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Increased Cooling Power with Nucleate Boiling Flow in Automotive Engine Applications

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1. Introduction

In modern automotive design, the increasing output of power delivered by ever more compact, i.e., less space consuming, engines requires a highly efficient liquid cooling system to keep the wall temperatures of the individual components on acceptable levels. A maximum possible cooling power is particularly needed on surfaces exposed to high thermal loads typically located in the periphery of the exhaust gas outlets in the cylinder head. Searching for a most intense transfer of heat across very confined surface areas, the modern layout of liquid cooling systems must make ever more use of the potential of the nucleate boiling regime to provide significantly enhanced heat transfer rates at acceptable wall superheats. It has been long known in the community of engine developers that nucleate boiling locally occurs in cooling jackets near the exhaust valves, especially when the engine is run under thermally critical operating conditions. These are typically reached when the vehicle is brought to a halt after operation at maximum engine load, so that a considerable amount of heat has to be removed with the help of the fan cooling on the air side only. Nonetheless, the traditional design concepts are essentially based on purely convective single-phase cooling, although under real maximum load conditions the peak temperatures would markedly exceed the allowed limits, if boiling did not occur. In view of this fact it appears very reasonable to make a more deliberate use of the boiling phenomenon for further optimization of the cooling system. The prominent goal of this novel concept is to provide a controlled transition from single-phase convection to the so called subcooled boiling flow regime, as soon as the local heat loads reach a certain level. The targeted regime of subcooled boiling flow offers the attractive feature that vapour is present only in a thin superheated near-wall layer, while the outer bulk flow region, where the bubbles collapse due to the subcooling, contains only liquid phase. Hence, there is basically no net production of vapour, and the cooling system practically remains in the single-phase regime. Going otherwise further beyond into the saturated boiling regime, associated with non-zero net production of vapour, the advection of a significant amount of vapour with the bulk flow, or a possible agglomeration of vapour in stagnant regions could be hazardous to a safe and stable operation of the liquid cooling system.

Relying on the incipience of nucleate boiling in the thermal layout requires appropriate wall heat flux models which cover the range from single-phase convection to the two-phase

boiling flow with acceptable accuracy. A realistic, physically sound model description of nucleate boiling flow is still challenged by very fundamental difficulties, as this highly complex phenomenon involves many sub-processes which are not fully understood yet. In particular, considering flow boiling on real life technical surfaces, it is basically impossible to describe mathematically the sub-process of bubble nucleation in a deterministic way. The complexity of the underlying contact physics between the gaseous, liquid and solid phases has thus far impeded any generally applicable parameterization of this sub-process. Hence, today's available model correlations bear a good deal of uncertainty due to a basically incomplete parameterization of the problem. As a result, the existing model approaches involve many tuning coefficients, which have to be appropriately adjusted from case to case. A survey of the basic model concepts proposed for subcooled boiling flow will be presented in this chapter.

To a great extent, the development of enhanced heat transfer concepts based on flow boiling was strongly driven by the need of compact high efficiency heat exchangers for use in spacecraft devices as well as in the field of chip cooling. Concerning the former application, much research work was focused on the effect of gravity, as buoyancy forces play an important role in the dynamics of the bubbles (Klausner et al., 2003; Kim et al., 2005; Bower & Klausner, 2006). The latter application has been mostly investigated in the context of "boiling in microchannels" (Kew & Cornwell 1997; Kandlikar, 2002; Thome, 2004; Cheng et al., 2007). This distinction has been made due to the small dimensions of the electronic devices, and the occurrence of the confined boiling regime, where the bubble diameters are of the size as the channel height, leading in general to higher heat transfer rates in comparison to the unconfined case.

Concerning the automotive engine application, several studies have been carried out investigating the convective boiling of typical engine coolants (Campbell et al., 1995; Kandlikar, 1998a; Kobor, 2003). Besides demonstrating the possible gain in cooling power, these investigations were strongly focused on the development, evaluation and further improvement of model correlations for the highly complex flow boiling heat transfer. In comparison to single-phase forced convection, the nucleate boiling heat transfer depends on many more parameters, which are often hard to determine, or whose relevance is not clear. Facing this fundamental difficulty, a safe and reliable thermal layout requires a most comprehensive knowledge on how specific operating conditions and system parameters typically found in engine cooling jackets can affect the subcooled boiling heat transfer. In particular, the actual state of the heated surface, be it characterized by a micro-scale surface topology, roughness or contamination, or by a macroscopic property like the orientation of the surface relative to the gravitational force, may have a significant influence on the boiling behaviour. Since automotive coolants are always multi-component mixtures, the effect of the composition of the working liquid has to be considered as well. Operating modes and operation time may also be important for the number of available active bubble nucleation sites (mostly surface imperfections like cavities or fissures). A selection of all these automotive engine relevant conditions and their potential relevance for the heat flux modelling shall be addressed and discussed in this chapter.

During the last decade, considerable boiling research was devoted to the development and investigation of so-called enhanced surfaces. Applying special, mostly porous, surface coatings can notably reduce the wall superheat required for the onset of nucleate boiling, and, beyond this point, it may intensify the process of bubble nucleation resulting in markedly enhanced boiling heat transfer rates. The prospective additional gain in total wall

heat flux makes this concept certainly attractive for a future application in automotive cooling as well. Some recent investigations on this specific approach will be addressed in this chapter as well.

2. Fundamental aspects of boiling flow heat transfer

Boiling flow represents one of the most challenging two-phase flow problems which to date could not be rigorously described based on physical principles. It involves many sub-processes associated with the nucleation, growth, detachment, coalescence, and collapse of vapour bubbles. These individual sub-processes can act on very different scales in time and space, and many of them are inaccessible to detailed experiments. Some basic mechanisms contributing to the total heat transfer can be still distinguished, which, despite their often unknown quantitative relevance, constitute at least a reasonable qualitative description of the whole phenomenon.

2.1 Basic mechanisms

The heat flux of boiling flow depends on numerous thermo-physical interactions between the liquid, the vapour phase, and the solid heater, as schematically shown in Fig. 1. The interaction between the liquid bulk flow and the non-boiling part of the heated surface, denoted by *1*, represents basically the convective single-phase heat transfer, generally termed macroconvection. The interaction between the bulk flow and the vapor bubbles, denoted by *2*, comprises the hydrodynamic forces of the flow acting on the bubbles and vice versa. These flow induced forces can strongly influence the bubble growth, the bubble detachment from the nucleation sites, as well as the paths of the rising bubbles. Moreover, a densely populated heated surface may change the near-wall flow conditions significantly. The bubble-liquid phase interaction, denoted by 3, is often subsumed as the so-called microconvection. It includes the entrainment of liquid into the wake of a rising bubble immediately after detachment, which mixes portions of cool liquid from the - mostly subcooled - outer flow region into the superheated wall layer. This bubble lift-off induced transport of cool bulk liquid towards the heated surface increases instantaneously the local convective heat transfer. At the same time, it cools down the area around the nucleation sites below the critical temperature for nucleation. This cooling down of the nucleation sites immediately after bubble lift-off is generally termed “surface quenching”. A quenched nucleation site needs some time to be re-heated beyond a critical temperature, where bubble nucleation is initiated again. The required heat is supplied by the solid heater via conductive heat transfer, denoted by *4*.

Aside from the hydrodynamical effect of the bubble motion on the liquid, the bubble-liquid interaction also includes the thermal effect of condensation at the vapour-liquid interface. This process occurs at the top of the bubbles, once the interface reaches a zone where the local temperature is below the saturation level. The evaporative counterpart of this mechanism of latent heat transport is realized by the interaction between the bubble foot and the heated surface, denoted by *5*. While the bubble is growing on its nucleation site, the latent heat needed for evaporation is continuously removed from the solid heater material, which causes a local drop of the temperature on the heated surface. The local cooling due to the removal of heat of evaporation during bubble growth, as well as the local cooling associated with the surface quenching after bubble detachment, are the main reasons why the surface temperature is highly non-uniform in time and space. This non-uniformity of the

surface temperature also leads to considerable thermal interactions (conductive heat exchange) between the individual nucleation sites inside the solid heater, denoted by *6*. It is noted that most of the boiling models used today do not account for these spatial and temporal fluctuations in the surface temperature, assuming in general a stationary and uniform wall temperature. For denser bubble populations occurring at higher wall superheats, the bubble-bubble interactions, denoted by *7*, such as coalescence or formation of columns, can play an important role. It is conceivable that this type of interaction makes simple single bubble considerations, on which many boiling models are based, highly questionable. The complexity shown in Fig. 1 does still not give the complete picture. Contact surface related aspects associated with surface tension, wettability, surface roughness, and porosity are known to have an important influence on the number of active nucleation sites needed for the formation of vapour bubbles. The so-called surface aging, which can be caused by chemical depositions on the heater surface leading to a long term deactivation of nucleation sites, also falls into this group. It is especially these surface related properties which impede a rigorous parameterization. Despite the numerous and physically complex effects which can be relevant in nucleate boiling, it is generally agreed that the marked increase in the heat transfer rate relative to the single-phase convection essentially results from two basic mechanisms:

- the microconvection induced by the motion of the bubbles, and
- the latent heat transport to and from the bubbles.

