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CRITICAL HEAT FLUX (CHF) AND POST-CHF HEAT TRANSFER IN HORIZONTAL AND INCLINED EVAPORATOR TUBES

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Abstract—The paper describes heat transfer experiments performed on evaporator tubes at various inclinations and over a parameter range typical of fossil-fired once-through boilers and waste heat recovery boilers. Particular attention is paid to the influence of tube inclination on the occurrence of the boiling crisis (dryout) and on heat transfer in the post-CHF regime. The paper identifies calculation models which can be used to determine the critical steam quality and wall temperatures in partially wetted and dried-out tube regions.

Key Words: heat transfer, boiler tubes, once-through boiler, waste heat recovery boiler, critical heat flux (CHF), post-CHF heat transfer, critical steam quality, wall temperatures

1. INTRODUCTION

In the design of the tubing in once-through boilers the temperatures and temperature gradients which are to be expected in evaporator tube walls during boiler operation have to be known. These parameters determine the material to be used and the allowable heat production in the boiler. The wall temperatures and temperature gradients, in turn, are governed by the system parameters, such as pressure or mass velocity, and by the wetting state of the heat-transferring surface (tube interior surface, heating surface). With a wetted tube interior surface heat is transferred to the fluid by convection. If, as the enthalpy of the flow increases, the heating surface temperature exceeds the saturation temperature of the fluid, boiling starts at the tube wall. This enhances heat transfer in such a way that the temperatures at the tube interior surface remain approximately constant up to the boiling crisis. This means that, as long as the heating surface is wetted, the maximum wall temperatures are only insignificantly higher than the saturation temperature of the flow. If the heating surface is not wetted, however, the heat from the tube wall is transferred directly to the steam. Because of the low transport coefficient of steam, heat transfer in this regime (post-CHF regime) is severely reduced, so that significantly higher heating surface temperatures are to be anticipated than with a wetted tube wall.

In addition to the system parameters, the steam quality of the flow at the boiling crisis affects the level of the tube wall temperatures in the post-CHF regime. If the boiling crisis (dryout) occurs at low steam qualities, higher wall temperatures must be expected than if the tube drys out at high steam qualities. The reason for this is that the average flow velocity rises with increasing steam quality, thus enhancing the cooling at the tube interior surface.

The models for calculation of the heating surface temperatures in the post-CHF regime presented in the literature to date are mostly based on experiments on vertical tubes, e.g. Collier (1972) or Hein & Köhler (1984). In this ideal case, flow patterns and heat transfer are not influenced by gravity effects. In practice, however, the tubing in boilers is frequently installed horizontally or inclined. For this reason, the effects of gravity in non-vertical tubes, e.g. phase separation in two-phase flow, are of importance to the design of such boilers. Since, however, no systematic investigations are available on this subject, experiments with various tubes and tube inclinations have been performed over a wide range of parameters. The results of these experiments enable us to quantify the effects of gravity on flow patterns and heat transfer in evaporator tubes.

2. EXPERIMENTS

The experiments were performed on a high-pressure loop [BENSON test rig, Hein *et al.* (1977)], figure 1. The test loop is operated with demineralized and degasified water from a water treatment

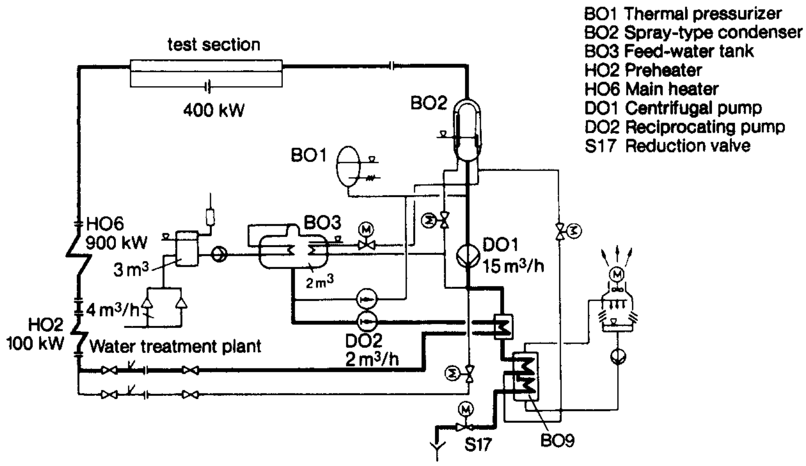


Figure 1. BENSON test facility.

