DKV-Forschungsbericht Nr. 71

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2005

978-3-932715-74-7

Forschungsberichte

des

Deutschen Kälte- und Klimatechnischen Vereins

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Wärmeübergang beim Behältersieden

von Kältemittel/Öl-Gemischen

an einem Glattrohr und einem Hochleistungsrohr

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Schlaitdorf

2005

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Kurzfassung

In dieser Arbeit wurde der Einfluß eines nieder-viskosen Kältemaschinenöls auf Basis von Polyolestern (POE-Öl) auf den Wärmeübergang beim Behältersieden untersucht.

Ein glattes Kupferrohr mit feinsandgestrahlter Oberfläche von 18,0 mm Außendurchmesser sowie ein Hochleistungsrohr GEWA-B mit 18,7 mm Außendurchmesser, ebenfalls aus Kupfer, dienten bei den Wärmeübergangsmessungen der Kältemittel R134a, R410A und R507 als Heizfläche. Das Hochleistungsrohr besitzt eine gewalzte Oberfläche mit umlaufenden hinterschnittenen Kanälen. Das Kältemittel strömt durch Poren in bzw. aus den Kanälen. Für die Wärmeübergangsmessungen mit dem Kältemittel R404A wurde nur das feinsandgestrahlte Glattrohr eingesetzt. Der Ölmassenanteil wurde bei allen Messungen zwischen 0,0 % und 10 % variiert, und es wurden Siedetemperaturen zwischen -28,6°C und +20,1°C untersucht.

Für alle am GEWA-B-Rohr untersuchten reinen Kältemittel wird der Wärmeübergangskoeffizient beim Erhöhen der Wärmestromdichte von 20.000 W/m² auf 40.000 W/m² vermindert. Diese Verminderung tritt vorwiegend und verstärkt bei normierten Drücken um $p^* \approx 0{,}1$ auf. Sie wird auf ein teilweises Austrocknen der hinterschnittenen Kanalstruktur des Hochleistungsrohres zurückgeführt. Für geringere normierte Drücke liegt das teilweise Austrocknen der Kanalstruktur ebenfalls vor, allerdings bewirkt die Erhöhung der Wärmestromdichte keine Verminderung des Wärmeübergangskoeffizienten, sondern nur eine geringere Steigung der Ausgleichskurve der gemessenen Wärmeübergangskoeffizienten im doppelt-logarithmischen $\alpha,\dot{q}$-Diagramm.

Das Kältemaschinenöl bewirkt sowohl Verbesserungen als auch Verminderungen des Wärmeübergangs, abhängig von Ölmassenanteil, Wärmestromdichte und Siedetemperatur. Am Glattrohr treten die Verbesserungen im Wärmeübergang vorwiegend für 3 % Ölmassenanteil auf. Für die R134a/Öl-Gemische betragen die Verbesserungen bis zu 20 %, für die R410A/Öl-Gemische bis zu 35 %, für die R507/Öl-Gemische bis zu 75 % und für die R404A/Öl-Gemische bis zu 45 %. Dagegen wird der Wärmeübergang durch das Öl am Hochleistungsrohr GEWA-B nur für die R134a/Öl-Gemische im Bereich des teilweisen Austrocknens verbessert. Die Verbesserungen werden mit sinkender Siedetemperatur größer und betragen bis zu 85% bei 0,5% Ölmassenanteil. Verminderungen treten an beiden Rohren vor allem im Bereich der nicht vollständigen Mischbarkeit des Öls im Kältemittel (Mischungslücke) auf, sie betragen bis zu 80 % bei 10 % Ölmassenanteil.

Für den Vergleich von Hochleistungsrohr und Glattrohr wird das Verhältnis ROE $\alpha_{\mathrm{GEWA\text{-}B\text{-}Rohr}} / \alpha_{\mathrm{Glattrohr}}$ gebildet, das die Steigerung des Wärmeübergangs am Hochleistungsrohr gegenüber dem Glattrohr beschreibt. Die Werte des Verhältnis ROE steigen mit sinkender Siedetemperatur und sinkender Wärmestromdichte für die reinen Kältemittel. Für die R134a/Öl-Gemische wird bei 0,5 % Ölmassenanteil das Verhältnis ROE größer und durch weitere Ölzugaben kaum verändert bzw. leicht verringert. Für die R410A/Öl- und die R507/Öl-Gemische werden die Werte des Verhältnis ROE vermindert.

