THEORETICAL AND EXPERIMENTAL STUDY ON DROPWISE CONDENSATION IN PLATE HEAT EXCHANGERS

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Abstract

This paper describes the outline of a simulation model for an oil-cooled flat-plate condenser and summarises the main results from its application. Mass and energy balances are solved for a specific Alfa Laval plate heat exchanger with three channels. Dropwise condensation on the corrugated plate surface is described using a modified model from the literature, and studied through high-speed imaging. Parameter variations show the most relevant parameters of the model. The influence of temperature difference, oil flow, pressure and some model parameters on the simulation results is analysed. The results of the simulations are compared to experimental data, showing that the model is useful over a certain range of parameters, but has to be improved for high heat fluxes to achieve a better fit to the experimental data.

1 Introduction

Dropwise condensation realises heat transfer coefficients which are about one order of magnitude higher than those for filmwise condensation (Koch, 1997). For a long time, however, no adequate surfaces for dropwise condensation of steam had been found which satisfied industrial requirements with respect to stability and cost. In the last decades new coating technologies have been developed, and today good prospects exist for the production of durable and affordable hydrophobic coatings (Rose, 2002). With these new technologies dropwise condensation may be implemented in industrial applications.

Proper design of industrial condensers requires the accurate knowledge of the underlying heat transfer coefficients. To fully exploit the potential of the enhanced heat transfer, the basis of such design calculations needs to be refined. This work outlines experiments and proposes a model that can form the basis of an adequate design procedure.

For performing the experimental tests for steam condensation on various coated plates, a heat exchanger test rig is set up at the Institute of Technical Thermodynamics of the German Aerospace Centre (DLR).

The dropwise condensation process with its highly unsteady nature, where around a million coalescences can occur in one second on a square centimetre of the condensing surface, can not be modelled in detail. The excellent agreement between theoretical results and heat-transfer measurements justifies the assumption of a steady heat transfer rate for a drop of given size and a steady distribution of drop sizes (Rose, 1976).

The suggested approach is based on the energy and mass balances for steam and coolant flow rate with the assumption of a constant drop size distribution and drop growth rate during a particular interval. For a better adaptation of the model to the conditions in industrial plate heat exchangers, and as an improvement over previous models, the effect of the shape of the condenser plate is considered.

2 Modeling and Simulation

Based on the mass and energy balances over the plate heat exchanger the calculation is refined by the determination of heat transfer for the condensation and for the cooling side depending on the local temperature conditions. Likewise, the heat conduction through the plate heat exchanger is considered. Particular emphasis is placed on the detailed consideration of the dropwise condensation process.
Steady state mass and energy balances for the condensing steam and for the coolant flow form a system of non-linear, ordinary differential equations. The temperature of the coolant (thermo-oil) flow and the steam quality are the dependant variables. The overall heat transfer coefficient *U* is calculated from the local condensation heat transfer coefficient, the coolant heat transfer coefficient and heat conduction through the coated plate, and therefore depends on the local temperatures and steam quality.
This model is compiled for a symmetric plate heat exchanger with only three channels; in the middle the condensing steam and on both sides the thermo-oil flows. The direction of the oil flow can be co-current or counter-current to the steam flow.

2.1 Model of the Plate Heat Exchanger

The present contribution proposes a model formulation for a plate heat exchanger operated with condensing steam and thermal oil in adjacent channels. The considered setup is depicted in Figure 1. The energy balances for oil and steam form a set of equations that can be solved numerically.
Equations (1) and (2) are the basic equations for the oil temperature and the steam quality. Mass flow 3 is the same as mass flow 1.

$$\frac{d\vartheta_1}{dz} = -\frac{dA_p}{\dot{m}_1 \cdot c_{p,1}} \cdot U_{1,2} \cdot (\vartheta_1 - \vartheta_s) \tag{1}$$

$$\frac{dx_S}{dz} = -\frac{2 \cdot dA_p}{\Delta h_V \cdot \dot{m}_2} \cdot U_{1,2} \cdot (\vartheta_s - \vartheta_1) = 0 \tag{2}$$

The heat transfer coefficient depends on the oil and steam temperatures, the steam quality and the oil flow, therefore it also depends on the variable *z*: $U_{1,2} = f\left(\vartheta_s, \vartheta_1, \dot{m}_1, x_S\right) = f(z)$.