plant which is fed into the pressurized part of the loop by a reciprocating pump. The water flows through an electrical heater which can be controlled to vary the thermodynamic condition of the fluid as it enters the test section. The test tube is also heated electrically to simulate boiler conditions. The steam formed in the heater and in the test section is condensed in a downstream spray-type condenser. From here, the water is passed via a centrifugal pump to the main cooler before being discharged through a reduction valve. Pressure in the loop is controlled by means of a thermal pressurizer.

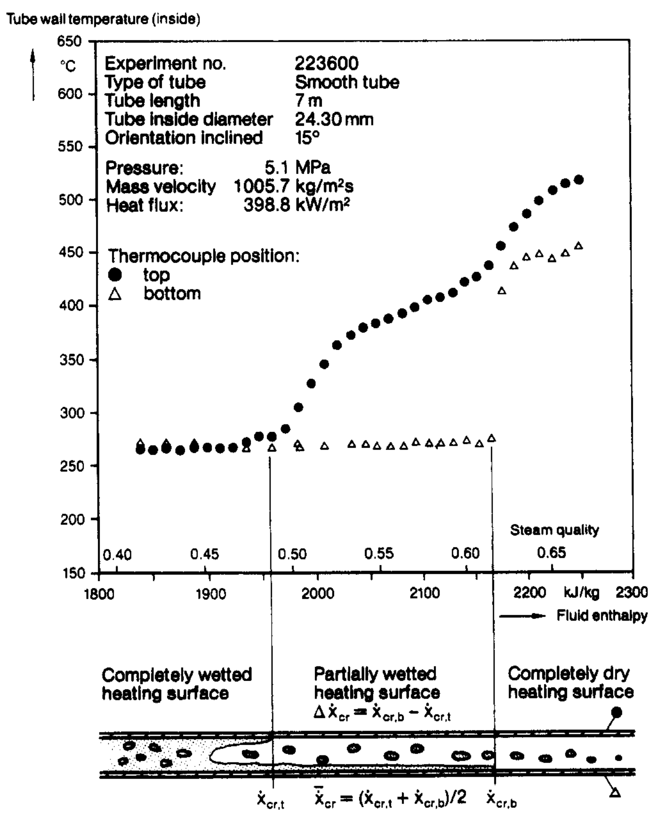


Figure 2. Wall temperatures of an inclined evaporator tube.

The parameter range covered by the experiments represents typical once-through boiler operating conditions, but also those of waste heat recovery boilers used in combined gas and steam turbine processes with coal gasification:

system pressure	$25 \leqslant p \leqslant 200$ b
mass velocity	$300 \leqslant \dot{m} \leqslant 2500$ kg/m^{2}s
heat flux	$75 \leqslant \dot{q} \leqslant 600$ kW/m^2
tube inside diameter	$d = 12.5$ and 24.3 mm
tube wall thickness	$s = 2.3$ and 4.6 mm
tube inclination	$\varphi = 0$, 15, 30 and 90°.

Tubes with a heated length of 7 m were used. In order to be able to measure axial temperature courses but also differences in temperature around the circumference, the outsides of the tube walls were instrumented with thermocouples at intervals of about 200 mm along three lines. For analysis of the experiments, the wall temperatures on the inside of the tubes were calculated from the temperatures measured on the outside.

3. CRITICAL HEAT FLUX

For evaluation of the experiments, the temperatures measured were plotted vs position along the tube which is related as an equivalent definition, to a certain enthalpy of the fluid, figure 2. From the shape of the curves it can be derived that, in the case of a horizontal or inclined tube, there are sections in which the heating surface is completely wetted, wetted only around a part of the circumference of the tube and fully dried out. The boiling crisis, which by definition occurs on the boundary between the wetted and non-wetted region, thus extends in the form of an ellipse over the circumference of the tube and along a certain tube length (in some cases over several metres).

