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Forced bulk boiling at high heat fluxes

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ABSTRACT

The influence of very high heat fluxes (10^5 to 10^6 W/m²) on the boiling behavior of R404A are investigated. By using a new evaporator design it is possible to change the evaporating effects even at these high heat fluxes to bulk boiling. This paper focusses on the experimental analysis of the new evaporator by varying the parameters geometry, mass flux, subcooling and fluid velocity. The results are compared with experimental works and mathematical models at lower heat fluxes. The results show, that fluid velocity, mass flux and geometry have high influence on the transferred heat flow, while the subcooling of the liquid phase has only a small effect.

1. INTRODUCTION

Plastics are one of the most important material groups for engineering. In 2009 the worldwide volumetrically amount of manufactured plastics was in the same scale to the amount of manufactured steel. Witt (2006) showed that most of the manufactured plastics are processed in molds. The cycle time of molding is mostly determined by the cooling time (Johannaber, 2007; Menning, 2008). To reduce the cooling time, molds are cooled with water flowing through cooling channels. Steinko (2008) pointed out that water cooling is not possible for geometries like small cylindrical objects due to the required minimum cooling channel diameter of 8 mm. These areas in molds are called hot-spots and they dominate the cycle time (Menges and Michaeli, 2007).

Several methods have been developed to cool hot-spots in molds. The most recent development is to cool the hot-spots by refrigerant. There are two methods published of refrigerant-cooling: Cooling with R744 in an open loop or cooling with R404A in a closed-loop (Knipping, 2010). Both methods allow minimum cooling channel diameters of 0.001 m. Challenging effects of using refrigerants in molds are high temperatures (up to 300 °C), geometrical influences (expansion valve about 0.2 – 1 m away from evaporator, surface topology), occurring high heat fluxes (up to 10^6 W/m²) and the design of the evaporator (redirectioning of the refrigerant flow in the small evaporator by 180°).

Boiling of refrigerants has been investigated by several authors exemplarily shown in table 1. The refrigerants are either pure HFC refrigerants or ternary HFC mixtures. Although several studies have been performed, none of them investigates heat fluxes as they appear in molds for plastic parts. To realize low molding cycle times the energy of the hot plastic parts has to be removed as fast as possible until the wall temperature of the processed plastic part is dropping below the removal temperature. Within the hot-spots the only way to accelerate the cooling time is to rise refrigerant mass flux. Within the test loop mass fluxes of 4800 kg/m²s and higher have been observed. Heat transfer at mass fluxes higher than 4000 kg/m²s have not been published so far.

Table 1: Recent exemplary studies of boiling refrigerants

author	year	refrigerant	mass flux kg/m ² s	heat flux kW/m ²	work	boiling behavior
Greco <i>et al.</i>	2004	R-404A, R-410A	290-1100	11-39	experimental	flow boiling
Kim <i>et al.</i>	2005	R-134a	285-1300	40-200	experimental	flow boiling
Sindhuja <i>et al.</i>	2008	R-407C	200-2000	5-80	experimental	flow boiling
Davide	2010	R-134a, R-125	200-600	9-53	analytical	
El-Nakla <i>et al.</i>	2011	R-134a	500-4000	50-90	experimental	film boiling
Chen <i>et al.</i>	2011	R-134a	60-480	6-90	experimental	flow boiling

To achieve quick response times of the expansion valve a magnetic expansion valve is used. After the magnetic expansion valve, the pre-expanded refrigerant is passed through a capillary tube to realize small cooling channel diameters. Depending on the inner diameter of the capillary and the occurring pressure loss, this section could nearly be described as a short tube orifice. Chen *et al.* (2004), Choi *et al.* (2004) and Nilpueng and Wongwises (2011) exemplarily investigated the behavior of refrigerant flow through short tube orifices, but none of them describes an expanded and (depending on the level of subcooling) partly evaporated refrigerant as working fluid.

Surface topology is an important criterion for the critical heat flux. The influence of surface topology on initial nucleation sites has been described by Luke and Gorenflo (2000) and Sarwar *et al.* (2007) for a wide range of roughnesses and materials. In technical applications, such as cooling hot-spots in molds the surface of the evaporator is manufactured either by deep hole drilling or with electrical discharge drilling (EDM drilling). Both manufacturing processes generate characteristically surface topologies which haven't been discussed so far.

