Condensation from Steam–Air Mixtures in a Horizontal Annular-Flow Channel

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Abstract

Experimental results are reported on the condensation of steam from steam-air mixtures in annular flow at a cooled annular tube. The range of investigation was varied from laminar to turbulent flow for $1{,}5 \times 10^3 < Re < 1{,}3 \times 10^4$ and inlet concentrations $0{,}59 < p_{steam} / p_{total} < 0{,}59$. The measurements performed at ambient pressure allowed to evaluate the local heat and mass transfer coefficients for various inlet lengths in the 2 m long annulus. The steam concentration was locally measured with a newly developed dew-point probe.

Near the inlet region the experiments showed a slightly higher heat flux at the bottom of the tube, compared to that at the top, although it is expected to be there smaller due to a thicker liquid film. Far downstream from the inlet region the heat transfer at the top was higher than that at the bottom. The reasons for these effects are discussed.

1. Introduction

Experimental investigations on partial condensation are i.e. reported by Marschall /1/ and Lehr /2/, who measured the heat- and mass-transfer from binary mixtures at the inner side of a cooled vertical tube. Steam-air mixtures inside a single vertical tube were investigated by Ackermann /3/. Different vapours in the presence of air were studied by Gerhart /4/, also in a vertical annulus with the inner tube cooled. Dallmeyer /5/ investigated partial condensation with special emphasis on the laminar and turbulent boundary layers at a vertical flat plate with longitudinal flow.

The work reported here investigated the mechanism of heat- and mass-transfer along a horizontal tube with the condensation of steam from a steam-air mixture in an annular-flow channel. The heat- and mass-transfer are mainly influenced by the condensing film, flowing around the inner cooled tube and by the gradient of the air fraction versus the annulus. In contrast to pure steam condensation, with partial condensation the steam is transferred by convection and diffusion and also a temperature profile in the gas phase exists.

2. Experimental Set-Up

The experimental set-up for measuring the heat-transfer with partial condensation is shown in fig. 1. The heat exchanger in this experimental set-up consists of two concentric tubes. The inner tube is cooled with water and the outer tube is insulated. The steam-air mixture, out of which the steam is condensing, flows through the annulus between the outer and the inner tubes. To make possible local measurements of the heat- and mass-transfer along the heat exchanger tube, having a total length of 2 m, an instrumented section 0,25 m long was installed, and the gas inlet was designed to be movable in the axial direction. This was achieved by placing a pipe in the gas annulus consisting of two concentric tubes.

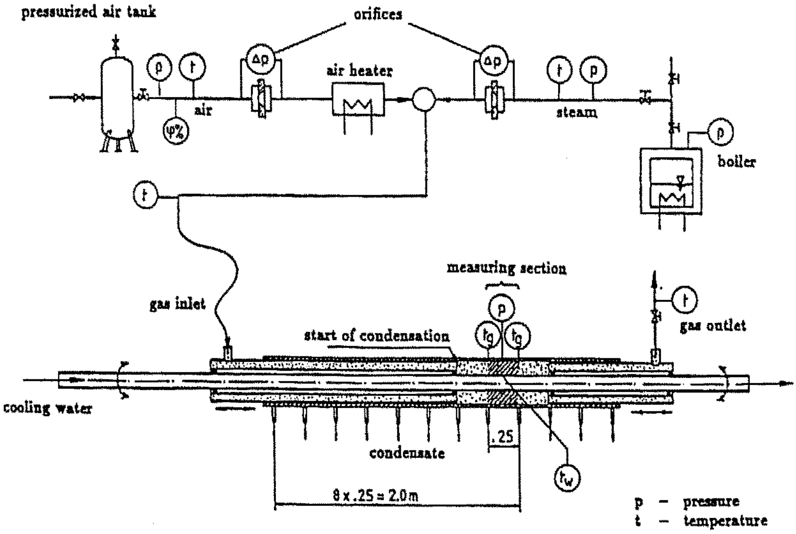

Fig.1: Experimental set up

Details of the test section are given in fig. 2 and fig. 3. Fig. 3 shows a cross section through the middle of the instrumented part of the test section. Here the cooled inner tube is surrounded by a turnable ring in which two measuring devices are installed: a pressure tap and a dew-point probe

