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Technical Note

A study of transient processes in a closed vessel due to injection of hot liquid sprays

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Abstract

An experimental study to investigate the transient processes which occur when hot liquid spray droplets are suddenly injected into a gas contained in a closed chamber was reported previously [Proc. Inst. Mech. Eng. 215E (2001) 79]. That work also presented a macroscopic (bulk) analysis model, which agreed reasonably well with the experimental data. This work presents an analysis based on the simultaneous solution of the energy and mass balance equations for a single droplet and the gas-vapour control volume associated with the droplet. Good agreement has been found between the experimental data and the results of the single droplet analysis. The values of the heat transfer Nusselt number and the mass transfer Sherwood number, found using this approach, are approximately equal to 1. This agrees well with the results obtained from the macroscopic analysis reported previously and studies on heat transfer from dense evaporative droplet sprays.

Keywords: Spray droplets; Heat transfer; Mass transfer; Analytical results

1. Introduction

Injecting liquids in the form of sprays is an efficient form of liquid–gas contacting, since a large interfacial area per unit volume is achieved [1], with resulting excellent heat and mass transfer performance between liquid sprays and surrounding gases. These processes are used in many engineering applications, such as spray condensers, evaporative cooling and for pressure control (particularly pressure suppression) in confined gas spaces, such as the containments of nuclear reactors. Generally two approaches may be used for the calculation of heat and mass transfer rates between spray droplets and surrounding gasses [2,3]. In the first approach, predictions are made for single droplets and this is then summed for all the spray droplets. In the second approach, estimations of macroscopic transfer rates are made on the basis of volumetric coefficients determined empirically under similar operating conditions.

In this work heat and mass transfer from hot water spray droplets injected from solid cone nozzles into air enclosed within a cylindrical chamber, and their effects on the pressure increase in the chamber are analysed. This problem was also considered in a previous study [4]. The aim of the previous investigation was to present new experimental data and to understand the basic mechanism of heat and mass transfer from hot water spray droplets into the surrounding gas. The second approach discussed above was used, and hence the emphasis was on the processes occurring in the bulk volume, rather than transfer processes for single droplets. It is the aim of this work to present a companion analysis based on the first approach discussed above, by simultaneously solving energy and mass balance equations for a single evaporating droplet and its associated

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gas/vapour phase. Similar problems have been analysed previously [5–7], but the investigations were either theoretical or the geometries used did not allow measurements of significant thermal parameters, such as pressure and temperature. The advantage of the present approach is that the geometry and the initial and boundary conditions are well defined, thus allowing direct comparison between theoretical results and numerical data. This is discussed below.

2. Experimental work

A diagram of the experimental apparatus is shown in Fig. 1. The spray chamber is constructed from an acrylic cylinder, of 121 mm internal diameter and 190 mm length, and mounted with its axis vertical. Spray nozzles are installed at one or more of the five positions provided in the upper cover plate, centrally and at four equi-spaced positions on a 60 mm diameter circle, and point downwards. Five different nozzle sizes and three different nozzle configurations were used to investigate their influence on the transfer processes in the vessel. Commercially available solid cone nozzles, manufactured from brass by the Delavan company, were used in the experimental work. Since the results for all nozzles are similar, only the results for nozzle BP10-90 are reported in this work. The manufacturer's data for this nozzle at the injected water flow rate of 20 g/s are: nozzle pressure drop 6.5 bar, spray angle 80° and Sauter mean droplet diameter 150 μm. The initial droplet velocity was then calculated as 35 m/s. Further details are given in the previous study [4].

The following parameters were measured: the nozzle injection pressure, spray chamber pressure, temperature of the injected water, and the initial air temperature in the spray chamber. The data were captured on a computer, which was also used to control the experimental work. Further details were given previously [4,8]. A selection of experimental results is presented in Figs. 2–5 in the form of spray chamber absolute pressure p against the injection time t. Further experimental results are given in the previous study [4].