Calculation of the tube wall temperatures requires the location and the axial length of the transition region with a partially wetted heating surface to be known, since the steam quality of the flow affects the temperature pattern in this region. Analysis of the experiments resulted in the finding that the mean steam quality of the flow in the transition region corresponds to the dryout steam quality to be expected in a vertical tube. This allows us to use correlations developed for vertical evaporator tubes, such as that of Kon'kov (1966), for horizontal or inclined tubes also.

The axial length of the transition region can be determined using the Froude number (Fr). This number is defined as the ratio of inertia to gravity forces in the flow and thus gives a measure of stratification. If the formulations derived by Taitel & Dukler (1976) and Wallis (1969) for horizontal tubes are extended, taking into account only those gravity force components that act perpendicular to the tube wall, the effect of the orientation of the tube on flow stratification can be described as

$$\mathrm{Fr} = \frac{\bar{x}_{cr} \cdot \dfrac{\dot{m}}{\sqrt{\rho_G}}}{\sqrt{g \cdot \cos\varphi \cdot d \cdot (\rho_L - \rho_G)}}, \qquad [1]$$

with $\bar{x}_{cr}$ being the mean steam quality in the transition region, ρ_L and ρ_G describing the density of water and steam, respectively, and g as the gravitational acceleration.

To verify the general applicability of the extended Froude number for the determination of the axial length of the transition region with a partially wetted heating surface, in figure 3 measured differences in steam qualities ($\Delta\dot{x}_{cr}$) at tube wall dryout at the top and bottom side are plotted vs this Froude number. According to the figure, pronounced stratification effects and thus an extended transition range is present if the $\mathrm{Fr} < 7$.

At $\geqslant 10$, the tube orientation has hardly any influence on the boiling crisis. The dependency established in the experiments can be described by the empirical formula

$$\Delta\dot{x}_{cr} = \frac{16}{(2 + \mathrm{Fr})^2}, \qquad [2]$$

the validity range of which is defined by the parameter range of the experiments, see section 2.

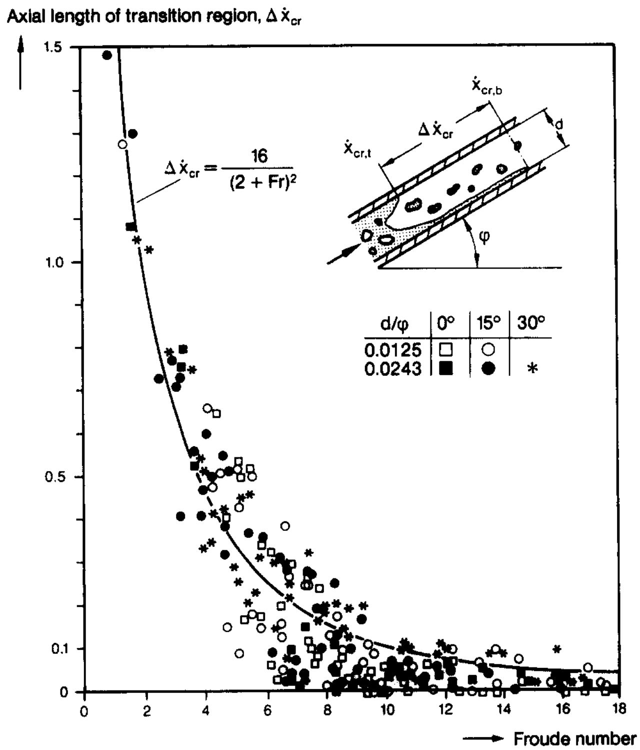


Figure 3. Dependency of the axial length of the transition region on the Froude number.

An investigation of the flow pattern at the tube position where the boiling crisis occurs by means of the method suggested by Taitel & Dukler (1976), figure 4, indicates annular flow to be most probable for most of the experiments presented here. As already evident from figure 3, the transition from symmetrical to asymmetrical annular flow takes place at Fr = 10, since stratification effects in the flow start influencing the boiling crisis at this number.

