Modelling of heating and evaporation of multi-component liquid films: recent developments

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Introduction

The modelling liquid film heating and evaporation and its various engineering applications have been widely discussed in the literature [6]. The application of the results of this modelling to spray cooling is perhaps the most widely known in engineering community. This cooling is widely used in metallurgy [8], electric vehicles to cool power batteries [1], electric motors [2], aerospace engineering [4], and electronics [3]. Particular attention has focused on spray cooling of high-power LEDs, where heat flux densities can reach 500 W/cm^2 (see Ref. [3]).

Although the practical importance of spray cooling is well known, many underlying physical processes that take place during its application are not well understood. Several studies indicate that the flow flux of the liquid is a key factor in determining the efficiency of heat removal during spray cooling [5, 9].

In this extended abstract, we will summarise recent results of the development on a simplified model that captures the key features of the processes in heated and evaporating liquid film and the validation of the results of this modelling. The main ideas of the model, described in this abstract, are based on the model of multi-component film heating/cooling and evaporation, described in [6, 7]. In that model, the analytical solutions to the one-dimensional heat transfer and component diffusion equations were used at each time step of the numerical code (analytical-numerical model). Some important limitations of the latter model, however, restricted its applicability with regard to many practically important engineering problems, including spray cooling. These were the assumption of a fixed wall temperature (Dirichlet boundary condition) and the absence of supply and removal of liquid from the film. These restrictions are removed in the new developments of the model briefly described below.

Materials and Methods

Model

The heat conduction equation inside the film, used in the analysis, is presented as:

$$\kappa_1 \frac{\partial^2 T}{\partial x^2} = \frac{\partial T}{\partial t}, \tag{1}$$

where $\kappa_1 = k_1/(c_1\rho_1)$ is the thermal diffusivity, k_1, c_1, and ρ_1 are the thermal conductivity, the specific heat capacity, and the density of the liquid, respectively, x is the distance from the surface.

We assume that the temperature gradient at the surface is specified as:

$$\left.\frac{\partial T}{\partial x}\right|_{x=0} = Q_0, \tag{2}$$

where Q_0 is proportional to the heat flux supplied to the film from the surface (Neumann boundary condition).

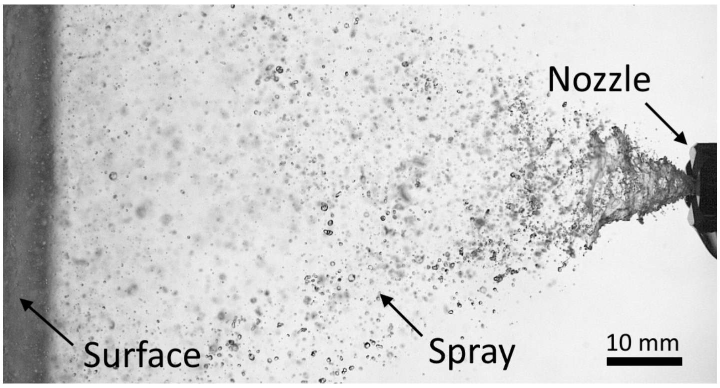

Figure 1. A photograph of a spray reaching the surface of the heat exchanger.

The boundary condition at the liquid film surface in the presence of a liquid supply to the film is formulated as the energy balance equation at this surface:

$$h(T_\mathrm{g} - T_\mathrm{s}) + \rho_\mathrm{l} L \dot{\delta}_{0e} + c_\mathrm{l}\rho_\mathrm{l}(T_\mathrm{spr} - T_\mathrm{s})\dot{\delta}_\mathrm{spr} = k_\mathrm{l} \left.\frac{\partial T}{\partial x}\right|_{x=\delta_0 - 0}, \tag{3}$$

where h is the convection heat transfer coefficient, T_g is ambient gas temperature, T_s is the surface temperature), ρ_l is liquid density, L is the specific heat of evaporation, $\dot{\delta}_e \leq 0$ is the rate of change of film thickness due to evaporation, T_spr is the sprays temperature, c_l is specific heat capacity of the liquid, $\dot{\delta}_\mathrm{spr} > 0$ is an increase in film thickness due to supply by spray, k_l is liquid thermal conductivity.

The analytical solutions to Equation (1) and the corresponding component diffusion equation in the liquids were used at each timestep of calculations. The predictions of the analytical solutions at the end of the timestep were used as initial conditions for the next timestep with adjusted values of input parameters.

The model predictions were validated using the results of in-house experimental data obtained using the setup described in the next section.