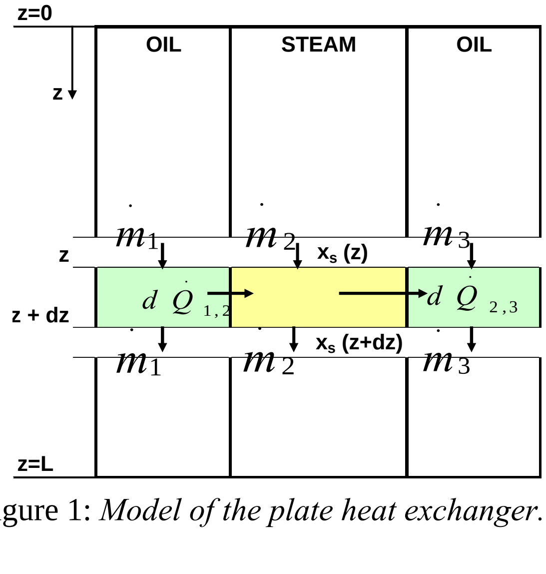

Figure 1: *Model of the plate heat exchanger.*

The following boundary conditions are used:

- Saturated steam at the entrance of channel 2: $x_S(z=0)=1$
- Same inlet temperature for both oil channels and same mass flow rate
- First and last plate adiabatic: $\dot{Q}_{0,1} = \dot{Q}_{3,4} = 0$

The overall heat transfer coefficient varies with the height z of the heat exchanger and is calculated from the condensation and the coolant (oil) heat transfer coefficients h_c and h_{oil} and the heat resistances of the plate and the coating, F_{pl} and F_c, as shown in equation (3).

$$\frac{1}{U} = \frac{1}{h_c} + F_{pl} + F_c + \frac{1}{h_{oil}} \tag{3}$$

With $F_{pl} = \frac{s_{pl}}{\lambda_{pl}} \qquad F_c = \frac{s_c}{\lambda_c}$ (4)

s_{pl} and s_c in equation (4) are the thickness of the condenser plate and of the coating and λ_{pl} and λ_c is the respective thermal conductivity of the plate and the coating.
Heat flux and heat transfer coefficient are connected as presented in equation (5).

$$\dot{q} = h_c \cdot \left(\vartheta_s - \vartheta_{wall,s}\right) = h_{oil} \cdot \left(\vartheta_{wall,oil} - \vartheta_{oil}\right) \tag{5}$$

The heat transfer coefficient of the thermal oil is calculated with an equation by Martin (2002). The calculation of the condensation heat transfer is described in detail in the following section.

2.2 Model for Dropwise Condensation

Various attempts have been made to estimate the heat transfer rate during dropwise condensation. Fatica and Katz (1949) were the first who proposed a model for the heat flux by assuming that all drops on a given area are of the same size and grow by condensation on their surface. Later, different researchers have dealt with the problem of drop size distribution, see for example Rose and Glicksman (1973), Tanaka (1975) and Wu and Maa (1976).
All these researchers developed their models on the basis of the mass and energy balances for a single drop with radius r. Integrating the velocity of the condensation process - condensate volume per unit time and unit area - over the radius interval from the minimum (emerging or nucleation) radius r_{min} to the maximum (sweeping) radius r_{max} (before it slides down) yields the following equation (assuming a contact angle of 90 degrees) for the heat flux

$$\dot{q} = \rho_c \cdot \Delta h_V \cdot \frac{\dot{V}}{A} = \rho_c \cdot \Delta h_V \cdot 2 \cdot \pi \cdot \int_{r_{\min}}^{r_{\max}} r^2 \cdot \dot{R}(r) \cdot N(r) \cdot dr \tag{6}$$

The drop growth rate $\dot{R}(r) = \frac{dr}{dt}$ depends on the drop radius, the steam pressure and therefore the condensation temperature as well as the wall subcooling $\Delta T = \vartheta_s - \vartheta_{wall}$. The drop size distribution $N(r) = \frac{n(r, r+\Delta r)}{\Delta r \cdot A}$ accounts for the number of drops of a specific radius r or radius range Δr.
Rose (1976) analysed various experimental data from different authors and found the following equations to describe drop growth rate and drop size distribution, see equations (7) and (8) below. Herein, $K_1 = 0{,}67$, $K_2 = 0{,}5$, $K_3 = 0{,}4$ are empirical constants.

