MODELING A DIRECT CONTACT HEAT EXCHANGER FOR A SUPERCRITICAL WATER LOOP

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Abstract

In the last thirty years, Direct Contact Heat Exchangers (DCHX) have found a great success in different power engineering applications. In fact, due to the direct contact of hot and cold working fluids, it is possible to reach very high mass and energy transfer efficiencies. Despite their high performance, it is still quite difficult to predict the correct heat transfer as a function of plant operation conditions, which constitutes a fundamental parameter to correctly operate heat exchangers. Therefore, in this work, a DCHX used in the Thermo-Hydraulic Laboratory of École Polytechnique de Montréal, has been studied. It consists of a vessel where superheated steam is cooled by mixing it with sub-cooled water via a nozzle that sprays the water under the form of tiny droplets (i.e., of about 200 μm in diameter). A thermodynamic model that takes into account the statistical distribution of droplets and their temperature evolution is developed. To this aim, the Droplet Distribution Function (DDF) based on Rosin-Rammler's equation is used. In the proposed model, the thermal energy exchange between liquid and steam takes into account both convection and evaporation heat transfer mechanisms. A comparison of model's predictions with experimental data shows very good agreement for steam pressures of 1.6 and 2.1 *MPa*, however at higher pressures the model over predicts the experimental trends.

Keywords: *Direct contact heat exchanger, droplet statistics, nozzle spray, convection, evaporation.*

1. INTRODUCTION

Direct contact heat exchangers are extensively used in numerous power applications, among other in nuclear power stations, cooling towers, petroleum, thermal and chemical plants.[1,2] Due to their high performance to transport both energy and mass the heat transfer from liquid droplets has been studied by many researchers. Marshall[2] studied heat and mass transfer from a liquid spray to air during air-drying processes by including the effect of the size of droplets. In his modeling approach, the droplet evaporation rate is explicitly included in the energy balance equation. Using experimental data, Marshall has proposed a correlation for the Nusselt number (Nu) as function of Reynolds (Re) and Prandtl (Pr) numbers. Srinivas et al.[3,4] presented a model by writing the conservation equations of spherical droplets. Based on their modeling approach, Srinivas et al. estimated the behavior of key physical properties, i.e., surface tension σ, droplet surface velocity v, Nu and surface shear stress τ, as a function of time and droplet angular position. Celata et al.[5] studied the trend of droplet temperature

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established in a condensing steam environment. As initial conditions, they assumed the droplets at sub-cooled liquid condition and the vapor at saturation. Assuming conduction heat transfer inside the droplet, Celata et al. have calculated the spray-droplet mean temperature by solving the energy conservation equation obeying the aforementioned initial conditions. Furthermore, to take into account the effect of liquid circulation inside the droplet, they introduced a coefficient as a function of Péclet's number which permitted a better agreement of model's predictions with experimental data to be achieved. Takahashi et al. [1] have compared the predictions of Celata et al. model with their own experimental data. Thus, they were able to show that the model was not adequate to evaluate the liquid temperature for non-dimensional distances lower than 6 (the non-dimensional distance is defined as $X = x/D$, where x is the distance from nozzle and D is the droplet diameter).

When liquid is sprayed into a gas atmosphere, it should be expected that droplets will have different physical dimensions; therefore the use of a Droplet Distribution Function (DDF) is mandatory. Moreover, the DDF becomes a key function when condensation and/or evaporation occur, because these two processes affect their physical dimensions. Using statistical moments as a function of time to compute DDF and including droplet collisions and break-up mechanisms, Beck and Watkins [6] were able to determine the evolution of droplet sizes. A relatively simple way to compute the DDF consists of using droplet size distribution laws, as those proposed by Rosin-Rammler or Nukiyama-Tanasawa [7], in conjunction with empirical correlations to estimate factors required by these laws. One of these factors corresponds to the Sauter mean diameter (D_{32}) that is given as [8]:

$$D_{32} = \frac{\sum_{i=1}^{n} n_i D_i^3}{\sum_{i=1}^{n} n_i D_i^2} \qquad (1)$$

It represents a ratio of the entire volume (index 3) occupied by the droplets to their entire surface area (index 2). Several correlations are available in the open literature that allows D_{32} to be determined. Lefebvre [9] has proposed a correlation as function of the geometry of the nozzle and thermodynamic inlet conditions of the liquid to estimate D_{32}.