However, in many cases no consensus has been reached on the quantitative contribution of each mechanism to the total heat flux.

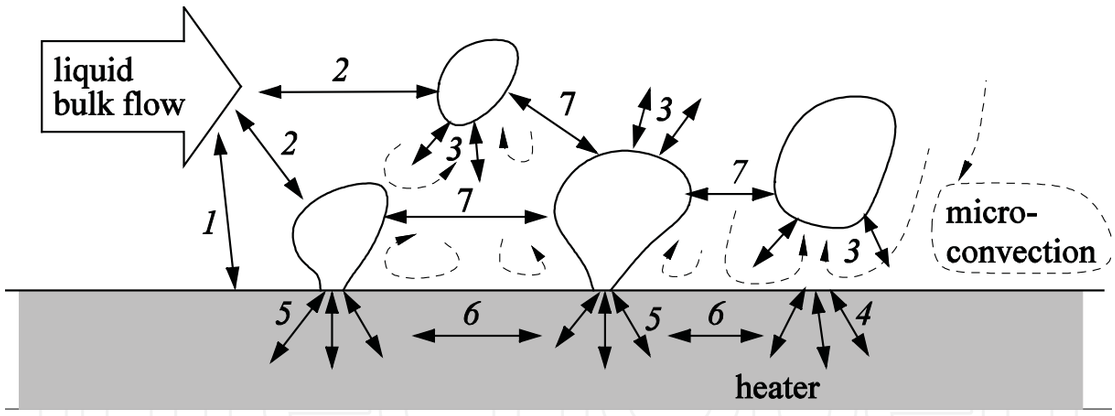


Fig. 1. Interactions between liquid phase, the vapour bubbles and the solid heater.

2.2 Onset of nucleate boiling and the subcooled boiling flow regime

A typical subcooled boiling flow configuration is schematically shown in Fig. 2a. The regime is basically characterized by a superheated wall with $T_w > T_s$, and a bulk temperature, which is lower than the saturation temperature, $T_b < T_s$. The formation of vapour bubbles starts, once the heated wall has reached the onset of nucleate boiling (ONB) temperature, $T_w = T_{ONB}$ at point *B*. Fig. 2b shows three boiling curves depicting the wall heat fluxes q_w versus the wall temperatures T_w for three different bulk velocities, with zero velocity - representing the pool boiling case - being the lowest. Two subranges can be distinguished in the subcooled boiling region: in the partially developed boiling (PDB) regime at lower wall superheats, the

bulk flow rate affects the heat transfer significantly. So the onset of nucleate boiling is shifted to higher wall superheats for higher flow rates *(B→B')*, as shown in Fig. 2b. In the fully developed boiling (FDB) regime at higher wall superheats, the boiling curves converge towards the pool boiling curve, which indicates that the flow becomes insignificant, once the wall superheat is sufficiently high. A further increase of the wall heat flux can heat up the bulk liquid to finally reach the saturation temperature, such that $T_b = T_s$. This denotes the transition from the subcooled to saturated flow boiling, where the bulk flow contains a significant fraction of vapour, as there is no subcooled region, where the bubbles could collapse.

The onset of nucleate boiling is a crucial point for the heat transfer characteristics, as it denotes the incipience of the formation of vapour bubbles initiating the desired substantial increase in the heat transfer rate. In the considered case of surface boiling, the ONB is associated with the activation of nucleation sites on the heated wall. Potential nucleation sites are mostly surface imperfections like cavities, fissures, or impurities.

Hsu (1962) was the first to develop an analytical condition for the onset of nucleate boiling at a given cavity. Hsu considered strongly idealized conditions shown in Fig. 3, assuming a cone-shaped cavity with an exact circular mouth of radius r_c, a quiescent liquid, whose temperature T_l decreases linearly with the normal distance to the wall y, and a bubble of spherical shape. Applying such simplifications, Hsu determined the temperature at ONB

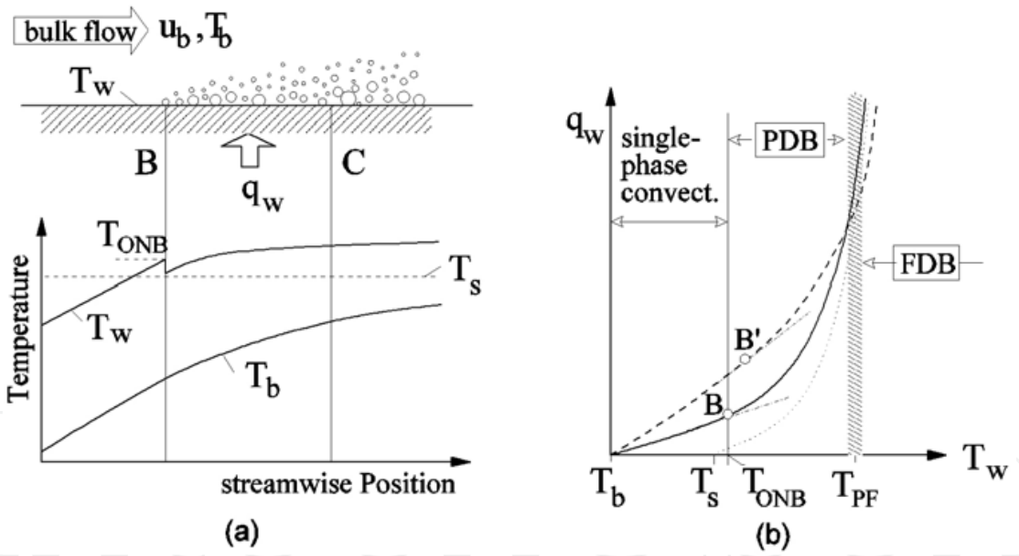


Fig. 2. Subcooled boiling flow: (a) variation of bulk and wall temperatures with the streamwise position; (b) boiling curves for different velocities of the bulk liquid; the dashed line, ---, denotes the highest velocity case, the dotted line, ·····, denotes the pool boiling case.

from the minimum wall superheat $\Delta T_{sat,min}$ required for the activation of a cavity of a given mouth radius r_c. This superheat at ONB is obtained as

$$T_{ONB} - T_s = \Delta T_{sat,min} = \frac{4\, T_s \sigma \sin\beta}{\rho_g h_{lg} r_c}, \tag{1}$$

and the corresponding heat flux reads

$$q_{ONB} = \frac{\lambda_l \rho_g h_{lg} \left(T_{ONB} - T_s\right)^2}{8 T_s \sigma \left(1 + \cos\beta\right)} \tag{2}$$

with λ_l being the thermal conductivity of the liquid, ρ_g the vapour density, h_{lg} the latent heat of evaporation, σ the surface tension, and β the liquid/vapour contact angle.

Hsu's concept, which is strongly idealized concerning the geometry of the cavity and the conditions of the liquid, remained the baseline also of later advancements in the modelling of the ONB, although it ignores completely the flooding of cavities. This occurs especially with well wetting liquids, where Hsu's criterion (1) significantly underpredicts the minimum superheat required for ONB. Basu (2002) accounted for the wettability of the surface by reducing r_c to an *effectively* available cavity size $r_{c,eff} = r_c \Phi$. The correction factor Φ is computed as an empirical function of the contact angle β. Φ decreases from unity to zero for increasing wettability associated with $\beta \rightarrow 0$.

Following Hsu's strongly simplifying analytical approach to model the bubble nucleation is basically not feasible for real technical surfaces, where it appears impossible to consider all relevant, mostly microscale effects like the microscopic surface topology, dynamics at the contact line between liquid, solid and vapour phase, corrosive abrasion or deposition, the instantaneous variation of the temperature inside the solid heater, or the content of dissolved gases. Many of these effects are to date not accessible by detailed experimental or numerical investigations. Therefore, in most approaches the ONB is not modelled explicitly, as this does not notably improve the overall accuracy of the predictions.

