

THE INFLUENCE OF THE THERMAL DIFFUSIVITY OF THE CONDENSER MATERIAL ON THE HEAT TRANSFER COEFFICIENT IN DROPWISE CONDENSATION

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Abstract

Heat transfer measurements were performed during the condensation of pure saturated steam on five vertical condenser surfaces with copper, aluminium, brass, bronze and stainless steel as basic materials. The surfaces of the different condensers were gold-plated to guarantee the same dropwise condensation conditions on all materials used. The saturation temperature has been varied between 50°C and 90°C at heat-flux densities up to 550 kW/m². The result is that the heat transfer coefficient grows with ascending thermal diffusivity, independently of the saturation temperature and the heat-flux density. This effect results from the finite thermal conductivity of the condenser material in connection with the distribution of droplets of varying size over the surface that causes a temperature field in the surface, varying with the time. Thus, in an area near the surface of the condenser, heat must be transported parallel to the surface. This event has been calculated with a numerical model, using an experimental droplet function. The agreement of the numerical results with the experimental data is satisfactory.

Introduction

High magnification visual observations of the condensation process shows that there are drops of different size on the condenser surface with diameters varying from less than 1 µm to about 1500 µm. Viewing any single spot on the surface, the drop size will vary in a steady and unsteady way, caused by condensation on the drops and by coalescence. With the assumption that the heat of evaporation is released only at the surface of the drops /1/ and is transported through the drops mainly by conduction /2/, there results a heat-flux density through the drops varying with the drop size, caused by the heat transfer resistance of the drops. As a consequence the surface temperature of the condenser must also change with the size of drops on it, causing fluctuations of the surface temperature. Such fluctuations have been registered by Ohtani et al. /3/, who showed that they are considerable. In a layer near the surface the fluctuations of the surface temperature will be equalized by conduction parallel to the surface.

As the surface temperature at one place changes with time, there exists not only conduction but also energy

storage acting in the layer near the surface. So there must be an influence of the thermal diffusivity of the condenser material on the heat transfer coefficient in dropwise condensation, because this heat transfer coefficient is usually defined as the quotient between the mean heat-flux density and the mean temperature difference between the saturated vapor and the surface.

However, it is a point of issue up to now, whether the thermophysical properties are of secondary significance - hardly to verify in an experiment - or are important parameters. Some experiments result in the conclusion that there is almost no influence /4,5/, others recorded a strong dependence of the heat transfer coefficient on the properties /6,7,8/. At this point it should be recorded that all investigations of this effect only paid attention to the change of the thermal conductivity and did not use the thermal diffusivity for their correlations. This will cause no error in the interpretation as long as the measurements taken into account are all carried out on metal condensers, as the product of density times specific heat capacity is almost constant for all metals - and so thermal conductivity and thermal diffusivity differ by a constant factor. There might arise some difficulties if non-metallic condensers are used.

The discussed influence has been investigated theoretically by two authors. Hannemann and Mikic /9/ calculated the heat transfer coefficient in dependence of the thermal conductivity by using an experimentally verified drop distribution function and the model of the constriction resistance beneath drops, developed by Mikic /10/. Their results are in good agreement with some experiments /6,7,8/.

Sadhil and Plesset /11/ investigated the influence of the thermal conductivity of the condenser material on the heat-flux through a single drop, sitting on a semi-infinite solid. Though this model is a strong simplification of the dropwise condensation process, it corresponds in its tendency to the above mentioned experimental results /6,7,8/.

Experimental investigation

Because of the different experimental results, own measurements have been carried out. The test apparatus was a closed loop consisting of an evaporator, heated with a thermostat, a steam pipe, the test chamber, a pipe for the remainder steam, a condensate pipe, a secondary condenser for the remainder steam and a throttle, through which the condensate returned to the evaporator. The test chamber is presented in Fig.1.