In this paper, a thermodynamic model has been developed to estimate the heat transfer rate in a DCHX currently in use at our laboratory. The main purpose of this thermal equipment, identified with a "Quenching Chamber" technical designation in Figure 1, is to cool superheated steam produced at the outlet of a test section used to perform choking flow experiments with water above supercritical conditions. As shown in the figure, cooling water is sprayed and mixed with steam coming from the test section. Since droplet sizes depend on the type of nozzle (i.e., its geometry) and the temperature of the injected water, we estimate the DDF using a methodology that is presented in the following section. Thereafter, the heat transfer from the steam to the liquid, both by convection and evaporation is introduced in the calculations. Finally, the predictions obtained applying the proposed approach are compared with experimental data collected under two system pressures during supercritical water experiments carried out in our laboratory.

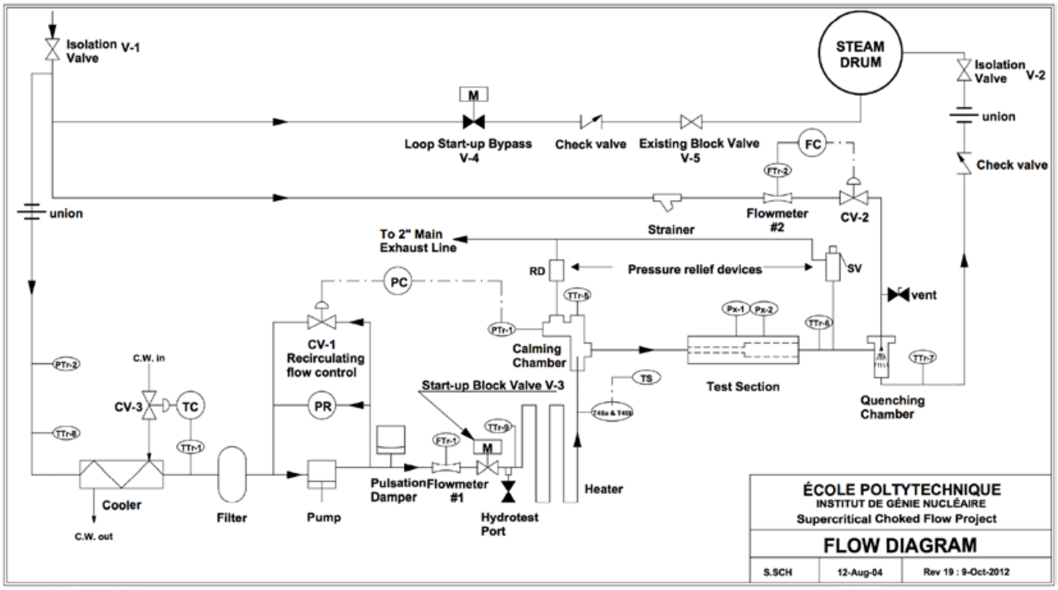

Figure 1. Portion of the flow diagram of the supercritical-water facility.

2. DROPLETS (Size) DISTRIBUTION FUNCTION

In order to predict heat transfer in DCHX's an appropriate estimation of droplet sizes is necessary; however, this evaluation is not an easy task. In fact, the droplet size distribution function depends on many factors, including the liquid temperature, the liquid flow rate and the geometrical characteristic of the nozzle [9-13]. To this purpose, several statistical laws are proposed in the open literature which provide distribution functions for liquid particles (e.g., log-normal, Nukiyama-Tanasawa, upper-limit, root-normal, etc.). One of the simplest laws available is the Rosin-Rammler cumulative distribution function, where the probability of having droplet volume fractions for droplets with diameters smaller than a given D is defined by [8]:

$$F(D) = 1 - \exp\left[-\frac{D}{D_{0.632}}\right]^q \tag{2}$$

where $D_{0.632}$ is a representative diameter for which 63.2% of the droplet population has diameters smaller than $D_{0.632}$ and q is a measure of the distribution width (i.e., skewness of the distribution function) . It is apparent that the application Equation (2) requires a previous computation of $D_{0.632}$ and q. Experimental studies performed by Lefebvre [9] have shown that for most types of sprays, the value of q varies between 2 and 2.8. Furthermore, Zhao et al. [10] have demonstrated that $D_{0.632}$ can be determined as a function of Sauter's mean diameter (Eq. 1) and q, as:

$$\frac{D_{0.632}}{D_{32}} = \Gamma\left[1 - \frac{1}{q}\right] \tag{3}$$

where Γ is the gamma function. Nevertheless, the use of Equation 3 necessitates the value of D_{32} given by Equation 1 which is not necessarily easy to handle. Therefore, in this work we have used an empirical correlation proposed by Ashgriz [8] who has presented all existing correlations for every type of nozzles, from air-blast to electrostatic ones, encountered in industrial applications. In particular, for a swirl nozzle similar to the type used in our experimental facility shown in Figure 1, we have selected a relationship developed by Lefebvre [9] that is given as:

$$D_{32} = 4.52\left(\frac{\sigma\,\mu_l^2}{\rho_v\,\Delta P_l^2}\right)^{0.25}(t\cos\theta)^{0.25} + 0.39\left(\frac{\sigma\,\rho_l}{\rho_v\,\Delta P_l}\right)^{0.25}(t\cos\theta)^{0.75} \tag{4}$$

where σ is the surface tension [N/m], μ_l is the dynamic viscosity of the liquid [Ns/m^2], ρ_l and ρ_v are the density of the liquid and the steam respectively [kg/m^3], ΔP_l is the pressure difference in the nozzle chamber [Pa], t is the sheet film-thickness outside of the discharge orifice [m], and θ is half the spray cone angle. The sheet film-thickness is calculated as [9]:

$$t = 2.7\left[\frac{d_o FN\,\mu_l}{\sqrt{\rho_l\,\Delta P_L}}\right]^{0.2} \tag{5}$$

where d_o is the discharge orifice diameter and FN is the flow number defined as:

$$FN = \frac{\dot{m}_l}{\sqrt{\rho_l\,\Delta P_l}} \tag{6}$$

with $\dot{m}_l$ the liquid flow rate given in kg/s. Finally, Zhao et al. [10] have shown that the lower and the upper limit of the distribution function can be estimated, respectively by:

$$D_{0.1} = D_{0.632}(0.1054)^{1/q} \tag{7}$$

and

$$D_{0.999} = D_{0.632}(6.9077)^{1/q} \tag{8}$$

In these equations, $D_{0.1}$ and $D_{0.999}$ represent values that produce 0.1 and 0.999, respectively when they are used in Equation (2).

Equation (4) shows that D_{32} depends both on thermodynamic properties and on geometric characteristics of the spray nozzle. Figure 2 compares the behavior of the DDF as a function of the working pressure (i.e., the steam pressure in the quenching chamber of Figure 1). As shown in Figure 2, the DDF significantly changes with pressure, even though the value of $q = 2.1$ is kept constant. The inlet water flow rate of the nozzle is also constant (= $2.25\text{x}10^{-5}$ m^3/s) while the temperature is 144.4^oC and 186.3^oC for 0.8 and 3.1 MPa, respectively. It is known that the sizes of the droplets decrease with increasing the working pressure, the mass flow rate as well as the liquid temperature [13]. Hence, it is apparent that the results shown in Figure 2 satisfy this general behavior.