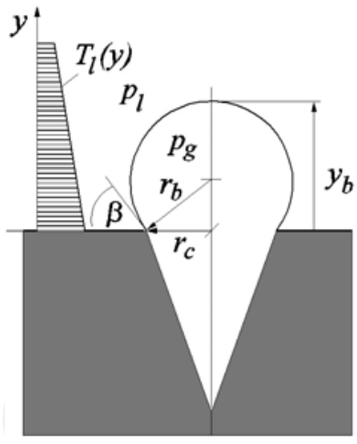


Fig. 3. Bubble nucleus at the onset of nucleate boiling (ONB) on an idealized cavity.

2.3 Models for the wall heat flux

In pace with the increasing computational resources it has become possible to simulate the conditions inside the very thin liquid microlayer at the wall, on which the growing bubble basically resides (Dhir et al., 2005). However, due to the high computational costs such detailed simulations are still restricted to configurations with very few bubbles on ideal surfaces. Therefore, the basically model-free direct simulation is way off the flow boiling on real technical surfaces (Mei et al., 1995a,b; Shin et al.,2005).

Some researchers even abandon the deterministic approach of mechanistic modelling. They consider boiling rather as a chaotic process (Shoji, 2004). This approach was proven as useful

to describe the chaotic fluctuations of the surface temperature associated with the non-linear interaction of bubble nucleation on neighbouring sites (Mosdorf & Shoji, 2004). As such the concept of boiling chaos gave important insight into individual sub-processes, but it did not yet deliver a full predictive wall heat flux model.

A great variety of wall heat flux models has been developed for use in engineering applications, and basically all of them adhere to the deterministic approach. The proposed models can be broadly grouped into two categories:

- *General empirical correlations*, which describe the wall heat transfer rates mostly as general power functions of non-dimensional groups.
- *Mechanistic models*, which attempt to capture the basic relevant mechanisms for the total heat flux, i.e., the hydrodynamic convective transport and the thermal heat transport associated with evaporation.

While the first concept relies completely on experimental data in deriving the non-dimensional model correlations, the latter is more analytical in that it accounts explicitly for the different physical mechanisms contributing to the total heat flux. Therefore, the mechanistic models in general perform better in transition regimes, where the relative contribution of the individual mechanisms may change substantially. Despite their more physical basis, the mechanistic models still involve a good deal of empiricism in the sub-models for the individual mechanisms.

2.3.1 General empirical correlations

In the FDB region, most empirical correlations model the total heat flux as a power function of the wall superheat generally written as

$$q_w = K\left(T_w - T_s\right)^m = K\Delta T_{sat}^m \tag{3}$$

where K and m are empirically determined model parameters. McAdams (1949) proposed this ansatz for water with K=4.77 and m=3.86. The so called Boiling number based model of Shah (1977)

$$q_w = \alpha_{fc} 230 Bo^{0.5} \Delta T_{sat} \tag{4}$$

falls also into this group. It basically models the total heat flux as a forced single-phase convective transport associated with the heat transfer coefficient α_{fc}, which is enhanced by boiling. The enhancement is represented by the Boiling number

$$Bo = \frac{q_w / \rho_l h_{lg}}{u_b} \tag{5}$$

which relates the "boiling velocity" given in the nominator to the velocity of the bulk flow. Kandlikar (1998b) modified Shah's correlation to

$$q_w = \alpha_{fc} 1058\ F_{fl}\, Bo^{0.7} \Delta T_{sat} \tag{6}$$

introducing a fluid-surface parameter F_{fl} representing the surface fraction covered by the liquid phase. The Boiling number based models are basically designed for the FDB regime, where they perform fairly well. On the other hand, they are quite inaccurate in the PDB

regime close to the ONB, where, according to Eqs. (4) and (6), the total wall heat flux vanishes as the superheat goes to zero.

2.3.2 Mechanistic models

The basic strategy of mechanistic models is to identify and to model the essential physical mechanisms which contribute significantly to the total wall heat flux. They mostly assume the heat flux to be composed of three components, namely, the single-phase convection q_{fc}, the evaporation heat flux needed for the phase change q_{ev}, and the sensible heating of the portion of liquid which fills the volume vacated by a departing or collapsing bubble q_{qu}. The total wall heat flux can then be written as

$$q_w = q_{fc} + q_{ev} + q_{qu}. \tag{7}$$

The forced convection component is commonly obtained using simply the classical textbook correlation by Dittus-Boelter, or more sophisticated correlations (Gnielinski, 1976).

The modeling of the evaporation heat flux q_{ev} faces the problem that it requires the number density of active nucleation sites, whose determination brings in again a great deal of empiricism. The same applies to the quenching component q_{qu}, regarding the determination of the quenching period, which depends on the time needed for the transient heat-up of the liquid filling the void of a departed bubble. Due to these difficulties in the modeling of the individual components associated with the phase change, most mechanistic models do not account for these fluxes separately. They rather combine them to one single ``pool boiling'', or ``nucleate boiling'' term, q_{nb}, such that Eq. (7) becomes

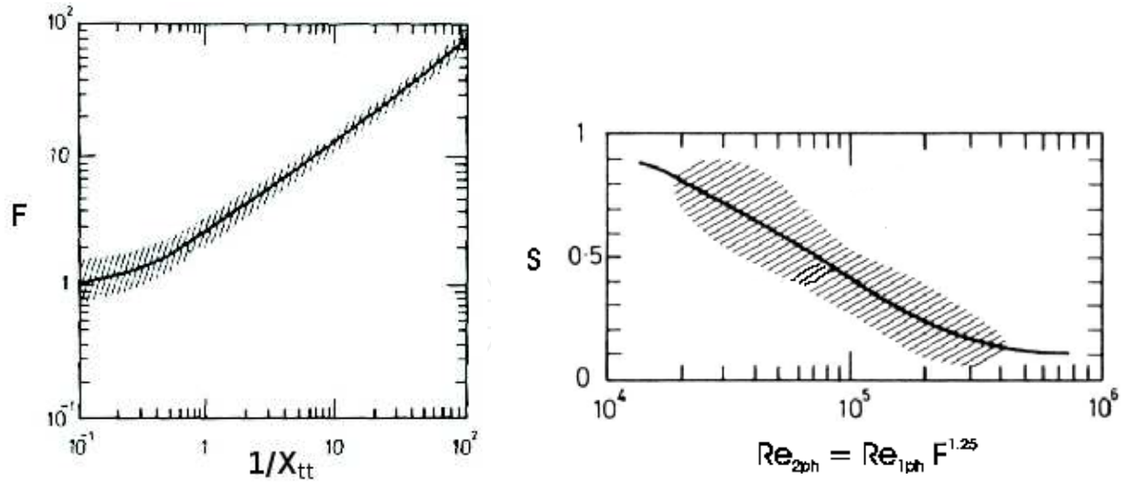
$$q_w = q_{fc} + q_{nb}. \tag{8}$$

A large number of models of this category have been proposed. In contrast to the Boiling number based methods discussed above, the explicit distinction between a convective and a nucleate boiling contribution brings about much freedom in selecting an appropriate model for each component. It also provides by definition the right asymptotic behaviour when approaching the limits of single-phase convection, where the nucleate boiling composition becomes zero, $q_{nb} \to 0$, as well as the limit of pool boiling, where the forced convection contribution goes to zero, $q_{fc} \to 0$. The latter limit is particularly troublesome for the Boiling number based empirical correlations, because, as seen from its definition in (5), the Boiling number goes to infinity for vanishing bulk velocity $u_b \to 0$, which makes this type of models incapable to provide a smooth transition between the flow boiling and the pool boiling regimes.

A pioneering superposition model, whose underlying concept is still widely used, was proposed by Chen (1966). Chen defined the total heat transfer coefficient as composed of a ``macroconvection'' coefficient of the two-phase flow and a ``microconvection'' coefficient associated with nucleate boiling, written as

$$\alpha = \alpha_{mac} + \alpha_{mic} = F\,\alpha_{fc} + S\,\alpha_{nb} \tag{9}$$

The heat transfer coefficient for the macroconvection is obtained from the Dittus-Boelter correlation using an increased two-phase flow Reynolds number Re_{2ph} incorporated through the factor F. As such, the factor $F = (Re_{2ph}/Re_{1ph})^{0.8}$ accounts for the enhanced convective heat



(a) (b)

Fig. 4. Best fit curves for correction factors in Chen model: (a) convection enhancement factor; (b) boiling suppression factor. The shaded regions denote the ranges of experimental data (reprinted from Chen (1966) with permission of ACS).

transport caused by the vapour bubble agitation. It was graphically obtained by Chen as dependent on the Martinelli parameter

$$X_{tt} = \left(\frac{1-x}{x}\right)^{0.9} \left(\frac{\rho_g}{\rho_l}\right)^{0.5} \left(\frac{\eta_l}{\eta_g}\right)^{0.1} \tag{10}$$

based on a best fit to experimental data as shown in Fig. 4a. Aside from the liquid-vapour phase density and viscosity ratios, X_{tt} essentially depends on the vapour mass fraction x. For small vapour fractions x associated with $1/X_{tt} \leq 0.1$, the factor F is assumed to be unity.