The present work focusses on investigating the effects taking place within a refrigerant-cooled hot-spot in molds. The refrigerant R404A is chosen because of the reachable low temperatures of evaporation at pressures between 0.1 to 0.5 MPa. The steel used for molds usually has very low heat conductivity so low temperatures of the coolant are helpful to reach processable heat transfer rates and cycle times. It has to be noted that this work doesn't focus on the detailed investigation of every effect inside the evaporator but to give a general introduction into this new kind of evaporator design for spot cooling and the industrial use of refrigerants in molds.

2. EXPERIMENTAL SETUP

2.1 Flow loop

Figure 1 (top) shows a schematic diagram of the test setup used to investigate heat transfer at high heat fluxes. The test setup consists of a hermetically sealed circuit with a continuously working compressor, a condenser, the test section assembly in which the refrigerant evaporates, several valves to regulate the flow, a mass flow meter and instrumentation. Subcooling is adjusted by a separate closed loop with a heater and an evaporator linked to the test loop through a heat exchanger. The water-heated post-evaporator is installed to make sure that only gaseous, superheated refrigerant is sucked by the compressor. The thermostatic expansion valve is used to regulate the evaporation pressure. The phase of the refrigerant can be checked by sight glasses installed at several positions.

Figure 1 (bottom) shows the state points of the loop in a pressure-enthalpy diagram. The refrigerant is compressed using a compressor (state point 1 to 2). To remove most of the refrigeration oil out of the loop an oil-separator is installed. The temperature of the compressed and uncondensed refrigerant is detected with a thermocouple (cp. figure 1 (top)). The condenser is of a forced-convective type cooling the refrigerant from state point 2 to 3. The phase of the refrigerant (bubble-freeness) is checked by a sight glass. Pressure and temperature are measured at the shown positions of the loop. The refrigerant then passes a liquid-cooled subcooler to control the subcooling of the liquid refrigerant (state point 3 to 4). After that a Coriolis-type mass flow meter detects the density, the temperature and the mass flow of the liquid refrigerant. Before entering the test-section the pressure and temperature of the liquid refrigerant are measured to detect pressure drop. To control the mass flow a magnetic expansion valve is used. State variables of state point 5 could not be measured precisely due to the test section assembly.

To prevent evaporation inside the fluid line it is mounted concentrically inside the suction line. This is necessary because of temperatures between 60 °C and 150 °C during the molding process. To generate realistic results this design was also used for the test-section. Because of the concentricity the temperature at the fluid line outlet could not be measured. The pressure drop inside the capillary tube is depending on its diameter (0.5 to 1.5 mm) and length and can't be measured due to the concentricity of the tubes as well. Leaving the test section, pressure and temperature of the evaporated refrigerant are measured and the vapor quality of the refrigerant is observed by a sight glass.

The enthalpy of state point 6 is calculated by subtracting the enthalpy difference of the post-evaporator from the enthalpy difference between the points 4 and 1. The enthalpy difference of the post evaporator is calculated by an energy balance, measuring the water flux with a flow meter and two thermocouples detecting the difference of the water at inlet and outlet of the post-evaporator. The pressure in the suction-line is controlled by a thermostatic expansion valve working as a bypass line to the test-section.