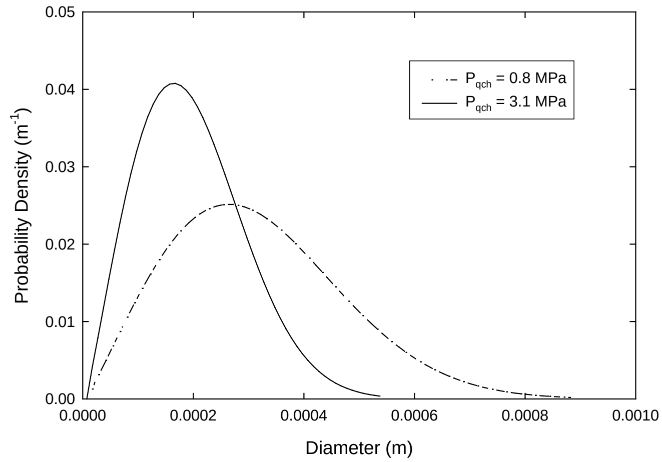

Figure 2. Behavior of the DDF with working pressure.

3. HEAT TRANSFER

To model the heat transfer between steam atmosphere and water droplets, it is assumed that the droplets are spherical and that the steam temperature does not change during the cooling process, i.e., its partial condensation affects the mass of steam in the "Quenching Chamber" of Figure 1 without changing its temperature. However, it must be pointed out that the temperature of the steam that enters into the Quenching Chamber (i.e., the DCHX) is not known *a priori*. However, the conditions of supercritical-water at test section inlet (see Figure 1) are accurately known. Therefore, the discharge across the test section should correspond to one of the following thermodynamic transformations:

- Constant enthalpy transformation or
- Constant entropy transformation.

Since the exact flow evolution is not known, it is assumed that the temperature of the steam at the outlet of the test section can be approached by an average value calculated as:

$$T_v = \frac{1}{2}\left(T_s + T_h\right) \tag{9}$$

T_S corresponds to the steam temperature for an isentropic evolution and T_h for an isenthalpic one. Thus, the knowledge of the initial water temperature, the steam temperature given by Equation (9) and the droplet size distribution from Equation (2) should make it possible to determine the heat transfer between the steam and the droplets. The overall procedure consists of applying Equations (7) and (8) to generate N droplets having diameters ranging from $D_{0.1}$ to $D_{0.999}$, hence:

$$N = \frac{D_{0.999} - D_{0.1}}{dD} \tag{10}$$

with $dD = 10^{-5}$ *m*. Thereafter, a temporal scale with a time step of to 10 *ms* is created in such a way that for each diameter and for each time step, properties such as the droplet size (i.e., when evaporation occurs), the drag coefficient, the droplet velocity, the residence time, the heat transfer coefficient, the temperature and the evaporation rate as a function of time are calculated. More details about the methodology is given in the following sections. It is important to mention, however, that possible effects associated to droplet collisions and/or droplet break-ups have been neglected. This hypothesis assumes that the number of droplets in the chamber is low enough; therefore, the probability of their mutual interaction is very low. Furthermore, their diameter is so small and the velocity of the steam is so low, that the surface tension overwhelms any probability of breaking them up.

3.1 DROPLET CONVECTION HEAT TRANSFER

The water that enters into the quenching chamber (Figure 1) is at sub-cooled state, while the steam may be at saturation or super-heated state; therefore, during the first interaction of droplets with the steam, heat is transferred to liquid only by convection. Let us assume that the temperature distribution inside a spherical droplet of radius R can be determined from a conduction heat transfer equation written in spherical coordinates as [11]:

$$\frac{\partial^2 T}{\partial r^2} + \frac{2}{r}\frac{\partial T}{\partial r} = \frac{1}{\alpha}\frac{\partial T}{\partial t} \tag{11}$$

where the temperature depends on the droplet radius r and on time t (i.e., the effects of others spatial coordinates are neglected). To solve this equation, the following boundary conditions are imposed:

- Uniform initial temperature distribution inside the droplet; $T(r, 0) = T_i$,
- Continuity of heat flux at the droplet surface; $-k_l \frac{\partial T}{\partial r} = h\left[T(R,t) - T_\infty\right]$,
- Symmetry at the center of the droplet; $\left.\frac{\partial T}{\partial r}\right|_{r=0} = 0.$

