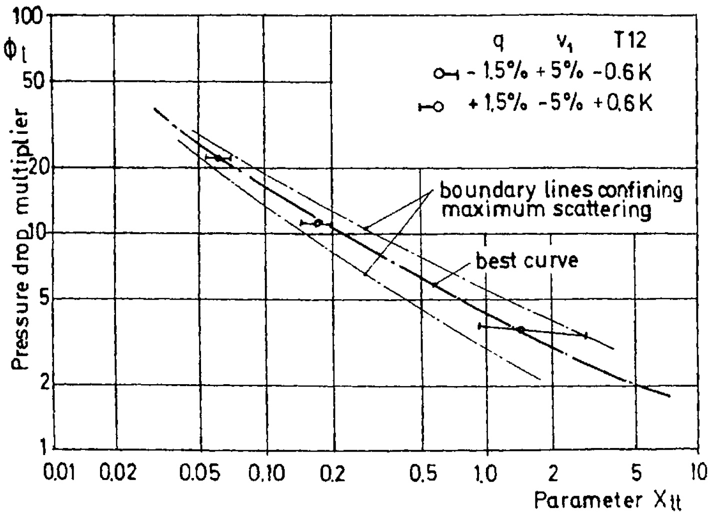

Fig. 4: Error calculation for 7-pin bundle data.

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Boiling Heat Transfer in the Transition Region from Bubble Flow to Annular Flow

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ABSTRACT

Experimental investigation were carried out in the transition region from bubble to annular flow up to fully developed annular flow. The measurements were performed with refrigerant R12 in a vertical inside cooled tube with an inner diameter of 14 mm, wall thickness of 0,5 mm and heat-flux controlled surface. The heated length of the tube was varied in the range of 1 to 5 m. The experimental data showed that with increasing heat flux, mass flow rate, pressure and L/D-ratio the ratio of the heat transfer coefficients increases. The influence of these parameters on the boundaries of the phase change were also investigated. The measurements presented in this paper show that the heat transport in the transition region as well can be satisfactorely calculated by the relation

$$\frac{\alpha_{2ph}}{\alpha_{fl_0}} = C_1 \left[Bo \cdot 10^4 + C_2 \left(\frac{1}{X_{tt}} \right)^r \right] \cdot \left(1 + \frac{D}{L} \right)^s$$

taking also into account the corresponding L/D-ratio of the boiling channel.

NOMENCLATURE

A	- test section area	m²
C	- constants	-
D	- test section diameter	m
L	- test section length	m
$\dot{Q}$	- total heat flux	W
$\dot{m}$	- mass flow rate	kg/m² s
p	- pressure	N/m²
$\dot{q}$	- heat flux rate	W/m²
r	- latant heat of vaporisation	J/kg
$\dot{x}$	- mass quality	-
α	- heat transfer coefficient	W/mK

η	- dynamic viscosity	kg/ms
ϱ	- mass density	kg/m³
ϑ	- temperatur	°C

Subscirpts

2ph	- two phase
fl	- liquid
fl_o	- liquid allone
D	- vapor
w_i	- inner wall
sat	- saturation
red	- reduced

INTRODUCTION

Forced two-phase flow with heat transfer occurs in many apparatus, chemical reactors and evaporators. The heat transfer in these flow boiling systems is affected by the thermodynamic and hydrodynamic parameters of the fluid and also by the distribution of liquid and vapor in the channel. A significant change in the heat transfer coefficient appears in the transition region from bubble- to annular flow. In this region the nucleate boiling changes into interface evaporation (-surface boiling). Although extencive investigations have been carried out, the most important thermo- and hydrodynamic parameters for the change are still unknown.

Special experimental investigations which were carried out in the above mentioned transition region from bubble flow with nucleate boiling to annular flow with surface evaporation. The measurements were performed with refrigerant R12 in a vertical inside cooled tube and heat flux controlled surface. The heat transfer to the evaporating fluid and the phase distribution in the channel were investigated. The experiments were carried out with variation of the parameters-system pressure, heat flux, mass flow rate and length to diameter ratio. For the determination of liquid-vapor phases in the channel an optical reflection probe war used. In addition to these probe-measurements the phase distribution was also investigated by the aid of high-speed cinematography in rectangular as well as vertical direction.