It consists of the test condenser, the steam chamber and the cooler. The test condenser is radially insulated to secure one-dimensional conduction through the condenser. For further details of the apparatus and its preparation see Waas /12/. With this apparatus dropwise condensation experiments with degassed steam on vertical condensers have been carried out. The saturation pressure of the steam varied between 100 mbar and 740 mbar, the used condenser materials were copper (99,99%), aluminium, brass, bronze and stainless steel. All condenser surfaces were gold-plated (4.5 μm) to get the same surface conditions on all test condensers, especially the same contact angles.

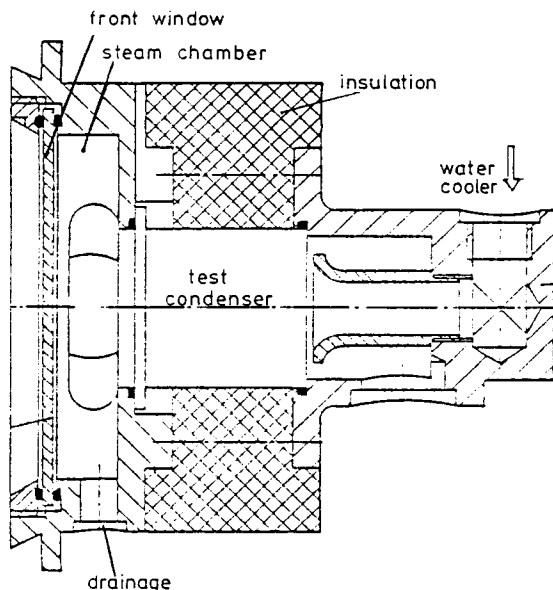


Fig. 1: Cross section of the test chamber

The heat transfer coefficient was calculated as the quotient of the mean heat-flux density and the mean temperature difference between the saturated steam and the condenser surface. The temperature of the saturated steam was measured directly with a thermocouple, the heat-flux density and the surface temperature were calculated by means of 2 to 4 temperature measurements within the condenser; the number of measurements depending on the length of the used condenser. The thermal conductivity of the condenser material, required for the calculation of the heat-flux density, have been measured in dependence of the temperature in a separate experiment.

In Fig.2 the results of the heat transfer measurements during dropwise condensation on copper are presented. Each point in this diagram represents the mean value of twelve measurements at steady state, with different thermocouples for each measurement to minimize the errors resulting from calibration and not exactly defined location of the thermo-couples /13/. The open symbols represent measurements with long condensers (35 mm), the half-filled symbols those with short condensers (25 mm).

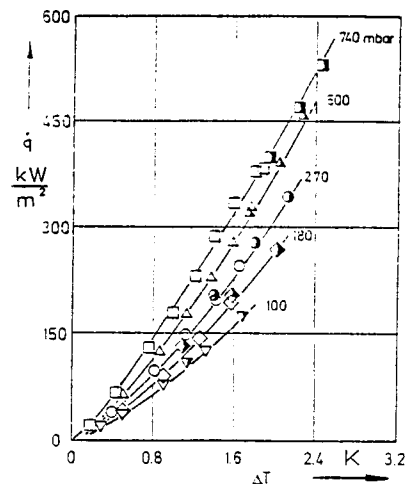


Fig. 2: Heat-flux density \dot{q} versus mean temperature difference ΔT during the condensation on copper with the saturation pressure as a parameter.

Fig. 2 points out that with increasing saturation pressure the slope of the lines increases, which means that the heat transfer coefficient $\alpha = \dot{q} / \Delta T$ increases while ΔT or \dot{q} remains constant. The second remarkable result is that the values at constant pressure are not lying on a straight line - as most authors concluded - but that the slope of the line at constant pressure increases with increasing subcooling of the condenser surface, which means that the heat transfer coefficient increases. These effects and their origin are presented and discussed in more detail in Waas /12/.

Similar results have been obtained for the other condenser materials /12/. With these data the heat transfer coefficient can be presented as a function of the thermal conductivity as it is done in Fig.3.