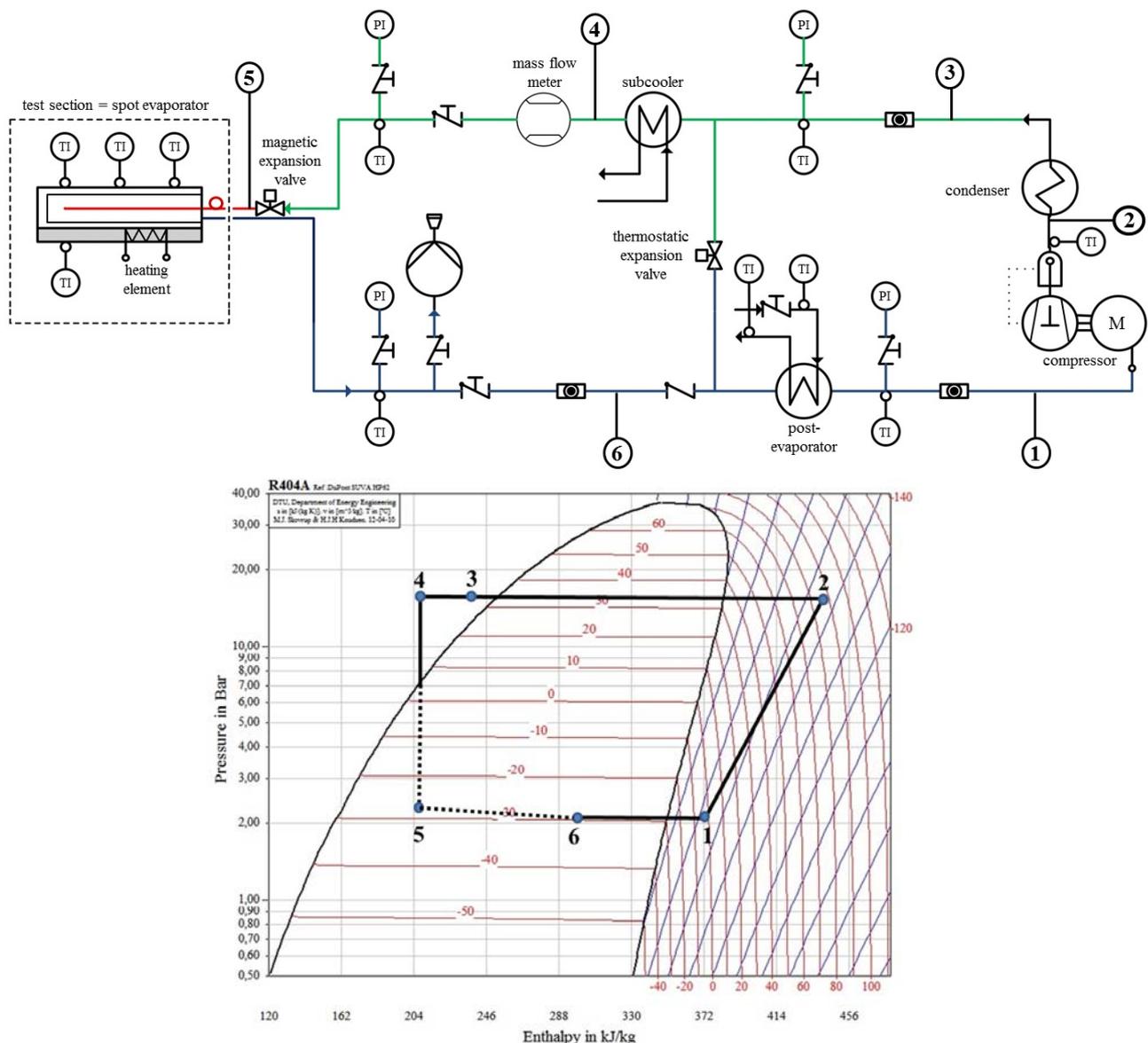


Figure 1: Experimental setup for heat transfer investigation at high heat fluxes. Top: Schematic of the test loop. Bottom: Pressure-enthalpy diagram (produced with CoolPack (Skovrup (2010)))

Figure 2 shows the assembly of the test-section referring to the components located within the dashed line in figure 1. It contains of the evaporator, a heating element, fluid and suction line and the adiabatic housing. The housing is made of polypropylene covered with a thin film of aluminum to reduce radiation heat transfer. The evaporator consists of aluminum and is insulated with Polytetrafluoroethylene (PTFE) to reduce thermal conduction. Heat flux changes are realized by varying the diameter of the evaporators and the power of the heating element. The heating element is an electrical heater and covers the aluminum cylinder at the perimeter.