Heat Transfer Mechanisms in Literature

In a vertical channel with a heated vapor-liquid flow two primary heat transfer sections with different vapor fractions and phase distributions can be differentiated. There is the region of bubble flow which encloses the subcooled boiling region with the heat transfer by nucleate boiling. This region is characterized by a low vapor fraction. The other section is the heat transfer of forced convection with annular flow and high vapor fraction and the adjoining region of convective heat transfer to the vapor flow. Many equations exist for the calculation of the heat transfer with clear transfer mechanism - bubble boiling or forced convection

boiling - also called "surface boiling". Most of them are empirical equations. The heat transfer with forced nucleate boiling is normally described as a function of the heat flux in the relation of

$$\alpha = C_o \cdot \dot{q}^n \tag{1}$$

In annular flow the vapor and the liquid flow in seperate parts in the channel. The heat transfer mechanism from the heated wall through the annular liquid film is clearly of a convective nature. The evaporation takes place at the interface of the liquid film to the vapor core. It is called forced convection boiling or "quiet boiling". In the literature on two-phase flow heat transfer with annular flow two groups of calculation equations can be found. The first one was developed by Dengler and Addoms /1/. The general form

$$\frac{\alpha_{2ph}}{\alpha_{fl_o}} = C_1 \cdot \left(\frac{1}{X_{tt}}\right)^n \tag{2}$$

can be applied to distinct annular flow. The two-phase flow heat transfer coefficient is related to the single-phase heat transfer coefficient and expressed as a function of the Martinelli-Nelson two-phase flow parameter X_{tt}. This kind of equation was later used in a slightly modified form for different fluids by Tong /2/, Benett /3/, Collier /4/ and others. Schrock and Grossman /5/ extended equation (1) by the Boiling-Number

$$Bo = \frac{\dot{q}}{\dot{m} \cdot r} \quad \text{to} \quad \frac{\alpha_{2ph}}{\alpha_{fl_o}} = C_2 \cdot \left[Bo \cdot 10^{-4} + C_3 \cdot \left(\frac{1}{X_{tt}}\right)^m\right]^n \tag{2a}$$

which fitted their measurements better than the form of eq.(1). This equation (2a) represents the second group of equations for the calculation of the annular heat transfer coefficient. This kind of equation was later used in a slightly modified form for different fluids by Chaddock and Bruneman /6/, Pujol and Stenning /7/ and others.

A comparison of these two statements is shown in figure 1. It presents the measurements and equations for the bubble flow and annular flow regions by Tong and by Schock and Grossman.

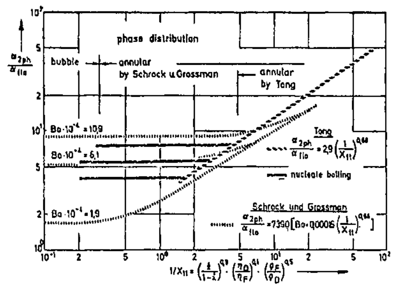


Fig. 1: Heat Transfer Coefficient ratio vs. Martinelli Parameter -Comparison of Tong and Schrock and Grossman -

According to Tong the ratio of the heat transfer coefficient with and without vaporisation at annular flow grows linear with decreasing Martinelli-Parameter (see Fig. 1). The ratio is not influenced by the heat flux. In the region $1/X_{tt} < 5$ nucleate boiling occurs. An influence of the Martinelli-parameter on the heat transfer coefficient ratio could not be observed. Schrock and Grossman discovered a dependence of the heat transfer coefficient at annular flow on the Boiling-number Bo - the ratio of total heat flux to evaporation mass flow. Their measurements yielded that at high values of the boiling-number nucleate boiling predominates whereas at low values the forced convection mechanism dominates up to hight values of the Martinelli-parameter. According to Tong nucleate boiling occurs in this region. Agreement with results in the literature can only be stated for the region of low X_{tt}-values which correspond to high mass vapor fraction.