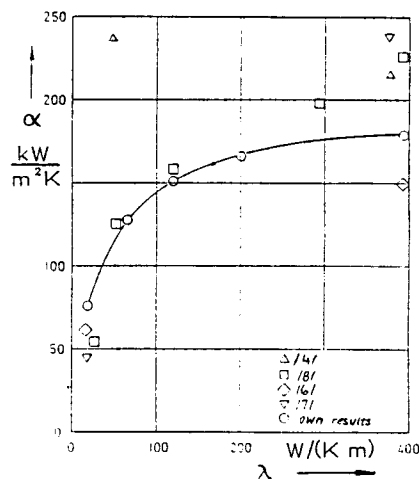


Fig. 3: Heat transfer coefficient versus thermal conductivity of the condenser material - a comparison of own results with those of other authors. ($\dot{q} = 0,1 \text{ MW/m}^2$, $p_S \approx 1 \text{ bar}$)

Own measurements in Fig.3 correspond to a saturation pressure of 0,74 bar and a heat-flux density of 0,1 MW/m². All the presented results of other authors have been obtained at 1 bar, the heat-flux density is of no interest, as the scatter of their results only allowed straight-line interpolation. The difference in the saturation pressure is of secondary significance, as in this pressure range the heat transfer coefficient is only a weak function of the pressure /12/. The diagram shows that there is a significant dependence of the heat transfer coefficient on the thermal conductivity, respectively on the thermal diffusivity; the quantitative results are in agreement with other authors. In the investigated pressure range, no systematic influence of the pressure on the above mentioned effect could be perceived. The contradictory results can be explained with the errors in the measurement - increasing with decreasing thermal conductivity /13/ - and with the use of condensers, which are rather thin, so that the temperature fluctuations are still present at the cooled side and almost no conduction parallel to the condenser surface takes place.

Theoretical investigations

Taking into consideration the heat of evaporation only, the resulting heat-flux can be calculated on the basis of the mass-flux of the condensate:

$$\dot{Q} = \dot{m} \Delta h = \dot{q} \dot{V} \Delta h \quad (1)$$

The volume flux \dot{V} can be described by the growth of single drops $\dot{R}(r)$ and by the dropsize distribution function $N(r)$. With these two functions the stationary heat-flux density in dropwise condensation can be calculated by the following equation:

$$\dot{q} = 2 \pi \rho \Delta h \int_{r_{\min}}^{r_{\max}} r^2 \dot{R}(r) N(r) dr \quad (2)$$

The dropwise distribution function is defined by

$$N(r) = \Delta n / \Delta r \quad (3)$$

with Δn being the number of drops per unit area with radii x with

$$r \leq x \leq r + \Delta r$$

and Δr the finite width of a radius interval.

Measurements of this drop-size distribution function with steam, condensing on vertical surfaces point out that it can be described in the observable region ($r \geq 10 \mu\text{m}$) by the following equation:

$$N(r) = C + D (r/r_b)^{-2.2} \quad (4)$$

with $C = 1.72 \text{ m}^{-3}$, $D = 0.49 \text{ m}^{-3}$ and $r_b = 0,8 \text{ mm}$

The growth of single drops can be calculated analytically under the assumption of simple boundary conditions /1/. The influence of the thermal conductivity of the condenser material on the heat transfer coefficient, taking into account the interaction of the drops, has been treated up to now by Hannemann

and Mikic /9/only. This model includes some simplifications concerning the energy transport through the drops, the effects of which are unknown.

Therefore, the following numerical model has been developed making it possible to calculate the growth of drops under the influence of neighbouring drops and modelling the heat transfer through the drops with a higher physical accuracy.

For the numerical calculation the complete radius interval of the drop-size distribution function from r_{\min} to r_{\max} is divided in m separate intervals. Then each interval i ($i = 1, 2, \dots, m$) contains a number of drops per unit area Δn_i which can be calculated by the aid of the drop-size distribution function.

