PARTIAL CONDENSATION FROM STEAM-AIR MIXTURES IN AN HORIZONTAL TUBE BUNDLE HEAT EXCHANGER

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ABSTRACT First results of a study of partial condensation from steam-air mixtures in a parallel flow heat exchanger with a bank of three horizontal cooling-tubes in a vertical row are presented. A new apparatus and a newly developed dew-point probe for measuring steam-concentrations in the heat exchanger are introduced. The temperatures in the bulk flow and the cooling-tubes were recorded. Flow patterns of the condensate dropping from the cooling-tubes and the behaviour of the condensate-film are observed optically through a window. Preliminary results for different air concentrations $0.1 < \dot{m}_L/\dot{m}_D < 2$ and results for a single horizontal tube are reported.

1. Introduction

Shell-side condensation is relevant to many important applications both in power engineering and in processing industry. Many experimental and theoretical investigations have been done in this field. A very good review is given by Marto.[1]

A special point of interest in condensation experiments is the influence of non-condensible gases on the heat and mass transfer rates. Already small amounts of inert gas in vapour can cause a gradient of the air-fraction close to the condensate and may reduce significantly the condensation heat transfer rates. Differently from pure steam condensation, during partial condensation the steam is transferred by convection and diffusion. A temperature profile in the gas phase develops. Very good summaries of this phenomena are given by Chisholm and Webb and Wanniarachchi.[2,3] A popular method to calculate shell-side heat transfer with influence of inert gas was developed by Colburn and Hougen.[4] An assumption for calculating condensation heat transfer is the knowledge of the the mass transfer coefficents for the diffusive and convective transport of the vapour through the boundary layer. Many experimental and numerical investigations have been reported in this field. Some of the recent investigations are made by Kutateladze et al. and Cavallini et al..[5,6]

A new area in this investigations is the study of horizontal parallel flow condensation heat exchangers. Results on condensation in parallel flow on a single horizontal tube have been reported by Nolte.[7] In horizontal flow the condensate in the heat exchanger is influenced by momentum and shear stress transfer perpendicular to gravity. This causes a very specific motion of the condensate film and the fluid leaving the cooling surface.

In a channel with two or more cylindrical, horizontal cooling surfaces in a vertical row, the condensate is dropping from the upper to the lower row. Thus is disturbing there the boundary layer and inundating the tube. The droplets between the tubes lead to direct condensation on themselves. The condensation rate at the droplets is a function of the droplet diameters and the droplet velocities. Therefore the air fraction in in the lower part of the channel is higher than in the upper part. In steam-air mixtures this vertical stratification is steady. Caused by the higher condensation rate in the lower part of the heat exchanger, the pressure gradient along the cooling tubes is higher than in the upper part. In this way the gas will flow from the upper to the lower parts of the channel. Thereby secondary turbulences can be stimulated. A special point of interest is the behaviour of the flow, when its Reynolds number is in laminar regime.

2. Apparatus and Procedure

The schematic of the experimental apparatus is shown in Fig. 1. Steam is generated from destilled water in a boiler by three 15-kW heaters. The vapour mass-flow is regulated by three control valves. The air is filtered from oil and dust and then heated by three 0.9 kW heaters. One of them is connected by a variable transformer so that the air-heater power can be variied continuosly from 0 -2.7 kW. The vapour mass -flow is measured in a V-Cone mass-flowmeter, the air mass-flow with an orifice plate. The test fluids pass from a mixer to the test section. The tubes are heated and insulated carefully, in order to avoid condensation taking place there.