The microconvection coefficient is obtained from the pool boiling correlation by Forster and Zuber (1955). Chen further modified this contribution introducing a suppression factor S in order to reflect the generally observed decrease in the nucleate boiling activity with increasing flow velocity. Chen graphically correlated the flow induced suppression factor S as a best-fit curve to experimental data dependent on the two-phase flow Reynolds number Re_{2ph}, as shown in Fig. 4b. In a later work, Butterworth (1979) provided an analytical best-fitting dependence for the parameter S written as

$$S = \frac{1}{1 + 2.53 \cdot 10^{-6} \, \mathrm{Re}_{2ph}^{1.17}}. \tag{11}$$

Chen's concept, which was originally proposed for saturated boiling flow of water, has become a well established, very popular approach. It has been modified and developed further by many other authors (Gungor and Winterton, 1986; Campbell et al., 1995; Kobor, 2003; Steiner et al. 2005). These generally termed Chen-type models mainly differ in the sub-model for the nucleate boiling component and in the determination of the boiling suppression factor S.

Chen's linear superposition concept was also generalized to a non-linear combination

$$\alpha = \left[\left(\alpha_{fc} F \right)^n + \left(\alpha_{nb} S \right)^n \right]^{1/n} \tag{12}$$

as suggested by Kutateladze (1963), Liu & Winterton (1991) with n=2, and by Steiner & Taborek (1992) with a general exponent n. For increasing values of n, the power-additive formulation inherently gives more weight to the nucleate boiling, where $\alpha_{nb} >> \alpha_{fc}$, which nicely reflects the vanishing influence of the liquid flow on the total heat flux in the FDB regime. At low superheats, the formulation pronounces the convective heat transfer, where $\alpha_{fc} >> \alpha_{nb}$, which basically has the effect of a flow induced suppression. Therefore, Steiner & Taborek (1992) model the factor S as independent of the flow rate. The degree of the flow induced suppression is inherently determined by the magnitude of the exponent n.

Campbell et al. (1995) were the first group who considered subcooled boiling flow in automotive cooling systems. They carried out experiments with a typical engine coolant composed of 50Vol% ethylene-glycol and 50Vol% water, and they adopted Chen's ansatz for the heat flux modelling. Due to the subcooling they ignored the enhancement of the macroconvection caused by the agitation of the bubbles assuming F = 1, which implies Re_{2ph} = Re_{1ph}. Based on their experimental results they proposed the following correlations for the flow induced suppression factor:

$$S = \begin{cases} 1.0, & \mathrm{Re}_{1ph} < 10^4 \\ -0.6 \log_{10} \mathrm{Re}_{1ph} + 3.4, & 10^4 \le \mathrm{Re}_{1ph} \le 4 \cdot 10^5 \\ 0.04, & \mathrm{Re}_{1ph} > 4 \cdot 10^5. \end{cases} \tag{13}$$

The model of Campbell et al. (1995) is, like most of the proposed superposition models, not well suited for application in the CFD of coolant flows in engineering devices, because the model correlation for the suppression factor depends on the bulk flow Reynolds number as a non-local parameter. On the other hand, suppression is basically a local effect, and it should therefore be modelled dependent of local flow quantities, which may be readily provided by the CFD solution. Moreover, dealing with CFD of geometrically complex flow configurations like those in coolant jackets, it is practically not possible to define a Reynolds number based on bulk flow conditions in a meaningful way. To circumvent these obvious shortcomings, Kobor (2003) developed a Chen-type approach, termed Boiling Departure Lift-off (BDL) model, which accounts for the dynamic effect of the near-wall flow field on the bubble detachment from the heated surface. In the BDL model the suppression factor is modelled as dependent on the ratio of two characteristic bubble diameters, the departure and the lift-off diameters, such that

$$S \propto \frac{d_D}{d_L}. \tag{14}$$

The diameters d_D and d_L are computed from local force balances at the instants of bubble departure from the nucleation site and bubble lift-off from the surface, respectively, relying on a concept of Zeng et al. (1993). The BDL model was introduced into the CFD software FIRE, where it performed very well in simulations of automotive coolant jackets. Steiner et al. (2005) extended the BDL model to the boiling of pure water, where they introduced an

additional suppression factor for the nucleate pool boiling component to capture explicitly the influence of the subcooling. The BDL model predicted the flow induced suppression fairly well, as exemplarily shown in Fig. 5 by a comparison against experiments and predictions from other models.

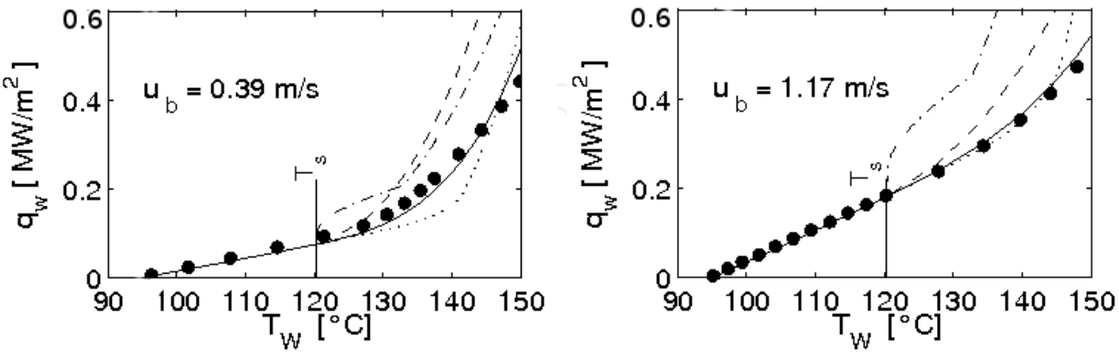


Fig. 5. Predicted flow boiling curves for water at p=2bar and two different bulk flow velocities: solid line '—', BDL, dashed line '- - -', Chen (1966), dash-dotted line '-·-·-', Shah (1977), dotted line '····', Kandlikar (1998b); (reprinted from Steiner et al. (2005) with permission of Elsevier).

The neglect of the influence of the bubble agitation on the convective heat transfer generally assumed in subcooled boiling flow loses its justification as the void fractions become higher near the wall, which is especially the case at high wall superheats combined with low liquid velocities (Maurus, 2003; Ramstorfer et al., 2008a). In order to account for the dynamic effects of the bubbles, while still remaining within the framework of a single-fluid formulation, Ramstorfer et al. (2008a) proposed to introduce a bubble-equivalent wall roughness. The increased wall roughness leads to a higher turbulence intensity in the near-wall region, yielding the enhanced convective heat transfer which is in reality due to the agitation of the bubbles.

3. Subcooled boiling flow under automotive engine conditions

In general, the models proposed for the wall heat flux in nucleate boiling flow are calibrated and validated for very specific conditions concerning working fluid, material and surface quality of the heater, as well as flow configuration. As such, the models, be they empirical or mechanistic, involve correlations and parameter settings, which are typically based on laboratory experiments using purified, degassed liquids, clean homogeneous surfaces with a specified finish and a defined, uniformly heated area. The experimental set-ups are mostly designed to ensure well defined stationary flow conditions in the test-sections. Experimental findings obtained with such particular set-ups may therefore not apply straightforwardly to real-life conditions in technical applications.

It has therefore become best practice to still apply the available well-established standard approaches to real engineering problems, however, using specially adapted model coefficients and/or introducing additional parameters to capture effects of particular importance for the actually considered case. Several of these effects with relevance for automotive cooling systems shall be discussed in the following.