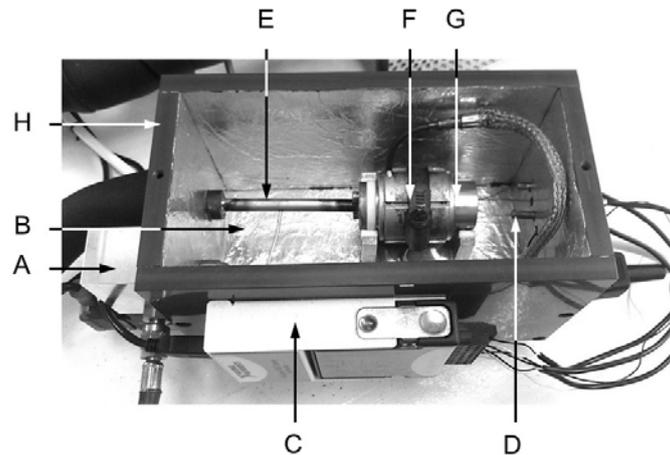


Figure 2: Test-section assembly with the components (A) magnetic expansion valve, (B) fluid line, (C) data acquisition for thermocouples, (D) thermocouples (4 pieces), (E) suction line, (F) heating element, (G) evaporator and (H) adiabatic housing made of polypropylene (top part removed for photography)

2.2 Data acquisition and uncertainties

12 temperature and 5 pressure sensors are installed in the experimental setup (cp. figure 1). Each reported measurement is obtained as an average of approximately 1200 points obtained over 10 minutes of steady-state conditions. The electrical power to the AC heating elements within the test section is measured using a combined voltage and current measurement system which has an uncertainty of 0.5 % of full scale. The refrigerant mass flow is measured with a Coriolis-type mass flow meter with an uncertainty of 0.1 % of the reading within the measurement range of 0 to 0.025 kg/s. Pressures at different points in the loop are measured using absolute pressure transducers with a range of 3.5 MPa and an uncertainty of 0.05 % of full scale. All temperatures are measured using calibrated J-type thermocouples with an uncertainty of ± 1 K.

2.3 Experimental procedure

The refrigerant used for the experiments is R404A; the oil circulation ratio (OCR) is less than 1 %. The mass flux is varied from 0.0014 to 0.0111 kg/s (5 to 40 kg/h). The pressure of the fluid is varied from 1.35 to 1.80 MPa, the suction line pressure is varied from 0.2 to 0.5 MPa. The different spot evaporators used for the experiments with their corresponding areas are listed in table 2. The electrical power of the heating element is adjustable from 0 to 200 W.

Table 2: List of used spot evaporators (cylindrical design)

diameter in mm	length in mm	area in $\text{m}^2 \cdot 10^{-4}$	diameter in mm	length in mm	area in $\text{m}^2 \cdot 10^{-4}$
1.50	41.83	1.99	4.30	32.99	4.60
2.50	42.26	3.37	5.60	33.92	6.21
3.00	37.47	3.60	5.70	35.11	6.54
3.20	42.16	4.32	6.20	33.78	6.88
3.50	42.33	4.75	6.60	33.07	7.20
4.30	41.59	5.76	7.00	33.69	7.79

To identify the process, numerous preliminary tests have been performed. Chen et al. (2004) and Marcel et al. (2009) observed that fluid-to-fluid modeling is sustainable while observing the critical heat flux of refrigerants and water. Corresponding on the works by El Nakla (2011) fluid-to-fluid modeling is also possible for lower heat fluxes and boiling phenomena. Thus it is assumed, that the well-known Nukiyama-diagram for water is also negotiable to refrigerants.

To achieve the presented results two different measurement methods have been performed. To determine the critical heat flux (CHF) measurement starts at subcritical heat flux with constant mass flux. Heat flux is then increased until wall temperature suddenly rises (ΔT_{CHF}). To determine the Leidenfrost heat flux (LHF) measurement starts at supercritical heat flux with constant mass flux. Heat flux is then decreased until wall temperature suddenly drops (ΔT_{LHF}) (figure 3).

The second measurement method is used to map the industrial process: Within the molds the heat flux is always constant. The only variable to force bulk or film boiling is the mass flux of the refrigerant. Thus heat flux is held constant during the second experiments and mass flux is varied until temperature drops or rises according to the described methods above.

The graph for wall temperatures is always characteristically formed: starting at wall temperatures of about 300 °C the wall is cooled by evaporating refrigerant. A characteristically phenomena of all experiments is the sudden rise of temperature derivation at 55 - 60 °C. Combined with the evaporation temperature of -25 °C this means wall-superheat of 80 K according to the Nukiyama-diagram. At this temperature wall superheat is assumed to drop below the Leidenfrost-Point and bulk boiling is starting. The sudden drop of temperature could be explained following the lower dotted line in figure 3. According to the Nukiyama-diagram wall temperature has to drop immediately which could not be observed during the experiments (figure 4). The reason for this could be the position of the thermocouple (about 1 mm behind the surface) and the inertia of the surrounding material (aluminum).