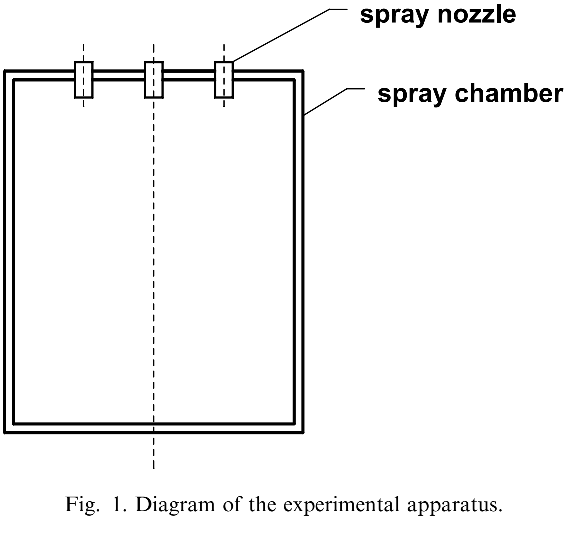

Fig. 1. Diagram of the experimental apparatus.

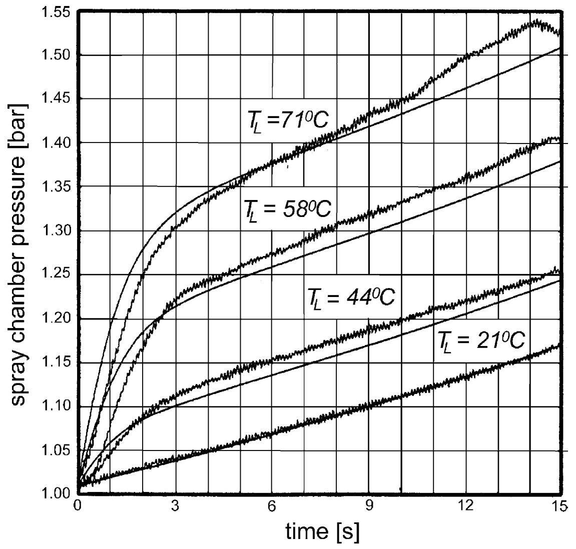

Fig. 2. Comparison of theoretical and experimental results for 1 central BP10-90 nozzle in air, water injection flow rate 20 g/s, $p_0 = 1.004$ bar and initial relative humidity 50%; influence of the liquid temperature.

3. Theoretical work

3.1. Development

As stated above, the theoretical analysis presented in this work is based on a simultaneous solution of the energy and mass balance equations for a single evaporating liquid droplet and its associated gas phase. It is assumed that a single droplet with the associated gas phase characterises the whole spray chamber, and hence that the transient energy and mass transfer processes there are representative of the transient energy and mass transfer processes in the whole chamber. Similar methods have been used previously [9,10]. The analysis assumes that there is no direct interaction between the droplets and the wall. Furthermore, a simple scoping analysis indicates that the heat transfer rate between the gas phase and the dispersed droplets is at least an order of magnitude greater that the heat transfer rate between the gas phase and any liquid film on the containing wall. Similar approach has been used previously [9]. However, the presence of the walls is taken into account when the

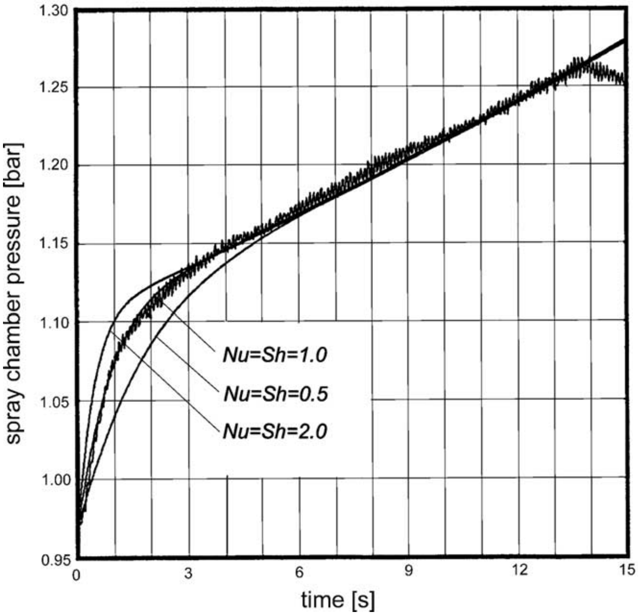

Fig. 3. Comparison of theoretical and experimental results for 1 central BP10-90 nozzle in air, water injection flow rate 20 g/s and injection temperature 62 °C, $p_0 = 0.975$ bar and initial relative humidity 50%; influence of the Nusselt and Sherwood numbers.

