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Tahsin Boyman
*University of Applied Sciences of Central Switzerland*

Peter Aecherli
*University of Applied Sciences of Central Switzerland*

Anton Steiner W. Wettstein
*AG fur Kaltetechnik*

# FLOW BOILING OF AMMONIA IN SMOOTH HORIZONTAL TUBES IN THE PRESENCE OF IMMISCIBLE OIL

**Tahsin BOYMAN[1, 3], Peter AECHERLI[1], Anton STEINER[2]**

[1] Lucerne School of Engineering and Architecture (HTA Luzern)
University of Applied Sciences of Central Switzerland
Technikumstrasse 21, CH - 6048 Horw, Switzerland

[2] W. Wettstein AG für Kältetechnik,
Mattenstrasse 11, CH - 3073 Gümligen, Switzerland

[3] Corresponding author: tboyman@hta.fhz.ch / Phone +41 41 3493245 / Fax +41 41 3493960

## ABSTRACT

The direct evaporation of the refrigerant inside the tubes allows an important minimization of the plant refrigerant charge. In this paper experimental data on the flow boiling of ammonia inside smooth horizontal tubes and the decrease of the heat transfer coefficient due to the presence of an immiscible oil are presented. The tests have been carried out with plain tubes (with 14 mm internal diameter) for mass velocities from 40 to 170 kg/(m$^2$s) at evaporation saturation temperatures from −10°C to +10°C with heat fluxes from 10 to 50 kW/m$^2$. The flow boiling heat transfer coefficient (average values for the evaporation from 0.15 vapor content to 1.0) ranges for an oil-free operation between 1’000 to 10’000 W/(m$^2$s) depending on the mass velocity. Experiments run with a frequently used immiscible synthetic oil (viscosity grade ISO-VG 68) with oil contents from 0 to 3% (by weight) showed a significant decrease of the flow boiling heat transfer coefficient mainly at high mass velocities.

## 1. INTRODUCTION

A considerable advantage of the refrigeration systems using in-tube evaporation 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 to the lack) of the miscibility of the oil with the refrigerant. Shen and Groll (2003) published a review on the influence of lubricants on the boiling of refrigerants, stressing the importance of the modification of the local flow pattern distribution as a result of the oil presence.

The experimental data for the boiling of the natural refrigerant "ammonia" inside plain horizontal tubes is scarce, especially for the evaporation of ammonia in the presence of oil. Ohadi *et al.* (1996) reviewed the available correlations for two-phase flow heat transfer of ammonia and concluded that most of the existing models for local heat transfer of ammonia are only applicable to fully turbulent flow regimes under oil-free conditions. Therefore they emphasized the necessity of research work on the effects of oil contamination.

The use of ammonia in direct expansion systems is limited due to its immiscibility with the conventional lubricating oils. The immiscible oil forms a layer with a poor conductivity, thus causing a strong reduction of the heat transfer coefficient. Shah (1975) reported on the behavior of oil both in non-boiling and boiling regions as well as on the detrimental effect of an immiscible oil (Arctic 300) on the mean heat transfer coefficients. The experiments of Chaddock and Buzzard (1986) with small percentages of an immiscible oil (mineral oil Capella D) show that the heat transfer coefficient is reduced by 50% to 90% below that of the oil-free flow boiling of ammonia.

The significant decrease of the heat transfer coefficient of ammonia in 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 velocities between 30 to 65 kg/(m$^2$s) and the tests by Zürcher *et al.* (1998) with polyalkylene glycols with mass velocities from 50 to 80 kg/($m^2$s) show that with oil contents up to 3% (by weight) no major negative effect of the miscible oil on the heat transfer coefficient occurs. However, in order to run 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.

The present work is an experimental contribution to the *direct evaporation of ammonia* with the frequently used *synthetic oil* Gargoyle Arctic SHC 326 which is *immiscible with ammonia*. The results of the mean heat transfer coefficient measurements during the flow boiling are summarized with a short description of the visual observations at critical stages of the in-tube boiling.

## 2. EXPERIMENTAL TEST FACILITY AND THE TEST CONDITIONS

The test facility has been operated as refrigeration plant with the single-stage vapor compression principle. The flow circuit diagram of the test facility is given in Figure 1, the schematic arrangement of the measurement points on the evaporator in Figure 2. The secondary sides of the condenser and of the evaporator have been connected in a closed water-glycol loop in order to recover the heat. The temperatures have been measured in the ammonia loop and in the water-glycol loop with thermocouples calibrated with an accuracy of ±0.1 K. The refrigeration system is equipped with different expansion devices which can be switched on alternatively. The experiments presented in this paper have been carried out with the manually controlled expansion valve in order to run under constant operating conditions.