This comparison shows that the different statements in the literature result from the unknown dependence of most thermo- and hydrodynamic parameters for the phase change and the heat transfer coefficient in two-phase flows. Therefore special experimental investigations were carried out in the transition region from bubble to fully developed annular flow by measuring the two-phase heat transfer coefficient and the phase distribution.

Experimental Equipment

The experimental investigations were performed with the pilot plant (Fig. 2) with refrigerant R12 in a vertical inside cooled tube with an inner diameter of 14 mm. The total length of the tube was varied from 3 to 5 m, the heated length from 1 - 5 m. The test section was heated by direct-current, resulting in a heat-flux controlled surface. Special attention was paid to the prevention of

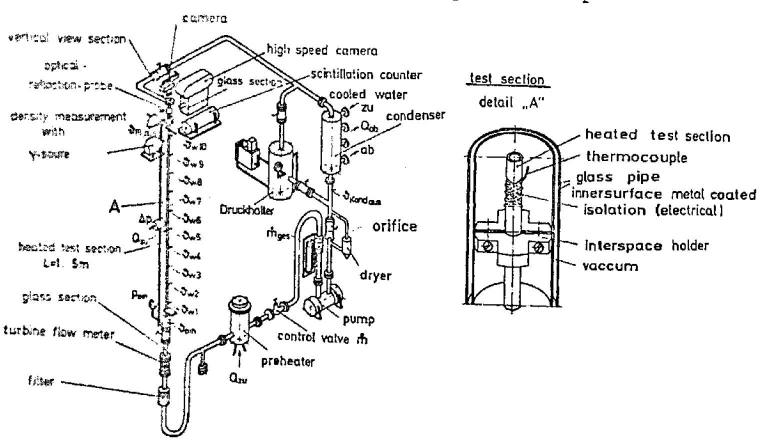


Fig. 2: Pilot Plant

heat loss. The test section was isolated after the principle of a thermos bottle (see fine detail "A" in Fig. 2)

The mass flow rate was measured with a turbine-mass-flow meter, the system pressure with a precision pressure gauge and the differential pressure with a special mercury gauge. The electrical total heat was calculated from the current and voltage measured by special instruments. The fluid- and wall-temperatures were measured with cromel-alumel-thermocouples. The total deviation for the temperature taking into account all mistakes occuring during calibration and measur ing was ± 0,42 °C or 1,4 % at the lowest measured temperature.

The fluid flowed slightly subcooled normally 3 K into the entrance of the test channel. The fluid was evaporated in the test section in the same manner as in a technical evaporizer. In this way a natural phase distribution could be obtained.

For the identification of the phase distribution and the flow pattern at the end of the test section an optical reflection probe, similar to that developed by J.M. Delhaye et al /8/ was used. The operation principle of this probe shall be explained briefly (see Fig. 3, left side). The probe consists of an U-shape fiber optical waveguide. The tip is pointed by an angle 90° and polished. At one end the probe pipe was connected to a light source, at the other end to a photocell. For the different refractive indeces of refrigerant R12 for liquid (n = 1,25) and gas (n = 1,02) the light was totally reflected when there was gas or vapor at the tip of the probe. Little reflection of light could be observed for liquid phase at the tip. That part