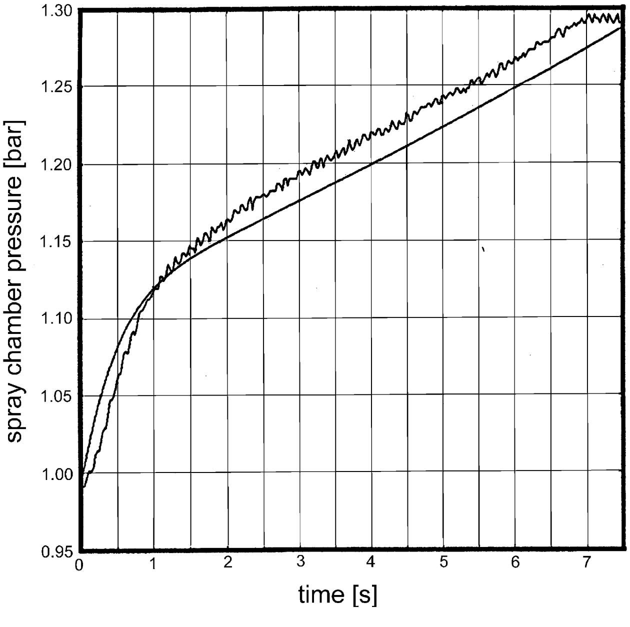

Fig. 4. Comparison of theoretical and experimental results for 2 side BP10-90 nozzles in air, water injection flow rate 20 g/s (total injection flow rate of 40 g/s) and injection temperature 60 °C, $p_0 = 0.992$ bar and initial relative humidity 50%.

average droplet residence time in the spray chamber is calculated from its initial velocity and the typical trajectory. Based on the nozzles spray angle and the geometry of the spray vessel, the typical droplet trajectory for the BP10-90 nozzle under present conditions is estimated as $l_T = 150$ mm.

A typical control volume for a single droplet, as shown in Fig. 6, is considered. The energy balance for a droplet can be written as [4]

$$m_D c_D \frac{dT_D}{dt} = h S_D (T_G - T_D) + h_{fg} \frac{dm_D}{dt} \quad (1)$$

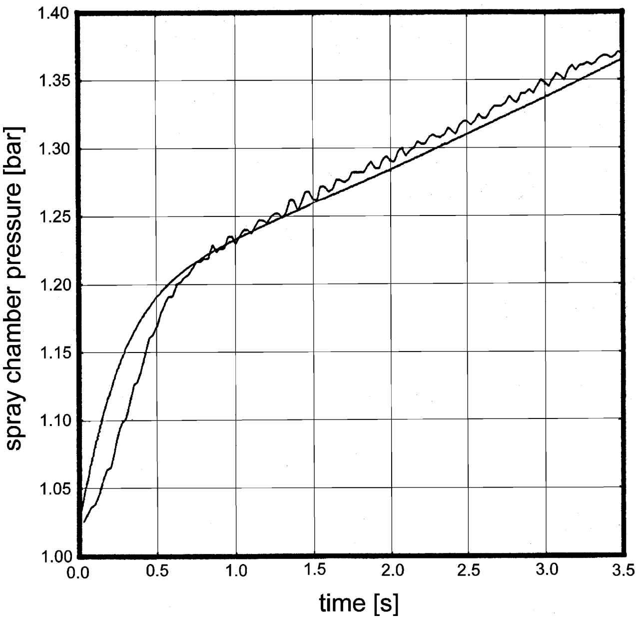

Fig. 5. Comparison of theoretical and experimental results for 4 side BP10-90 nozzles in air, water injection flow rate 20 g/s (total injection flow rate of 80 g/s) and injection temperature 65 °C, $p_0 = 1.022$ bar and initial relative humidity 50%.