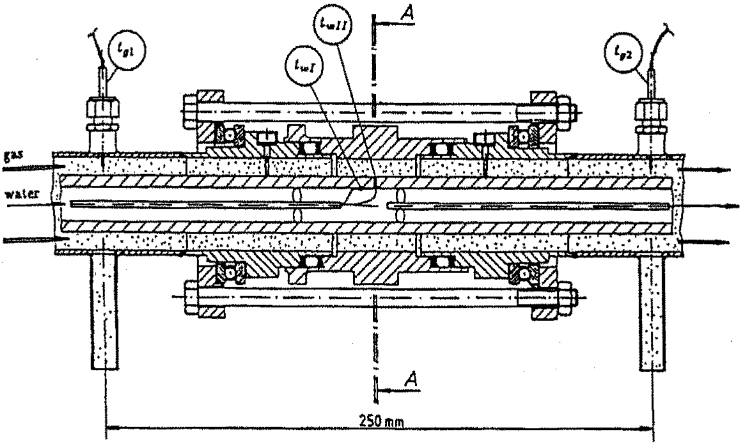

Fig.2: Test section

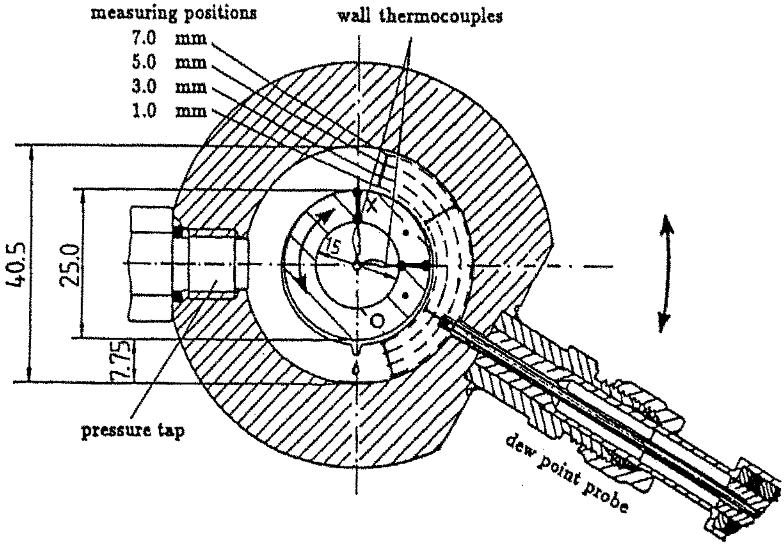

Fig.3: Cross section A-A through test section(Fig.2)

for determining the steam-air fraction in radial and circumferential direction. The aim of the newly developed dew-point probe is to obtain the local concentration of none-condensables in a condensing atmosphere. The method is based on the determination of the dew-point temperature and the total pressure of the gas mixture. The design of the dew-point probe is shown in fig. 4 and fig. 5. The measuring sensor of the dew-point probe consists of a miniature thermo-couple sol-

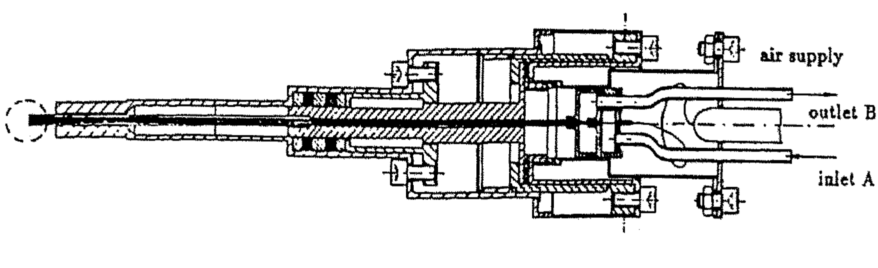

Fig.4: Dew point sensor

dered to a copper foil about 0,1 mm thick that separates the steam atmosphere from the channel system (fig. 5). The copper foil can be cooled from its rear side by air. To do this, the inner capillary tubes of the channel system are connected to an air supply. The two outer capillary tubes form an insulating space, to avoid condensation at the outer tube wall of the probe.