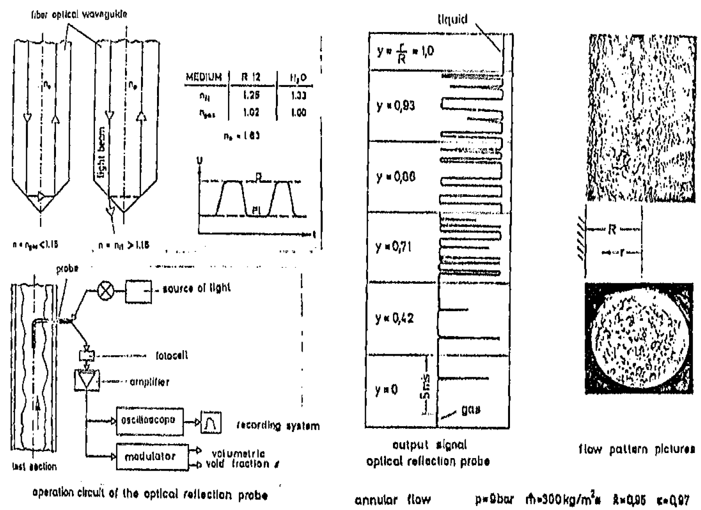


Fig. 3: Operation principle and measuring results by optical reflection probe.

of the light which went back in the fiber optical pipe was converted into voltage by a photocell. This voltage signal could be recorded by an oscilloscope or line printer and then be interpreted for the phase distribution.

In the middle of Fig. 3 this probe signal is shown, presented as a function of the dimensionless radius y at the example of annular flow. The volumetric void fraction can be determined from the ratio of the integrated signal-time to the total measuring time. A comparison with calibration values carried out with the γ -beam-attunuation method showed a good agreement.

In addition to these probe measurements the phase distribution was also investigated by the aid of high-speed and one-picture cinematography in rectangular as well as vertical direction to the flow. An example for annular flow is also shown in Fig. 3.

Experimental Results

The local experimental heat transfer coefficient was calculated from the ratio-heat flux rate to the operative temperature difference resulting from the inner surface temperature and local saturation temperature

$$\alpha = \frac{\dot{q}}{\vartheta_{w_i} - \vartheta_{\text{sätt}}(p)} \tag{3}$$

The mass vapor fraction was calculated from an energy balance over the test section.

$$\dot{X} = \frac{\dot{Q}_{zu} - \dot{m}\cdot A\cdot c_p\,(\vartheta_{\text{sätt}} - \vartheta_{ein})}{r\cdot\dot{m}\cdot A} \tag{4}$$

The Martinelli-Parameter is defined as

$$X_{tt} = \left(\frac{\rho_D}{\rho_{fl}}\right)^{0,5}\cdot\left(\frac{\eta_{fl}}{\eta_D}\right)^{0,1}\cdot\left(\frac{1-\dot{X}}{\dot{X}}\right)^{0,9} \tag{5}$$

With these experimental investigations the influence of the thermo- and hydrodynamic parameters was measured with variation of the

system pressure	$9 < P < 26$ bar
heat flux rate	$0{,}5 < \dot{q} < \dot{q}_{Bo}$ W/cm²
mass flow rate	$300 < \dot{m} < 1200$ kg/m² s
L/D-ratio	$70 < L/D < 360$

Measuring results of heat transfer are shown in Fig. 4. They are presented in a diagram with

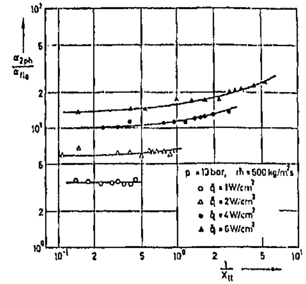


Fig.4: Heat transfer coefficient with and without boiling in dependence of Martinelli-Parameter

the dimensionless parameter of the heat transfer coefficent with and without boiling as a function of the Martinelli-Parameter. In the investigated region the curves rise slightly. First they run horizontal at fully developed nucleate boiling and then gradually change into a statement like Tong's in the region of annular flow. The transition from nucleate boiling at bubble flow to the annular flow with forced convection with interface evaporation is an ascending gradient function without a break in the curve. The ratio of heat transfer coefficient is proportinal to the Boiling - Number Bo.

A comparison between these measurements and the values, calculated with the equation of Pujol and Stenning is presented in Fig.5.

By varying the influencial parameters for the transition region from bubble- to annular flow the following dependence was determined.