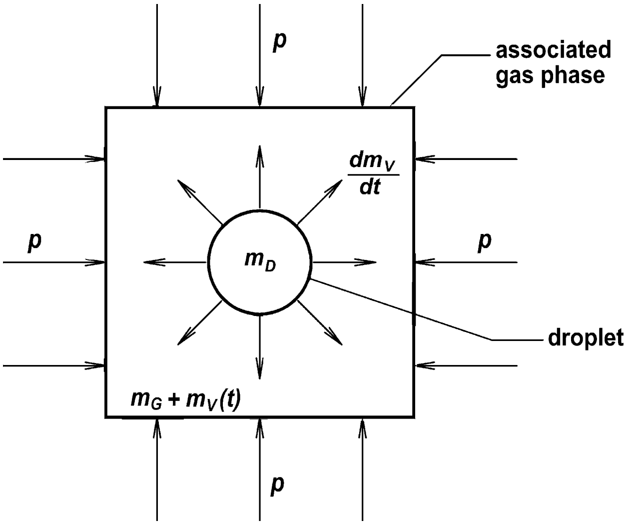

Fig. 6. Diagram of the control volume surrounding a single droplet.

where m_D is the mass of the droplet, T_D is the droplet temperature, T_G is the temperature of the gas/vapour control volume surrounding a single droplet, S_D is the droplet surface area, t is time, c_D is the specific heat of the droplet, h is the droplet-gas heat transfer coefficient and h_{fg} is the specific enthalpy of evaporation.

The energy balance for the gas/vapour control volume surrounding a single droplet is

$$\frac{\mathrm{d}U_{CV}}{\mathrm{d}t} = hS_D(T_D - T_G) - p\frac{\mathrm{d}V_{CV}}{\mathrm{d}t} + h_V\{T_D\}\frac{\mathrm{d}m_V}{\mathrm{d}t} \tag{2}$$

where U_{CV} and V_{CV} are the internal energy and the volume of the gas/vapour control volume respectively, p is the total pressure, m_V is the mass of the vapour in the control volume, and h_V is the specific enthalpy of the vapour.

The rate of change of the control volume internal energy can be re-written in terms of specific internal energies of the gas, u_G, and the vapour, u_V, as

$$m_G \frac{du_G}{dt} + m_V \frac{du_V\{T_G\}}{dt} + u_V\{T_G\} \frac{dm_V}{dt} = hS_D(T_D - T_G) - p\frac{dV_{CV}}{dt} + [u_V\{T_D\} + pv_V\{T_D\}]\frac{dm_V}{dt} \quad (3)$$

where m_G is the mass of gas in the control volume, and v_V is the specific volume of the vapour.

The mass of the gas/vapour mixture in the control volume, m_M, is given as

$$m_M = m_G + m_V \quad (4)$$

where m_V is a function of time. The rate of change of the mixture internal energy is then given as

$$m_M \frac{du_M}{dt} + u_M \frac{dm_M}{dt} = m_G \frac{du_G}{dt} + m_V \frac{du_V}{dt} + u_V \frac{dm_V}{dt} \quad (5)$$

where

$$u_V = c_{VV} T_G \quad (6)$$

where c_{VV} is the specific at constant volume of the vapour, u_M is the specific heat internal energy of the gas/vapour mixture

$$u_M = c_{VM} T_G \quad (7)$$

and where c_{VM} is the specific heat at constant volume of the gas/vapour mixture, defined as

$$c_{VM} = \frac{x_{GCV} M_G c_{VG} + x_{VCV} M_V c_{VV}}{x_{GCV} M_G + x_{VCV} M_V} \quad (8)$$

and where c_{VG} is the specific at constant volume of the gas, x_{GCV} and x_{VCV} are the mole fractions of the gas and the vapour in the bulk of the control volume respectively, and M_G and M_V are the molecular masses of the gas and the vapour respectively.

Substituting Eqs. (4)–(8) into Eq. (3), and after some re-arrangements

$$m_M c_{VM} \frac{dT_G}{dt} = hS_D(T_D - T_G) + \frac{dm_V}{dt}(c_{VV}T_D - c_{VM}T_G) - p\frac{dV_{CV}}{dt} + pv_V\{T_D\}\frac{dm_V}{dt} \quad (9)$$

The change of droplet mass due to evaporation can be written in terms of mole fractions of vapour at the droplet surface and in the control volume bulk, neglecting the solubility of the gas in water [11] as

$$\frac{dm_D}{dt} = -\frac{\rho_G}{M_G} D_{ab} Sh \pi a \frac{x_{VD} - x_{VCV}}{1 - x_{VD}} \quad (10)$$

where ρ_G is the density of the gas, D_{ab} is the diffusivity of water vapour in air, Sh is the Sherwood number, x_{VD} is the mole fraction of vapour at the droplet surface (at the droplet temperature T_D) and x_{VCV}, the mole fraction of vapour in the control volume, is based on the control volume temperature T_G.

