 15 K to 20 K

3. DATA REDUCTION

As mentioned above, ***Sections 1*** and ***2*** of the evaporator are geometrically identical (each 12 m long) (see Fig. 1). During the present tests the ***Sections*** were operated in parallel or in series allowing measurements with the required mass flux.

The temperatures were measured in the evaporator every 2 m, both in the refrigerant and in the water-glycol loops. The typical temperature distributions (or temperature profiles) of the refrigerant and of the water-glycol at defined positions of the double-pipe are shown in Figure 3 for the parallel operation mode and in Figure 4 for the serial one. The heat exchanger was operated in counter-current mode: in Figures 3 and 4 the water-glycol inlet is on the right hand side, the refrigerant inlet on the left.

The temperature profiles and mass flow rates were used for the data reduction. Matching of the beginning of the Sector S3, which is marked by a significant increase in the refrigerant temperature, with the heat flux decrease on the water-glycol side (i.e. the change of the slope of the water-glycol temperature profile) was used for checking the steady-state thermal conditions and the thermal balance of both fluids. Cross-checking of the heat transfer rates Q_{dot} on the refrigerant and on the water-glycols loops showed an accuracy of ±3% for R723 tests. This accuracy was ±2% during ammonia tests [Boyman *et al.* (2004)].

During the tests with lower mass fluxes [up to 140 kg/(m² s)] the evaporator was operated with ***Sections 1*** and ***2*** in parallel. Almost identical temperature profiles confirmed that both ***Sections*** were running under similar thermal and flow conditions, thus allowing the use of the average values of the temperatures for the data reduction (corresponding to “Sector S1” in Figure 3). The vapor quality at the evaporator inlet was kept at approximately 0.10.

During the tests with high mass fluxes [from 150 kg/(m² s)] ***Sections 1*** and ***2*** were operated in series (Figure 4). The refrigerant was still introduced with a vapor quality of approximately 0.10 into the Evaporation Sector S1 (corresponding to ***Section 1*** in Figure 1), in which it evaporated up to an intermediate vapor quality and then flew through a connection pipe equipped with valves (which caused some slight additional pressure drop) before entering the Evaporation Sector S2. In Sector S2 the evaporation of the refrigerant went from the intermediate vapor quality up to dry-out conditions. It must be pointed out that in serial operation mode the Evaporation Sectors S2 and S3 constituted together the ***Section 2*** of the evaporator.

In both modes of operation (in parallel or in series) the Evaporation Sector S3 marks the sector in which the last step of the evaporation takes place in the “mist flow” regime followed by superheating. From the onset of dry-out, the inner tube wall is no more wetted by the liquid refrigerant and the heat transfer coefficient decreases rapidly: the result is a steep increase of the refrigerant temperature and a significant change of the slope of the water-glycol temperature profile.

The heat rate ($\dot{Q}$ or Q_{dot}) transferred in each Sector Sx (Sx = S1 or S2) of the evaporator and the average heat flux are computed with data from the water-glycol side:

$$\dot{Q}_{WG\_Sx} = \dot{m}_{WG} * c_{p\_WG} * (\Delta T_{WG\_Sx}) \qquad \text{(Eq. 1)}$$

(with WG as abbreviation for Water-Glycol)

Heat flux ($\dot{q}$ or q_{dot}) is defined as heat transfer rate per inner surface area of the plain tube.

The “Sector averaged” heat transfer coefficient $U_{Sx\_ref=ri}$ over the Sector Sx equals:

$$U_{Sx\_ref=ri} = \frac{\dot{Q}_{WG\_Sx}}{2 * \pi * r_i * l_{Sx} * \Delta T_{m\_log}} \qquad \text{(Eq. 2)}$$

with:

$\dot{Q}_{WG\_Sx}$: heat transfer rate according to Equation 1,

ΔT_{m_log} : logarithmic mean temperature difference based on the temperatures at the inlet and outlet stations of the considered Sector,

r_i : inner radius of the plain tube (which was taken as reference radius), and

l_{Sx} : length of the considered Sector

(in case of the serial operation mode with $l_{S1} = 12$ m, and l_{S2} computed with the temperature profiles and in case of the parallel operation mode with $l_{S1} = l_{S2}$ computed with the temperature profiles).