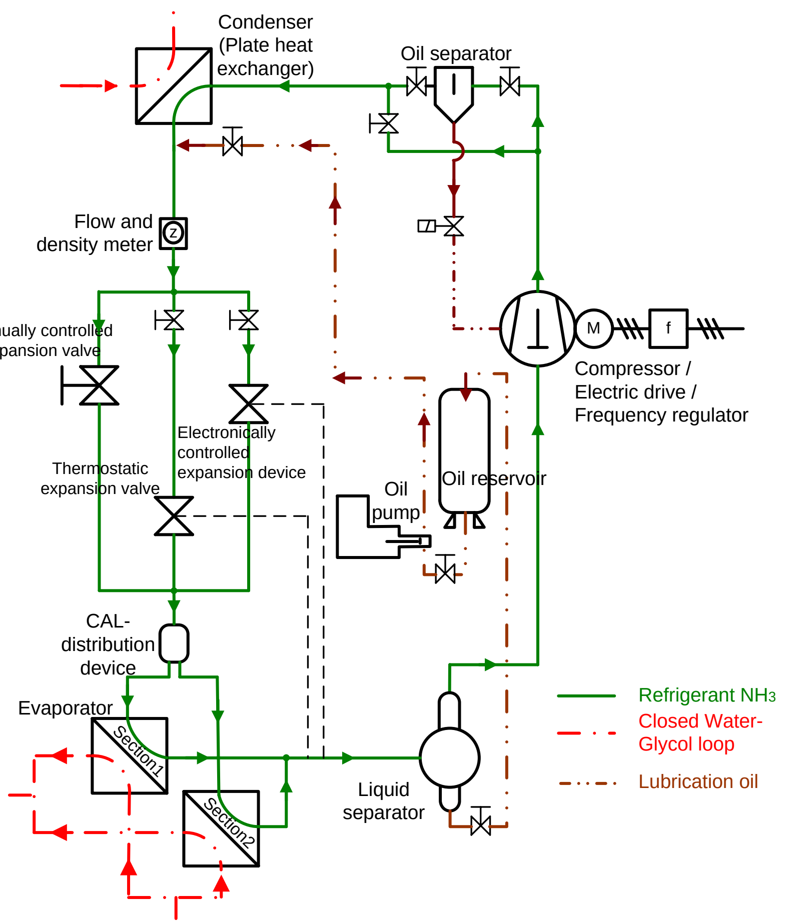


Figure 1: Flow circuit diagram of the test facility

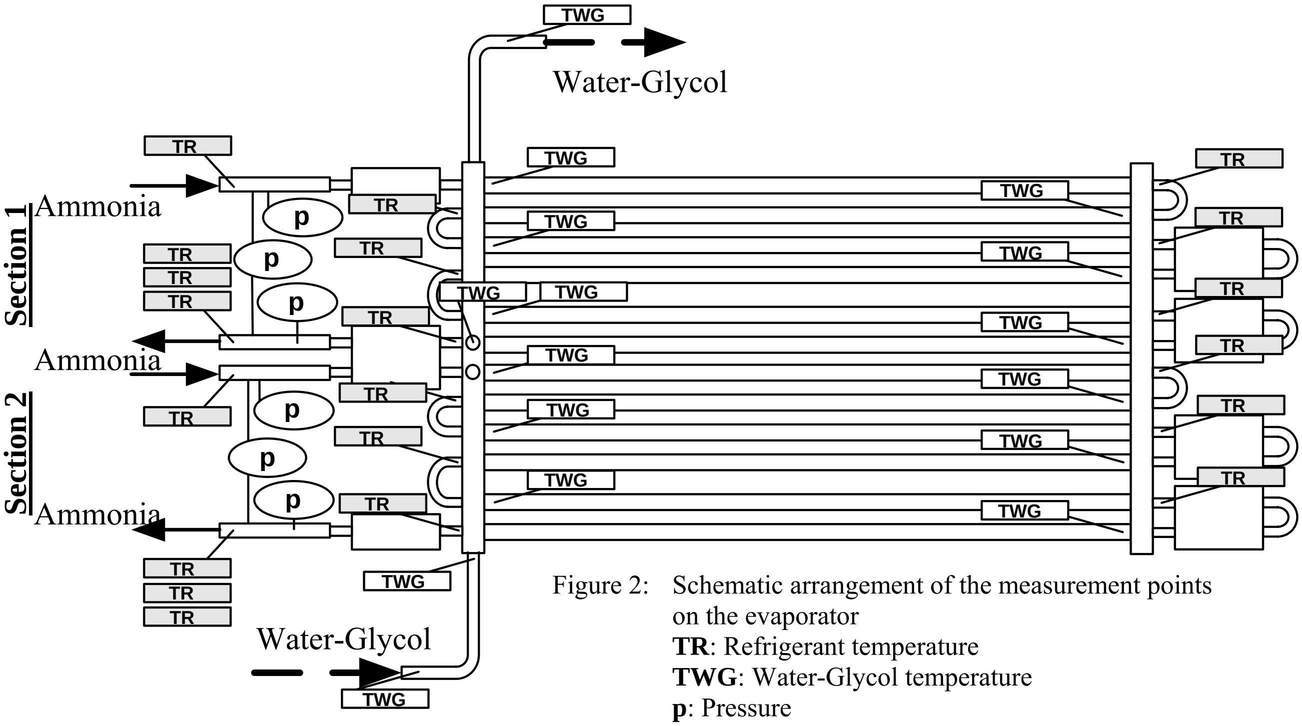


Figure 2: Schematic arrangement of the measurement points on the evaporator
**TR**: Refrigerant temperature
**TWG**: Water-Glycol temperature
**p**: Pressure