Without cooling the thermo-couple at the copper foil shows the true temperature of the steam-air mixture. As soon as the air-cooling is started, the temperature at the copper foil falls. Fig. 6 shows three typical temperature curves, recorded with a continuous line recorder. Curve 1 was detected

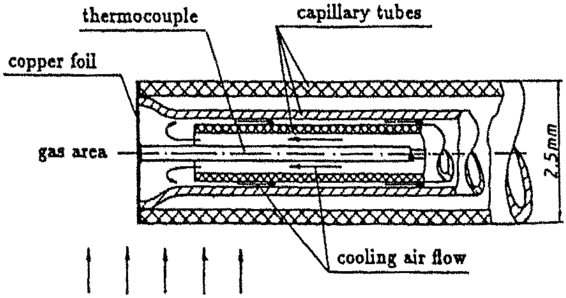

Fig.5: Top of the dew point sensor

in a pure steam flow, curve 3 in a pure air flow, and curve 2 in a mixture of steam and air. Curve 2 shows after the start of cooling at first a smooth decrease in the temperature until condensation begins on the outside of the copper foil and at this moment a sudden strong change in the temperature-time curve can be observed. This marks the dew-point temperature from which the steam concentration in the mixture can be calculated.

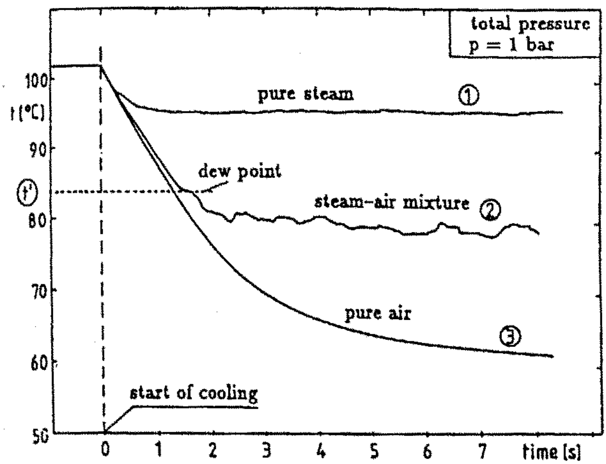

Fig.6: Temperature signals from dew sensor,
(1) pure steam, (2) steam-air-mixture, (3) pure air

The varying heat- and mass-transfer in the axial direction is mainly influenced by the decreasing steam flow rate and by the temperature- and air-fraction-profile in the radial direction in the gas phase. The total heat-transfer from the gas to the cooled wall is controlled by three phenomena.

1. Heat flux due to the phase change of the condensing steam.

2. Heat transport due to the convection in the gas annulus.

3. Convective heat transport in the liquid fil—

Details for evaluating the measured data and deducing from them, the heat- and mass-transfer coefficients can be found in /6/.

In contrast to the Nusselt-Wasserhaut-theory the film surface temperature cannot be assumed to be constant in our experiments, and also the influence of the condensate cooling cannot be neglected. The Nusselt-number is formed with the widths of the annulus and the thermal conductivity of the none-condensable gas.

$$Nu = \frac{\alpha_d(2s)}{\lambda_g} \tag{1}$$

The Sherwood-number is defined correspondingly.

$$Sh = \frac{\beta(2s)}{D} \tag{2}$$

The flow conditions are characterized by the Reynolds-number

$$Re = \frac{w_g(2s)}{\nu_g} \tag{3}$$

and the amount of steam in the none-condensable gas is expressed by the steam-air fraction.

$$\psi_{in} = \frac{p_{steam}}{p_{total}} \tag{4}$$

Due to the condensation of the steam on the way of the mixture through the annular channel the volumetric flow rate is decreasing, and by this also the velocity. The grade of flow reduction is a function of the steam-air-fraction ψ_{in} at the inlet of the channel and of the Reynolds-number of the incoming mixture Re_{Ge}. Both variables influence the heat- and mass-transfer coefficient. Fig. 7 conveys an impression of the velocity decrease in the channel for high and for medium steam-air-fraction. For low steam-air-fraction the velocity stays nearly constant, of course.