The convective heat transfer coefficient h is estimated using the Ranz and Marshall [2] correlation expressed as:

$$\mathrm{Nu} = 2 + 0.6 \times \mathrm{Re}_v^{1/2} \times \mathrm{Pr}^{1/3} \tag{12}$$

where Re_v is the steam Reynolds number based on an observer moving with respect to the steam at the velocity of the droplets. Thus, this equation requires a previous knowledge of the droplet velocity that is obtained by solving the following differential equation [8]:

$$\rho_l \frac{\pi D^3}{6}\frac{d}{dt}v_D = 3\pi\mu_v D\, f\left(v_v - v_D\right) + \rho_l \frac{\pi D^3}{6} g \tag{13}$$

where v_D is the droplet velocity, v_v is the velocity of the steam, f is a drag factor and g is the acceleration of gravity. As usual, the drag factor is calculated as a function of a drag coefficient C_D and the Reynolds number as:

$$f = \frac{C_D \,\mathrm{Re}_v}{24}$$

$$\begin{cases} C_D = \dfrac{1}{24}\mathrm{Re}_v\left(1+0.15\times\mathrm{Re}_v^{0.687}\right) & if \quad \mathrm{Re}_v \le 10^3 \\ C_D = 0.44 & if \quad \mathrm{Re}_v > 10^3 \end{cases} \tag{14}$$

Close to the nozzle, the Reynolds number Re_v is higher than 10^3.[6]

Knowing the boundary conditions, Equation (11) is now solved using the method of separation of variables. Schneider[12] provides the space-averaged solution of the heat transfer problem with the mean temperature $\overline{T}$ obtained from:

$$\overline{T} = T_\infty + \frac{(T_i - T_\infty)}{4/3\pi R^3}\int_0^R 4\pi r^2 T(r)dr \tag{15}$$

The complete solution of Equation (11) is expressed as[12]:

$$\frac{\overline{T} - T\infty}{T_i - T_\infty} = \sum_{n=1}^{\infty}\frac{6}{\varsigma_n^2}\frac{\left[\sin\varsigma_n - \varsigma_n\cos\varsigma_n\right]^2}{\left[\varsigma_n - \sin\varsigma_n\cos\varsigma_n\right]}\exp\left(-\varsigma_n^2\mathrm{Fo}\right) \tag{16}$$

where Fo is the Fourier number and ς_n are the roots of the following equation:

$$1 - \varsigma_n\cot\varsigma_n = \mathrm{Bi} \tag{17}$$

and Bi the Biot number. In this work we have used only the first five roots of this equation. These values were taken from the Schneider[12]. The heat transferred by convection at a given instant t_D can then be estimated as:

$$Q_{conv} = \int_0^R 4\pi r^2 \rho_l c_{p,l}\left[T(r,t_D) - T_i\right]dr \tag{18}$$

It is obvious that under some particular conditions the heat transfer problem can be simplified. For instance, when Fo > 0.2 the solution of Equation (11) can be approximated by considering only the first term of Equation (17); hence:

$$\frac{\overline{T} - T_\infty}{T_i - T_\infty} = \frac{6}{\varsigma_1^2}\frac{\left[\sin\varsigma_1 - \varsigma_1\cos\varsigma_1\right]^2}{\left[\varsigma_1 - \sin\varsigma_1\cos\varsigma_1\right]}\exp\left(-\varsigma_1^2\mathrm{Fo}\right) \tag{19}$$

with ς_1 the first root of Equation (17). In turn if Bi << 0.1 then the lumped parameter approach can be applied, this gives:

$$\frac{\overline{T} - T_\infty}{T_i - T_\infty} = \exp\left(-\frac{t}{\tau_T}\right) \tag{20}$$

with

$$\tau_T = \frac{4/3\pi R^3\rho_l c_{p,l}}{4\pi R^2 h} = \frac{1}{3}R\frac{\rho_l c_{p,l}}{h} \tag{21}$$

3.2 DROPLET CONDENSATION HEAT TRANSFER

When droplets reach saturation conditions, they start evaporating. Marshall [2] has studied the evaporation of sub-cooled droplets in a hot-air environment. Assuming that the heat transferred to a droplet from the steam by convection is used by the droplet to vaporize itself, then for a single droplet the following energy balance equation can be written:

$$\pi D^2 h\left[T_v - T_{sat}\right]dt = h_{fg}\, dm_{evap} \tag{22}$$

with h_{fg} the latent heat of vaporization. This equation allows the rate of mass transferred to the steam to be calculated as:

$$\dot{m}_{evap} = \frac{dm_{evap}}{dt} = \frac{\pi D^2 h\left[T_v - T_{sat}\right]}{h_{fg}} \tag{23}$$