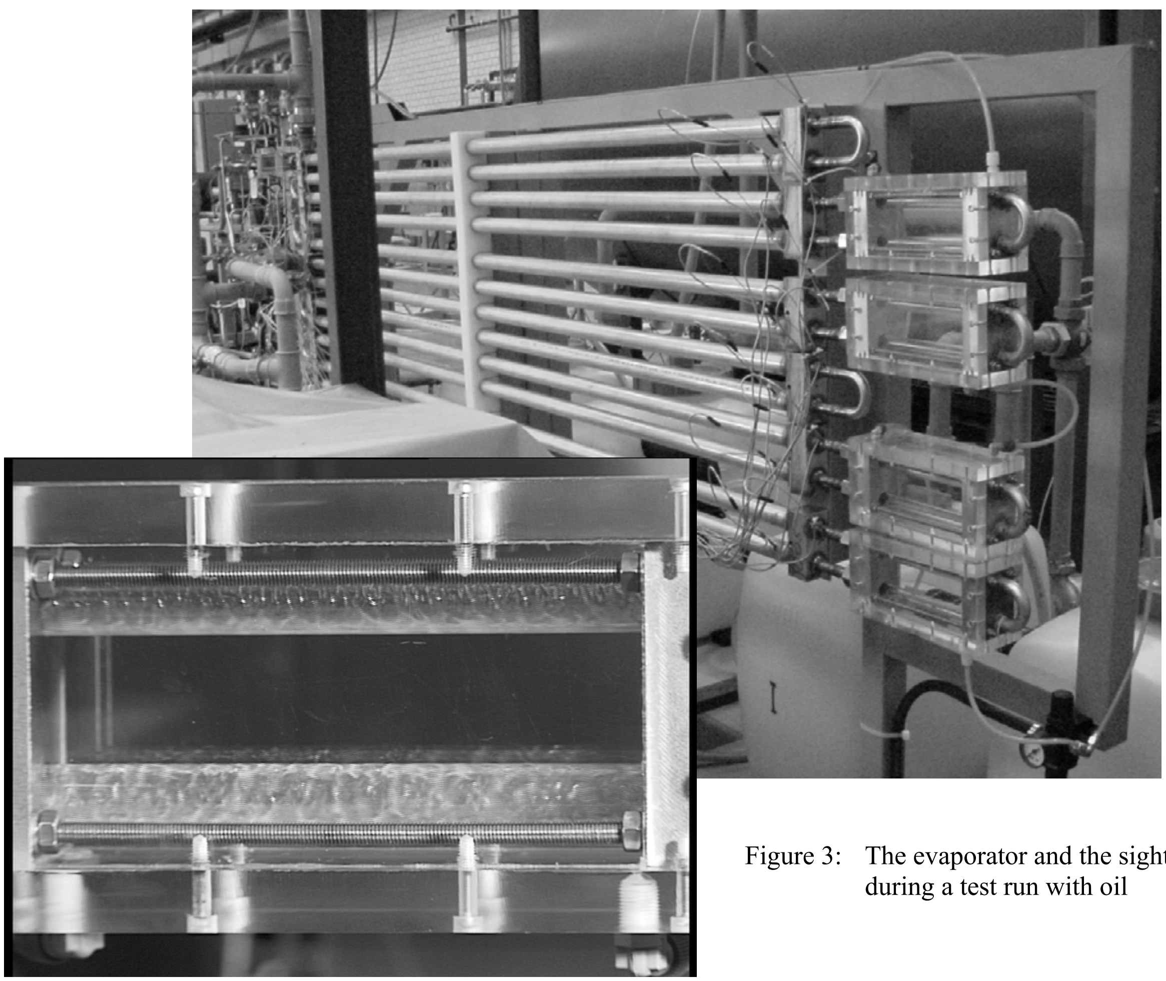

Figure 3: The evaporator and the sight glass during a test run with oil

The evaporator of the test facility consists of two sections each 12 m long. They can be operated in parallel or in series, thus allowing the operation in a wide range of mass velocities. Each section is a serpentine with horizontal straight tube sections of 2 m (plain stainless steel tube with 14 mm internal diameter) connected with U-bends with the same internal diameter. Water-glycol is fed in the concentric outer tube in counter current flow direction. The test facility is equipped with special tubular sight glasses with the same internal diameter as the evaporator tubes for the observation of the flow patterns (see Figure 3). The flow patterns at the inlets, at the outlets and at two intermediate stations of the both coils have been systematically observed.

The plant is provided with a condenser of welded plate type, thus considerably reducing the ammonia charge. A standard reciprocating compressor with air-cooled cylinder head blocks is driven electrically with frequency regulator allowing the adequate adjustment of the plant operating conditions coupled with the ammonia flow rate.

The tests for the present work have been performed with the commonly used immiscible oil Gargoyle Arctic SHC 326 (fully synthetic oil on the basis of polyalphaolefin and alkyl-aromatics, viscosity grade ISO-VG 68).

The oil content in the ammonia loop has been adjusted through a special circuit and a pump from 0 up to 3% (by weight). The oil injected prior to the 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 "liquid ammonia – oil" are measured after the condenser online by means of a Coriolis type instrument.

The experimental conditions for this study are:

- Evaporation temperature (at the evaporator inlet): $t_{sat\ evap\_in} = -10$ to $+10°C$
- Mass velocity: $G = 40$ to $170\ kg/(m^2 \cdot s)$