3.2. Outline of the numerical solution

Eqs. (1), (9) and (10) form a system of coupled first order differential equations, which were solved using the fourth order Runge–Kutta method [12]. The equations were supplemented by an equation for pressure calculations. The gas partial pressure p_{Gt} at any time t was determined from the ideal gas law. The total pressure p_t at any time t was calculated as

$$p_t = p_{Gt}\left(1 + \omega_t \frac{R_G}{R_V}\right) \quad (11)$$

where ω_t is the specific humidity, and R_G and R_V are the gas and the water vapour specific gas constants respectively.

The average droplet lifetime t_D and the average number of droplets present in the spray chamber N_D are calculated as shown in [4], with the typical control volume associated with a single droplet, $V_{CV} = V_{SC}/N_D$, where V_{SC} is the volume of the spray chamber.

The transient processes occurring during a typical droplet lifetime were then modelled. When this lifetime elapsed a new, 'fresh' droplet with the specified initial temperature and radius was introduced into the control volume and the whole process was repeated until the total time span was reached.

4. Results and discussion

Calculations were carried out for a similar range of conditions reported in the previous work [4], thus allowing a direct comparison between the bulk analysis reported there and the single droplet analysis presented in this work. Values of the Nusselt number, *Nu*, and Sherwood number, *Sh*, were specified and the governing equations solved. It should be noted that the widely used assumption $Nu = Sh$ was used in this work. This relationship can be proved for low Reynolds numbers, and is widely used for low and moderate Reynolds numbers [11]. The best values of the Nusselt and Sherwood numbers were determined by trial and error by comparing the theoretical results with the experimental data. The appropriate initial conditions were specified, and the calculations were carried out for Delavan BP10-90 nozzle, as a typical representative of the range of nozzles investigated.

Figs. 2–5 indicate that, for the appropriate value of *Nu* and *Sh*, there is a reasonable agreement between the experimental data and the theoretical solutions. Furthermore, these figures also show that, for the conditions investigated in this work, the best values of *Nu* and *Sh* are $Nu = Sh = 1$. Fig. 2 compares theoretical and experimental results on the variation of the spray chamber pressure with time, and the influence of the liquid temperature. Fig. 3 shows the influence of the Nusselt and Sherwood number on the theoretical results. Finally, Figs. 4 and 5 compare the results for different nozzle configurations.

It should be noted that the bulk analysis reported previously [4] can be used to calculate the corresponding droplet to gas Nusselt number on the basis of the typical droplet properties given by the manufacturer's data. Such calculations indicate that the Nusselt number is equal approximately to 1.

This value of the Nusselt number may seem surprising, considering that it is below the lower limit of $Nu = 2$ for convective heat transfer from a single sphere. However, it has been shown [13] that for dense evaporative sprays the effect of clustering and evaporation reduces the Nusselt number to levels below 2.

Figs. 2–5 indicate that for a wide range of conditions there is a reasonable agreement between theoretical results and experimental data. Nevertheless, the theoretical results for the spray chamber pressure generally overpredict the experimental pressure for short injection time and generally overpredict the experimental pressure for long injection time. The reason is that for short injection times, when the evaporation rates are high, the Nusselt and Sherwood numbers should be even lower than 1, and that for longer injection times the Nusselt and Sherwood number should be greater than 1.

5. Conclusions

The macroscopic (bulk) analysis model, derived in the previous study [4], agrees reasonably well with the experimental data, and leads to a corresponding droplet-to-gas Nusselt number equal approximately to 1. However, the bulk analysis does not fully incorporate the effect of evaporation from the spray droplets. This limitation has been overcome in the present work using an analysis based on the simultaneous solution of the energy and mass balance equations for a single droplet and the gas-vapour control volume associated with the droplet. Good agreement has been found between the experimental data and the results of the single droplet analysis. The values of the heat transfer Nusselt number and the mass transfer Sherwood number, found using this approach, are equal approximately to 1. This agrees well with the results obtained from the macroscopic (bulk) analysis and with other work on heat transfer in dense evaporative droplet sprays.

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