The increase of the pressure results in an increase of the heat transfer coefficient, as is shown in Fig. 6. The best agreement were found with the equation which takes into account the Boiling-number Bo.

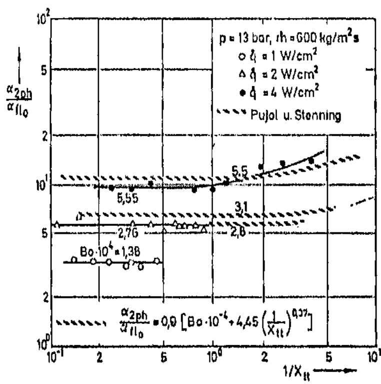


Fig. 5: Heat transfer ratio as function of Martinelli-parameter comparison with Pujol and Stenning

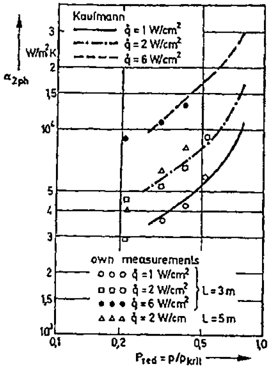


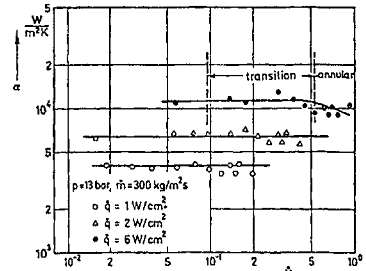
Fig. 6: Heat Transfer coefficient as a function of reduced pressure.

This experimental observation - an increase of the heat transfer coefffficient at increasing pressure - was also found by Kaufmann /9/ in his experiments. For the calculation of the heat transfer coefficient as a function of the pressure exist many empirical equations in forms of

$$\alpha_{2ph} = C \cdot \dot{q}^{n} \cdot F$$

with $F = f\ (p_{red})$ for example developed by Danilova /10/ or Haffner /11/, which were carried out for nucleate boiling in channel flows. For pressure lower than 13 bar the measurements can be recalculated with the equation by Danilova and for higher pressure the equation of Haffner gave the minimal divergence. This can indicate that in the transition region the mechanism of nucleate boiling dominates.

With increasing of heat flux at constant pressure and mass flow rate a distinct increase of the heat transfer coefficient can be observed (Fig. 7).



heated length = 3 m

heated length = 5 m

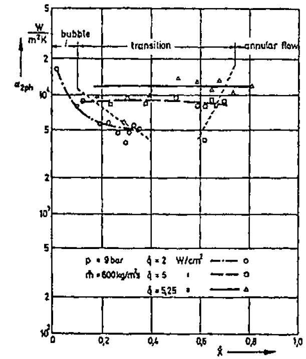
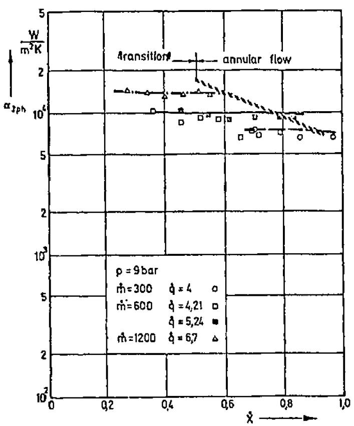


Fig. 7: Influence of the heat flux rate on the heat transfer coefficient as a function of mass quality.

The boundaries of the phase distribution and the flow pattern which were investigated with the described measuring procedures are also presented. The transition region extends up to the vapor fraction $\dot{x} \cong 0,55$ at 3 m measuring length. A dependence of the heat transfer coefficient on the quality can not be stated. The measurements with the 5 m test section gave similar results as those at 3 m. The boundary of the phase distribution is also marked, but at this length there can be observed an influence of the heat flux rate to the boundaries as a function of mass quality. The increase of the heat flux expands the transition region.
The influence of the mass flow rate on the heat transfer coeffi-



Two phase flow heat transfer coefficient in dependence of mass vapor fraction-influence of mass flow rate –

Fig. 8: Influence of the mass flow rate on the heat transfer coefficient as a function of mass quality

cient is shown in Fig. 8. An increasing mass flow signifies an increase in the heat transfer coefficient. The boundary of the transition moves to a mean vapor quality.