- Refrigeration capacity: $Q_{dot\ evap} = 15$ to $30\ kW$
- Heat fluxes at evaporator: $q_{dot\ evap} = 10$ to $50\ kW/m^2$
- Vapor void fraction (=vapor quality) (at the evaporator inlet): $x_{evap\_in}$ = approx. 0.15
- Oil content in the ammonia-oil mixture in the evaporator: $\xi_{oil} = 0$ to 3% (by weight)
- Overheat at the evaporator exit: $\Delta T_{overheat} = 15\ K$ to $20\ K$

## 3. DATA REDUCTION AND TEST RESULTS

The temperatures have been measured in the evaporator every 2 m both in the ammonia and in the water-glycol loops. These temperature profiles (see Figures 4 and 5) as well as the matching between the beginning of the ammonia overheating (sector S3) and the heat flux increase on the water-glycol side have been used for checking the steady thermal conditions. The cross checking of the heat powers on the ammonia and on the water-glycols loops showed an accuracy of ±2%.

The Sections 1 and 2 of the evaporator are geometrically identical (each 12 m long). During the tests with lower mass velocities [up to 90 $kg/(m^2 \cdot s)$] the evaporator has been operated with Sections 1 and 2 in parallel. The tests 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 4).

During the tests with high mass velocities [from 80 $kg/(m^2 \cdot s)$] the Sections 1 and 2 have been operated in series (Figure 5). The ammonia has been introduced with approx. 0.15 vapor quality into the Sector S1 (corresponding to Section 1 in Figure 2) and evaporated up to an intermediate vapor quality, went through a connection pipe provided with a valve (thus causing some pressure drop) before entering the Sector S2. The Sector 2 is the fragment of the heat exchanger where the evaporation up to the saturated vapor condition takes place and has a variable length. In the last fragment (labeled as Sector 3 in Figure 5) the ammonia will be overheated (In the serial operation mode the Sectors 2 and 3 correspond together to the Section 2 in Figure 2).