The following assumptions have been made for the calculation of the heat transfer:

- The temperature field surrounding a drop with the radius r_i is controlled by drops of size r_j , with $j < i$ and $j = 1, 2, \dots, m$. The surrounding of the smallest drops ($i = 1$) is considered as adiabatic.
- The surrounding of each drop of one radius interval is influenced in the same manner by smaller drops, so that the calculation of the heat-flux through one representative drop of the interval is sufficient.
- The considered drop r_i is always fixed centric on the face of a cylinder with the radius r_{ki} that represents a section of the condenser surface. The lateral surface of the cylinder shall be adiabatic. The opposite face of the cylinder is kept at a constant temperature. The length of the cylinder is large, compared with the maximum drop radius r_{\max} ($r_{\max} \leq 0,1$).
- The drop is represented in the numerical calculation by a cylinder, the height of which equals its radius. At the free surface of the drop a heat transfer coefficient is present that is calculated by the aid of the kinetic theory /1/.
- The heat transfer through the drops is governed mainly by the stationary temperature field in the drops and in the condenser. Such unsteady effects must not be considered. Only conduction is taken into consideration as transport process.

The assumption a) includes that the area influencing the growth of the considered drop is small, as own calculations and those of Sadhal and Plesset /11/pointed out. In addition it is presumed that the drop-size distribution is randomly and that most time of the drop growth happens without the influence of a large drop in the neighbourhood. This assumption seems to be justified by viewing motion pictures of the phenomena.

Point b) is a consequence of the steady mean drop-size distribution function that is considered randomly in space.

The representation of a drop by a cylinder causes less than 5% error in the resulting heat-flux in comparison to a hemispherical drop but it simplifies the numerical calculation considerably.

The assumption e) reduces the influence of the thermal diffusivity on the heat transfer to that of the thermal conductivity. It seems to be justified as the change of the radius on the one hand during the growth of a drop, undisturbed by coalescence, is very small and so the event is quasi-steady. On the other hand, most coalescence happens between drops of greatly different size, so that the small one disappears -leaving free area for nucleation - and the larger one grows by a small amount that rarely influences its own growth and its temperature field. However, the final check of this assumption is possible only by a comparison between the experimental and the theoretical results.

Assumption a) allows the averaged heat transfer coefficient $\bar{\alpha}_{i-1}$ on the circle ring area around a drop with the parameter r_i to be calculated from the heat-flux that passes through drops of the size j with $j < i$. The basic model is represented in Fig. 4.

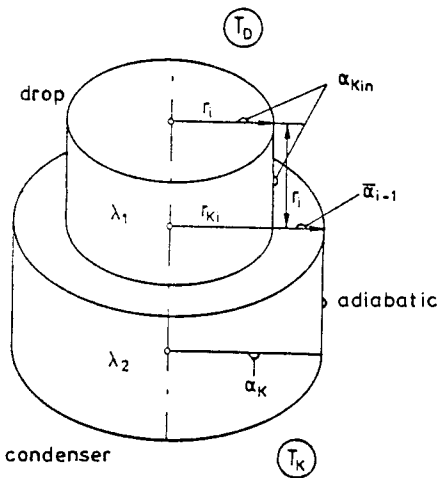


Fig. 4: Model for the calculation of the drop growth and the heat transfer coefficient with assumed drop size distribution function.

The ratio of the drop-radius and the condenser-radius can be calculated with the drop-size distribution function, which gives the quotient of the area covered by drops of the size i to the area covered by drops equal and smaller than r_i (including the uncovered area). The part of the area that is covered by drops is

$$S = \pi \sum_{i=1}^m \Delta n_i r_i^2 = \pi \int_{r_{min}}^{r_{max}} N(r) r^2 dr \quad (5)$$

The ratio of the radii is then given by

$$\frac{r_i}{r_{ki}} = \left[\frac{\pi N(r_i) r_i^2 \Delta r_i}{1 - s + \pi \int_{r_{min}}^{r_{max}} N(r) r^2 dr} \right]^{0.5} = \left[\frac{\Delta n_i r_i^2 \pi}{1 - s + \sum_{j=1}^i (\Delta n_j r_j^2 \pi)} \right]^{0.5} \quad (6)$$