$$\dot{R}(r) = \frac{\Delta T}{2 \cdot \rho_{fl} \cdot \Delta h_V} \cdot \frac{1 - \frac{r_{\min}}{r}}{K_1 \cdot \frac{r}{\lambda_{fl}} + K_2 \cdot \frac{T_s}{O} + F_{pl} + F_c} \tag{7}$$

$$N(r) = \frac{1}{3 \cdot \pi \cdot r_{\max}^3} \cdot \left(\frac{r}{r_{\max}} \right)^{-\frac{3}{8}} \tag{8}$$

The minimum and maximum radius r_{min} and r_{max} and the surface O are calculated from equation (9).

$$r_{\min} = \frac{2 \cdot \sigma \cdot v_f}{\Delta h_V \cdot \Delta T} \qquad r_{\max} = K_3 \cdot \left(\frac{\sigma}{g \cdot (\rho_{fl} - \rho_g)} \right)^{-0,5}$$

$$O = \rho_g \cdot \Delta h_V^2 \cdot \frac{\kappa + 1}{\kappa - 1} \cdot \left(\frac{2 \cdot \sigma}{R_G \cdot T_s} \right)^{0,5} \tag{9}$$

Previously published correlations assume a surface inclination of 90° with an idealized spherical drop shape. Considering a plate heat exchanger with corrugated channels, the influence of the shape of the surface must also be evaluated. Experimental data of Koch et al. (1997) are used to calculate the dependence of the heat transfer coefficient on the angle of inclination of the heat exchanger surface, see Figure 2. A polynomial equation is fitted to the experimental data.

$$\frac{h_c(\beta)}{h_c(90°)} = \text{-}1{,}43 \cdot 10^{-8} \cdot \beta^4 + 5{,}54 \cdot 10^{-6} \cdot \beta^3 \text{ - } 7{,}72 \cdot 10^{-4} \cdot \beta^2 + 4{,}59 \cdot 10^{-2} \cdot \beta + 1{,}26 \cdot 10^{-3} \tag{10}$$

2.3 Simulation Results

In the following, some results from the simulation of the plate condenser with counter-current oil flow are presented.

Figure 3 (left graph) shows the variation of the operational parameters temperature difference between steam and oil and saturation pressure that have also been varied in the experiments. With increasing saturation pressure the heat flux increases. Also with increasing the temperature difference higher heat fluxes can be reached.

To check the influence of the model parameters, the drop radii were also varied. The value of the minimum drop radius has almost no influence on the total heat flux. Only when it reaches the same order of magnitude as the maximum radius, it influences the calculated results. This can be clearly seen in Figure 3 (right diagram). The influence of the value of the maximum radius is much more significant; thus its estimation, and particularly the estimation of parameter K_3, have to be performed carefully in order to obtain good simulation results.

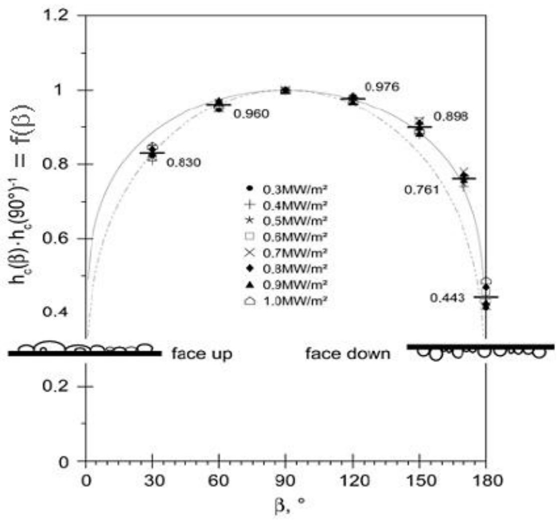

Figure 2: *Dependence of the heat transfer coefficient on the surface inclination (Source: Koch 1997).*

