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Numerical Modelling and Thermal Analysis of Mist Jet Impingement in Laminar Regime

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Abstract

Jet impingement has been traditionally one of common thermal engineering solution for cooling and heating of solid surfaces, deployed in wide range of industrial applications. Flow and thermal characteristics of single-phase jet impingement is fairly well-known owing to an extensive analytical, experimental and numerical studies. Relying on convection, as the main heat transfer mechanism contributing in jet impingement, Nusselt number increases with jet velocity and additionally by turbulent eddies influence. Therefore, thermos-fluid engineers aim for increased heat transfer rate at higher Reynolds number, while higher flow and turbulence intensity corresponds to complication of manufacturing and operational requirements. Inclusion of latent heat in thermal exchange process is also applied as one of convection enhancement techniques. This phase changing thermal phenomenon introduces new streams of investigation to the field, including boiling and partial evaporative impinging jets. Air carrying liquid micro-droplets have been remarked as mist with enhanced cooling effect, originated by partial evaporation of liquid. This paper develops and utilises a CFD model of air-water mist impinging jet to investigate associated thermal enhancement originated by mist evaporation. Species transport model is examined as accurate and resource effective approach, for the continuous-dispersed thermos-fluid problem in laminar regime. Another key element of numerical model is the coupled heat and mass transfer scheme for evaporative scheme. Operating and characteristics parameters are set according to earlier experimental study, and validated by comparison of average convective heat transfer coefficient.

Introduction

Multi-phase techniques of thermal enhancement are widely investigated and successfully applied to various engineering designs, promoting capacity of cooling and heating devices. As the most applicable phase-change process, liquid-gas flow utilises additional latent heat during evaporation process. This could be a convective boiling process where liquid is the primary phase or spray/mist thermo-fluid process with liquid droplets considered as secondary dispersed phase. This classification of multi-phase thermal processes would assist to set requirements of experimental or numerical investigation. Mist jet impingement is the subject of interest for current study which should be remarked as mixture of air and water micro droplets, impinging and cooling a hot surface. Micro scale dimension of droplets and premixing process of droplets with air suggest two facts. First, the influence of dispersed phase could be accounted as phase averaged properties when particle concentration is low enough. Second, slip velocity between micro droplets and carrier phase (air) is negligible.

Terekhove and Pakhomov [1] developed one of the earliest numerical two-phase model of gas-vapour-drop mist flow over a flat iso-thermal plate. This study uses particle-source-in cell model (PSI-Cell) to simulate flow and thermal influence of particle presence while validating numerical results for various flow rate and mist percentage against previous experimental measurements. They concluded significantly higher Nusselt number by inclusion of, up to 4%, mist into flow. Terekhove and Pakhomov [2] accordingly extended their numerical model using a two-velocity Euler approximation and also increasing flow rate to a turbulent, air-droplet flow regime in a duct. Mist is considered as a low volume fraction ($\alpha_l < 10^{-4}$) and mean particle diameter in order of 100 μm where mean Reynolds number range was set to 10^4-10^6. This work was more toward improving knowledge of numerical modelling (e.g. heat transfer due to direct contacts of drops with the wall, the stochastic motion of drops, and non-isotropic turbulent fluctuations) rather than adding a new thermal conclusion to literature. Su et al. [3] conducted an experimental study of air-nitrogen mist impingement on a superheated surface with Reynolds number range of 3,000-20,000 and reported augmentation factors in range of 1.2–2.8 times, compare to air jet. They considered droplet momentum as a key parameter as atomizer nozzle was applied in this experiment. Accordingly, both flow and thermal characteristics are found affected by droplet significantly deviated from mean flow parameters of continuous phase. Garbero et al. [4] developed and utilised a numerical model for combined micro/macro scale analysis of velocity and temperature profiles near the surface exchanging heat with impinging micro droplets. Eulerian-Lagrangian scheme was used for macro scale case study to simulate nozzle-generated droplets and for parametric analysis. A VOF based model was additionally used for analysis of single and multiple impinging droplets cooling the surface. Mist droplets, investigated in this study, has diameter range of 25-200 μm and Reynolds was increased up to 2.75×10^5. Their results indicated 20-40% and 150% heat transfer improvement, for micro and macro scale models, respectively. Pakhomov and Tereshko [5] numerically studied a gas-droplet mist impingement in turbulent regime. The RANS based model, considered a low fraction (less than 2%) mixture as an Euler-Euler framework and implemented essential source terms to establish a two-way coupling between gas and dispersed phase. They also extended their investigation to a pulsed mist-jet and examined influence of pulsing frequency on heat transfer characteristics. Thermal enhancement up to 45% is reported by this work and best frequencies for various H/D ratios are identified. Mist enhanced convective heat transfer was analytically and numerically investigated by Kumari et al [6]. This work introduces implementation of droplet evaporation method for mist-assisted convective heat transfer problems. They monitored 23% of thermal enhancement by examining mist carrying droplet of size 10-50 μm and mist-air weight fraction of 1%. XiaoMing et al. [7] carried out an experimental study of mist jet with fixed and swirling configuration and measured thermal enhancement originated by additional evaporation effect. Utilising a mixing chamber, compressed air and water spray were separately supplied and accordingly outlet mixture was a mixt carrying 40 µm water droplet. Thermal measurement is carried out across heater where heat flux was assumed to be constant for each test (6699-7850 W/m^2). This experiment was one of few investigation studying influence of low percentage (<1%) water droplet and reports enhancement up to 3.5 times oriented from such low percentage of mist. Indicating enhancement potential of mist with low percentage, CFD investigation are motivated to simulate such phase-change thermo-fluid phenomenon. Elwekeel and Abdala [8] developed, validate and utilised a CFD model for assessment of mist-jet over range of parameters including Reynolds, mist fraction and jet shape. This model was developed for a turbulent jet and considered mist particles as dispersed phase simulated by DPM. They applied Schiller Naumann scheme for drag and accounted for both convective and evaporative mass transfer from droplets which resulted in a relatively expensive model (0.5-1.1 million cell with coupled DPM solver). Another CFD model, for investigation of low-fraction mist jet impingement, was developed by Pakhomov and Terekhov [9]. This model utilises a RANS closure where both continuous (air) and dispersed (droplets) phases have been simulated using Eulerian approach with two-way coupling interaction. This model was shown as a robust and valid model for complicated jet impingement flows, including pulsed flows.