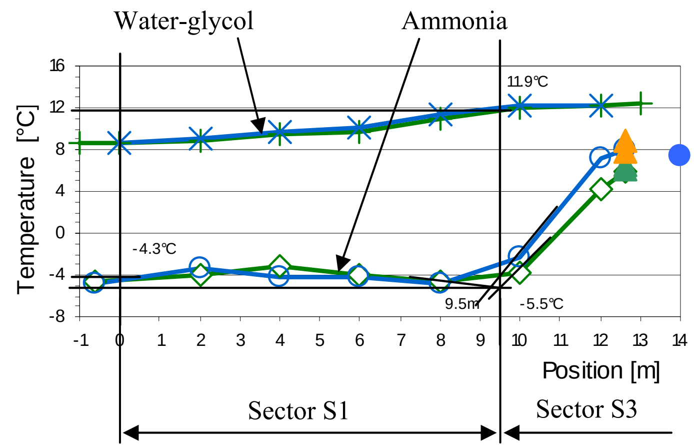


Figure 4: Temperature profiles along the evaporator coils during test runs with evaporator sections operated in parallel (Evaporation from $x_{evap\_in} = 0.15$ to 1.0 in Sector S1, Overheating in Sector S3)

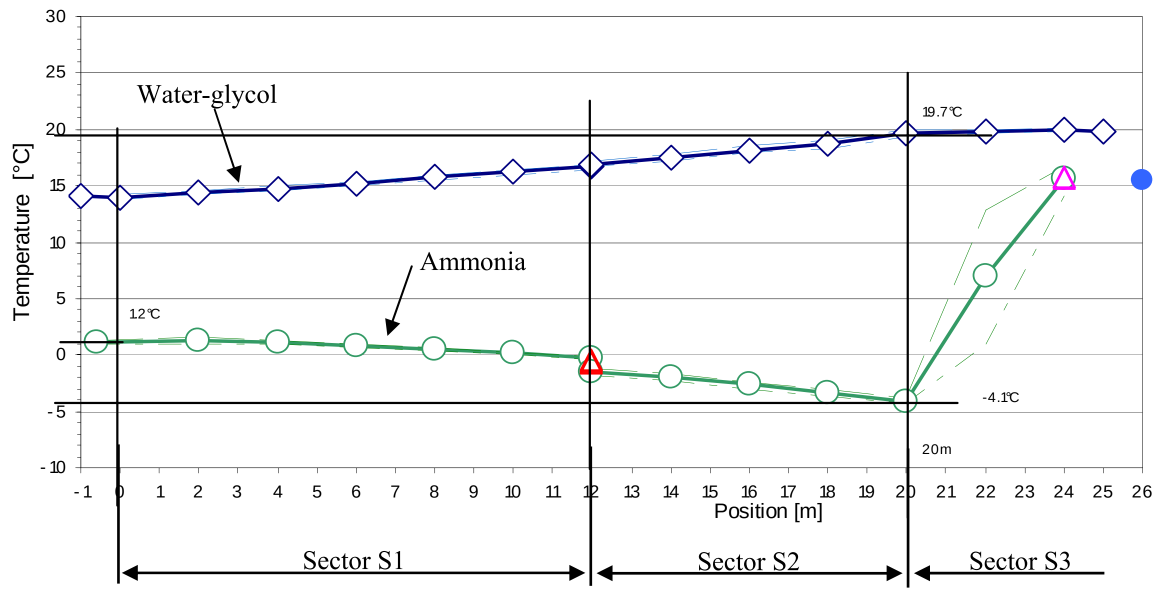


Figure 5: Temperature profiles along the evaporator coils during test runs with evaporator sections operated in series (Evaporation from $x_{evap\_in} = 0.15$ to 1.0 in Sectors S1 and S2, Overheating in Sector S3)

The heat rate transferred in each sector of the evaporator and the average heat flux have been computed with the data from the water-glycol side:

$$Q_{dot\_WG\_sector} = m_{dot\_WG} * c_{p\_WG} * (\Delta T_{WG\_sector})$$

(with WG as abbreviation for Water-Glycol)

The global heat transfer coefficient averaged over the sector is:

$$U_{ref\_ri} = Q_{dot\_WG\_sector} / ( 2 * \pi * r_i * \ell_{sector} * \Delta T_{m\_log} )$$

with:
$\Delta T_{m\_log}$ : logarithmic mean temperature difference on the inlet and outlet stations of the considered sector and
$r_i$ : internal radius of the plain tube (which has been considered as reference radius).