Die lokalen Wärmeübergangskoeffizienten werden an den beiden Rohren für die jeweils untersuchten Kältemittel/Öl-Gemische in Abhängigkeit von Ölmassenanteil, Wärmestromdichte und Siedetemperatur diskutiert. Am feinsandgestrahlten Glattrohr ist der Öleinfluß beim Sieden der Kältemittel/Öl-Gemische sehr gering vom Umfangswinkel abhängig. Dagegen zeigt der Öleinfluß am Hochleistungsrohr GEWA-B eine nicht zu vernachlässigende Abhängigkeit vom Umfangswinkel. Beim Sieden der Kältemittel/Öl-Gemische am GEWA-B-Rohr ist das teilweise Austrocknen wie auch das teilweise Verstopfen der Kanäle mit Öl vom Umfangswinkel und dem Ölanteil abhängig.

Die Messungen haben gezeigt, daß die Mischbarkeit des Öls in einem Kältemittel den Wärmeübergang beim Behältersieden stark beeinflußt: Im Bereich der Mischungslücke wird der Wärmeübergang beim Behältersieden stark vermindert. Im Bereich der vollständigen Mischbarkeit des Öls im Kältemittel zeigen sich dagegen zum Teil enorme Verbesserungen im Wärmeübergang beim Behältersieden. Bei Kältemittelgemischen ist nicht die Mischbarkeit des Öls in dem jeweiligen Kältemittelgemisch relevant, sondern vorwiegend die Mischbarkeit des Öls mit den jeweiligen Komponenten dieses Kältemittelgemisches.

Korrelationen aus der Literatur zum Einfluß des Öls auf den Wärmeübergang beim Behältersieden an Glattrohren können für die untersuchten Kältemittel/Öl-Gemische nicht ohne Anpassung verwendet werden. Der Einfluß der Mischungslücke einer Komponente von Kältemittelgemischen ist bislang nicht untersucht worden. Es wird ein Modell zur Berechnung des Wärmeübergangs beim Behältersieden von Gemischen aus Kältemaschinenöl und Kältemittelgemischen vorgeschlagen. Dieses Modell berücksichtigt auch den Bereich der Mischungslücke der Komponenten von Kältemittelgemischen mit dem Öl.

In der Literatur ist bislang kein Modell zum Wärmeübergang beim Behältersieden von Kältemittel/Öl-Gemischen an Hochleistungsstrukturen bekannt. Für die Berechnung des Wärmeübergangskoeffizienten von reinen Kältemitteln am Hochleistungsrohr GEWA-B wurde das Modell von Chien um den Einfluß des Druckes erweitert. Eine Überlegung zum Einfluß des Öls auf den Wärmeübergang am GEWA-B-Rohr wird beschrieben, die Aussagen über das teilweise Austrocknen der Kanalstruktur zuläßt.

Abstract

This thesis describes the experimental and theoretical results of pool boiling heat transfer measurements at the outside of horizontal tubes to different refrigerant-oil mixtures.

The schematic set-up of the test facility for the heat transfer measurements of boiling refrigerant-oil mixtures is shown in Fig. 10 (s. page 59). The evaporator consists of two concentrical glass cylinders to allow visual observation of the boiling liquid. The refrigerant-oil mixture is contained inside the inner prestressed glass cylinder. This cylinder with its flanges is surrounded by an outer glass cylinder. Ethanol flows in the annulus between the cylinders and their flanges. The ethanol is kept at constant temperature by a cryostat.

The test tube is mounted at the lower flange of the inner cylinder. It contains an electric heating cartridge and 4 thermocouples for measuring the average wall temperature (ϑ_W). Further thermocouples inside capillary tubes indicate the liquid temperature below the test tube ($\vartheta_{F,a}$), the temperature in the two-phase region (ϑ_S) above the test tube and the temperature of the vapour (ϑ_D). The glass cylinders are installed with an inclination of 15° to the horizontal line to achieve a greater liquid height above the test tube which is installed exactly horizontal.

The test tubes are both a sandblasted smooth copper tube and an enhanced copper tube named GEWA-B, with fins rolled over (compare Fig. 14 at page 64 and Fig. 15 at page 65) manufactured by Wieland-Werke AG, Ulm, Germany. The geometric parameters of both tubes are shown in Fig. 11 (s.

page 61) and are listed in Tab. 7 (s. page 62). The enhanced tube could be described as porous structure with micro-channels.