3.1 Composition of the liquid

The boiling of multi-component mixtures can be strongly affected by diffusion as well as differing volatilities of the individual components. The latter leads to an enrichment of the vapour/liquid interface with the less volatile component, so that the local mixture at the interface has an effectively higher saturation temperature, $T_{s,int} > T_s$, and the evaporation rate decreases. This non-azeotropic behaviour of the mixture is in principle due to the finite speed of diffusion, which impedes a prompt balancing of the deficit of the more volatile component at the interface. Accordingly, its effect on the boiling activity is typically accounted for by incorporating a diffusion-induced suppression factor F_D into the model correlations which are basically derived for pure liquids (Wenzel & Müller-Steinhagen, 1995; Kandlikar 1998a). The suppression factor $F_D \leq 1$ reduces the nucleate boiling component, analogous to the factor S in Eqs. (9) or (12), and it essentially depends on the effective increase in the saturation temperature, $T_{s,int} - T_s$.

The working liquid in automotive cooling systems generally consists of two main components, water and the antifreeze component ethylene-glycol. The volumetric mixing ratio is mostly 50/50Vol%. Figs. 6a und b show two sets of flow boiling curves experimentally measured for three different mixtures at the Reynolds numbers Re_b = 5500 and 22000 (Breitschädel, 2008). The bulk velocities corresponding to the two Reynolds numbers are about u_b = 0.1 and 0.4 m/s, respectively. It becomes obvious that in both cases the 60/40Vol% mixture, i.e., the liquid with the smallest fraction of the more volatile component (water), exhibits the lowest total wall heat fluxes at a given wall superheat $\Delta T_{sat} > 0$ in the nucleate boiling regime. The boiling curves obtained with the BDL model (Kobor, 2003) are also shown in Figs. 6a,b to demonstrate the predictive capability of a typical Chen-type model which does not account explicitly for non-azeotropic effects of the mixture. The predictions also reflect the experimentally observed tendency for the considered mixtures. It appears to be sufficient to model the effect of the varying water/ethylene-glycol ratio in terms of a corresponding variation of the material properties of the mixture. This basically means that the binary system can be treated as an azeotropic mixture, and a diffusion induced suppression needs not to be introduced.

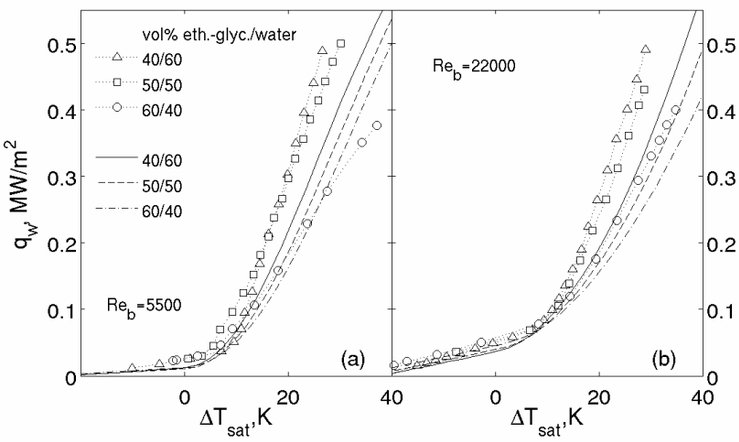


Fig. 6. Flow boiling curves for varying coolant compositions: (a) Re_b = 5500; (b) Re_b = 22000; experimental data denoted by symbols, BDL model predictions by lines; pressure: p = 1.5 bar (from Steiner et al., 2008).

3.2 Microgeometry of the heated surface

The microgeometry of technical surfaces is generally described in terms of the surface roughness, which is mostly expressed as an average roughness height R_a given in μm. Being a rather crude measure, R_a certainly does not represent the effectively boiling relevant microgeometry, which may be constituted by many different types of geometry elements, such as plateaus, peaks, valleys, cavities, etc. . It is generally accepted that only those surface elements can act as stable bubble generating centres (active nucleation sites), which are not completely filled with liquid after bubble departure. Therefore, an increase in the average surface roughness may lead to an increase in the boiling heat transfer only if the higher roughness is associated with additional active and stable nucleation sites. Numerous - mostly pool boiling - experiments have investigated this effect (see, e.g., Jakob & Fritz, 1931; Corty & Foust, 1955; Kuhihara & Myers, 1960). They showed that the heat transfer rate in general rises as the surface roughness is increased. The quantitative extent, however, strongly depends on the considered surface quality. The variation of surface roughness affects the nucleate boiling heat transfer most pronouncedly when considering high quality finished surfaces, where the average roughness is of the order of $R_a \approx 1\ \mu m$ and lower, as it was recently confirmed in pool boiling experiments of Jones et al. (2009) (see Fig. 7). In contrast, typical surface roughness in automotive cooling systems is markedly higher, where the mostly applied iron or Aluminium as cast surfaces have an average roughness height $R_a \approx 40 \mu m$ and higher. Breitschädel (2008) carried out experiments for automotive cooling conditions, varying the roughness of the aluminium surface between R_a=2μm ("smooth"), R_a=45.7μm ("standard"), and R_a=130μm ("rough").

As seen from the boiling curves in Fig. 8, which were measured after different operation times, both the smooth and the rough surfaces exhibit almost the same heat transfer conditions at a primary stage, because the earliest measured boiling curves of both surfaces almost coincide. The considerable concentration of large cavities on the very rough surface does evidently not provide additional active nucleation centres compared to the smooth,

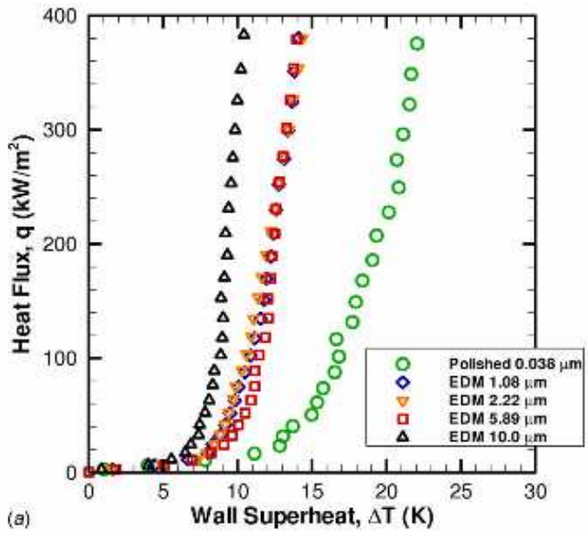


Fig. 7. Pool boiling curves for water on Electrical Discharge Machined (EDM) surfaces with different roughness R_a (reprinted from Jones et al. (2009) with permission of ASME).

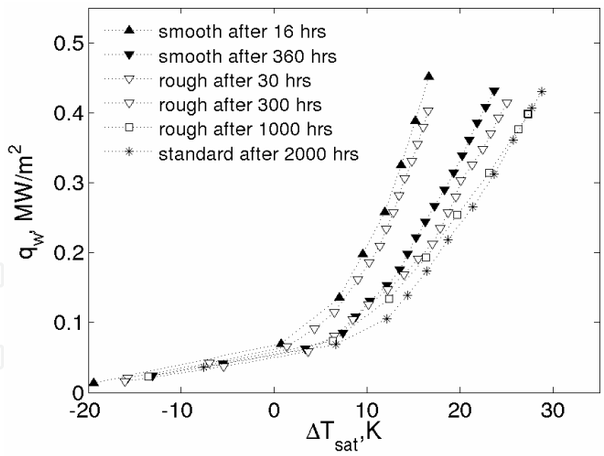


Fig. 8. Flow boiling curves for varying surface roughness after different operation times; bulk liquid velocity u_b = 0.476 *m/s*; mixture 50/50Vol% ethylene-glycol/water; pressure p = 1.5 *bar* (from Breitschädel, 2008).

polished surface. With increasing operation time, both types of surfaces exhibit a notable degradation of the heat transfer rate, as seen from the shift of the boiling branches towards higher wall superheats. This tendency points to a successive deactivation of initially active nucleation sites, so that they finally approach the boiling curve of a long-term measurement using the standard surface. The shown trends illustrate again the dominant role of the microgeometry, which finally levels out any potential primary surface roughness effect on the boiling heat transfer during long-term operation. For the considered surface types, the effect of the macroscopic surface roughness needs not to be explicitly considered in the modelling of the wall heat flux.

The potential increase in the boiling heat flux attainable with special superficial layers of the heater has become an important research topic investigating the boiling behaviour on so-called *enhanced surfaces*. Depending on their fabrication, these specially designed surfaces may be structured (Memory et al., 1995; Kim & Choi, 2001), e.g., with microfins, with pores connected by subsurface gaps, or unstructured such as porous coatings (Afgan et al., 1985; Kim et al., 2002; Rainey et al., 2001, 2003).