The heat transfer coefficient for flow boiling of ammonia $h_{NH3}$ is given as the average for the evaporation from $x_{evap\_in}$= 0.15 (approx.) to $x_{evap}$ = 1 according to the following equation:

$$[1/(h_{NH3} * r_i )] = [1/( U_{ref\_ri} * r_i )] - [(1/k_{steel}) * \ln(r_o / r_i)] - [1/( h_{WG} * r_o)]$$

with:
$k_{steel}$ : thermal conductivity of stainless steel and
$r_o$ : outer radius of the plain tube.

The tests with lower mass velocities [up to 90 kg/(m$^2$·s)] were carried out with the evaporator sections in parallel, both sections running under similar thermal and flow conditions. Thereafter it was possible to use the temperature values averaged from both sections (corresponding to “Sector S1” in Figure 4) for the data reduction.

For high mass velocities [from 80 kg/(m$^2$·s)] the evaporator sections were connected in series. The flow boiling heat transfer coefficients $h_{NH3}$ given in Figure 6 are the area based averages of $h_{NH3}$ values from each sector (i.e. sectors S1 and S2).

The oil content (“mass flow rate of the injected oil” compared to the “ammonia-oil mass flow rate”) was varied from 0.1% to 3%. The tests were run in two phases, Phase 2 taking place approx. 500 hours after Phase 1. During Phase 1 the oil content could be tuned between 1 to 3%: in a first step with evaporator sections connected in parallel without oil injection and later with oil injection. After test runs with oil injection the plant was operated for approx. 10 hours with a high ammonia flow rate without any oil injection. In the second period of the Phase 1 both evaporator sections were connected in series and tests performed in the same order, i.e. in the first period without and thereafter with oil injection. During the test series labeled as Phase 2 it was possible to adjust the oil injection rate more accurately between 0.1 and 3% with a modified pump.

The results of the experiments are given in Figures 6 and 7. The average values of the heat transfer coefficient are summarized in Figure 6 as sets of tests without or with oil injection, with evaporator sections connected in parallel or in series. During the tests without oil injection with lower mass velocities (i.e. test runs in the “parallel connection mode”) a 30% decrease of the heat transfer coefficient is observed from Phase 1 to Phase 2, most probably as a result of the slight oil traces in the evaporation coils which could not be completely eliminated. A similar decrease was not noticed in the “serial connection mode”. The tests with oil contents from 1 to 3% showed a significant reduction of the heat transfer coefficient mainly at high mass velocities. The details of this rapid decrease were investigated during the Phase 2 (see Figure 6, oil contents from 0.1 to 0.4%).