Current paper proposes a new simple and robust CFD model which is optimised, in so many numerical aspects, and successfully applied on an axisymmetric solver. Assuming diffusive-convective behaviour of dispersed phase, a species transport model is deployed for solution of phase fraction across domain. This was primarily done to avoid use of empirical correlation for drag and heat transfer between separated Eulerian phases and later found drastically reducing demand on computation resources, compared to Euler-Euler and Euler-DPM models. Geometry and operating condition of current model has been configured according to an earlier experimental study, carried out by Nadim and Chandratilleke [10].

Problem definition

Typical geometry of round jet is investigated as mixture of air and micro-droplets of water (referred as mist) is the cooling fluid. Current study mainly emphasis on augmentation effect, originated from evaporation and would not carry out any geometrical parametric analysis. Hence, dimensions and ratios are maintained constant, as identified. Figure.1, illustrates schematic of mist jet arrangement which has been numerically simulated.

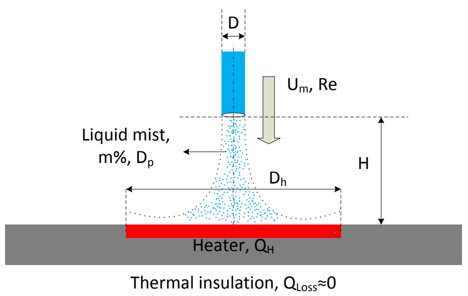


Figure 1. Schematic of investigated round jet and key dimensions

Flow rate is adjusted to ensure fully laminar flow (Re<400) where diameter of jet nozzle is 4.5 mm. Heater diameter (D_h) and Nozzle to heater space (H) are set at 45mm and 12mm respectively. Given dimensional ratios allow comparison of dry jet impingement case with earlier experimental measurements. Mist characteristics are key parameters of current study, determining mass transfer and accordingly evaporative heat transfer enhancement. Mist with micro droplet of water (D_p≈µm) and low mass fraction (F_m<0.5%) are investigated in case studies of this paper. These parameter are realistically adjusted based on an experiment [10] which utilised piezoelectric elements for mist generation in mixing chamber. Benchmarking the experimental analysis, mean particle diameter of 1 µm is set where mass fraction of water to air is varying between 0.1 - 0.25%.