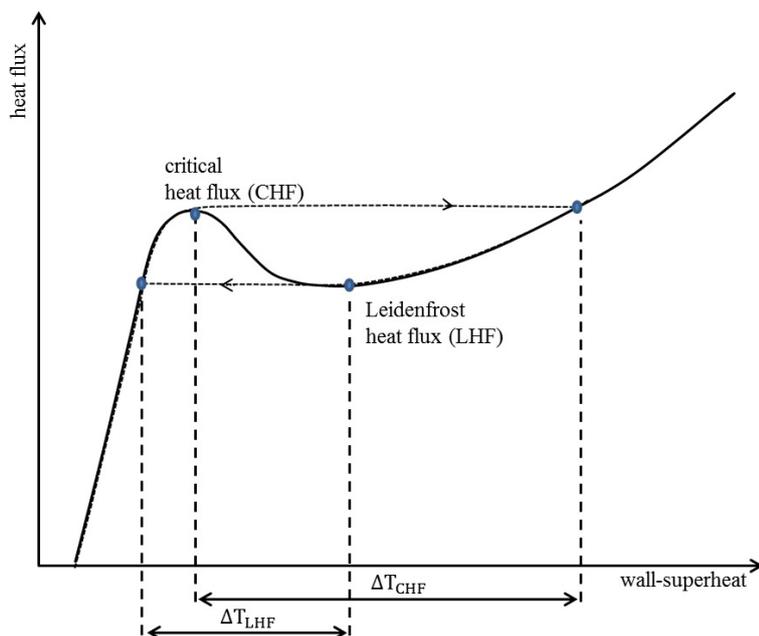


Figure 3: Nukiyama-diagram (schematic) with proceeded measurement runs

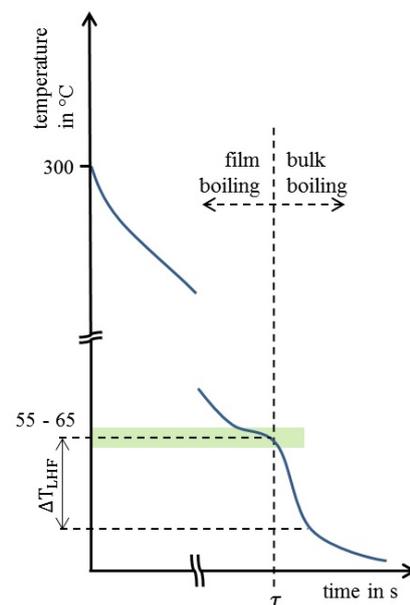


Figure 4: Characteristic graph (experimental data) for wall-temperatures

3. EXPERIMENTAL RESULTS

3.1 Mass flux, critical heat flux and Leidenfrost heat flux

In order to identify the influence of the mass flux on CHF and LHF a curve was constructed from the measured data as shown in figure 5. It is shown that CHF and LHF's derivation is much higher in the area of 0 to 5000 kg/m² s than in higher mass flux areas. The reason for this can possibly be found in the design of the spot cooling evaporator. High mass fluxes have only been reached with small evaporator diameters. Small evaporator diameters mean, that the area ratio between capillary and suction line is 2.5 (compared to more than 30 at larger evaporator diameters). It can also be observed, that LHF is always lower than CHF even at high mass fluxes and heat fluxes. According to the Nukiyama-diagram this means, that the characteristic form is also sustainable for high heat and mass fluxes.

As shown in figure 6 the cooling time τ (cp. figure 4) strongly depends on the refrigerant mass flux. To get these results the aluminum cylinder has been charged with an amount of energy (40 kJ) which is estimated to be a typical amount of energy for a hot-spot in molds. The higher the mass flux and thereby the possible enthalpy of the refrigerant, the less time is needed to cool down the aluminum cylinder. The graph also shows that cooling time depends on evaporators' diameter. The reason for this phenomenon is both, the bigger evaporator volume and the smaller heat flux due to the bigger heat exchanger area. Depending on the assumption, that in the regime of film boiling heat transfer is mainly of the convective type, a bigger heat exchange area also accelerates cooling.