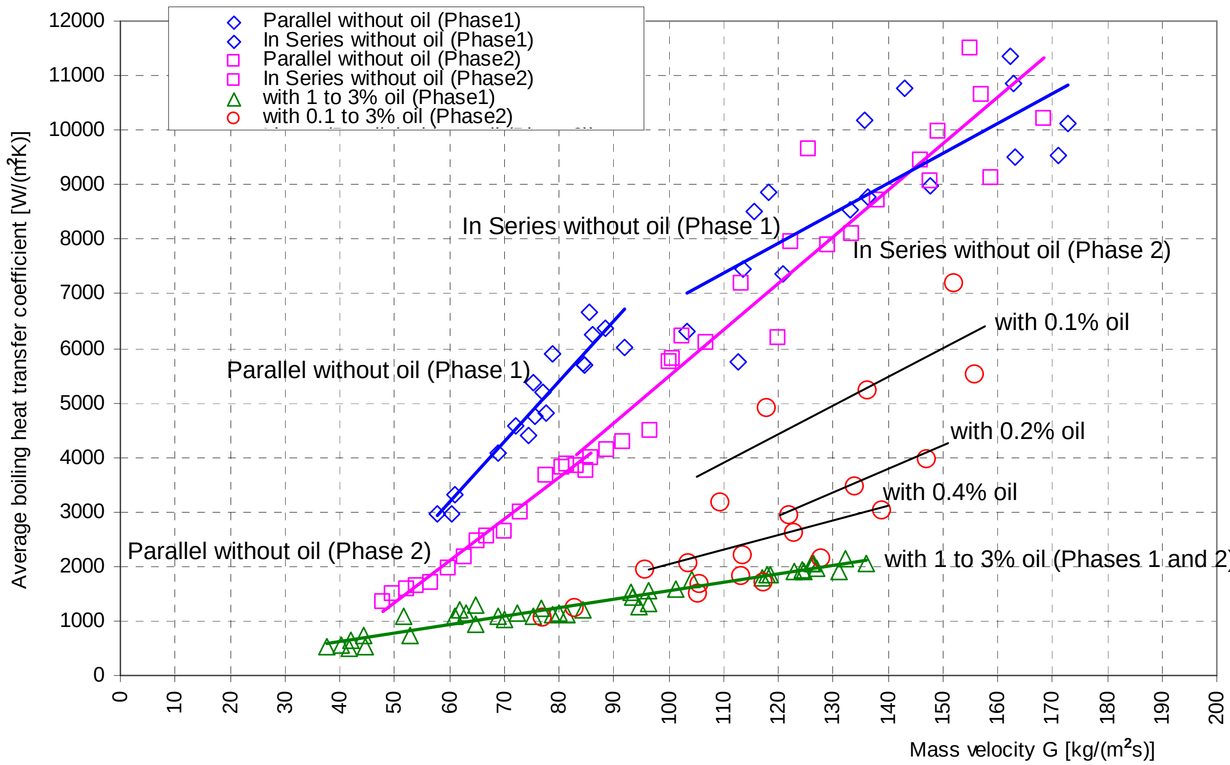


Figure 6: Average values of the heat transfer coefficient $h_{NH3}$ for flow boiling of ammonia versus mass velocity for different oil contents in ammonia

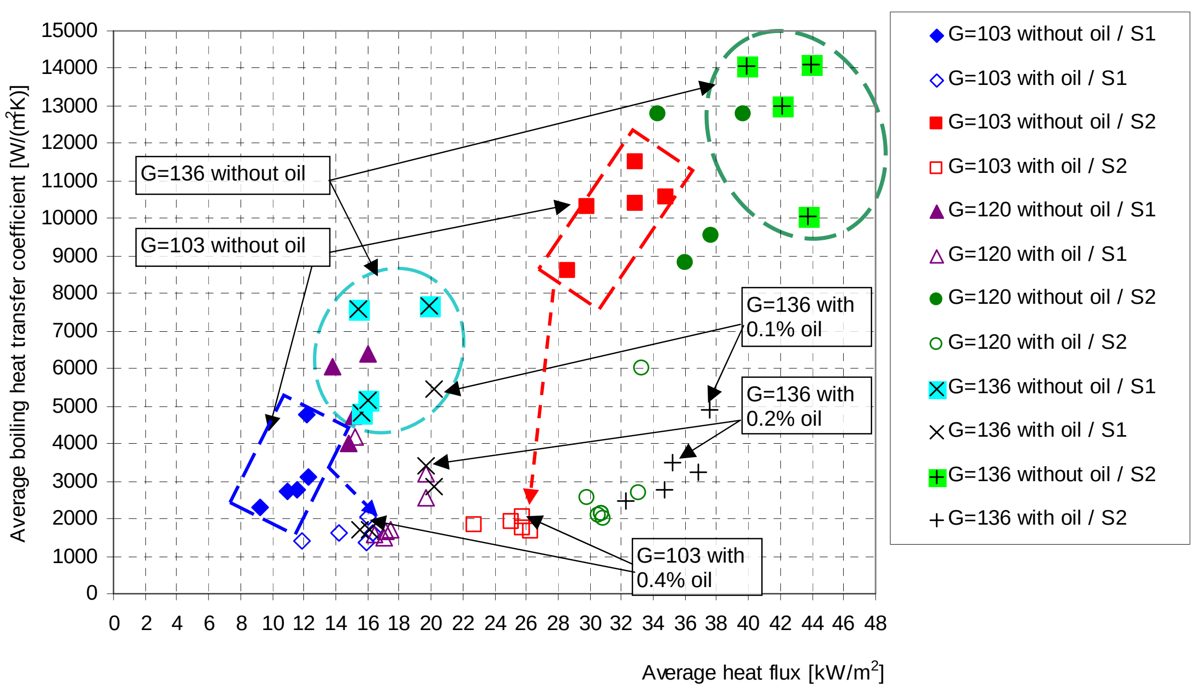


Figure 7: Average values of the heat transfer coefficient for flow boiling of ammonia in the sectors S1 and S2 {for selected mass velocities G [kg/(m²s)]} versus average heat flux in the corresponding sector of the evaporator