With horizontal orientation of the cooled tube where the film of the condensate is forming on, the temperature and the steam-air-fraction versus the circumference of the cooled tube or the channel is of interest. Fig. 8 shows the circumferential temperature distribution at the surface of the liquid film t_F and at the outer ($t_{W\,II}$) and at the inner ($t_{W\,I}$) surface of the cooled tube. On the right side of this figure, the steam-air-fraction at the surface of the liquid film (ψ_{DF}), and in the bulk of the steam-air-flow (ψ_{DG}) at a position of 0,875 m after the inlet for a Reynolds-number of Re_{Ge} = 7800 and with 95 % steam at the inlet of the channel.

The heat flux from the steam-air-mixture to the tube by condensation and by convection, averaged over the circumference of the tube, is a rather complicated function of the initial steam-air-fraction and of the Reynolds-number at the inlet versus the flow path. Fig. 9 shows that high steam-air-ratio combined with low Reynolds-numbers at the inlet of the channel can lead to low heat fluxes at the

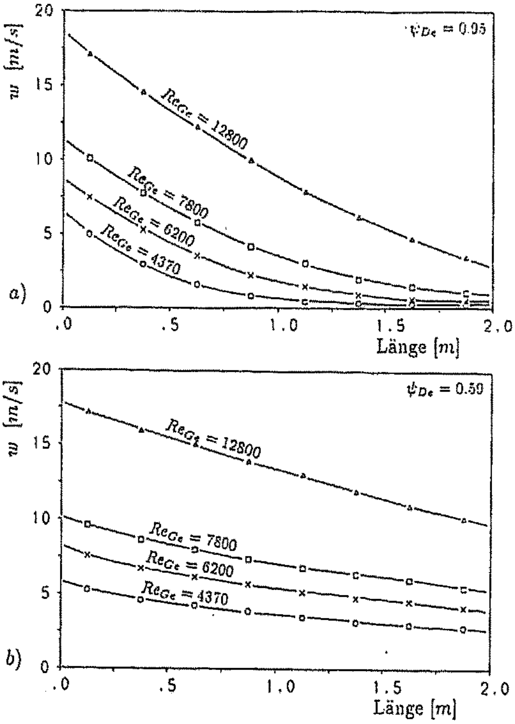

Fig.7: Velocity of steam-air-mixture versus channel length;
a) ψ_{De}=0.95; b) ψ_{DR}=0.59

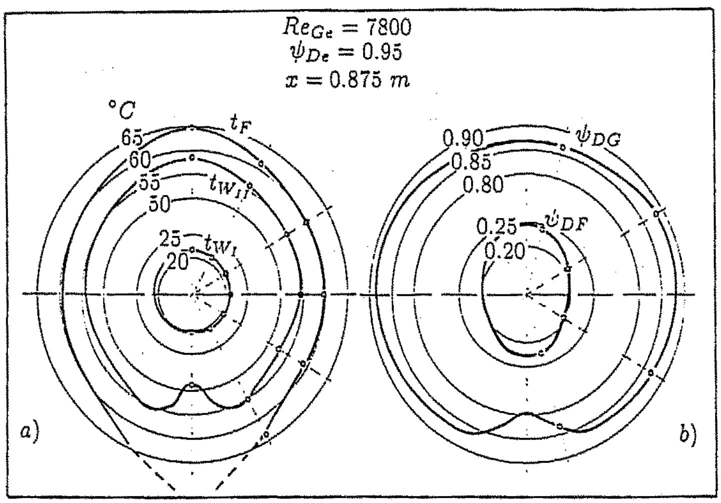

Fig.8: Temperature- and concentration-profile around the tube;
a) temperature at film surface (t_F) and the outer (t_{WI})and inner (t_{WI}) wall of the cooled tube;
b) steam concentration at liquid film surface (ψ_{DF}) and outside of boundary layer(ψ_{DG})