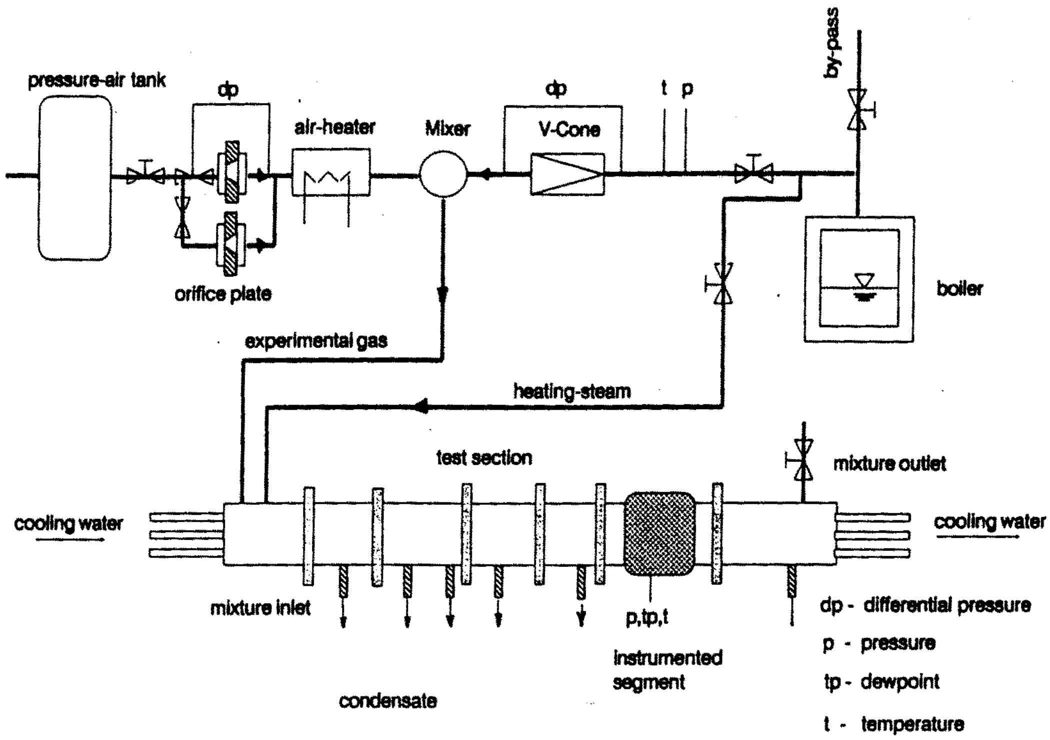


Fig. 1: Schematic of the test apparatus

Test section

The dimensions of the tubes and the shell are indicated in Fig. 2. The total length of the duct is 2m. The tubes and the shell are made of stainless steel. For investigations in a parallel flow condensation heat exchanger the gradients of the mixture temperature, the vapour concentration, the temperatures of the cooling tubes, the flow pattern of condensate film and the condensate drops must be known. Therefore large scale instrumentation of the whole test channel would be necessary. To avoid this problem the used here heat exchanger is built up by eight different parts, which are connected with flanges. Only one segment, shown in Fig. 3, is instrumented with a thermocouple, a dew point probe and a pressure transducer. The probes are held by a disc which can be rotated around its axis. The thermocouple and the dew-point probe can traverese the test section in horizontal direction. The diameters of the probes are small enough - 4 mm for the dew-point probe, 1 mm for the thermocouple - not to disturb the flow. The other side of this section is closed by a window which allows the observation of the condensation process. The window is heated by hot air to avoid condensation or thermal dissipation at this place. Both, the window and the rotating disc are flush mounted in the channel so that they do not disturb the flow.

Five segments with different lengths (three 500 mm segments, two 200 mm segments and one 100 mm segment) are only equipped with condensate outlets. Fig.