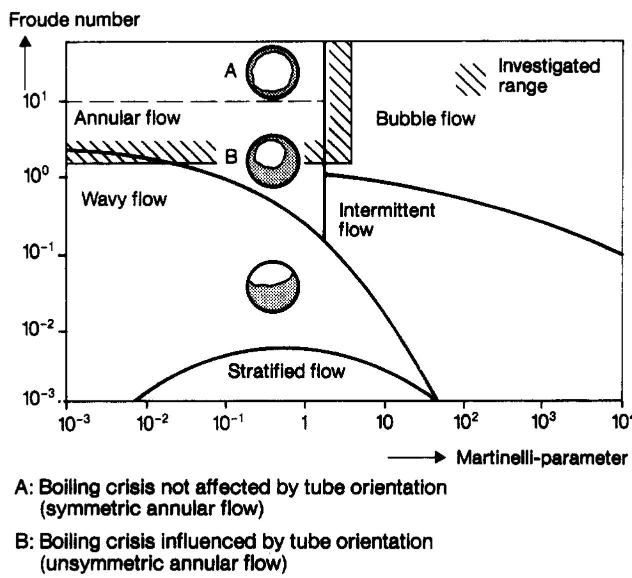


Figure. 4. Flow pattern at the location of the boiling crisis.

Finally, the steam qualities in the flow at the axial positions at which tube dryout at the top and bottom side occurs ($\dot{x}_{cr,t}$ and $\dot{x}_{cr,b}$) can be calculated as follows:

$$\dot{x}_{cr,t/b} = \bar{\dot{x}}_{cr} \pm \frac{\Delta \dot{x}_{cr}}{2}, \qquad [3]$$

"$-$" top side of tube
"$+$" bottom side of tube.

Thus, two points essential to the determination of the axial tube wall temperature courses are known.

4. POST-CHF HEAT TRANSFER

The tube wall temperatures in the region of a partially wetted heating surface are generally lower than those measured with a vertical tube under otherwise identical conditions, figure 5. This is due not only to the redistribution of heat when the tube circumference is only partially wetted, but also to the lower steam temperature and the improved heat transfer in this region. For the determination of the maximum temperatures and temperature gradients in the tube wall, it is sufficient merely to calculate the temperatures at the top side of the tube, as the highest temperatures occur at this location. The bottom side of the tube remains wetted in the entire transition region, so that the maximum temperature gradients are likewise to be expected in this zone.

To determine the heat redistribution in the tube wall from the non-wetted tube top side to the wetted tube bottom side—under steady-state conditions—Poisson's differential equation must be solved:

$$\lambda \cdot \left(\frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \cdot \frac{\partial T}{\partial r} + \frac{1}{r^2} \cdot \frac{\partial^2 T}{\partial \varphi^2} \right) + q''' = 0, \qquad [4]$$

where T denotes the wall temperature, r and φ are the coordinates in the radial and azimuthal directions, respectively, λ indicates the conductivity of the material and q''' represents the specific heat production in the tube wall.

A relaxation method such as that described, for example, by Eckert (1966) can be used for this purpose. This method calculates the temperature field stabilizing in the tube wall as a function of the local conditions, e.g. angle of wetting and heat flux, figure 6. The largest temperature gradients occur in the region of the boundary between the wetted and unwetted part of the tube surface.

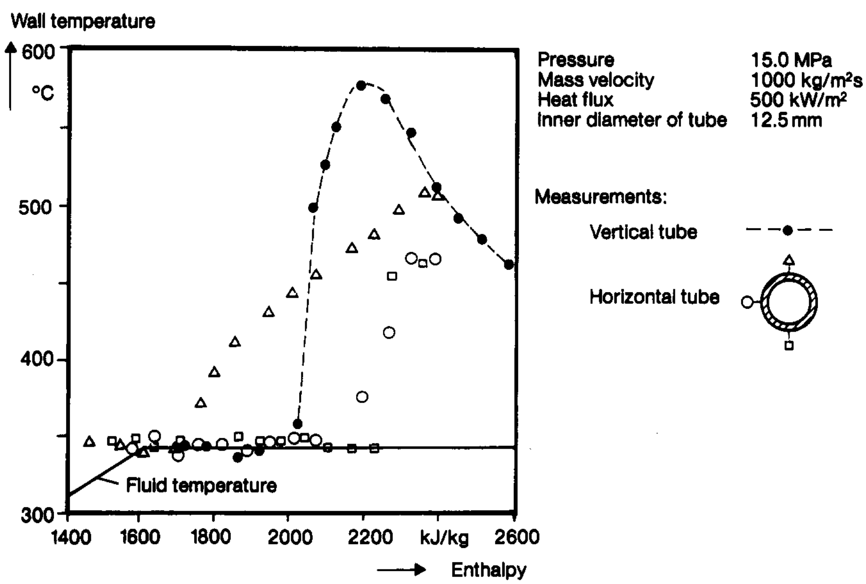
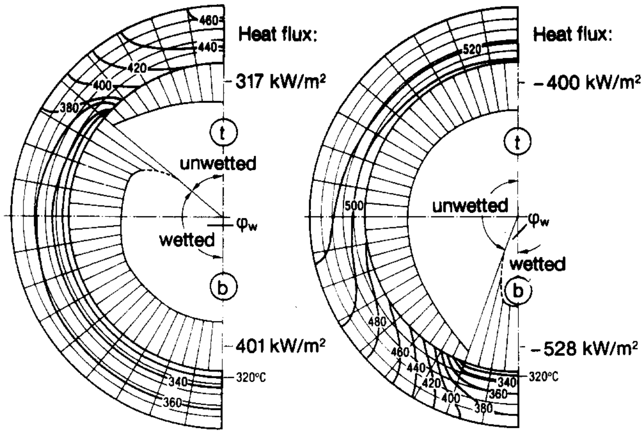