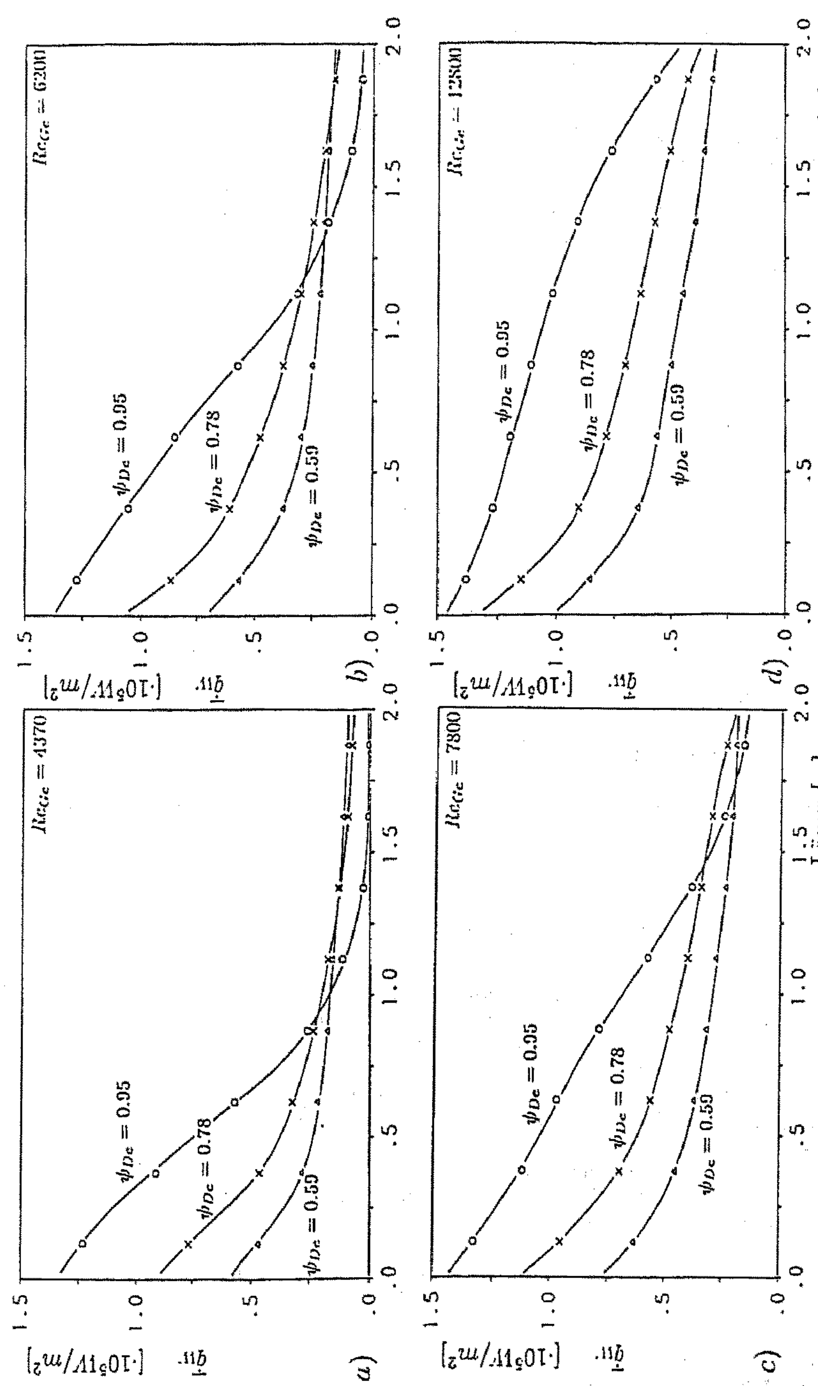

Fig.9: Local heat flux density, averaged versus the circumference of the channel

end of the channel due to the fact that by condensation the mixture loses most of its volume, resulting in very low velocities in the second half of the channel. In this part the heat-transfer is mainly or only controlled by convective transport. So the heat flux with high initial steam-air-fraction may become lower at the end of the flow channel than that with initial medium steam content.