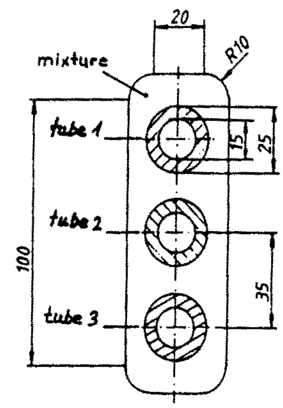
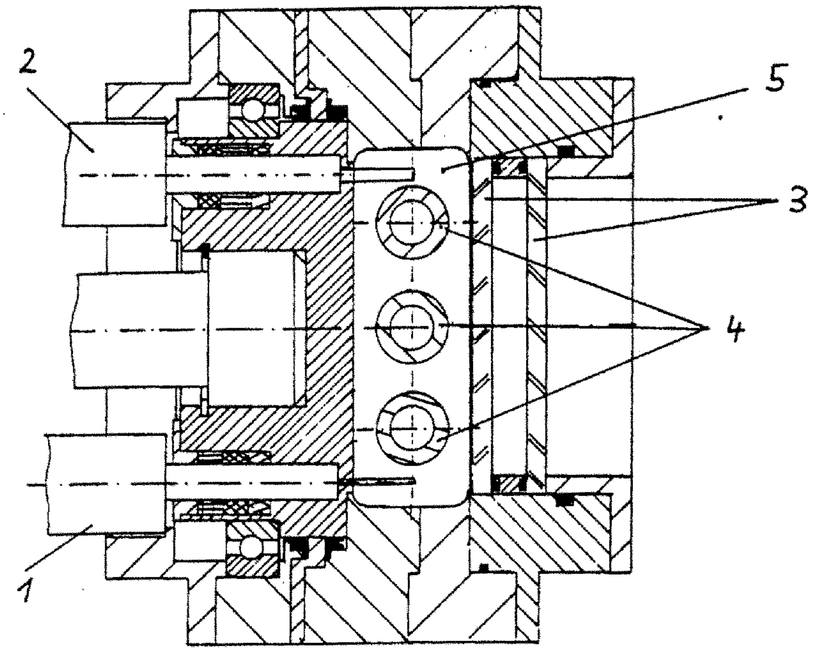


Fig. 2: Dimensions of the test channel

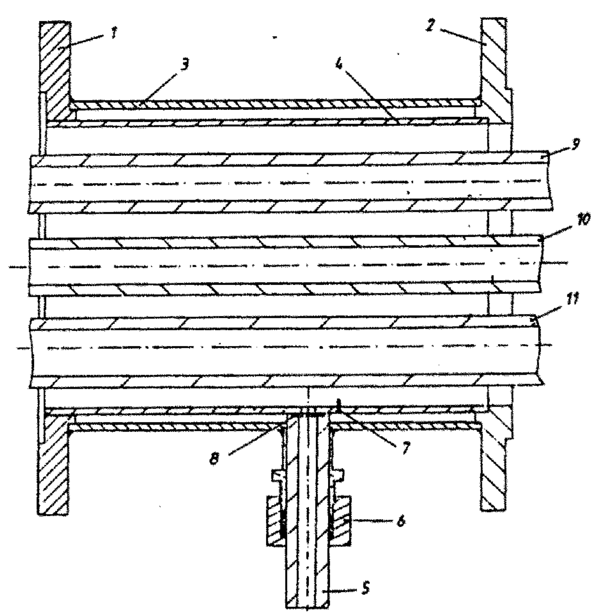
4 shows one of the segments. They can be fixed in front of the instrumented part in any number and succession. In this way the gradient of temperature and concentration over the whole test section can be measured with a single thermocouple and a single dew-point probe. The segments are heated with steam flowing through the space between test channel and outer tube. In front of the segments a mixture-inlet part is



No.	Name
1	Thermoelement
2	Dew-point probe
3	Windows
4	Cooling-tubes
5	Investigated channel

Fig. 3: Instrumented part

mounted. There the gas passes a flow straightener and reaches the test-section parallel with the cooling tubes. The inlet temperature of the mixture is measured by two thermocouples.
Each of the cooling tubes is instrumented by four NiCrNi thermocouples of 0.5 mm diameter. The hot-junctions of the thermocouples are flush-mounted with the tube walls, as indicated on Fig 5. The inner ones are soldered in a groove, the outer ones in a drill hole perpendicular to the tube. The four thermocouples are installed in a capillary mounted inside the tube. The thermocouples are located in the center of the instrumented section. Therefore the gradient of the temperature along the tubewalls can be measured with a single thermocouple. The cooling tubes can be rotated 45° around their axis, to measure the temperature



No.	Name
1,2	Flanges
3	Outer-tube
4	Investigated channel
5	Condensate outlet
6	Threaded joint
7	Flight
8	O-seal
9 - 11	Cooling-tubes

Fig. 4: Segment of the heat exchanger

gradient along the perimeter of the tube with two thermocouples. In each tube a thermocouple is mounted in a slide which can be moved along the whole test section, to measure the temperature gradient in the cooling water.