Figure 5. Comparison of wall temperatures in horizontal and vertical evaporator tubes.



Average heat flux	$\bar{\dot{q}} = 399$ kW/m²	407 kW/m²
Redistribution of heat	$1 - \frac{\dot{q}_t}{\bar{\dot{q}}} = 0.2$	0,02

Figure. 6. Temperature distribution in the tube wall with different wetting angles.

If this is near the top side of the tube, the local heat flux at this point is reduced by the azimuthal heat flux (figure 6, left side). If the wetting boundary is farther away from the top side of the tube, i.e. in the case of a small wetting angle, there is hardly any influence of heat conduction on the heat flux at the top side to be observed (figure 6, right side).

If heat redistribution over the tube circumference is taken into account in the calculation of the temperatures at the tube inner surface, lower values will be obtained than when a vertical tube model is used, figure 7. In all cases, the temperature of the steam was calculated from an energy balance, so that the comparison also accounts for the thermal non-equilibrium between the steam and water phase in the flow. It is apparent from the diagram that the redistribution of the heat cannot be the sole reason for the lower wall temperatures at the tube top side in the post-CHF

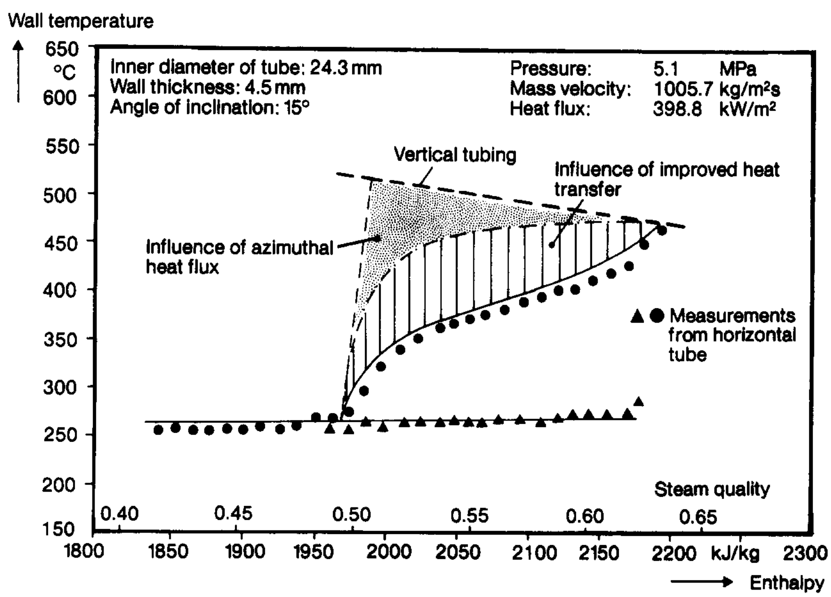


Figure 7. Prediction of tube wall temperatures accounting for azimuthal heat flux and improved heat transfer in the region of partially wetted circumference.