From the measured values of heat flux, temperature differences, and an amount of condensed steam, the Nusselt- and the Sherwood-numbers can be calculated. Fig. 10 gives an impression how for a constant Reynolds-number at the inlet these dimensionless numbers vary versus the channel length depending on the steam-air-fraction at the channel inlet. After approximately 1 m flow path most of the steam is condensed out of the steam-air-mixture and the curves for different inlet-steam-air-fractions fall together.

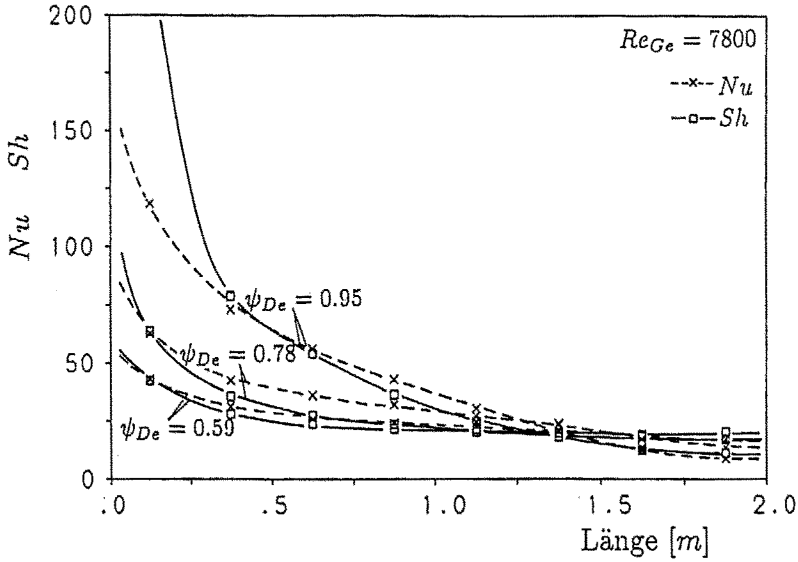

Fig.10: Local Nusselt- and Sherwood Numbers (averaged versus circumference)

The Sherwood-number, however, is also a function of the circumferential position over the perimeter of the tube. For high inlet steam-air-fractions the Sherwood-number at the top of the horizontal tube is at first higher than at the bottom. This can be easily explained by the varying film thickness of the the condensate versus the circumference of the tube. On the other hand the difference in the Sherwood-number versus the circumference can also result from turbulence effects at the lower part of the channel. Droplets falling down from the film on the cooled tube promote turbulence.

Comparing the results with the predictions of the film theory /1-5/ gives interesting information how the real fluid-dynamic situation differs from that assumed in the film theory. In fig. 11 the correction factor B defined in the film theory

$$B = \frac{Sh(1-\psi_{DF})}{Sh_0} = \frac{\beta(1-\psi_{DF})}{\beta_0} \tag{5}$$

is plotted versus the length of the channel. In Equ. 5 the subscript 0 indicates the situation when the steam mass flow to the tube is small compared to the convective transport. At the inlet of the channel the experiment shows a higher correction factor B than the theory predicts. This is an entrance effect, because for Sh_0 a correction - i.e. according to Hausen /6/ should be taken in account in the film theory. This Hausen correction predicts higher values up to 10-20 % compared to fully developed flow.

The upper diagram of fig. 11 represents an experiment in which the local Reynolds-numbers of the steam-air-mixture were always greater than 3000. Therefore throughout the whole channel turbulent flow existed. In the lower diagram of fig. 12 the Reynolds-number of the flow decreased so much that it was well below 1500 at the outlet.