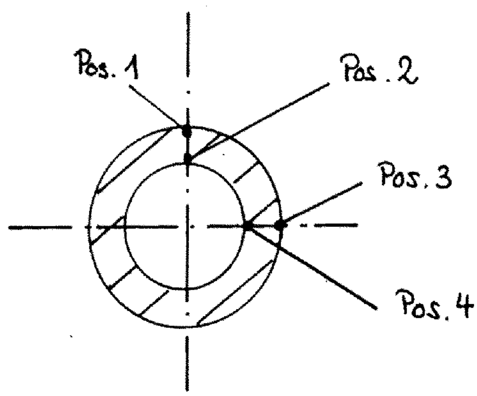


Fig. 5: Locations of the thermocouples in the cooling-tubes

Dew point probe

The dew point probe mentioned earlier is shown in Fig. 6 was developed for the application in this kind of condensation experiments. The temperature range in which the probe can be used is limited by the gasket material. All parts of the probe which are in contact with the measuring fluid are made of stainless steel. The top of the probe has a diameter of 4mm, that flow-structures are basically not disturbed. (see Fig. 7). The probe consists of a thermocouple whose hot junction is welded in a round foil of 0.1 mm thickness and 4mm diameter. The foil is cooled by nitrogen-gas which is flowing through a capillary system. For the measurements the effect of condensation heat is used

No.	Name
1	Gas area
2	Foil
3	Thermocouple
4	Capillaries

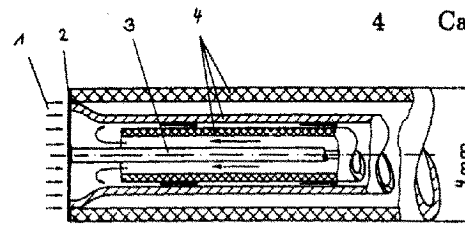
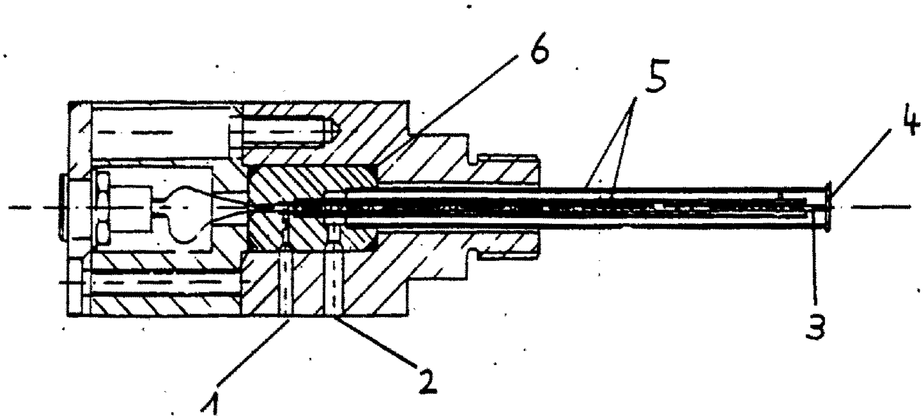


Fig. 7: Top of dew-point probe

to detect the dew-point temperature. The measuring can be executed, when the temperature of the probe is about five degrees above the dew-point temperature. Then the cooling can be started by opening the ball valve in the nitrogen channel. The foil is cooled by the gas and the temperature is recorded by a voltmeter of a rate of at least 50 Hz. When the foil reaches the dew point temperature condensation begins and the temperature gradient suddenly changes because of the condensation heat which works against the cooling effect. The temperature at which the gradient changes indicates the dew-point temperature. When the dew-point temperature was reached the cooling can be stopped. The next measurement can be executed after the condensate at the foil has evaporated. The dew-point is calculated using the equations (1) and (2).