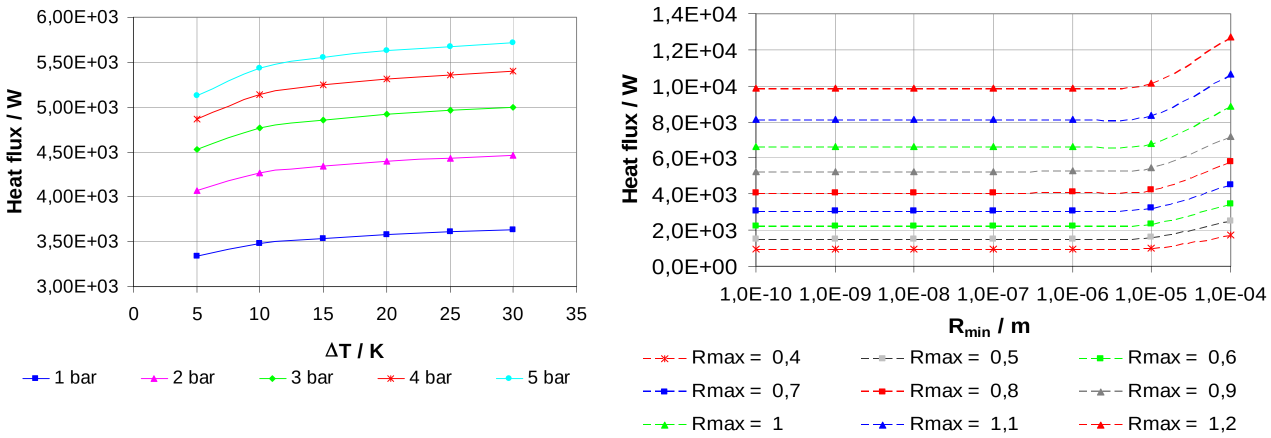

Figure 3:
Left: *Calculated heat flux over temperature difference with saturation pressure as parameter*
Right: *Calculated heat flux over minimum radius of the simulation model with the maximum radius as parameter.*

3 Experiments

Experiments with coated heat exchanger plates were performed to quantify the influence of the pressure and the wall subcooling on the performance. Plates with hydrophobic Ni-P-PFA coatings were tested at various pressures and different oil temperatures and volumetric flow rates, and were compared with uncoated plates. An Alfa Laval TS6 plate heat exchanger was used for the experiments. The test rig allows visual inspection of the condensation process (see Figure 4) and thus analysis of the relevant phenomena. A flow diagram of the experimental set-up is shown in Figure 5.
Visual observation of the drop lifecycle on coated plates showed excellent dropwise condensation behaviour. High-speed videos were recorded, showing the dependence of drop growth rate and drop size on the operating conditions. Overall heat transfer measurements show an increase in heat flux of up to 20 %, which means an enhancement of about 100 % with respect to the condensation heat transfer coefficient, see Figure 6.
Figure 7 shows both simulated and measured data for a steam pressure of 2 bar. For dropwise condensation, simulation results seem to fit the measured data very well. For lower oil temperatures (= higher temperature differences between oil and steam) the calculated data predict higher heat fluxes than the measurements. The same effect can be seen for higher steam pressures.

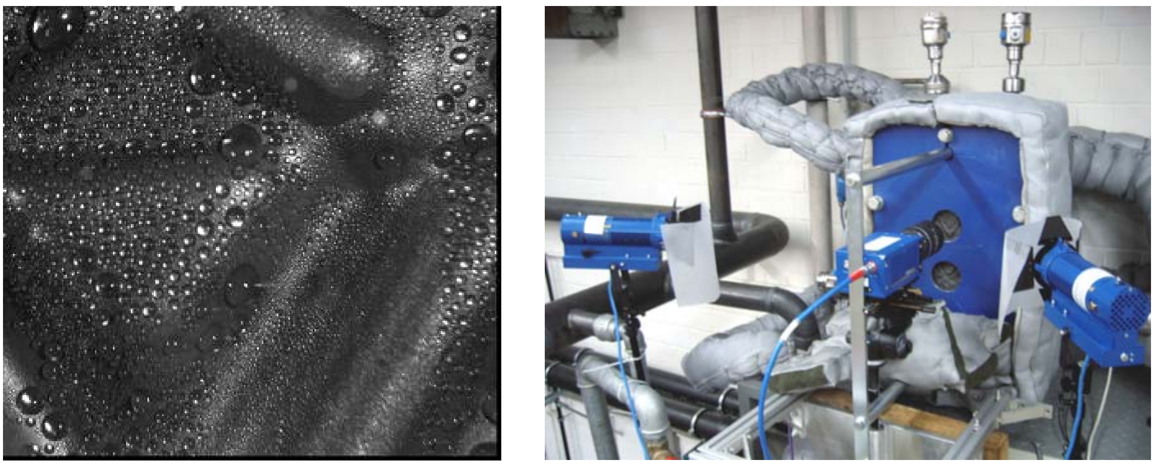

Figure 4: Left: *Dropwise condensation on a coated heat exchanger plate*
Right: *Test condenser with windows for visual observation and high-speed camera*

This allows the conclusion that an additional model parameter is to be considered. A possible explanation is that the condensate drainage may become a limiting factor with high condensation rates. The resulting decrease of unwetted area represents an additional heat transfer resistance, this in particular at the bottom part of the heat exchanger.
The measurements reveal that with dropwise condensation a substantial enhancement of heat flux is accomplished. However, the extent of this enhancement is limited by the significantly lower coolant heat transfer coefficient. For low temperature differences it causes a limitation of the wall subcooling and thus a limitation of the condensation heat transfer coefficient. Comparison of simulation and measured data also shows that the enhancement of the heat flux for dropwise condensation in a coated plate heat exchanger compared to filmwise condensation without turbulence, as is described in Nußelt theory, is well reproduced by the simulation model.

