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Experimental investigation of force balance at vapour condensation on a cylindrical fin

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ABSTRACT

Experimental investigation of vapour condensation on single cylindrical fin has been carried out under various gravity conditions: $1g_0$, $1.8g_0$ and $0.05g_0$. For the first time, a balance of forces acting on the condensate flow is analyzed. The fin surface is divided in seven areas, each one being characterized by the main force acting on the liquid motion: gravitational vs surface-tension pressure gradient. The magnitude of the areas does not depend on the gravity level. This confirms the correctness of the fin surface segmentation, as used by various authors, to simplify the condensation modelling. It has been shown that the most intensive vapour condensation takes place in the areas, where the surface-tension pressure gradient is higher than the gravity force. Analysis of the heat load shows that each region affected by the surface-tension pressure gradient provides 10–15% of the total heat load, a contribution that is not negligible and should be taken into account. In case of low gravity, the most intensive condensation takes place at the corner of the fin tip, where the curvature gradient is maximum, and on its cylindrical part. The heat loads in these regions are comparable.

1. Introduction

The enhanced heat transfer tubes for vapour condensation is widely used, e.g. in refrigeration, air conditioning, petroleum and food industry, etc. A widely used intensification method is the finning of the tube surface. The enhancement is provided not only by increasing the contact area between the vapour and the colder surface but also by the action of surface tension forces, which help to move a condensate from the fin additionally to other forces (gravitational, shear stress, ...). If a liquid-vapour interface is curved, a pressure difference across the interface must be present to establish a mechanical balance. The balance is described by the curvature (κ_{tot}) of the liquid-vapour interface within the Young-Laplace equation:

$$P_l - P_v = \sigma \kappa_{tot} \tag{1}$$

Any curvature variation along the interface induces a pressure inhomogeneity within the liquid which leads to fluid redistribution. This pressure variation is known as the surface-tension pressure gradient (*STPG*). The fin shape influences the condensate film flowing along the fin surface. This leads to the arising of *STPG*, which effects on the film flow. The expression for the *STPG* can be obtained by differentiating the Eq. (1) with respect to the film surface arc length (s):

$$dP/ds = \sigma \cdot d\kappa_{tot}/ds \tag{2}$$

It means that *STPG* is presented in the condensate flowing along any fin, because each fin shape has corners and/or rounding. This fact reveals two main problems in the theoretical description of vapour condensation on the finned surface. First of all, the fin shape becomes one of the parameters in the modelling that complicates the simulation, considering the big variety of possible profiles. Secondly, there are some difficulties with the validation of existing models.

Many efforts can be found in the literature focusing on the investigation of condensation on the outside of horizontal finned tubes. Comprehensive overview can be found in the book of Rifert and Smirnov [1] and some review papers [2,3]. Three classes of prediction models exist based on various approaches: semi-empirical, numerical and surface segmentation. Semi-empirical models are based on the already obtained experimental data and on a conception of the condensate film behavior [4,5]. They provide a very good prediction only for fluids and fin shapes contained in the database used. Numerical models using Navier-Stokes equations together with the heat balance equation [6,7], not only predict the heat transfer but also the behavior of condensate film on

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Nomenclature

Symbols

Cp	specific heat
f	focal length
g	gravitational acceleration
g_0	normal gravitational acceleration, 9.8 m^2/s
G	mass flow
J	net mass flux
l	length
M	molar mass
P, p	pressure
q	heat flux
Q	heat removed per unit time
R	thermal resistance
R_0	universal gas constant, 8.3144598 J/(mol K)
s, S	arc length
T	temperature
T_{cp}	temperature of cold plate
T_{Ph}	temperature of fin-sample base
T_{tc}	air temperature
U	overall heat transfer coefficient
X	abscissa
Y	ordinate

Greek symbols

α	heat transfer coefficient
δ	thickness
Δ	difference
φ	angle of tangent direction
κ	curvature
λ	thermal conductivity
θ	turn angle of surface arc
ρ	density
σ	surface tension

Subscripts

as	axial symmetry
c	capillary
cond	condensation side
f	fin
fs	fin surface
hp	heat pipe
l	liquid
tot	total
sat	saturation
v	vapour
w	water

Acronyms

HTC	heat transfer coefficient
STPG	surface-tension pressure gradient
TEA	Thermo-Electric Assembly
TEM	Thermo-Electric Module

the finned surface. Unfortunately, they require a very detailed description of the fin profile and time consumption of the calculation is very high. A third class of models is based on the subdividing of the condensation surface into areas with different approaches of the heat transfer [8–10]. This way of modelling is very adaptive to any fin shape. However, the results are very sensitive to the segmentation algorithm. Such variety of different modelling ways and various calculation algorithms are the result of the lack of any reliable information about condensate flow on the fin. Furthermore, almost all models provide the same level of prediction uncertainty of the average heat transfer coefficient within 15–20%, which complicates their verification.

Mainly, experiments are carried out on long tubes (several meters) in order to make an "averaging" on fins and minimize the effect of fin imperfections on the condensation. Indeed, the same fin shape cannot be reproduced along the tube because of non-perfections of the manufacturing processes, of worn cutting tools used, etc. Therefore, a large number of fins is used in the experiments. Generally, the tube is cooled by water flowing inside. Temperatures of water at both inlet and outlet are measured. Heat removed from the tube per unit time Q is measured by the standard calorimetric method:

$$Q = Cp_w \cdot G_w \cdot \Delta T_w \quad (3)$$

Here Cp_w is water specific heat, G_w is mass flow of water and ΔT_w is temperature difference at both tube ends. Temperature and pressure of the vapour are maintained stable. An averaged heat transfer coefficient is evaluated using the Wilson plot method based on analysis of overall thermal resistance of the global heat transmission process [11–13]. The overall heat transfer resistance can be written in steady state as:

$$1/U = 1/\alpha_{cond} + R_{tube} + 1/\alpha_w \quad (4)$$

Here, U is the overall heat transfer coefficient, α_{cond} is heat transfer coefficient (HTC) on vapour side, R_{tube} is the tube thermal resistance, and α_w is HTC of the water flow. R_{tube} is inversely proportional to the tube thermal conductivity and, therefore, is constant (neglecting any temperature dependencies). Typically, α_{water} is varied by manipulating with the water flow and the overall HTC is measured in the experiments. Although Eqs. (3) and (4) are rather standard, they still allow emphasizing the inconsistency of literature data obtained with this method and are also used for the verification of the existing condensation models. Fig. 1 demonstrates schematically the condensation experiment with the tube having rectangular finning. Condensation HTC is evaluated from Eq. (4). Detailed description of the experimental procedures and data reduction algorithms can be found in the papers of Fernandez-Seara et al. [14] and Ji et al. [15]. Such type of experiments provides an accuracy of about 15–25% of the average heat transfer coefficient measurement. This makes the obtained data to be very useful in engineering. However, the provided accuracy is not enough to validate models which have similar uncertainty of prediction.

The local data about the condensate behavior on the fin and the liquid flow between fins are very important for improvement of existing models and validation of the approaches made. Some scientific teams tried to use the original concepts of experiments to obtain any information about condensation on the fins [16–18]. Hirasawa with colleagues [16] has performed experimental simulations on collecting condensate between two rectangular fins. In their geometry, the liquid film was flowing along a vertical open channel. The channel width was decreasing from top to bottom. There, the flow dynamics of the liquid flow from the fin to the space between the fins has been observed and described. The local thinning in the area between the thin film and the meniscus has been observed. It has been shown in [16] that the thinning of the liquid film on the fins produces a local acceleration of the