Using enhanced surfaces provides in general a higher number of active nucleation sites, which leads to lower minimum wall superheats required for the onset of nucleate boiling (ONB). Beyond the ONB, mostly higher boiling heat transfer rates are observed as compared to the unmodified smooth surfaces. The intensified boiling activity is commonly explained by the hypothesis that, on enhanced surfaces, the bubble nucleation occurs predominantly in subsurface, hence higher superheated, microchannel-like dendritic cavities, which are also more likely to entrap a gaseous rest after bubble departure than the superficial cavities on the unmodified surfaces. The potential of this concept to enhance the boiling activity for use in automotive cooling was investigated by Ramstorfer et al. (2008b). They experimentally examined flow boiling of an automotive coolant composed of 60 and 40Vol% water and ethylene-glycol, respectively, using heated surfaces with two different types of coatings. The first type of coating ("coating S1") was produced by spraying molten mild steel on the ground cast-iron surface. The second type of coating (``coating S2'') was fabricated by sintering a highly porous layer of iron particles on the cast-iron ground body (see Fig. 9).

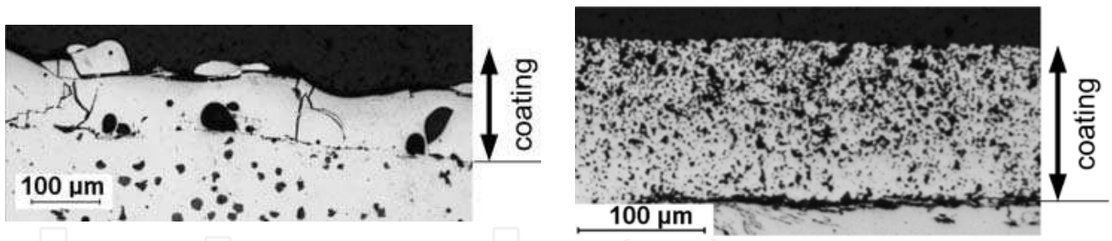


Fig. 9. Cross-sectional photomicrographs of the superficial layers of the heaters: left subfigure coating S1, right subfigure coating S2 (reprinted from Ramstorfer et al. (2008b) with permission of ASME).

The comparison of individual flow boiling curves shown in Fig. 10 reveals significantly higher boiling heat transfer rates for the coated surfaces, which can be clearly observed from the markedly reduced wall superheats ranging up to 15 *K*. It is interesting to note that coating S1 performs best, although its porosity is much lower than that of coating S2. This can be attributed to the particular microstructure of the coating S1. The coating S1 is not a typical granular-porous, nor a channel-porous layer, as it consists for the most part of a solid metallic base matrix, which is penetrated by a few, but comparatively deep cavities. These deep cavities are evidently most capable to entrap vapor, hence, to act as bubble nucleation centres, as it was also observed by Qi et al. (2004).

Ramstorfer et al. (2008b) also demonstrated that the superposition ansatz generally defined in Eq. (8) can be well extended to boiling on enhanced surfaces. Considering a vertical heated surface, they modelled the convective component q_{fc} following an ansatz of Churchill (1972) proposed for mixed (natural + forced) convection. For the nucleate boiling component q_{nb} they used the classical pool boiling correlation of Rohsenow (1952) with experimentally fitted model parameters. The predicted wall heat fluxes on the coated surfaces agree very well with the experiments over a wide range of bulk velocities, as seen from Fig. 11.

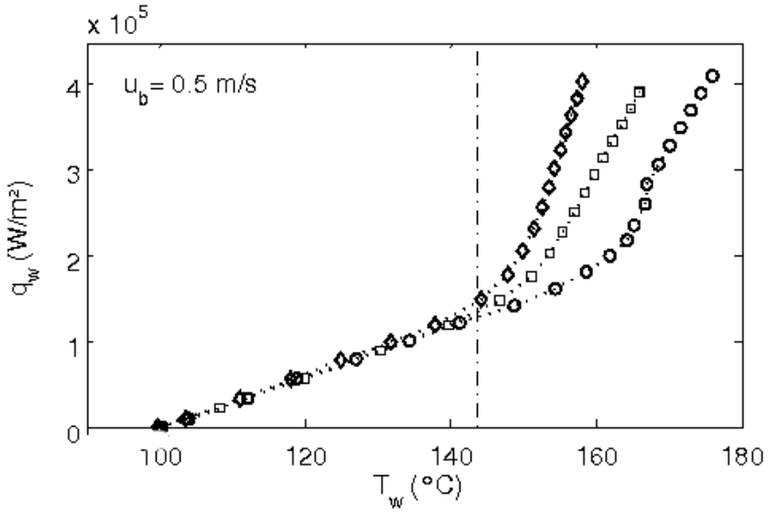


Fig. 10. Flow boiling curves of a 60/40Vol% water/ethylene-glycol mixture on different surfaces: ○ plain cast iron; ◊ coating S1; □ coating S2; saturation temperature T_s = 143 °C (dash-dotted line) ; subcooling $\Delta T_{sub} = T_s - T_b = 43$ *K* (reprinted from Ramstorfer et al. (2008b) with permission of ASME).

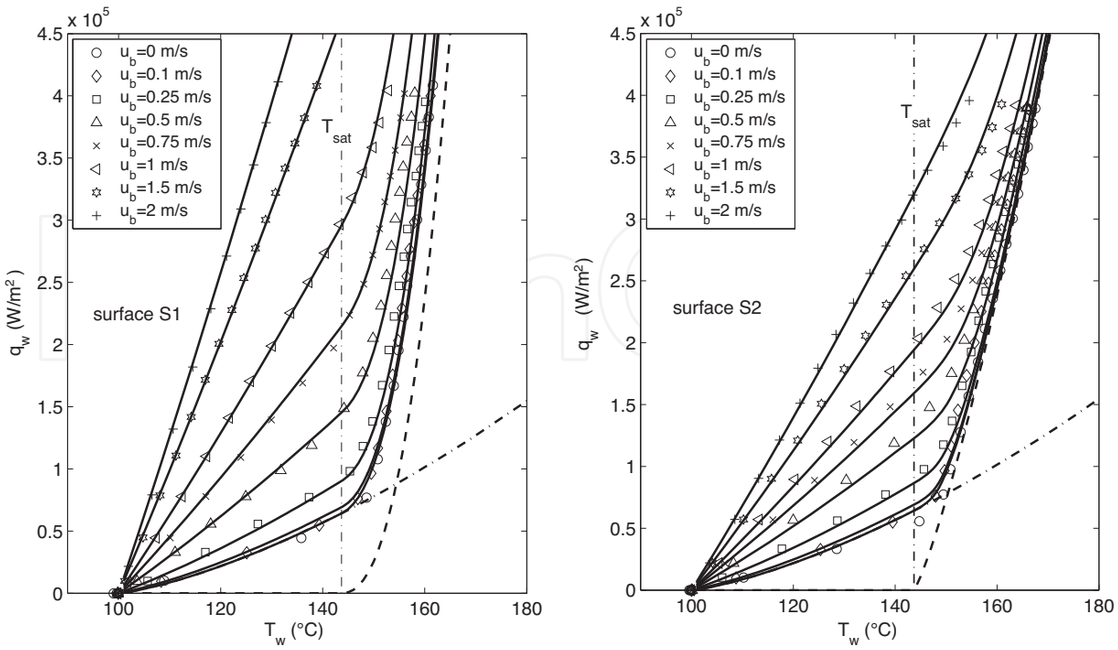


Fig. 11. Predicted flow boiling curves for the porous surfaces S1 and S2 at different velocities of the bulk flow: full line, "—", model prediction; dash-dotted line, – · – · –, natural convection; dashed line, "– – –", pool boiling curve; the symbols denote the measurements reprinted from Ramstorfer et al. (2008b) with permission of ASME).

Based on these findings, it can be noted that the use of enhanced surfaces represents a promising concept to increase the efficiency of automotive cooling systems. However, before its technical realization, several open questions need to be answered. Critical issues related to the manufacturing of the surface, the long term stability of the coating, or the effect on the mechanical resilience of the base structure, call for further detailed investigations. The modelling for the wall heat flux can rely on standard approaches using an appropriate parameter setting for the considered surface. These model parameters may vary considerably for different combinations of coatings and working liquids.