The saturation temperature was varied from +20.1°C to -28.6°C and oil mass fractions from 0 % up to 10 %. The measurements were performed with decreasing heat fluxes from 80 000 W/m² to 1 000 W/m² in order to avoid hysteresis effects.

Before each measurement started, the tube had been heated for about 12 hours with the maximum heat flux. Although the author knows about the importance of the hysteresis effect especially for refrigerant-oil mixtures, the aim of this study was the influence of the oil on the fully developed pool boiling heat transfer. The oil mass fraction w is defined as the mass of oil per total mass of liquid refrigerant-oil mixture. It is determined by weighing the oil mass and the refrigerant mass before filling in.

The measurements on the smooth tube were performed for refrigerant-oil mixtures with R134a, R410A, R507 and R404A. The measurements on the enhanced tube were performed for refrigerant-oil mixtures with R134a, R410A and R507. Only one lubricating oil was used. It was a low-viscous polyolester-based oil named Reniso Triton SEZ32 by Fuchs DEA Schmierstoffe GmbH & Co KG, Mannheim, Germany. The equations for calculating the properties of the refrigerant-oil mixtures are listed in appendix A.2 (s. page 195).

Each refrigerant (R134a, R410A, R507 and R404A) has a range of immiscibility with this lubricating oil. The ranges of immiscibility of the refrigerant R134a and the refrigerant mixtures R410A, R507 and R404A are shown in Tab. 5 (s. page 55) and Fig. 7 (s. page 56). It should be noted that the observed ranges of immiscibility of the refrigerant mixtures and the oil differ from the ranges given by the oil manufacturer. For the components R125, R32 and R143a of the refrigerant mixtures the manufacturer of the oil did not give any information about the range of immiscibility with the oil. The observed ranges of these immiscibilities are shown in Tab. 6 (s. page 55) and Fig. 7 (s. page 56).

The refrigerant-oil mixture could change from total miscibility to immiscibility or vice versa when boiling occurs. In this case two opposite effects are in superposition as schematically shown in Fig. 5 (s. page 53). The measurement point is moved from the region of immiscibility into the region of miscibility because the fluid temperature ϑ_{fluid} next to the heated surface $\vartheta_{fluid} = (\vartheta_S + \vartheta_W) / 2$ is higher than the temperature of the pool ϑ_S. The effective oil mass fraction increases, however, due to oil enrichment in the boiling refrigerant. So we do not know whether the fluid next to the heated surface is in the region of total miscibility or in the region of immiscibility, or if the fluid is always changing between these two regions.

The pool boiling heat transfer coefficients for the pure refrigerants on the smooth tube show the well known behaviour. The heat transfer increases with increasing saturation temperature and increasing heat flux. The slope of the boiling curves is greater than the free convection curves.

For the R134a-oil mixtures the same behaviour is found. Up to 5 % oil mass fraction there is only small influence of the oil. But at 10 % oil mass fraction the heat transfer gets worse, see Fig. 18 (s. page 75). The influence of the saturation temperature on the pool boiling heat transfer is much greater than that of the oil, for oil mass fractions up to 5 %. The heat transfer coefficients as function of the oil mass fraction show a minimum at 0.5 % and a maximum at 3 % oil mass fraction due to the saturation temperature and the heat flux. For high saturation temperatures and high heat fluxes at 3 % oil mass fraction the pool boiling heat transfer coefficients are greater than of the pure R134a. The enhancement due to the oil is up to 20 % and the deterioration at 10 % oil mass fraction is up to 60 %, compare Fig. 19 (s. page 76).

The lines of the pool boiling heat transfer coefficients for the R410A-oil mixtures on the smooth tube are changing from straight lines to curves for oil mass fractions greater/equal 1 %. The slope of the curves decreases with increasing heat flux for lower saturation temperatures. The influence of the saturation temperature decreases with increasing oil amounts. The influence of oil mass fractions up to 10 % is less than the influence of the saturation temperature between -28.6°C and +20.1°C. But the influence of the oil on the pool boiling heat transfer decreases also for decreasing saturation temperature, see Fig. 20 (s. page 78). The ratio of the heat transfer coefficients of the refrigerant-oil mixtures to those of the pure refrigerant increases for decreasing saturation temperature. At +20.1°C the heat transfer decreases with increasing oil mass fraction. The deterioration is up to 70 %. At saturation temperatures less/equal +0.2°C a maximum in the heat transfer coefficients is observed at 3 % oil mass fraction. The heat transfer enhancement due to the oil is up to 35 %. For example at 10 000 W/m² the heat transfer coefficient at -19.9°C with 10 % oil mass fraction is about 8 % greater than the heat transfer coefficient of the pure R410A, compare Fig. 21 (s. page 79).