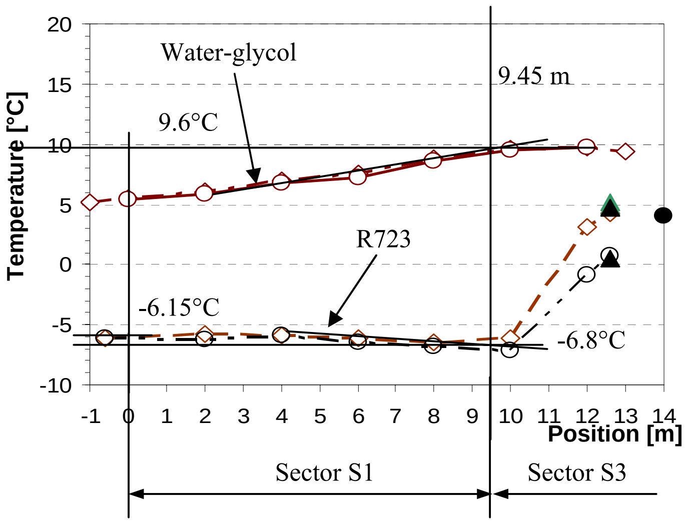

Figure 3: temperature profiles along the evaporator coils during test runs with evaporator ***Sections 1*** and ***2*** operated in parallel (Evaporation from $x_{evap\_in} = 0.10$ to dry-out in Sector S1, Superheating in Sector S3)

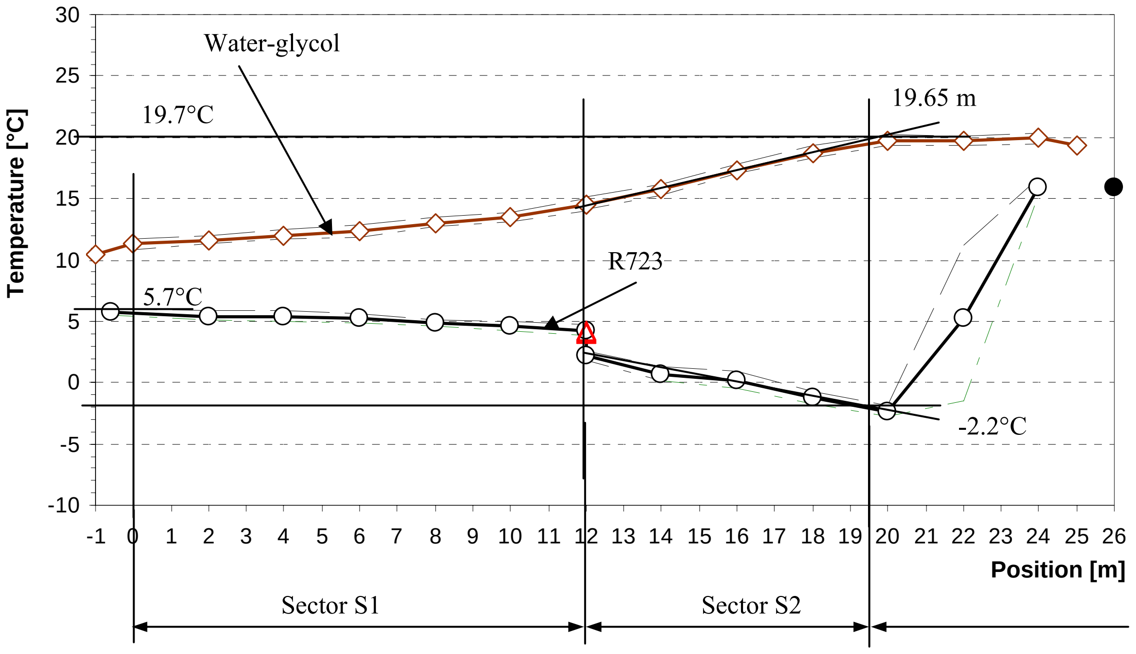

Figure 4: temperature profiles along the evaporator coils during test runs with evaporator ***Sections 1*** and ***2*** operated in series (Evaporation from $x_{evap\_in} = 0.10$ to dry-out in Sectors S1 and S2, Superheating in Sector S3)

The “Sector averaged” heat transfer coefficient h_{R723_Sx} for flow boiling of refrigerant R723 for each Sector Sx (with vapor quality going from $x_{evap\_Sx\text{-}in}$ to $x_{evap\_Sx\text{-}out}$) was calculated according to the Equation 3:

$$\frac{1}{h_{R723\_Sx} * r_i} = \frac{1}{U_{Sx\_ref=ri} * r_i} - [\frac{1}{k_{steel}} * \ln\{\frac{r_o}{r_i}\}] - \frac{1}{h_{WG} * r_o} \qquad \text{(Eq. 3)}$$

with:

k_{steel} : thermal conductivity of stainless steel, and

r_o : outer radius of the plain tube.

The heat transfer coefficient h_{WG} for water-glycol was calculated according to Gnielinski as in VDI-Wärmeatlas [7. edition, p. Gb7] for the annular flow conditions.