3.3 Orientation of the heated surface

Since nucleate boiling by nature involves the motion of a low-density vapour phase in a high-density liquid carrier phase, the dynamics in the thermal boundary layer may be strongly influenced by the buoyancy forces, especially at low flow rates of the bulk liquid. In such a case the orientation of the superheated surface relative to the direction of gravitational acceleration is of major importance. This aspect is mostly ignored by the nucleate boiling models though. Klausner et al. (2003) investigated the influence of buoyancy on the detachment process of the bubbles in much detail, computationally as well as experimentally. They carried out experiments with the perfluorocarbon liquid FC-87 for varying streamwise inclination angle of the heated surface from 0° to 360°. They found the influence of the surface orientation to be mainly dependent on wall superheat, liquid/vapour density ratio, and the flow velocity. Furthermore, they observed that the critical heat flux, where film boiling sets in, was reduced by almost an order of magnitude for a certain inclination at low flow rates. Roughly the same extent of reduction in the

critical heat flux was measured by Kim et al. (2005) for saturated pool boiling with water comparing upward against downward facing surfaces.

The potential effect of surface orientation was also investigated for automotive cooling conditions (Steiner et al., 2007). Fig. 12 shows the experimental results obtained for the subcooled boiling of a 50/50Vol% mixture of water and ethylene-glycol on a downward facing heated surface. The boiling curve in the lower left subfigure displays the measurements at three distinct positions, A, B, and C, as schematically shown in the subfigure above. The full line denotes the arithmetic mean of the three positions. Beyond a certain heat flux, marked by the transition point TP, the boiling curves at position B, and more pronouncedly at position C, deviate significantly towards higher wall superheats as compared to the almost straight boiling branch at A. The obvious streamwise decrease of the local heat transfer coefficient can be attributed to a transition from nucleate boiling to partial film boiling. This can be seen from the photographs in the right column of Fig. 12, displaying three distinct conditions corresponding to the points I, II, and III on the averaged boiling curve. Beyond the transition point (at conditions II and III), the vapour phase evidently tends to form ever larger film-like agglomerates, which cover a substantial part of the surface at the downstream end of the heater.

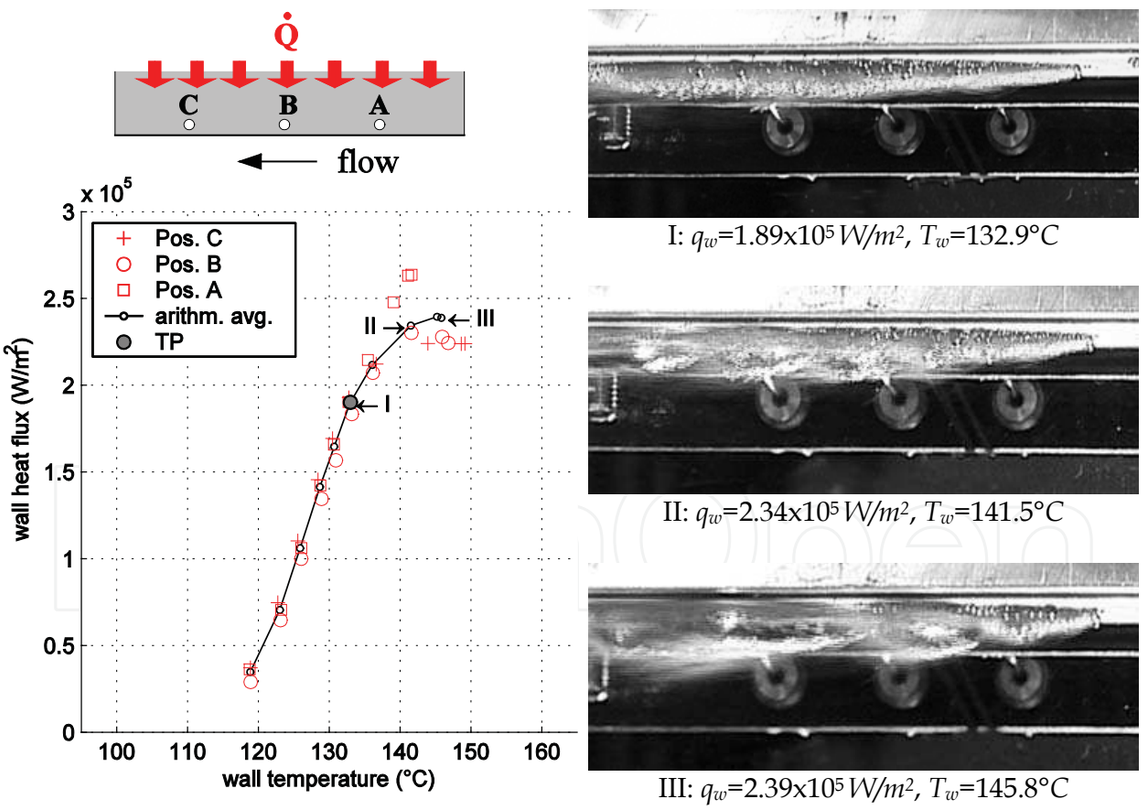


Fig. 12. Left column: wall heat flux q_w vs. wall temperature T_w measured at three positions for the bulk velocity $u_b = 0.2m/s$ and subcooling $T_s - T_b = 18°C$; right column: photographs of the two-phase region at the three distinct conditions I, II and III as specified in the boiling curves in the left column. The direction of flow is right to left (from Steiner et al., 2007).

Based on a dimensional analysis, a criterion for the observed transition from nucleate to partial film boiling was derived in terms of a non-dimensional correlation for a transitional Boiling number

$$Bo_{trans} = C\ \mathrm{Re}_L\ Ec_{sub}^{-n} \tag{15}$$

dependent on the Reynolds number $Re=u_b\rho_l L/\eta_l$ with the length scale $L=[\sigma/g(\rho_l-\rho_g)]^{0.5}$, and the Eckert number $Ec_{sub}= u_b^2/c_{p,l}(T_s-T_b)$ representing the effect of subcooling. $C=8.5\cdot 10^{-11}$ and n = 0.63 are empirically determined model parameters. As such, correlation (15) provides via the Boiling number Bo_{trans} an important estimate for the transitional wall heat flux $q_{w,trans}$, beyond which a transition to partial film boiling has to be expected, which may eventually lead to a hazardous increase of the wall temperature.

Aside from the issue of ensuring acceptable wall superheats, the transitional wall heat flux $q_{w,trans}$ also represents an upper limit for the applicability of standard wall heat flux models generally used in subcooled boiling flow. The boiling contributions in these approaches are mostly correlated and calibrated assuming fully developed nucleate boiling on upward or sideward facing surfaces, so that they expectedly reach their limit, once partial film boiling sets in. This limitation is illustrated in Fig. 13, comparing experimental data measured for a downward facing surface with the predictions of the BDL model using a parameter setting proposed for automotive engine coolants (Kobor, 2003). The model predictions agree in general fairly well with the experiments below the transitional heat flux $q_{w,trans}$ computed from (15), while they exhibit considerable deviations above.

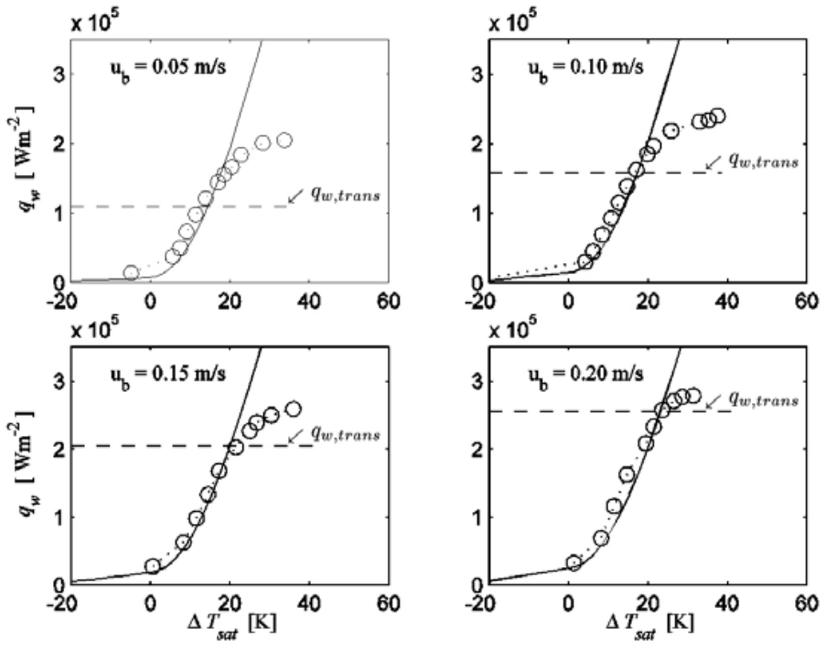


Fig. 13. Flow boiling curves of a 50/50Vol% mixture of water and ethylene-glycol on a downward facing heated surface at different velocities of the bulk flow u_b. The open circles,○, denote the experiments, the solid lines, '—', denote the predictions of the BDL model, the dashed lines, '- -', denote the transitional wall heat flux $q_{w,trans}$ computed from Eq. (15). The subcooling $\Delta T_{sub} = T_s - T_b = 25K$ is constant; (from Steiner et al., 2007).