$$ln(p_s/mbar) = 19.0160 - \frac{4064.95}{(T/°C) + 236.25} \qquad (1)$$

$$\frac{m_L}{m_D} = \frac{p - ps}{ps} \times \frac{R_D}{R_L} \qquad (2)$$



No.	Name
1	Gas inlet
2	Gas outlet
3	Thermocouple
4	Foil
5	Capillaries
6	O-seal

Fig. 6: Dew-point probe

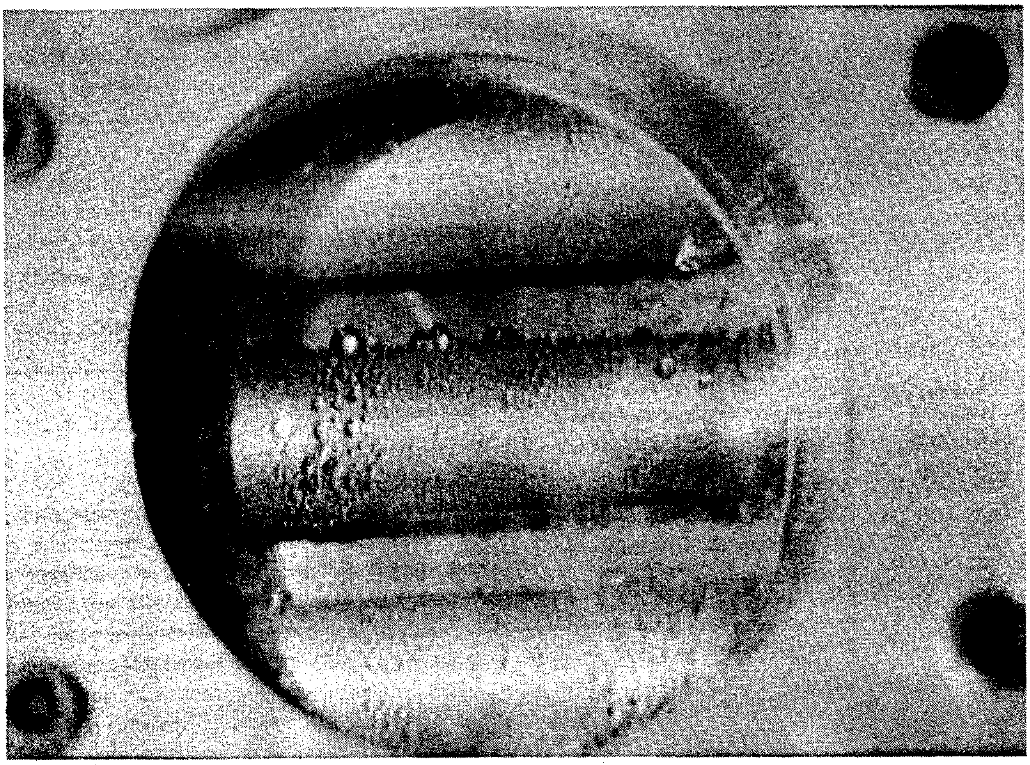


Fig. 8: Process of initial dropwise condensation on three horizontal cooling tubes

3. Experimental Results and Discussion

Experimental data has been obtained with $T_i = 100°C$, $\dot{m}_D/\dot{m}_L = 0 - 1.4$ and $v = 8m/s$ at the gas-inlet of the heat-exchanger.

Observations of Condensation

After initial dropwise condensation film condensation occures on the tubes. Fig. 8 shows a photograph taken from the heat exchanger during dropwise condensation. The photograph was taken at a distance of 2m from the mixture inlet. Drops hanging on the bottom of the tubes, fields with dropwise condensation, and fields wiped free from condensate by big drops moving over the tubes can be seen.

The flow patterns of the condensate can be divided in three groups: The first group occurs condensation at low bulk stream velocities. The drops have the same characteristics as in quiescent vapour. With increasing velocities, the big drops hanging at the bottom of the cooling-tubes begin to tremble. The drops vibrate for at least 10 sec. and then leave the tube in vertical direction. In this second flow pattern group there is no increase of drop velocities, as shear stress the bulk-flow is too small for moving the drops. In the third group, the shearstress from the bulk flow is big enough to move the drops hanging at the cooling tubes. Drops driven by the flow about 10 cm along the tubes have been observed. They leave the tubes with a high vertical velocity component.
An interesting area for investigations in parallel flow heat-exchangers will be the last group of condensation pattern, where shearstress and – perpendicular to it – gravity produces high velocities in the condensate film or condensate drops.