For Sector S1, $x_{evap\_S1\text{-}in}$ was computed from the subcooled liquid conditions before the expansion device using the related condensation and evaporation pressures [for S1 $x_{evap\_S1\text{-}in}$= 0.10 (approx.)].

Thus, for parallel operating conditions the average global boiling heat transfer coefficient h_{R723_av} is based on the values measured over the Sector S1. For serial operating conditions (i.e. the ***Sections 1*** and ***2*** operated in series) the average global boiling heat transfer coefficient h_{R723_av} is established using the “Sector averaged” coefficients and represents the area averaged global coefficient computed according to Equation 4:

$$h_{R723\_av} = \frac{h_{R723\_S1} * A_{S1} + h_{R723\_S2} * A_{S2}}{A_{S1} + A_{S2}} \qquad \text{(Eq. 4)}$$

Consequently, the coefficient h_{R723_av} is corresponding to the evaporation path from x_{evap_in} = 0.10 (approx.) to dry-out.

The oil content (i.e. “mass flow rate of the injected oil” compared to the “refrigerant plus oil mass flow rate”) was varied from 0.1% to 2% using a special oil injection pump with variable hub. Calculation of the oil content was performed using the oil level decrease in the oil reservoir for given time intervals. The oil content calculations were cross-checked with density measurements from the Coriolis flow meter, placed between the condenser and the expansion device. The steadiness of the oil injection was monitored continuously. After test runs with oil injection the plant was operated for approx. 4 hours with a high refrigerant flow rate without any oil injection. The efficiency of the purging was checked with tests under the same conditions.

4. TEST RESULTS

The results of the experimental work with R723 are summarized in Figure 5: the average global values of the boiling heat transfer coefficient (h_{R723_av}) are given as sets of tests without and with oil injection, with Evaporator ***Sections*** connected in parallel [for lower mass fluxes up to 140 kg/(m^2 s)] or in series [for mass fluxes higher than 150 kg/(m^2 s)]. The tests with oil contents from 0.14 to 0.30% resulted in a significant reduction of the heat transfer coefficient: approx. 40% for lower and 30% for higher mass fluxes. With oil contents higher than 1% the decrease is approx. 60% to 45%.

In comparison to the tests performed with ammonia and the same fully-synthetic oil, the decrease of the boiling heat transfer coefficient is slightly diminished, but the impact of oil on heat transfer is still evident. The test results published by our group [Aecherli and Boyman (2004)] as continuation of -and addendum to- our prior publication [Boyman *et al.* (2004)] show the deterioration of the boiling heat transfer coefficient of ammonia in the presence of oil. For the sake of completeness the results of the ammonia tests with the PAO with ISO-VG 68 (Gargoyle Arctic SHC 326) are given in Figure 6: it can be deduced that the impact of oil is much stronger in case of ammonia in comparison to R723.