With increasing amounts of oil the influence of the saturation temperature on the pool boiling heat transfer of R507-oil mixtures on the smooth tube decreases. For increasing saturation temperature the influence of the oil mass fraction on the heat transfer increases, compare Fig. 22 (s. page 81). The ratio $\alpha_{mix}/\alpha_{ref}$ increases for decreasing heat flux and decreasing saturation temperature as shown in Fig. 23 (s. page 82). The enhancement in heat transfer due to the oil is up to 75 % and the deterioration is up to 80 %. For R507-oil mixtures with oil mass fractions greater/equal 1 % an unexpected behaviour of the heat transfer coefficients for higher heat fluxes at saturation temperatures greater/equal +0.2°C was observed. Normally the heat transfer coefficient increases with increasing heat flux. However, for example with increasing heat flux from 40 000 W/m² to 50 000 W/m² the heat transfer coefficient of the R507-oil mixture with 3 % oil mass fraction decreases about 10 % at +10.1°C, see Fig. 22 (s. page 81). This behaviour starts for 1 % oil mass fraction at +20.1°C and 60 000 W/m², for 3 % oil mass fraction at +0.2°C and 40 000 W/m² and at 30 000 W/m² for 5 % oil mass fraction. This behaviour is caused by the immiscibility of the POE oil with the component R143a of the refrigerant mixture R507 for oil mass fraction greater 1 %. So the oil mass fraction next to the heating surface increases into the range of immiscibility caused by an increase in heat flux. This deteriorates the heat transfer at saturation temperatures greater/equal +0.2°C.

The pool boiling heat transfer of R404A-oil mixtures on the smooth tube shows the same behaviour as for R507-oil mixtures. It was expected that the decrease in heat transfer due to the oil is greater for R404A-oil mixtures than for R507-oil mixtures because of the higher mole fractions of the component R143a. But the influence of the oil on the pool boiling heat transfer for R404A is less than for R507; compare Fig. 24 (s. page 84) and Fig. 25 (s. page 85). This means that the values of the ratio αmix/αref for R404A differ less from 1.0 than for R507. This was found especially for parameter values at which an increase in heat transfer due to the oil for the R507 has been measured. The increase in pool boiling heat transfer of R404A due to the oil reaches up to 45 % and the decrease is up to 80 %. But the effect of immiscibility of the component R143a for oil mass fractions greater than 1 % at saturation temperatures higher than +0.2°C is much the same.

The surface roughness of the smooth tube was measured again after the completed heat transfer measurements. The roughness was decreased during the period of the measurements from R_a = 0.247 µm to R_a = 0.19 µm. So it seems that the oil reduces the roughness of the boiling surface for certain conditions, which have not been investigated here.

For all pure refrigerants the pool boiling curves on the enhanced tube are different from those of the smooth tube. The influence of heat flux for the enhanced tube is less. The values of the pool boiling heat transfer coefficients on the enhanced tube are much higher than for the smooth tube. Normally, for the GEWA-B tube the heat transfer increases with increasing heat flux. However, for reduced pressures about $p^* = 0.107$ ($\rho' / \rho'' = 59$) the values of the heat transfer coefficients at 20 000 W/m² are higher than the values at 40 000 W/m². Other authors in the literature e.g. Wang [92] and Chien [42] also observed this effect which is explained by a partial dry-out of the micro-channels. This partial dryout could occur if more refrigerant evaporates than liquid refrigerant flows back into the micro-channels of the enhanced tube surface. So the heat transfer coefficients decrease. It should be recognized that this partial dry-out is different to the dry-out at the critical heat flux. The partial dry-out is observed for reduced pressures less than about 0.17. The partial dry-out effects the reduction of the heat transfer coefficients for increasing heat flux from 20 000 W/m² to 40 000 W/m² for the reduced pressures between 0.17 and 0.10. For lower reduced pressures the slope of the boiling curves in the $\alpha,\dot{q}$-diagrams in the range of heat fluxes less than 20 000 W/m^2 is less than for higher reduced pressures. This is caused by the start of the partial dry-out at lower heat flux and a lower increase of the heat transfer coefficients, compare Fig. 32 (s. page 97).