Numerical model

Appropriate solver is chosen and modified to handle a multi-phase flow and heat transfer problem where coupled heat and mass exchange terms are the key element, determining thermal improvement. As mentioned earlier, DPM and Eulerian approach have been mostly applied for mist problem where simulation results are highly sensitive against incorporated momentum and heat exchange empirical schemes. Current model simulates mass fraction of water (in liquid and vapour form) and air using species transport equations. Such convective-diffusive scheme requires no drag scheme for phase coupling (here set as species) and suits current application since with given range of mass fraction, fluid may be assumed as mixture. Continuity and momentum equations are respectively incorporated as:

$$\nabla\left(\rho_m \vec{V}_m\right) = 0 \tag{1}$$

$$\nabla\left(\rho_m \vec{V}_m.\vec{V}_m\right) = -\nabla p + \bar{\bar{\tau}} + F \tag{2}$$

Wherein both equation mixture density is calculated as incompressible ideal gas. Stress tensor is included in Equation.2, as its general form which here is assigned for a laminar flow. Fraction for each specie (here air, water liquid and water vapour) is calculated through following equation having convective, diffusive and source terms as:

$$\nabla\left(\rho_i \vec{V}_m.Y_i\right) = -\nabla.\vec{J}_i + R_i \tag{3}$$

This equation calculates mass fraction (Y_i) of specie "i", advected by mixture velocity field and having diffusion flux of calculated as:

$$\vec{J}_i = -\rho_i D_{i.m} \nabla.Y_i - D_{i,T}\frac{\nabla T}{T} \tag{4}$$

$D_{i.m}$ and $D_{i.T}$ respectively represent mass and thermal diffusion coefficient, for specie.

Energy equation for mixture is determined as:

$$\nabla\left(\vec{V}_m(\rho E + p)_m\right) = -\nabla\left(k_{eff}\nabla T\right) - \nabla\left[\sum_j h_j \vec{J}_j\right] + S_h \tag{5}$$

Here, thermal properties are calculated based on mixture law and additional transport term is included to account for enthalpy change due to diffusion. This completes all the schemes required for thermo-fluid analysis of domain, except for phase change. The phase change phenomenon is evaporation which has to be included in species and energy equation as appropriate source terms (R_i, S_h).

Evaporative exchanges

Current model considers evaporative mass transfer rate as source terms to be included in liquid and vapours species. Knudsen-Hertz equation is applied, as a mechanistic scheme to quantify evaporative interfacial mass flux as:

$$\Gamma = \sqrt{\frac{M}{2\pi R T_{sat}}} \frac{h_{lv}(T - T_{sat})}{T_{sat}\left(\frac{1}{\rho_v} - \frac{1}{\rho_l}\right)} \qquad (6)$$

This enables possibility of local and accurate estimation of evaporation rate, using thermal condition and material properties (M: molecular weight, h_{lv}: latent heat, $\rho_{l,v}$: density of phases). Mass transfer source term for liquid and vapour species will be then:

$$\dot{m}_{lv} = -\dot{m}_{vl} = \Gamma I_{AC} \qquad (7)$$

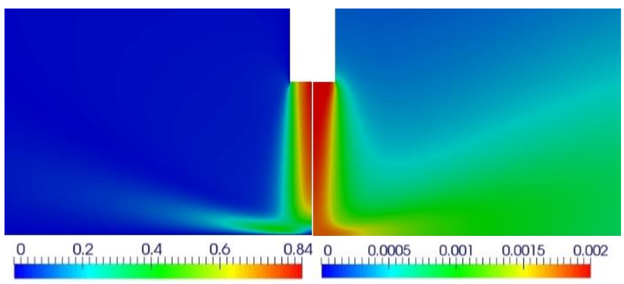
Where I_{AC} is interfacial area concentration and will be locally calculated with reference to particle diameter assuming droplets having spherical shape. With a known mass transfer rate and constant latent heat (almost uniform pressure across domain), energy source term is finally calculated as $S_h = -\dot{m}_{lv} h_{lv}$.

Domain and Solver Setup

The primary and well-known concern, in establishing geometry for computational domain of jet impingement, is to avoid outlet boundary conditions (Neumann or Dirichlet) imposing false mathematical stream on flow domain. Carrying out sensitive analysis, length and height of domain are respectively set as 4.5 time of heater radius ($D_h/2$) and 3.5 times of offset distance (H). Applying an axisymmetric solver, a 2D grid (19286 cells / grid step size of 10^{-4}-1.2×10^{-3} m). The course mesh toward outlets gets more refined as approaching toward area of interest and maximum refinement is applied on fluid cell adjacent to heater, with wall cell distance in the range of 2.54-2.6×10^{-5} m. Applying a coupled solver with flow Courant number of 10, stability of solution is found to be improved by adjustment of liquid water and energy equation. However, due to low mass fraction of water ($F_{lw\text{-}max}$=0.002) applying under-relaxation factor to liquid water or energy equation could undermine evaporation rate while solution is converged. Fully developed velocity profile with 0, 0.001 and 0.002 mass fraction of water is applied as inlet B.C and heater was given iso-thermal conditions. Outlets are assigned constant pressure adjusted as 1 bar.