It is apparent that this mass transfer plays an important role in the physics of the problem. In fact the value of $\dot{m}_{evap}$ provides a direct indication of the amount of thermal energy that is transferred along the phase change process taking place in the droplets. Equation (23) shows that $\dot{m}_{evap}$ depends on the diameter of the droplet that decreases with increasing the evaporation rate and the heat transfer coefficient, both are functions of time. Therefore, it cannot be assumed that the evaporation rate will be the same for each droplet at each time step.

3.3 THERMAL POWER TRANSFERRED IN DCHX SYSTEMS

For a droplet having known dimension and thermal conditions, the knowledge of the heat transfer between a single droplet and the steam permits the total power exchange to be estimated as follows:

$$\dot{Q}_{total} = \sum_{i=1}^{N} \dot{N}_i Q_i \tag{24}$$

where $\dot{Q}_{total}$ is the total thermal power exchanged in the DCHX, $\dot{N}_i$ is the rate of droplet population having a diameter D_i and Q_i the thermal energy transfer to droplets of diameter D_i . For a given droplet diameter, the droplet population rate is determined by:

$$\dot{N}_i = p_i\left(D_i\right)\dot{N}_{total} \tag{25}$$

with a probability density p_i function expressed as:

$$p_i\left(D_i\right) = \frac{N_i}{N_{total}} = \frac{\dot{N}_i\,\delta t}{\dot{N}_{total}\,\delta t} \tag{26}$$

It is obvious that this probability density function corresponds to the derivative of the cumulative probability given by Equation (2); it can be evaluated by assuming that under steady state conditions the total flow rate of droplets can be estimated by the following expression:

$$\dot{N}_{total} = \frac{6}{\pi}\frac{\dot{V}_{liq}}{D_m^3} \tag{27}$$

where D_m is the mean statistical diameter of the droplets. Then, from Equation (24) the total thermal power becomes:

$$\dot{Q}_{total} = \sum_{i=1}^{N} \dot{N}_i Q_i = \dot{N}_{total} \sum_{i=1}^{N} p_i(D_i) Q_i \qquad (28)$$

with N the number of droplets given by Equation (10).

4. COMPARISON OF THE PREDICTIONS OF THE MODEL WITH DATA

In this section the prediction obtained using the proposed modeling approach is compared with experimental data obtained using the facility shown in Figure 1. Data were collected for three values of steam pressure (P_{qch} = 1.6 *MPa*, P_{qch} = 2.1 *MPa* and P_{qch} = 3.1 *MPa*) over a wide range of cooling water flow rates. Figure 3 shows a comparison of the predictions of the model with data. These data do not take into account thermal losses to the environment thought the thermal isolation of the quenching chamber. In fact, these losses are estimated by considering both heat conduction across the thermal insulation of a steel cylinder and external and internal convective heat transfer resistances; it has been has been estimated to be less than 240 *W*.

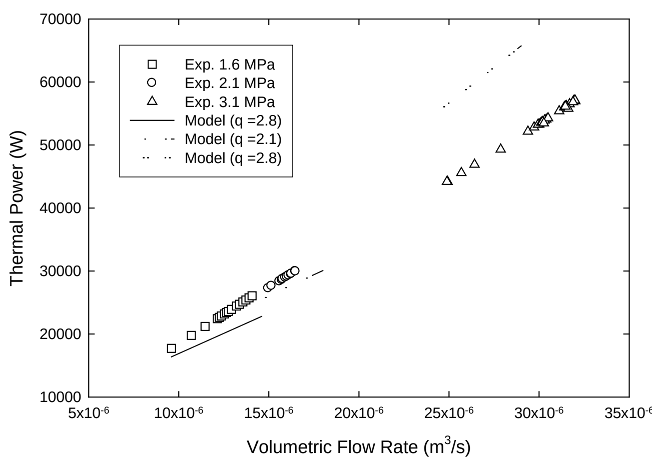

Figure 3. Comparison of predictions of the model with data.