The R134a-oil mixtures show that the absent liquid in the channel is reducing the values of the heat transfer coefficients at ϑ_S = +10.1°C (p^* = 0.102) and ϑ_S = +20.1°C (p^* = 0.141). The heat transfer of the R134a-oil mixture with 0.5% oil mass fraction is improved for all investigated saturation temperatures. This is due to sufficient liquid in the micro-channels for the thin-film evaporation in the menisci, compare Fig. 26 (s. page 88). Increasing oil mass fraction above 0.5% effects a decrease in heat transfer. The values of the ratio $\alpha_{mix}/\alpha_{ref}$ increases for decreasing saturation temperature. For -28.6°C the heat transfer coefficients of the R134a-oil mixture with 5% oil mass fraction shows higher heat transfer coefficients than the pure R134a. The deterioration in heat transfer due to the oil is up to 60% for 10% oil mass fraction at +20.1°C and the improvement is up to 80% for 0.5% oil mass fraction at -28.6°C, see Fig. 27 (s. page 89).

Due to the higher reduced pressure of R410A than of R134a at the same temperatures the pool boiling heat transfer coefficients of R410A on the GEWA-B-tube are higher than those of R134a. This is also the reason why boiling occurs at lower heat flux and much lower temperature differences for R410A than for R134a; compare Fig. 28 (s. page 91). Small amounts of oil increase the heat transfer, as for R134a. The increase in heat transfer of R410A with increasing heat flux for the same saturation temperature is less than for R134a, but it seems to be the same for the same reduced pressure p^*. In general, the heat transfer decreases with increasing oil mass fraction. Only for medium heat fluxes little heat transfer improvements of less than 10% are observed for 0.5% oil mass fraction. The deterioration in heat transfer due to the oil is to 85% for 10% oil mass fraction and 10 000 W/m^2 at -28.6°C in the range of immiscibility, see Fig. 29 (s. page 92). With exception of the range of immiscibility the influence of the saturation temperature is less than the influence of the oil mass fraction.

For R507-oil mixtures the range of reduced pressure is the same as for R410A. Hence the behaviour of the pool boiling heat transfer of the pure R507 on the enhanced tube is comparable to that of the R410A including the partial dry-out. Also, for pool boiling of R507-oil mixtures, the heat transfer decreases with increasing oil mass fraction. Especially for saturation temperatures higher than +0.2°C the heat transfer is deteriorated extensively due to the immiscibility of the component R143a of the refrigerant-mixture with the POE oil, compare Fig. 30 (s. page 94). The behaviour is as described above for the smooth tube. The deterioration in heat transfer due to the oil is up to 90%. Only for the saturation temperature -28.6°C and medium heat fluxes an improvement in heat transfer due to the oil of less than 10% is observed for oil mass fractions up to 1%. Outside the range of immiscibility of the R507 with the POE oil the influence of the oil mass fraction is greater than the influence of the saturation temperature. The values of the ratio $\alpha_{mix}/\alpha_{ref}$ increases with decreasing saturation temperature and decreasing heat flux, see Fig. 31 (s. page 95).

To compare the enhanced tube with the smooth tube the *ratio of enhancement* has been calculated. The *ratio of enhancement* is defined as $\alpha_{enhanced\ tube}/\alpha_{smooth\ tube}$ and has been calculated from the experimental data with $\Delta\vartheta_{smooth\ tube}/\Delta\vartheta_{enhanced\ tube}$, compare Eq. (19) (s. page 67). The *ratio of enhancement* is increasing with decreasing heat flux and decreasing saturation temperature for the pure refrigerants.

For the pure R134a the values of the *ratio of enhancement* are between 2 and 13 for 80 000 W/m^2 at all saturation temperatures and for 4 000 W/m^2 at -10.1°C, respectively. For the maximum in heat transfer of the R134a-oil mixture on the enhanced tube with 0.5% mass fraction the *ratio of enhancement* has also a maximum, see Fig. 33 (s. page 98). The values of the *ratio of enhancement* are up to 15. An increasing oil mass fraction decreases the *ratio of enhancement*. For oil mass fractions greater/equal 3% the *ratio of enhancement* is about the same as for the pure R134a for high heat fluxes. It is less than the values of the pure R134a for heat fluxes less than 10 000 W/m^2.