The horizontal dashed lines in Fig. 13 also illustrate the strong flow rate dependence of the transitional limit $q_{w,trans}$. Accordingly, at the lower bulk velocities, $q_{w,trans}$ drops down to a level which is an order of magnitude lower than the critical heat flux in the corresponding pool boiling case on an upward facing surface. The quantitative extent of this reduction is well in line with the findings of Klausner et al. (2003) and Kim et al. (2005) mentioned above.

3.4 Aging

Aging is a phenomenon which may strongly affect the long-term activity of the nucleation sites on the heated surface. It therefore represents a great challenge and persistent source of inaccuracy for all boiling models. Due to its potential influence on the liquid-vapour-solid interfacial interactions at the nucleation sites, it may be highly relevant for the entire boiling process, starting from the onset of nucleate boiling (ONB). Aging subsumes the gradual, mostly slow changes of the working fluid, as well as those of the heated surface, which in general lead to less favourable conditions for the incipience of nucleate boiling. As such, aging is a process acting over long time scales, which is manifested in a steady decrease of the boiling heat transfer rate observed during long periods of operation time (weeks or months). Especially under technical flow conditions it is often impossible to clearly identify and eventually eliminate all relevant causes for aging. It can have many – single or multiple - causes, such as a continuous flooding of cavities, depositions on the surface, corrosion and/or mechanical erosion of the surface material, chemical reactions in the liquid phase, etc.. The quantitative impact of aging on subcooled boiling can be exemplarily seen from Fig. 14, showing experimental data obtained for one given liquid/surface combination, a mixture of ethylene-glycol and water on an as-cast aluminium surface, which is typically met in cooling jackets of modern internal combustion engines. The shown curves were measured over a period of two weeks. The temperature, pressure and velocity of the bulk liquid were always kept the same. Within the considered range of wall heat fluxes, the first

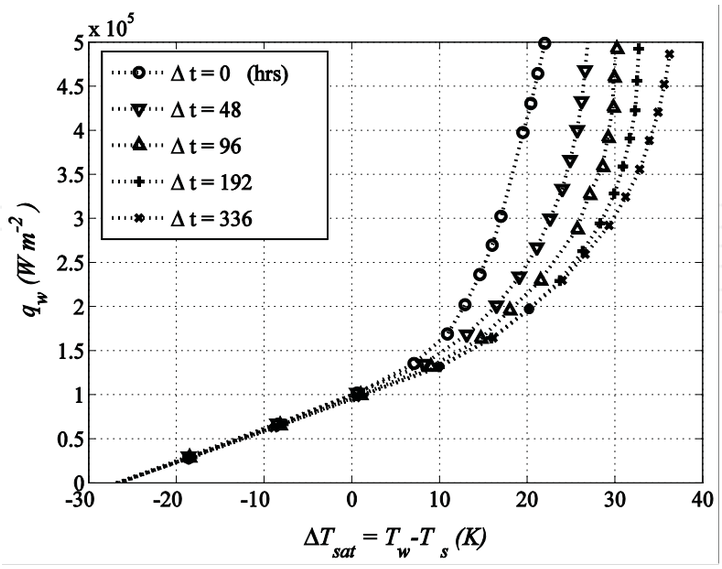


Fig. 14. Boiling curves of a 50/50Vol% mixture water and ethylene-glycol; the subcooling and the velocity of the bulk liquid are $\Delta T_{sub} = T_s - T_b = 25$ *K* and $u_b = 1$ m/s, respectively (from Kobor, 2003).

and the last curves differ up to 15 *K* in wall superheat $\Delta T_{sat} = T_w - T_s$. It could be shown that the aging effect observed here is partly caused by a continuous flooding of the cavities on the surface, which reduces the number of active nucleation sites. The other part could be attributed to depositions on the heated surface originating from the employed coolant liquid. The observed significant shift in the boiling curves strongly suggests that the aging conditions of the heated surface and the working fluid must not be overlooked in the interpretation of boiling flow measurements and in the specification of the model parameters based on such data. This caveat is particularly relevant for boiling of aqueous liquids on real technical surfaces.

7. Conclusions

The enhancement of heat transfer rates based on a controlled transition from pure single-phase convection to subcooled boiling flow appears to be a promising approach for application in automotive cooling systems. A reliable and save thermal management requires a most comprehensive knowledge of how certain operation and system conditions may affect the boiling behaviour. Therefore, we put our focus on a selection of engine relevant conditions and their possible impact on the modelling of the wall heat flux. This led us to the following resume.

As for the influence of the mixing ratio of the two main components of the coolant, water and ethylene-glycol, the heat transfer rates in the boiling regime tend to decrease when the fraction of the more volatile water component is smaller. The tested wall heat flux model, which basically assumes the coolant as an azeotropic mixture, reflected the observed tendency very well. The effect of the mixing ratio can be evidently captured with sufficient accuracy in terms of the material properties of the mixture. For the considered range of engine relevant mixing ratios and subcooled boiling flow conditions, non-azeotropic effects, such as the increase of the effective saturation temperature due to the depletion of the more volatile component at the liquid/gas interfaces, appeared to be of minor importance.

The effect of the macroscopic surface roughness turned out to be very limited in time. Long-term experiments confirm the dominant role of the microstructure of the surface, which finally leads to approximately the same boiling behaviour of all considered surface finishes. Based on this observation it may be concluded that the effect of the surface finish in terms of a roughness height may be disregarded in the wall heat flux model.

The use of porously coated, “enhanced”, surfaces appears also attractive for application in automotive cooling. The scope of most studies on this subject is, however, in general strongly limited to the particularly considered type of coating and working liquid. Making use of this concept requires therefore further detailed investigations especially devoted to porous superficial layers, which can be technically realized in engine cooling systems. The standard wall heat flux models can be well extended to enhanced surfaces, when an appropriately adapted parameter setting is used.

Concerning the effect of the surface orientation, the case of a downward facing surface heated from above is expectedly the most critical one. Since the buoyancy force counteracts the bubble lift-off from the surface, a transition from nucleate boiling to partial film boiling can occur well below the critical heat flux associated with an upward facing surface. The observed strong dependence of this transitional heat flux on the velocity and subcooling of the bulk liquid could be cast into a non-dimensional criterion for the corresponding transitional Boiling number. Applying exemplarily the BDL model for predicting the wall

heat fluxes, it could be further shown that this standard Chen-type superposition approach is capable to produce acceptably accurate predictions up to the transitional heat flux without any special modifications accounting for the effect of orientation.

Aging is probably one of the most critical phenomena, especially when using aqueous working liquids typically found in automotive cooling systems. The phenomenon may be sustained by many complex chemical/physical sub-processes, which are hard or even impossible to control under real technical conditions. The boiling curves obtained after different operation times, or operations modes, may be shifted by 15 *K* and even more in the wall superheats. It therefore often requires long-term experiments to obtain reliable results, which exhibit no notable change in time, so that they can be used for model evaluation and calibration.

8. Acknowledgements

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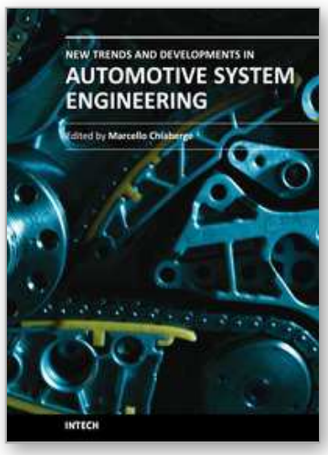

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In the last few years the automobile design process is required to become more responsible and responsibly related to environmental needs. Basing the automotive design not only on the appearance, the visual appearance of the vehicle needs to be thought together and deeply integrated with the â€œpowerâ€ developed by the engine. The purpose of this book is to try to present the new technologies development scenario, and not to give any indication about the direction that should be given to the research in this complex and multi-disciplinary challenging field.

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