The results presented in Figure 3 show that for low steam pressure (P_{qch} = 1.6 *MPa*, P_{qch} = 2.1 *MPa*), in general the agreement of the model with the data is very good. However, this is not necessarily true at high pressure. In fact, it seems that the proposed model significantly overestimate the thermal power. Several aspects, which are not taken into account in the present model, can probably explain this behavior. One of the possibly factors concerns the skewness of the DDF shown in Figure 2 (skewness that has been optimized in order to fit the data). In fact, this figure indicates that for a constant skewness coefficient $q = 2.1$, the pressure strongly affects the overall shape of the DDF. Therefore, the same calculations were repeated using three values of q ($q = 2.1$, $q = 2.4$ and $q = 2.8$,); the results are shown in Figures 4 to 6. In fact, for a constant value of Sauter's diameter D_{32} (Equation 1), $D_{0.632}$ increases with increasing q (Equation 3). Consequently, the probability of obtaining droplets with big diameters also increases (Equation 2). However, as shown in these figures, this is not a sufficient condition that can completely explain the discrepancy between the predictions obtained with the model. As expected from Figure 2, higher is the pressure the higher is the effect of *q* on the DDF; however, this behavior is not able to completely correct the predictions. These results show that the statistics of droplet size distribution is an important factor that has a strong control on heat transfer both at high and low steam pressures (Figure 4 to 6). Nevertheless, the influence of statistics on heat transfer changes with pressure. At low pressure, the heat transfer increases with increasing *q* (Figure 4 and 5); on the other hand, at high steam pressure, the heat transfer increases with decreasing *q*. This behavior may be caused by the physics of the droplets.

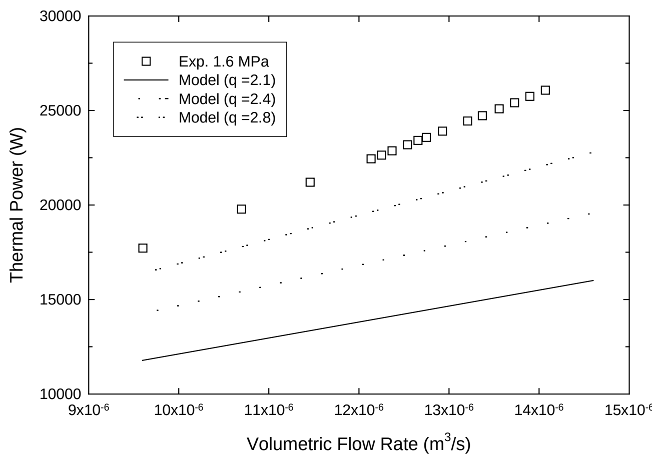

Figure 4. Heat transfer rate as a function of liquid volumetric flow rate and *q*.

In fact, the physical effects of droplet size on heat transfer are not easy to understand. For instance, if a comparison between the behavior of tiny and big droplets is performed, at first glance it can be argued that tiny droplets should evaporate faster than big ones. This is not necessarily true because the residence time of big droplets can be substantially reduced due to the increase of the Reynolds number and consequently the drag coefficient (Equation 14). Therefore, it is not obvious that small droplets will increase the overall performance of DCHX's. Therefore, to two outcomes are possible: 1) the best heat transfer should occur for a large population of droplets having diameters close to a mean statistical value; 2) there is an apparent trade-off between droplet population and size distributions (i.e., velocities) that can be responsible of the inversion effect observed on the heat transfer as a function of *q*.

The observed discrepancies cannot be necessarily related to the simplifying assumptions used to develop the model. The fact that droplet interactions and break-up are neglected must not affect its overall performance. The effect of surface tension increases with decreasing droplet surface temperature, which it can be assumed that decreases with decreasing the steam pressure, i.e., lower steam saturation temperature. Thus, an increase in the droplet surface temperature should increase the probability of break-up, which is necessary to obtain higher heat transfer rates, i.e., higher surface area and droplet population. It must be pointed out, that our model without taking into account droplet break-up mechanisms shows the right trends (i.e., higher heat transfer rate at high pressures and lower heat transfer rates at low pressures). Since the model over predicts heat transfer at high pressures, including a droplet break-up mechanism should further deteriorate its performance. Nevertheless, the fact that when pressure increases an inversion effect of *q* on the heat transfer is observed requires a further study about the statics that controls the droplet population distribution.