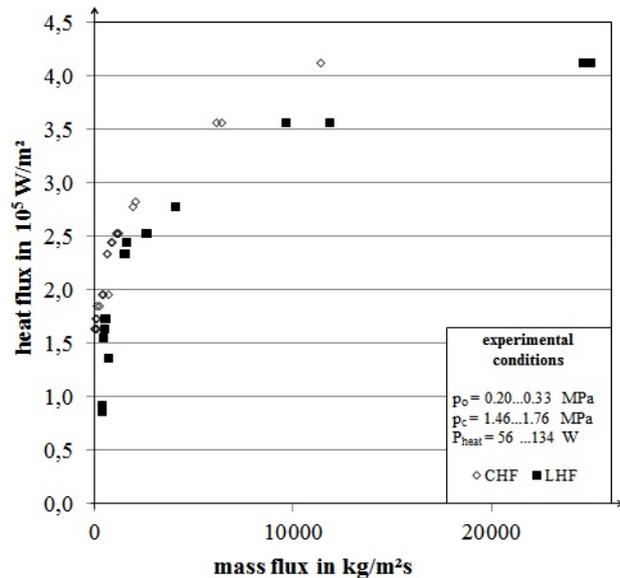


Figure 5: CHF and LHF as a function of mass flux

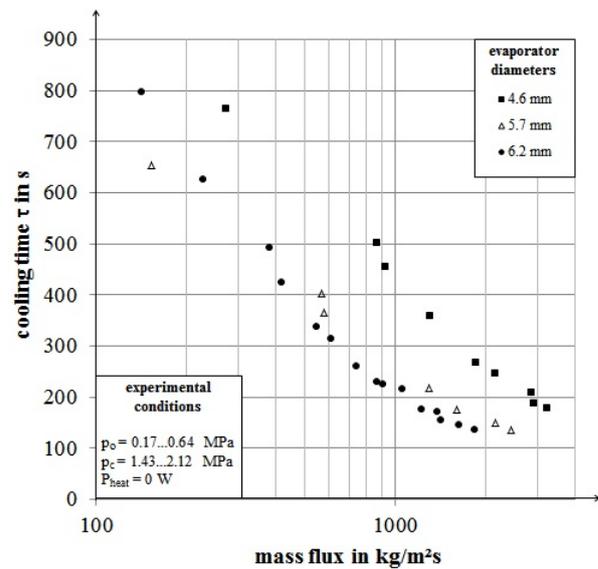


Figure 6: Cooling time as a function of mass flux

3.2 Subcooling

The influence of subcooling is shown in figure 7. It can be stated, that subcooling has only a little effect on the critical heat flux at given geometries. At the same time, mass flux is ascending with rising level of subcooling. This effect can be explained looking at the experimental setup and the thermo physical properties of the refrigerant. After the magnetic expansion valve the liquid, non-evaporated refrigerant is lead through a capillary tube. This causes a loss of pressure and the refrigerant partly evaporates. Thus there is a mixture of liquid and evaporated refrigerant inside the capillary tube. It is assumed, that the liquid refrigerant covers the whole hydraulic diameter of the capillary due to inner atomic van-der-Waals forces. This leads to the conclusion that the evaporating refrigerant "behind" causes an acceleration of the liquid in the capillary tube.

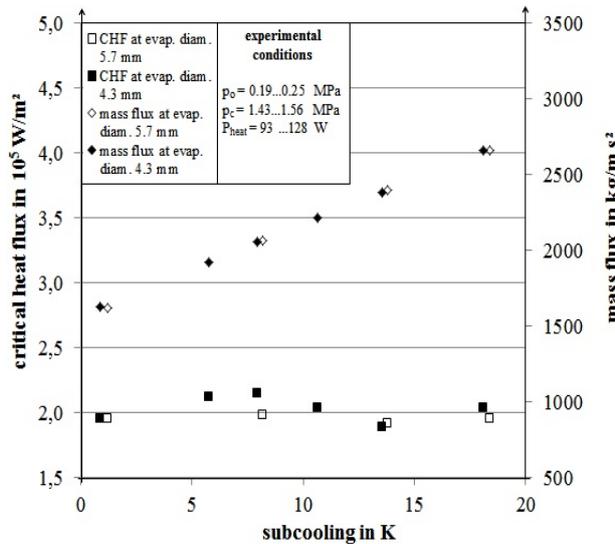


Figure 6: Influence of subcooling on CHF and mass flux at different diameters of the evaporator