Measurements of the heat-flux

The heat flux at the cooling-tubes is calculated using equation (3) with the temperatures measured by the thermocouples soldered in the tube walls.

$$\dot{q_W} = \frac{2}{d_{II} ln \frac{d_{II}}{d_I}} \times \lambda_W \times (t_{WII} - t_{WI}) \qquad (3)$$

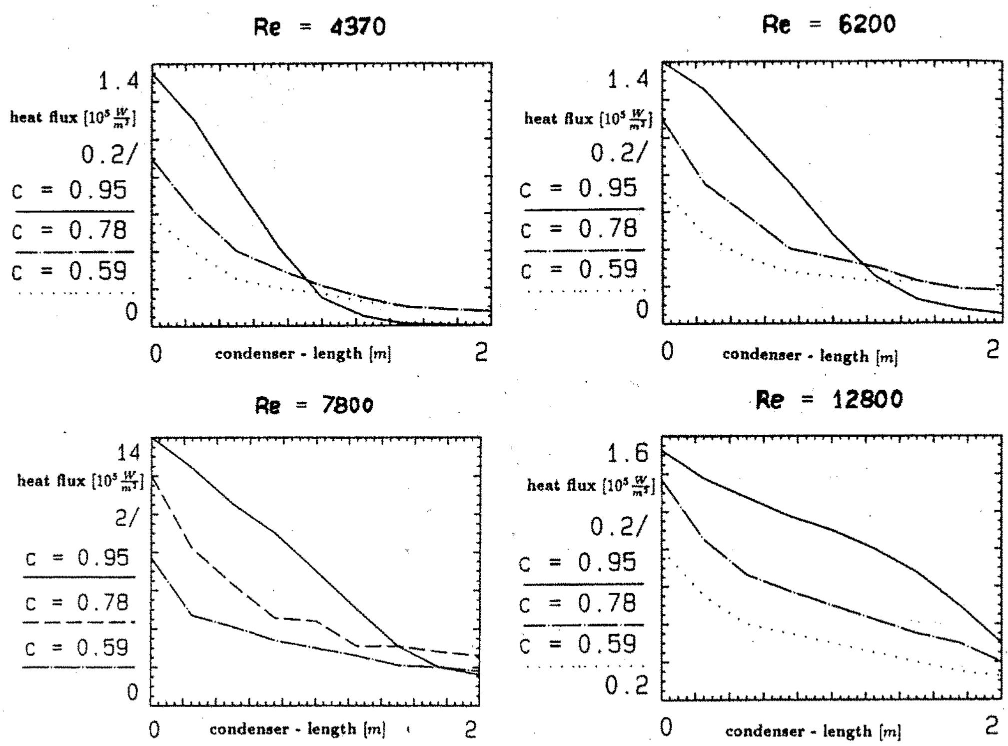


Fig. 9: Heat-flux densities along a single horizontal tube

The heat flux is a function of the Reynolds number of the mixture and the local steam concentration. Vapour condensation give rise to a decrease of thr flow velocity and vapour concentration along the tubes. Fig. 9 shows the distribution of heat-flux for three different steam concentrations and Reynolds numbers along a single horizontal tube.