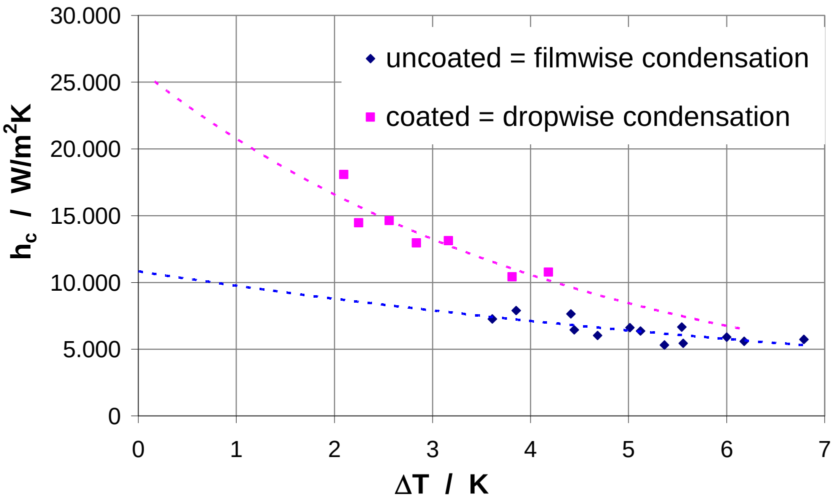

Figure 6: *Filmwise and dropwise condensation heat transfer coefficient from experimental data in the Ni-P-PFA-coated plate heat exchanger.*

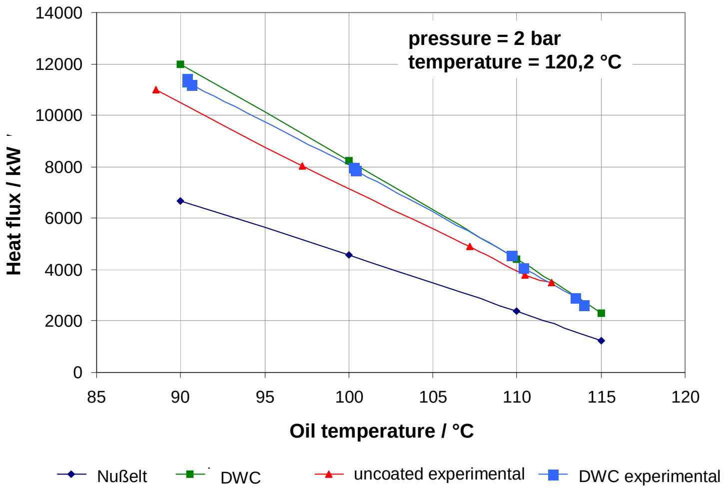

Figure 7: *Simulation and experimental results for heat flux with dropwise condensation on corrugated plates.*

4 Conclusions

A simulation model based on the mass and energy balances for a plate heat exchanger has been established. Comparisons with measured data reveal a reasonable accuracy of the calculations for both filmwise and dropwise condensation. A further improvement of the model is expected with extensions that account for another effect on the heat transfer, such as drainage limitations, the vapour shear or pressure drop effects.

The model was assessed in parameter variations, where the influence of temperature difference, oil flow, pressure and some model parameters on the condensation performance was analysed. Effects of the important operational parameters pressure and temperature difference on the heat flux have shown that the model is useful over a certain range of parameters, but has to be improved for high heat fluxes to achieve a better fit to the experimental data. Concerning the model parameters, the maximum condensate droplet radius (before it slides or falls down) has to be known better for good modeling results. For the nucleation radius only the order of magnitude is relevant.

It was shown that with a modern high contact angle surface, such as the Ni-P-PFA coating, considerably higher heat fluxes can be obtained while they stay in the dropwise condensation regime, compared to previous surfaces, which quickly transit to the filmwise condensation regime.

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