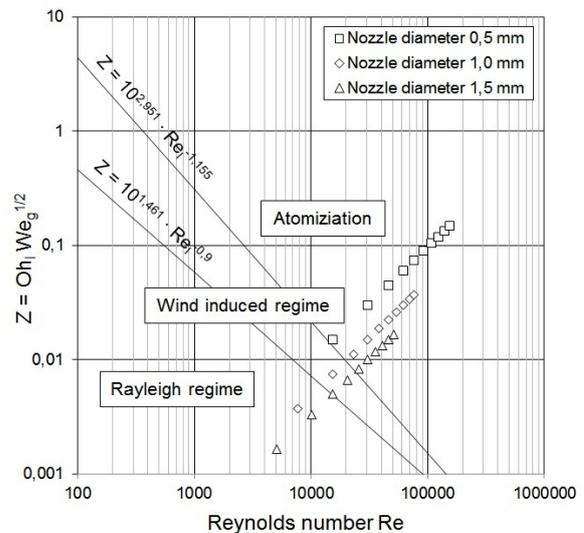


Figure 7: Influence of nozzle diameter on spraying behavior

3.3 Spraying behavior

Calculating the CHF using the method of Preusser (1979) for ternary mixtures and comparing it to the experimental data at given values of wall-superheat it can be stated that the experimental results are always above the calculated. Especially at low rates of subcooling this effect takes place. The reason for this can be the new evaporator design which forces bulk boiling even at high heat fluxes.

As mentioned in chapter 1 the capillary tube could nearly be described as a short tube orifice or aperture causing a spray of refrigerant. Wozniak (2003) and Richter (2012) showed that spraying behavior of an aperture is strongly depending on Reynolds number of the fluid. The fluid velocity is the main parameter taking influence to the Reynolds number. As mentioned in section 3.2 fluid velocity rises due to accelerating evaporation effects in the capillary tube while having only low rates of subcooling. Thus the spraying behavior of the aperture changes from Rayleigh's regime into atomization as shown in figure 7. Within the regime of atomization the diameter of the fluid drops are smaller than in Rayleigh's regime. At the same time the velocity of the flying drops is rising following conservation of momentum.

Within the field of film boiling a layer of evaporated refrigerant covers the heating area. Due to convective flow effects this layer has a reset force avoiding fluid drops to reach the hot surface. With increasing fluid drop velocity the drops are able to pass this layer and reach the hot surface changing the heat transfer effects from film boiling to bulk boiling.

4. CONCLUSIONS

To reduce the cycle time of molded plastic parts, evaporating refrigerants can be used for cooling. To analyze the influence of high heat fluxes (10^5 to 10^6 W/m²) a new evaporator design has been developed. This design forces the heat transfer even at this heat fluxes from film boiling to bulk boiling. The measured critical heat flux was higher than the calculated due to this effect. During the experiments, the mass flux was varied from 1000 to 22000 kg/m²s corresponding to flow rates of 0.0014 to 0.0111 kg/s (5 to 40 kg/h). The discharge pressure of the refrigerant was varied from 1.35 to 1.80 MPa, the suction line pressure was varied from 0.2 to 0.5 MPa, corresponding to an evaporation temperature of -31 to -6 °C. The subcooling of the liquid refrigerant was varied from 0 to 25 K.

It was shown that mass flux has strong influence on critical heat flux and Leidenfrost heat flux, while subcooling has no effect on these. Subcooling has effect on evaporation of refrigerant inside the capillary tube and therefore at the fluid velocity. Fluid velocity influences the spraying behavior and the heat transfer mechanisms due to changing Reynolds numbers.

NOMENCLATURE

EDM	electrical discharge machining	(-)
PI	pressure transducer	(-)
TI	temperature transducer	(-)
M	motor	(-)
PTFE	Polytetrafluoroethylene	(-)
AC	alternate current	(A)
OCR	oil circulation ratio	(%)
CHF	critical heat flux	(W/m ²)
LHF	Leidenfrost heat flux	(W/m ²)
T	temperature	(K)

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