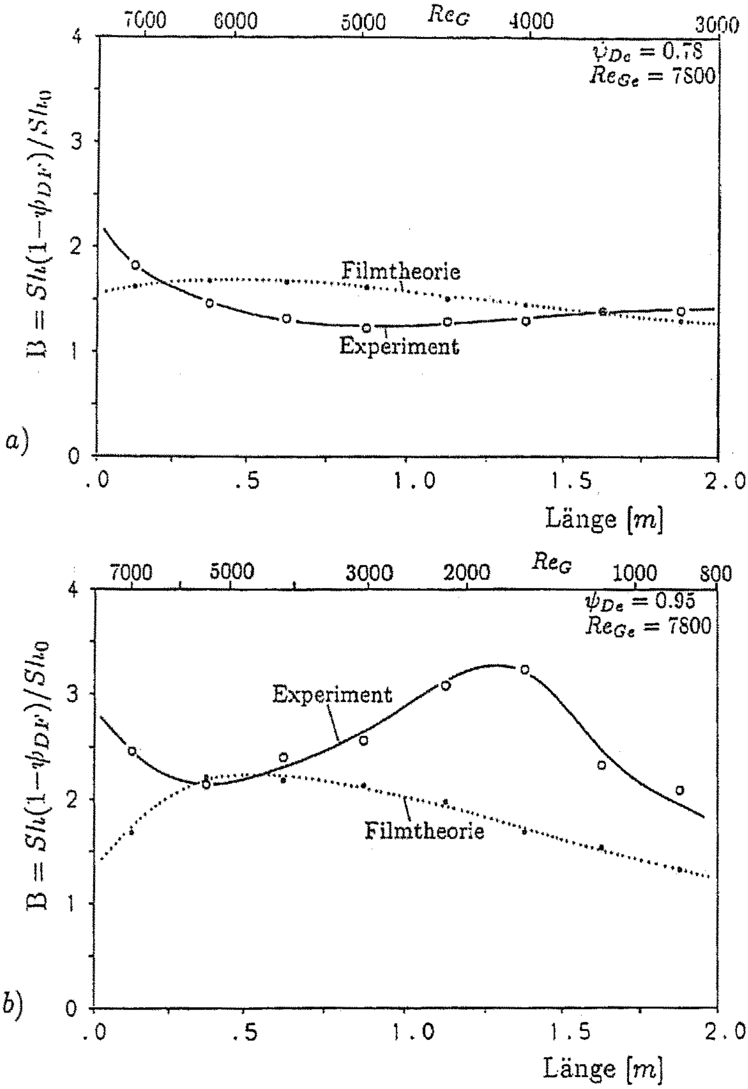

Fig.11: Correction factor B (equ.5) versus channel length;
a) Re_{Ge} >3000; b) 7800> Re_{Ge} >800

In the first case - with Reynolds-numbers higher than 3000 - the film theory predicts that the correction factor at first is rising and then falling again. The experiment shows the opposite behaviour, namely at first a falling and then an increasing correction factor. Excluding the inlet length and the outlet region, the experimental correction factor is always lower than the film theory predicts if the Reynolds-number in the channel does not fall below 3000, as fig. 11 in the upper diagram exemplary shows. There may be a good reason derived from Schlichting's /7/ theory for sucking off boundary layers on aeroplane wings. By sucking off the boundary layer the turbulence in the vicinity of the aeroplane wings is reduced. The mass flow rate of the vapour towards the cooled surface of the tube can be regarded as such a "sucking off" effect, because by phase change the volume of the vapour disappears. So this mass flow of vapour perpendicular to the main flow direction has the effect of damping the turbulence and by this of reducing the Nusselt- and Sherwood-number.

With low Reynolds-numbers - lower diagram in fig.11 - the experiment gives higher correction factors B than the film theory. Here the turbulence produced by the droplets separating from the horizontal tube may have an effect.

4. Conclusion

Partial condensation out of a mixture of condensable and none-condensable gases is different in vertically and horizontally orientated flow channels. With horizontal orientation of the cooling tube on which the liquid film of the condensate forms the droplets, separating and falling down from the tube, cause a concentration-profile in the gas-mixture in circumferential direction of the channel. In the lower part the liquid film at the tube wall is thicker and by this deteriorating the heat transfer to the tube wall, however, the separating droplets moving in a diagonal direction driven by drag-forces of the flow and by the gravity force are acting as additional heat sinks, because their temperature is lower than the dew-point temperature of the ambient mixture.

In addition the separating droplets enforce turbulence in the gas-mixture which also improves heat- and mass-transport.

5. References

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