Experimental setup

Experimental investigations of heat transfer during the interaction of a spray with a heated surface were carried out on a specialised test bench equipped with a system supplying pressurised distilled water through a nozzle fitted with a solenoid valve. A stainless-steel heat exchanger measuring 70×70 mm with a thickness of 20 μm was positioned at a fixed distance from the nozzle and heated by a direct current power source, as shown in Figure 3.

Measurements of the temperature distribution on the surface were performed using infrared thermography on the reverse side of the heat exchanger (suface). Positions of the surface, liquid film and spray are shown in Figure 2.

The experiments included three series of studies: the first investigated pulsed surface cooling during short-term injection of a fixed liquid volume at a constant pressure drop under various thermal conditions; the second analysed the removal of heat from the surface during continuous supply of water to the film; the third focused on the removal of the liquid film from the surface after the cessation of liquid supply at a steady mass flow rate and specified heat fluxes. In all cases, the surface temperature dynamics was recorded.

Results and Discussion

Initially the surface was dry. The balance of heat supply from the heater and heat removal from the surface by convection led to the establishing of a fixed surface temperature equal about 61.7°C. The time evolution of the experimentally observed surface temperatures is shown in Figure 3 as red circles. Zone I in this figure refers to the last 10 s of the period of heating of the

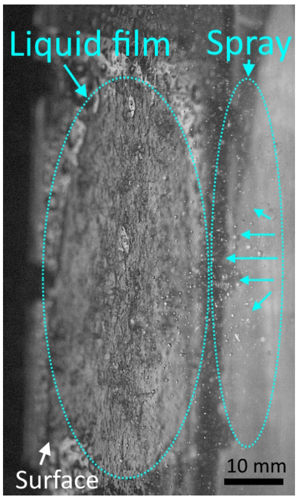

Figure 2. The same as Figure 3 but the photo was taken from a different direction and with a different focus.

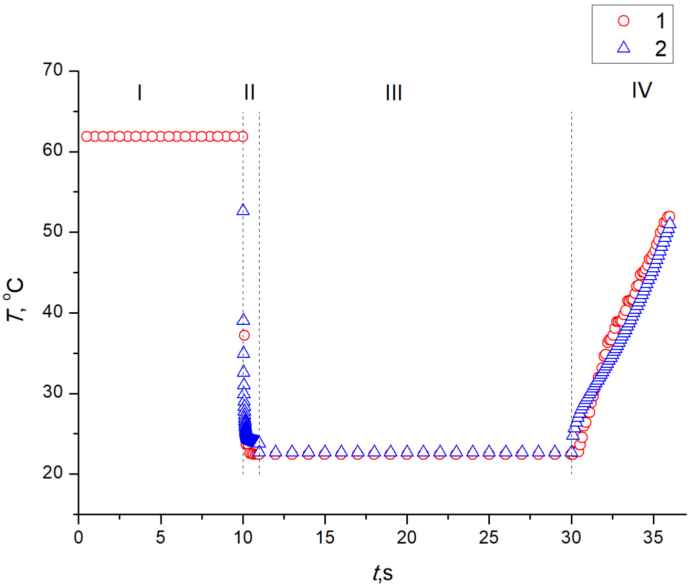

Figure 3. Observed (1) and predicted wall temperatures versus time.

dry surface.

At about 10 s spray is injected and water covers the whole surface. This leads to a rapid drop of surface temperature from around 61.7°C to around 22.4°C. This period of surface cooling is shown as zone II in Figure 3. At the next stage (zone III in Figure 3) continuous supply of water by spray and water removal from the liquid film lead to an establishment of almost constant film thickness and almost constant wall temperature of about 22.4°C. At the end of zone III, water supply to the film stopped but removal of water from the film continued. This led to a decrease in film thickness and an increase in wall temperature.

The results of temperature measurements during this process are shown in zone IV in Figure 3).

The results of modelling of these processes, using the approach described in the previous section, are shown in the same Figure 3. Good agreement between the experimental and modelling results are clearly seen in this figure.

Also, the model was generalised to consider the effect of possible formation of salt crystals in the salt solution. The predictions of this generalised model were shown to be reasonably close to experimental observations.

Conclusion

The results of recent developments of the model of muti-component liquid film heating and evaporation are presented. These developments include considering the effect of continuous supply of liquid to the film, the effect of continuous supply of heat to the film via the surface, and the effect of possible formation of salt crystals in the salt solution. The new developments are based on a combined analytical and numerical approach when the analytical solutions to the heat transfer and component diffusion equations are implemented in the numerical code. The predictions of the new model are validated based on the in-house experimental results.

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