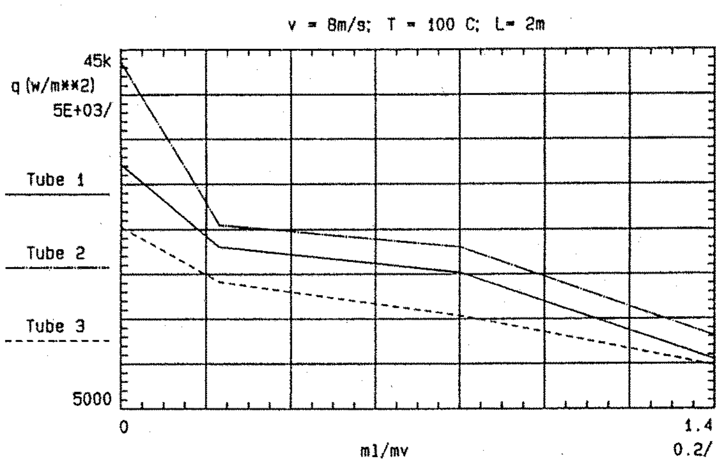


Fig. 10: Heat flux densities after a flow-length of 2m at the three cooling tubes

In contrast to a single tube, the heat transfer coefficients in a tube bundle are influenced by condensate inundation. The condensate dropping from the upper to the lower tubes causes there additional convective heat transfer as well as additional heat conduction resistance. Gravity, and perpendicular to it, shearstress cause a complex flow of the condensate which increases the convective heat transfer from the condensate film to the tubes.

In Fig. 10 heat flux as function of the tube row is shown for the inlet velocity $8m/s$ and different air concentrations. It can be observed, that for all air concentrations the heat flux density at the middle tube 2 is higher than on the upper tube 1. Here the influence of the increasing convective heat transfer by condensate inundation is decisive. On the lower tube, however, the additional heat conduction resistance due to vapour inundation will be predominante.

Conclusion

A new apparatus for investigating parallel flow condensation heat exchanger, with three horizontal coolingtubes in a vertical row is introduced. A dew-point probe developed for the use in this kind of condensation experiments is described.

Optical observations and preliminary results are presented. The authors will continue the extensive measurements and will develop equations to calculate condensation for a wide range of vapour concentrations and Reynolds numbers in a horizontal parallel flow condensation heat exchanger.

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References

1. P.J. MARTO, "Heat Transfer and Two-Phase Flow during Shell-side Condensation", *heat transfer engineering, vol.5, nos. 1-2*, (1984).

2. D. CHISHOLM, " Modern Developments in Marine Condensers: Noncondensable gases": An Overview in *Power Condenser Heat Transfer Technology*, eds. P. J. Marto and R. H. Nunn, pp 95-142, Hemisphere, New York, (1981).

3. R.L. WEBB and A.S. WANIARACHCHI, " The Effects os Noncondensable Gases in Water Chiller Condenser -Literature Survey and Theoretical Predictions, *ASHRAE Trans., vol. 86, pt. 1, pp. 142-159*, (1980).

4. A.P. COLBURN and O.A. HOUGEN, " Design of Cooler Condensers for Mixtures of Vapors with Noncondensing Gases", *Ind. Eng. Chem., vol. 26, pp.1178-1182*, (1934).

5. S.S. KUTATELADZE, I.I. GOGONIN and V.I. SOSUNOV, "The influence of condensate flow rate on heat transfer in film condensation of stationary vapor on horizontal tube banks", *Int. J. Heat Mass Transfer, Vol. 28, No. 5, pp. 1011-1018*, (1985).

6. A. CAVALLINI, S. FRIZZERIN, L. ROSETTO, "Condensation of R-11 vapor flowing downward outside a horizontal tube bundle", *Proc. of the Eights Int. Heat Transfer Conference San Francisco*, CA U.S.A, (1986).

7. G. NOLTE, "Kondensation aus Dampf-Luft Gemischen im horizontal durchströmten Ringkanal", *Dissertation TU München*, (1989).

Used symbols

c		concentration air/steam mass flow
d_{II}	$[m]$	outer tube diameter $[m]$
d_I	$[m]$	inner tube diameter $[m]$
$\dot{m}_L$	$[\frac{kg}{s}]$	air mass flow
$\dot{m}_D$	$[\frac{kg}{s}]$	steam mass flow
p	$[\frac{N}{m^2}]$	pressure
p_s	$[\frac{N}{m^2}]$	sturation pressure
q_w	$[\frac{w}{m^2}$	heat flux
R_D		Gas constant steam
R_L		Gas constant air
T	$[K]$	Temperature