The *ratio of enhancement* for the pure R410A is about 2 for 80 000 W/m² at all saturation temperatures. It increases up to 17 for 4 000 W/m² at -28.6°C. For high heat fluxes the influence of the oil mass fraction is negligible; compare Fig. 34 (s. page 100). However, for heat fluxes less than 40 000 W/m² the *ratio of enhancement* decreases with increasing oil mass fraction e.g. from 17 at 4 000 W/m² and -28.6°C of pure R410A to 5 at 5% oil mass fraction. In the range of immiscibility for oil mass fractions greater/equal 5% at saturation temperatures less/equal -19.9°C the *ratio of enhancement* is about 1.

For the pure R507 the *ratio of enhancement* is between 2 at 80 000 W/m² for all saturation temperatures and 18 at 4 000 W/m² and -28.6°C. The *ratio of enhancement* decreases with increasing oil mass fractions, see Fig. 35 (s. page 101). At the saturation temperature +20.1°C for the heat fluxes 60 000 W/m² and 80 000 W/m² the *ratio of enhancement* is less than 1 for oil mass fractions greater/equal 1%. For 40 000 W/m² the *ratio of enhancement* is less than 1 for oil mass fractions greater/equal 3% at +20.1°C. At 10% oil mass fraction the *ratio of enhancement* is about 1for the heat fluxes 60 000 W/m² and 80 000 W/m² for all temperatures. These low values of the *ratio of enhancement* for high heat fluxes are caused by plugging of the enhanced surface structure by the oil.

During the experimental work of this thesis many important visual observations were made. In the following some special observations during filling in the oil from the oil cylinder into the evaporator and during the heat transfer measurements are discussed.

When filling in the oil for the measurements with the R134a-oil mixtures the oil accumulates on the top of the liquid-vapor interface and is dissolved from there into to the liquid R134a-oil mixture. During the heat transfer measurements at -19.9°C with the enhanced tube GEWA-B some schliere are observed below the test tube. It was assumed that oil or oil-enriched liquid is flowing out of some pores at the bottom side of the tube. This observation was made for 5% oil mass fraction at 10 000 W/m² and for 10% oil mass fraction at 10 000 W/m² and at 4 000 W/m².

The oil at first swims on the liquid-vapor interface when filling into the pure R410A and the R410A-oil mixture respectively. At the interface between the liquid refrigerant-oil mixture and the oil some very turbulent processes have been observed, see Fig. 36 (s. page 103). The liquid next to the interface has a blue appearance during these processes. From this liquid-liquid interface drops are falling down and move to the lower flange as shown in Fig. 37 (s. page 103). The drops consists of oil or oil enriched liquid. When the drops, after a little while, are below the test tube they move up and swim on the liquidvapor interface. The temperatures of the R410A-oil mixture and of the oil are both about +20°C. The densities at +20°C are for the POE oil ρ_{oil} = 999 kg/m³ and for the refrigerantmixture R410A ρ_{R410A} = 1091 kg/m³. Because of its lower density the oil should swim on top of the refrigerant-oil mixture. However, the oil stays still at the lowest point of the evaporator.

During filling in the POE oil for the measurements of the R507-oil mixtures a little amount of the oil is first accumulated and swims on the liquid refrigerant-oil mixture. From this accumulated oil small drops are sinking down when filling in oil up to 1% oil by mass. These drops are flowing down towards the lower flange on the bottom of the inner glass cylinder. Next to the test tube the drops move up as shown in Fig. 37 (s. page 103). When filling in the oil for higher oil concentrations than 1% by mass a film flows downwards to the lower flange along the bottom of the inner glass cylinder. The oil is accumulated below the test tube as second liquid phase and does not come up. The densities at +20°C are for the POE oil ρ_{oil} = 999 kg/m³ and for the refrigerant-mixture R507 ρ_{R507} = 1080 kg/m³. Because of its lower density the oil should swim on top of the refrigerant-oil mixture; nevertheless the oil stays still at the lowest point of the evaporator. During the heat transfer measurements with the smooth and the enhanced tube at 5% oil mass fraction at +10.1°C and +20.1°C for heat fluxes less/equal 10 000 W/m² an accumulation of a second liquid phase on the top of the tube was observed. It was assumed that the second liquid phase is oil or an oil-enriched liquid phase. Little drops of this second liquid phase swim (Ø ≈ 0.2 mm) in the R507-oil mixture during the measurements at +10.1°C with 5% oil mass fraction. For 10% oil mass fraction and 4 000 W/m² schliere are observed at saturation temperatures less/equal +10.1°C. At the bottom side of the enhanced tube oil or an oil enriched liquid phase flows out of some pores as observed also for the R134a-oil mixture.