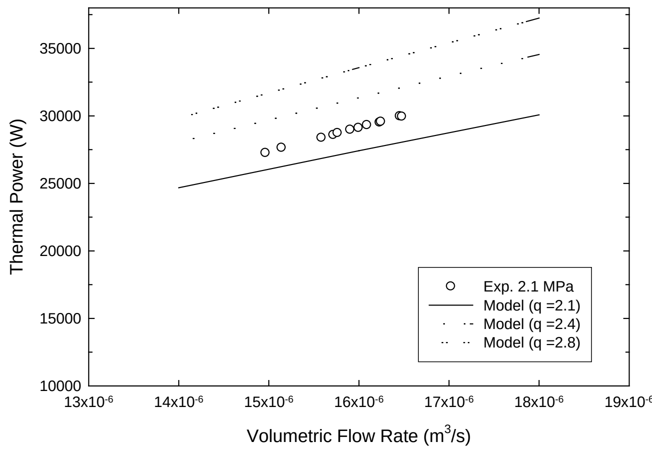

Figure 5. Heat transfer rate as a function of liquid volumetric flow rate and *q*.

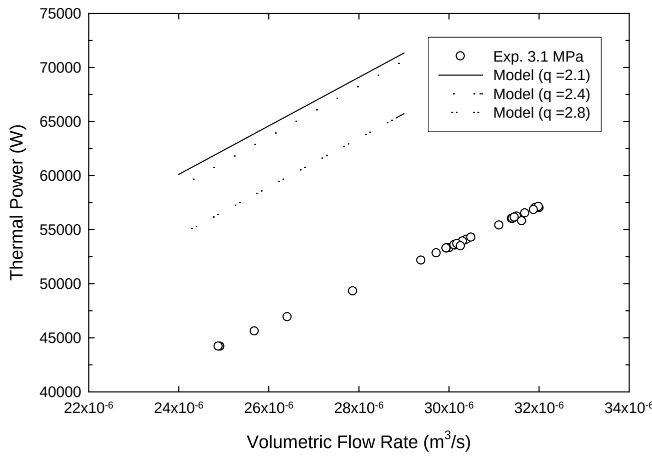

Figure 6. Heat transfer rate as a function of liquid volumetric flow rate and *q*.

5. CONCLUSION

A thermodynamic model used to estimate heat transfer in a DCHX has been proposed. The model takes into account the statistical distribution of droplet sizes and two heat transfer modes: convection and evaporation. The predictions of the proposed model are compared with experimental data collected in a quenching chamber of a supercritical water facility under operation at École Polytechnique de Montréal. In general, it has been observed that for low steam pressures ($P < 2.1$ MPa) a very good agreement between predictions and data are obtained. For high pressures ($P > 3.0$ MPa) the model over predicts the experimental values.

The analysis of the statistical distribution function used to estimate droplet diameters has shown that it cannot completely correct the observed trends. In fact, even though the skewness of the DDF is very sensitive to steam pressures, the shape of the distribution is unable to completely overcome the discrepancies observed at high pressures. In addition, it seems that taking into account some neglected physical phenomena such as droplet interaction and break-up cannot necessarily improve the model efficiency. Nevertheless, additional work is still required to explain the behavior of the model at high pressure. Despite the fact that at high steam pressures the performance of the model can be questionable, it provides us the possibility to analyze the heat transfer behavior of droplets having different diameters. In fact, it has clearly been determined that the overall heat transferred in this type of thermal unit is partially governed by mean-statistical size of the droplets. Therefore, a correct optimization of DCHX's must take into account the use of convenient criteria to choose the geometry of spray nozzles. It is apparent the best nozzle must produce the largest droplet population having almost uniform size distribution.

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