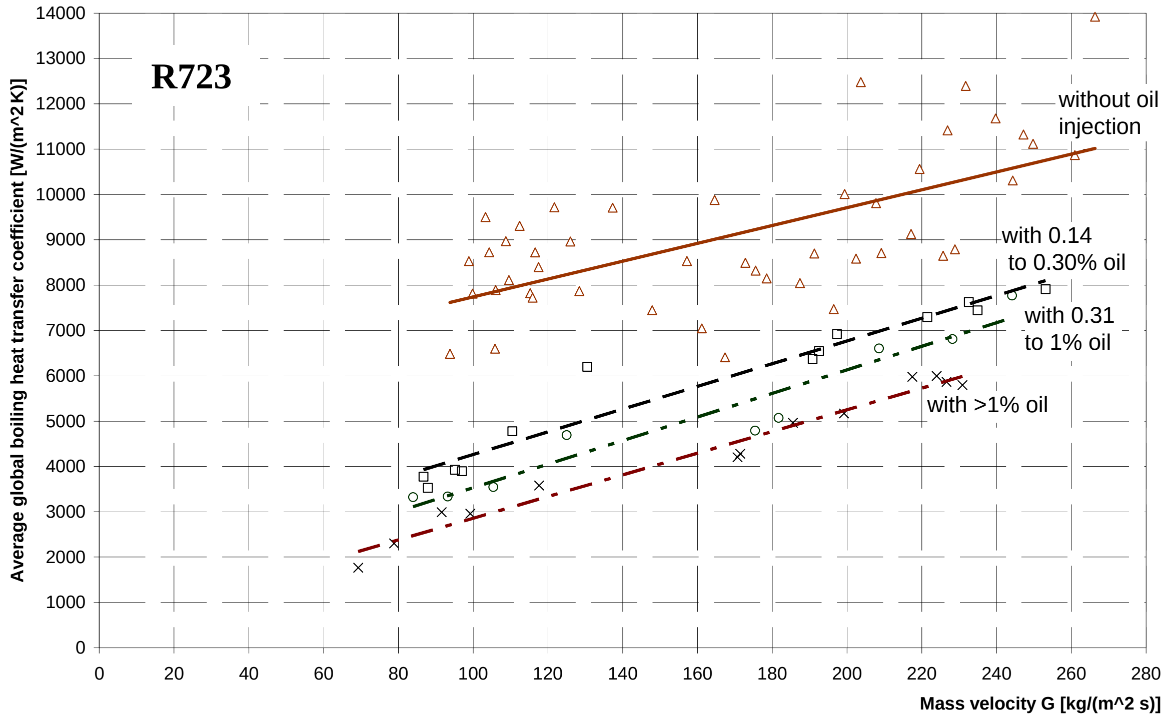

Figure 5: average global values of the boiling heat transfer coefficient h_{R723} for flow boiling of R723 versus mass flux for different oil contents.

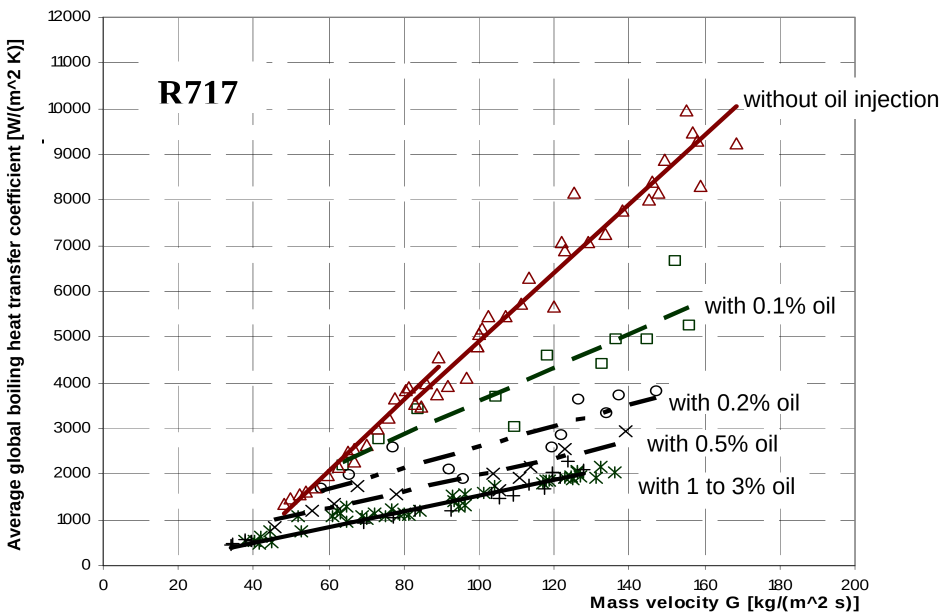

Figure 6: average global values of the boiling heat transfer coefficient h_{NH3} for flow boiling of ammonia (R717) versus mass flux for different oil contents.

The visual observations made through the sight glasses (Fig. 2) during tests with oil injection showed that in the Sector S1 the stratified-wavy flow pattern was predominant. On the other hand, the flow of the liquid refrigerant in the Sector S2 was almost annular, thus fostering the formation of a thin layer of oil. The significant reduction of the average heat transfer coefficient especially at high ammonia mass fluxes is mainly the result of the high insulating effect of the oil film in the Sector S2. In the case of R723 this effect is slightly reduced as a result of the miscibility of oil with the refrigerant.

5. CONCLUSIONS

- During the flow boiling of R723 without oil (or with no oil injection) average boiling heat transfer coefficients up to 12'000 W/(m^2K) have been measured with refrigerant mass fluxes of 260 kg/(m^2s). In comparison to ammonia the dependence of the heat transfer coefficient of R723 on mass flux is weaker.
- Small oil content (of fully-synthetic PAO oil with ISO-VG 68) causes a substantial reduction of the heat transfer coefficient (30 to 40% reduction). In the case of higher mass flow rates the impact of the oil was still observed, mainly in the evaporation zones with annular flow.
- However, compared to ammonia (R717), the impact of oil on the decrease of the average boiling heat transfer coefficient of the azeotropic mixture R723 (h_{R723}) was more attenuated. This can be interpreted as a result of the higher solubility of the oil in the refrigerant R723.
- There is no more decrease of heat transfer coefficient h_{R723} when the oil content exceeds 1%w/w.
- It is of particular importance to separate the oil efficiently at the discharge line of the compressor, mainly in plants with ammonia and R723 where high mass fluxes are strived for.

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