When filling in the oil before the heat transfer measurements of the R404A-oil mixtures a small amount of oil swims at the phase interface. Some little droplets sink down and flow along the bottom of the cylinder towards the lower flange. The oil or oil enriched liquid phase is accumulated below the test tube and does not flow up as for the R410A- and the R507-oil mixtures. The liquid looks cloudy for boiling the R404A-oil mixtures with 10% oil mass fraction with heat fluxes greater/equal 10 000 W/m². The nucleation sites are almost on the upper side of the smooth tube active, no boiling occurs at the bottom side. The liquid is clear for heat fluxes less/equal 4 000 W/m². A second liquid layer is at the liquid-vapour interface. Some droplets from this layer swim in the bulk liquid. The second liquid phase is also accumulated on the top of the tube.

In the following one possible explanation for the sinking oil during filling in is explained by a “fictive” phase diagram of ternäre mixtures of the components R125, R143a and oil, compare Fig. 6 (s. page 54). This is a “fictive” phase diagram because neither the molar mass of the oil nor the ranges of immiscibility of the ternäre mixtures of R125, R143a and oil due to the parts of R125 or R143a are known. When filling in the oil and also when the refrigerant-mixture evaporates there is locally an oil enriched phase. The oil enriched phase is in the range of immiscibility. Because of this the oil enriched phase is separated in two refrigerant-oil mixtures, a R125-oil mixtures with little amounts of R143a and a R143a-oil mixture with little amounts of R125. These mixtures of refrigerantmixture´s component and oil have different densities. For the refrigerant-mixture R507 the density of the component R143a at +20°C is ρ_{R143a} = 951 kg/m³ and the density of the component R125 is ρ_{R125} = 1220 kg/m³; compare Fig. 38 (s. page 105). As Burke [76] reported, R125 has a much better miscibility with PAG and POE oils than R143a, compare also Fig. 7 (s. page 56). When the POE oil contacts the refrigerant-mixture R507 the component R125 dissolves faster and more of the POE oil than the component R143a. The R125-oil mixture is heavier than the rest of the liquid; the R125-oil drops are sinking. When filling in oil for mass fractions up to 1% the component R143a can dissolve the POE oil so the density of the drops decreases and the drops come up after some time. When filling in oil for higher oil concentrations than 1% by mass the R143a is not able to dissolve any more oil. So the added oil can only be dissolved in the R125. The heavier R125-oil mixture remains below the rest of the refrigerant-oil mixture. The temperature in the evaporator is decreased for the first measurement by reducing the temperature in the condenser of the test rig. So after a little while refrigerant-oil mixtures starts boiling at the flange with its higher temperature than the fluid. The total POE oil is dissolved in the R507 during this process. For the refrigerant mixture R410A the density of the component R32 is ρ_{R32} = 981 kg/m³. However, R32 could dissolve the oil but not as fast and as much as R125. Hence for small amounts of oil the drops of the R125-oil mixture first sink and after a certain time when also R32 is dissolved they flow up as described above. Presumably this is the reason why a second liquid phase is accumulated on the top of the tube when boiling of R507-oil mixtures at higher saturation temperatures occurs as described above.