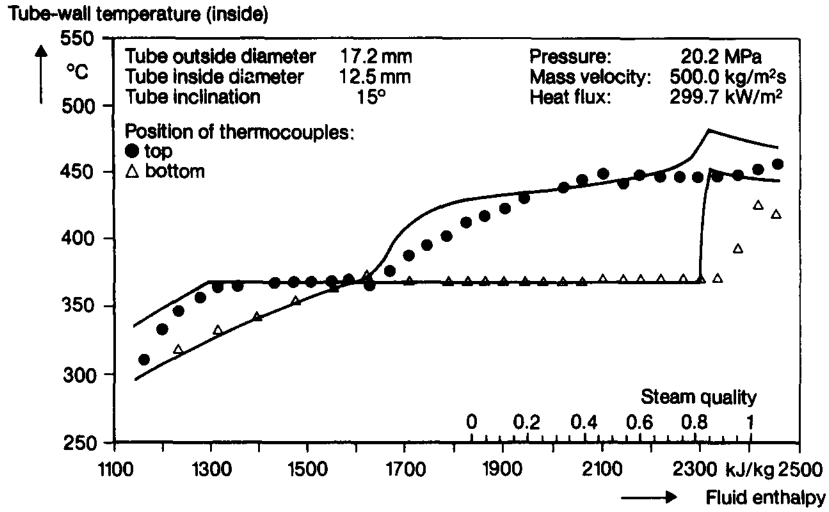


Figure 8. Prediction of wall temperatures in an evaporator tube with a computer code developed from the experiments.

regime. Calculations have shown that, even assuming saturation temperature for the steam phase, the heating surface temperatures predicted are still too high. Thus, the only possible explanation remaining is that of improved convective heat transfer between the tube wall and the fluid.

In the wetted surface region, heat is transferred from the tube wall to the water film, and steam is formed. This process is particularly pronounced in the vicinity of the boundary between the wetted and unwetted part of the tube wall since, because of redistribution, heat fluxes are particularly high in this zone. The steam thus generated enters into the core flow, causing both an increase in flow velocity and additional turbulence to the flow. Since the wetting angle in the tube represents a measure of steam production, and also describes the distance of the wetting boundary from the top side of the tube, the improvement in heat transfer is a function of this angle. Analysis of the experiments showed that the heat transfer coefficient at the tube top side at the beginning of the transition region is twice as high as in the post-CHF region of a vertical tube. This factor decreases continuously to a value of 1, the smaller the wetting angle becomes. If this finding is incorporated into calculations performed for comparison with measured tube wall temperatures, a good approximation is obtained, as shown by the corresponding curve in figure 7.

A computer program which enables us to calculate the wall temperatures to be expected in an evaporator tube of any given orientation has been developed on the basis of the theoretical fundamentals described and has been verified against the results of the experiments performed. The predictions of the measured tube wall temperatures resulted in a good approximation in all cases, figure 8. From this, the conclusion can be drawn that the assumptions made for the purpose of model development are confirmed as valid, at least for the parameter range investigated.

5. SUMMARY

For the design of the tubing in once-through and waste heat recovery boilers knowledge is required about the temperatures and temperature gradients to be expected in the evaporator tube walls during the boiler operation. Calculation models presented to date are based predominantly on findings obtained with vertical tubes and thus make no allowance for the effects of gravity. In practice, however, the tubing is frequently installed horizontally or inclined, so that the influence of gravity on the flow pattern and heat transfer has to be taken into account. In order to obtain information on this topic experimental studies were performed on tubes of different inclinations over a wide range of parameters.

The experiments showed that the boiling crisis in horizontal and inclined tubes can extend in the form of an ellipse over a certain tube length. The mean steam quality in the flow in this region corresponds to that which is to be expected at the boiling crisis in a vertical tube under the same conditions. The axial length of the transition region can be calculated with the aid of an extended Froude number.

The tube wall temperatures in partially wetted heating surface regions are generally lower than those which occur in a vertical tube at the same fluid enthalpy. The reasons being the azimuthal redistribution of the heat in the wall material, the lower thermal non-equilibrium and the improvement in heat transfer as a result of enhanced flow turbulence. If these findings are compiled into a computer program which is then used to predict measured tube wall temperatures, a good approximation is obtained over the entire parameter range investigated.

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