The calculation starts with $i = 1$ and for each drop-class the mean heat transfer coefficient α_i is calculated so that with increasing indices of the interval the growth of the drops can be determined in dependence to the influence of the surrounding. One problem in the numerical treatment of this process is that none of each ratio of the radii can be reproduced as a quotient of integers, when the maximum integer is limited, as it is necessary in the numerical treatment. That is why the width of the intervals has been adapted to the numerically representable ratios of the radii. The numerical calculation yields the two-dimensional temperature field in the drop and the condenser for the different drop-sizes, the heat-flux through the drop and, thus, the growth-rate of the drops. After integrating over the whole drop-size distribution the mean heat-flux density can be evaluated, respectively the heat transfer coefficient in drop-wise condensation, formed with the mean temperature difference between the saturated vapor and the mean surface temperature.

The minimal radius r_{min} has been determined by the Kelvin-Helmholtz-Equation, the maximal radius r_{max} has been chosen to 0.8 mm, as it has been measured in the experiment. The condensation coefficient, necessary for the "kinetic heat transfer coefficient" on the free surface of the drops has been fixed equal to unity. In Fig. 5 the result of this calculation is presented in comparison with the experimental data.

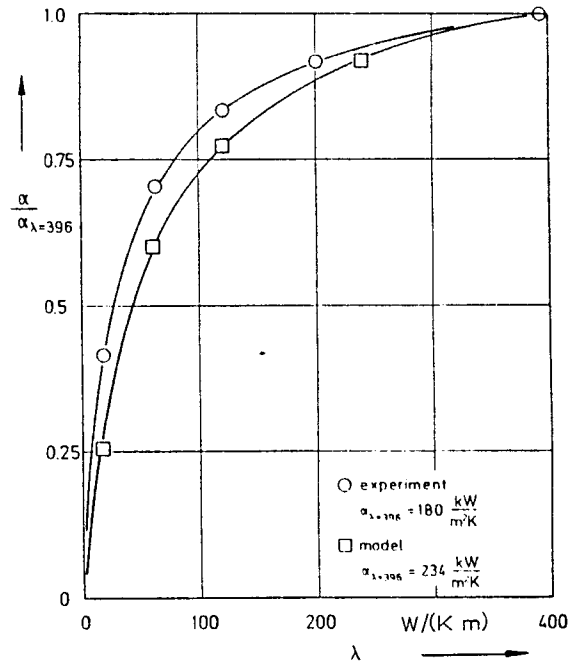


Fig. 5: Non-dimensionalized heat transfer coefficient versus thermal conductivity - a comparison between own measurements and the numerical model

The absolute values of the heat transfer coefficients in the experiment and the theory differ slightly from one another, yielding for copper $180 \text{ kW/m}^2\text{K}$ in the measurement and $234 \text{ kW/m}^2\text{K}$ in the numerical calculation. That is why the actual heat transfer coefficient divided by that of copper is plotted as a function of the thermal conductivity. The agreement between the experiment and the numerical model is satisfactory, in regard to the consideration that the drop-size distribution function is unknown for diameters below $10 \text{ }\mu\text{m}$, and the presumed function (4) probably will not be correct in the whole range.

Conclusion

The finite thermal properties of the condenser material in conjunction with the drop-size distribution, which causes inhomogenities in the surface heat-flux, yields an important influence on the heat transfer coefficient in dropwise condensation.

The satisfactory agreement between the experimental and the theoretical results - the latter obtained with a steady state model - leads to the assumption that the above mentioned influence is mainly caused by the thermal conductivity of the condenser material rather than by the thermal diffusivity. It should be kept in mind, however, that this result is verified up to now only with steam condensing on metal surfaces and that there might be a slightly different behaviour when using other vapors and other condenser materials.

Nomenclature

(in addition to the recommended one)

$N(r)$	dropsize distribution function
p_s	saturation pressure
r	drop radius
Δr	radius interval
$\dot{R}(r)$	growth function of drops
s	part of the surface covered by drops
ΔT	mean temperature difference
\dot{V}	volume flow rate

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