The phase distribution and the boundaries of the change in flow pattern are affected by the heat transfer- and mass flow rate and also by the geometrical factor of L/D-ratio as a comparison with the photographs showed. At constant quality at the exit of the test section there is less entrainment in the longer section - 5 m - than in the 3 m test section. The boundary from bubble- to churn-flow moves only slightly with increasing test section length, while the boundary from churn- to annular-flow is clearly observable at low qualities. Moreever the phase distribution at longer test section is more developed. It is more correspondent to the flow pattern which can be described theoretically. A similar tendency in the shifting of the flow pattern boundary under the influence of heat flux and mass flow rate was determined by Sato /12/ with water-vapor system at low pressure.

The examination of the L/D-ratio and its influence on the heat transfer coefficient were carried out with the variation of the test section and heated length. The results, presented in the ratio of heat transfer coefficient with and without evaporization in dependence on the Martinelli-parameter, show (Fig. 9) an increase of the ratio of heat transfer coefficients with increasing L/D-ratio.

The result of the experimental examinations is an equation, which corresponds to the equation developed by Pujol and Stenning, and which takes into account the heat transfer coefficient without evaporization, the Boiling-number, the Martinelli-parameter and the geometrical factor (1 + D/L).

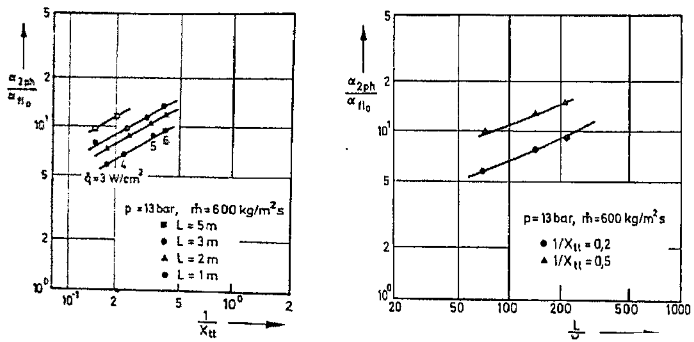


Fig. 9: Influence of L/D-ratio on the ratio of heat transfer coefficient with and without evaporization

$$\frac{\alpha_{2ph}}{\alpha_{fl_0}} = C_1 \left[Bo \cdot 10^4 + C_2 \left(\frac{1}{X_{tt}}\right)^r \right] \cdot \left(1 + \frac{D}{L}\right)^s \qquad (6)$$

Equation (6) can be used in the transition region from bubble to annular flow up to fully developed annular flow. The constants with which these measurements can be recalculated are

$$C_1 = 0{,}85 \qquad C_2 = 4{,}5 \qquad r = 0{,}35$$

$$s = \left(\frac{1}{X_{tt}}\right)^t \qquad t = 0{,}41$$

Conclusions

The heat transfer and phase distribution in vertical flows through a direct heated test channel of different lengths were studied. The heat transfer coefficient increases with increasing heat flux, mass flow rate, pressure and L/D-ratio. An influence of the vapor mass quality on the heat transfer coefficient in the transition region from bubble to annular flow can not be found. The phase change from bubble to churn flow in dependence of the mass quality is only minimally affected by the heat flux, mass flow rate and test section length, whereas the phase boundary from churn- to annular-flow is affected by these factors. With growing heat flux the boundary moves to higher mass quality while an increase of the mass flow rate and test section length results in a shift of the boundary in the direction of lower mass quality. Equation (6) can be used for the calculation of the two-phase flow heat transfer coefficient in the transition region from bubble- to annular flow.

Acknowledgement

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