The experimental pool boiling heat transfer coefficients of the refrigerant-oil mixtures on the smooth tube are compared with correlations from the literature. For the enhanced tube no correlation for boiling of refrigerant-oil mixtures exists. First, the correlation recommended in the VDI-Wärmeatlas [4] for pure refrigerants is used. This correlation is based on the reference heat transfer coefficient α_0 and the exponent n. The exponent n describes the influence of the heat flux as a function of the reduced pressure. The reference heat transfer coefficient α_0 is measured or calculated at a reduced pressure of $p^* = p_s / p_{cr} = 0.1$, a heat flux of 20000 W/m² and an arithmetic mean roughness $R_{a0} = 0.4$ µm. For the pure refrigerant R134a the measured reference heat transfer coefficient is $\alpha_{0,R134a}$ = 3 910 W/m² K, for the refrigerant mixture R410A $\alpha_{0,R410A}$ = 3 850 W/m² K, for the refrigerant mixture R507 $\alpha_{0,R507}$ = 3 310 W/m² K and for the refrigerant mixture R404A $\alpha_{0,R404A}$ = 3 050 W/m² K. The calculated heat transfer coefficients according to VDI-Wärmeatlas agree within ±20% with the measured heat transfer coefficients for oil mass fractions up to 1% for all investigated refrigerants.

For higher oil mass fractions the heat transfer coefficients differ by more than 50%. The calculated exponents n agree within ±0.1 with the fitted exponents from the experimental data for oil mass fractions up to 1% for all investigated refrigerants. For R134a, this agreement is also for oil mass fractions up to 5%. Higher oil mass fractions lead to lower values of the exponent n.

For the correlation of Rohsenow [71] the solid-liquid parameter C_{SF} is fitted to the experimental data for all refrigerant-oil mixtures.

Some special correlations for pool boiling heat transfer of refrigerant-oil mixtures are given in the literature. One of them is the correlation by Chongrungreong and Sauer [21]. The calculated heat transfer coefficients according to the original correlation are lower than the measured heat transfer coefficients. Chongrungreong and Sauer assume an influence of the heat flux which is too low. Substituting the original exponent n of Chongrungreong and Sauer by the correlation of VDI-Wärmeatlas the agreement between the calculated and measured heat transfer coefficients is considerably improved.

Jensen and Jackman [23] developed a correlation for pool boiling heat transfer of refrigerant-oil mixtures. The calculated heat transfer coefficients agree within ±20% for oil mass fractions up to 1%. For higher oil mass fractions the calculated heat transfer coefficients may deviate by more than ±40% from the experimental data.

The influence of oil on the pool boiling heat transfer of refrigerant mixtures is also affected by the ranges of immiscibility of the oil with the refrigerant mixture´s components. This is obvious e.g. for the R507-oil mixture with 10% of oil by mass if we know that the component R143a is only able to dissolve 1% of oil by mass. However, until now no model is able to quantify this influence. In this thesis a new prediction model is suggested to quantify the influence of an oil on the pool boiling heat transfer if the influence of the oil on the pool boiling heat transfer of the refrigerant mixture´s components is known.

For refrigerant mixtures the different miscibility of the mixture and its components with oil makes the prediction the pool boiling heat transfer of refrigerant-oil mixtures difficult. In this thesis the ratios $\alpha_{mix} / \alpha_{ref}$ of the refrigerant-oil mixture´s components are superposed according to the molar concentration. The results of the molar superposed ratio $\alpha_{mix} / \alpha_{ref}$ of R143a-oil mixtures and R125-oil mixtures agree within ±20%for oil mass fractions up to 5%. It was further assumed that the refrigerant mixture R404A is a mixture of the components R507 and R134a. For this assumption the molar superposed ratio $\alpha_{mix} / \alpha_{ref}$ of R507-oil mixtures and R134a-oil mixtures agree within ±30% with the experimental ratio $\alpha_{mix} / \alpha_{ref}$ of R404A.

A possible influence of the dynamic viscosity and the density of the liquid on the pool boiling heat transfer from a smooth tube is discussed.

Chien [42] describes a model to calculate the pool boiling heat transfer coefficients of enhanced surfaces with micro-channels and pores. In the present thesis the model of Chien has been extended to consider the influence of the saturation pressure. Saturation temperature and temperature difference between tube wall and saturated refrigerant are necessary for the extended model. Results of the simulation with this new model are the number of menisci and the number of active pores. Both are necessary to distinguish the *flooded mode*, the *suction mode* and the *dried-up mode* of the evaporation on the enhanced tube; compare Fig. 40 (s. page 107). Also the contribution of latent heat transport, the bubble growth time and the waiting time are calculated. The so calculated pool boiling heat transfer coefficients of the pure refrigerants agree within ±50% with the experimental data in the range of *suction mode*.

The detailed results of the simulation with the extended model for the pure refrigerants show the possible influence of the oil on the pool boiling heat transfer from the enhanced tube as a function of the reduced pressure.