The significant reduction of the average heat transfer coefficient (averaged over the entire evaporation path from vapor content of 0.15 to 1.0) especially at high mass velocities is mainly the result of the high insulating effect of the oil film in the Sector S2. This effect can be followed in Figure 7: the average values of the heat transfer coefficients in the sectors S1 and S2 are presented versus the average heat flux (both values averaged in this case over the corresponding sector of the evaporator). For selected mass velocities G [G = 103, 120 and 136 kg/(m$^2$s), i.e. test runs in "serial connection mode"] the averages were computed individually for each sector (labels S1 respectively S2 in Figure 7). It was observed that the coefficient specific for the Sector S1 decreased less significantly than that one specific for the Sector S2. For example, with G = 103 kg/(m$^2$s) the value of $h_{NH3}$ in the Sector S1 lies between 2'000 and 5'000 W/(m$^2$K) and drops slightly below 2'000 W/(m$^2$K) while 0.4% oil is injected. This is far below the decrease from approx. 10'000 W/(m$^2$K) to 2'000 W/(m$^2$K) in the Sector S2 for the same conditions.

The conclusion of Shen and Groll (2003) stressing the importance of the modification of the local flow pattern distribution as a result of the oil presence can be confirmed. The visual observations made through the sight glasses (Figure 3) during tests with oil injection showed that in the Sector S1 the stratified-wavy flow pattern was longer predominant. On the other hand the flow of the liquid ammonia in the Sector S2 is annular, thus fostering the formation of a thin layer of oil with meandering rings. This oil layer is circulating at a lower velocity than the ammonia vapor, but at the present stage it is not possible to give neither its precise thickness nor its velocity.

## 6. CONCLUSIONS

- During the flow boiling of ammonia without oil (or with no oil injection) average heat transfer coefficients up to 10'000 W/(m$^2$K) have been measured with refrigerant mass velocities of 160 kg/(m$^2$s).
- But a small oil content of 0.1% (immiscible oil with ISO-VG 68) causes a considerable reduction of the heat transfer coefficient (In the case of high mass velocities approx. half of the values without oil).
- The decrease of the heat transfer coefficient is more significant in the sector of the evaporator where boiling with annular flow pattern occurs.
- There is no more decrease when the oil concentration exceeds 1%.
- It is of particular importance to separate the immiscible oil effectively at the discharge of the compressor, mainly in the plants where high mass velocities are strived for.

## REFERENCES

1. Aecherli P. (2002): Ammoniak-Kälteanlage der HTA Luzern - Einfluss des nichtlöslichen Öles Gargoyle Arctic SHC 326 im Verdampfer, FH-Diploma thesis in Thermal Engineering, HTA Luzern
2. Boyman et al. (1997): Experiments and experiences with small vapor compression refrigeration plants with ammonia and miscible oil, *Heat Transfer Issues in Natural Refrigerants, Proc. of IIR conference in Nov. 1997,* College Park (USA), pp. 203-220
3. Chaddock J., Buzzard G. (1986): Film coefficients for in-tube evaporation of ammonia and R502 with and without small percentages of mineral oil, *ASHRAE Trans*., Vol. 92, Part 1A, pp. 22-40
4. Gross, U. (1994): Einfluss von Oel auf die Zwangskonvektion im Verdampferrohr, *Ki Luft- und Kaeltetechnik*, Nr. 6, pp. 269-274
5. Ohadi M. M., Li S. S., Radermacher R., Dessiatoun S. (1996): Critical review of available correlations for two-phase flow heat transfer of ammonia, *Int. J. Refrig*., Vol. 19, No. 4, pp. 272-284
6. Shah M. M. (1975): Visual observations in an ammonia evaporator, *ASHRAE Trans*., Vol. 81, Part 1, pp. 295-306
7. Shen B., Groll E. A. (2003): A critical review of the influence of lubricants on the boiling of refrigerants, *Proc. of the 21st IIR Int. Congress of Refrigeration in Aug. 2003,* Washington D.C. (USA), Paper ICR0032
8. Zürcher, O., Favrat D., Thome, J.R. (1998): Evaporation de mélanges d'ammoniac et d'huile dans des tubes, *Rapport final, Office fédéral de l'énergie (Berne) (Avril 1998).*

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