Overview of Thermo-Fluid Results

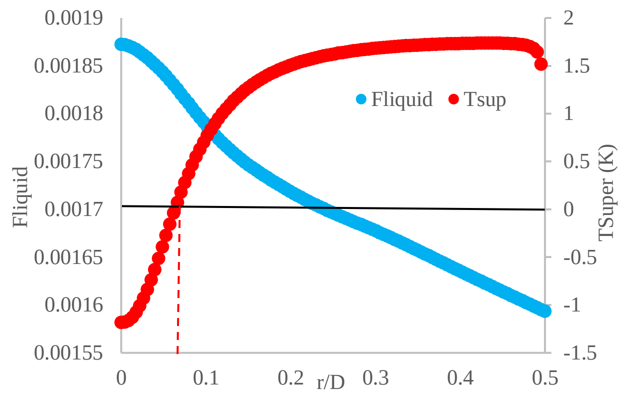
Figure.2 represents contours of velocity magnitude and mist (liquid water droplets) fraction for a laminar a case with slightly superheated heater. Inlet temperature is maintained constant at 300 K and saturation temperature is assumed to be 373.15K in atmospheric pressure.



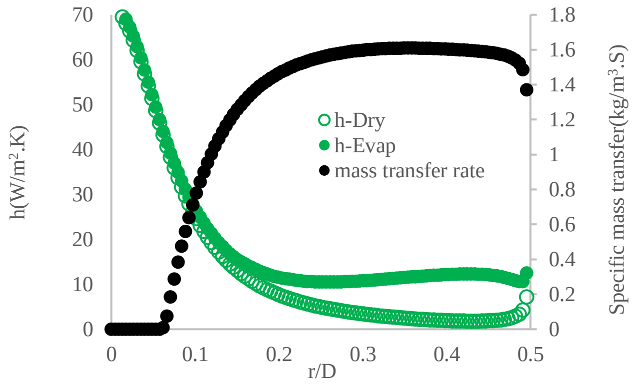
(a) Velocity (m/s) (b) Mist fraction (-)

Figure 2. Field overview; Re=177, F_{wl}=0.2%, T_{Sup}=1.85 K

Maximum mist fraction of 0.2% at inlet is slightly dispersed as enters exits from nozzle and approaches heater. Therefore, evaporation rate and accordingly resulted thermal enhancement (compared to dry identical case) will be factor of superheated degree (T_{sup}=T-T_{sat}) of fluid and available mist for evaporation. Figure.3 shows process of evaporation for the same review case, illustrated in Figure.1. Mist fraction which is 0.002 at inlet is affected by dispersing and reached to 0.00187 as jet flow impinges the stagnation point; however, fluid temperature, adjacent to heater is lower than saturated temperature and hence evaporation process is yet to be triggered.



(a) Mist fraction and fluid temperature at fluid vicinity adjacent to heater



(b) Evaporation rate and thermal enhance

Figure 3. Evaporation process and key parameters in heater area

Location at which fluid temperature reaches to saturated temperature is marked with a dashed line, in Figure.3-a. Associating this threshold with incipient of mass transfer (indicated in Figure.3-b) explains effectiveness of mist presence at temperature range higher than saturation.

Heat transfer coefficient of mist jet begins to deviate, from dry case values, after evaporation incipient point and as a results, higher average heat transfer coefficient is measured for mist cases.

	h_{Avg}(W/m^2.K)			Enhancement (%)	
Re	**Dry**	**F_{lw}=0.1%**	**F_{lw} =0.2%**	**F_{lw} =0.1%**	**F_{lw} =0.2%**
44	4.9	8.2	10.4	70%	110%
88	9.3	13.5	15.9	40%	70%
132	15	18	20.5	20%	40%
177	20.5	23.8	24.8	20%	20%
219	27.4	28.1	28.9	0%	10%
400	44.6	44.9	45.2	0%	0%

Table 1. Average value of heat transfer coefficient for dry and mist jet impingement, varying Reynolds, T_{sup}=1.85 K

Ratio of mist to dry cooling heat transfer coefficient (thermal enhancement) would be higher for lower flow rates, as superheat degree and mist percentage is constant. Table.1 tabulates values of heat transfer coefficient and thermal enhancements for various Reynolds numbers in the range of laminar flow. Given small values of mist presence are found effective on heat transfer improvement, even for small degree of heater superheat. Value of mist fraction applied in current work is reported to be achieved by ultrasound mist generator arrangement [10] and could be considered as realistic value for potential designs. Isothermal boundary condition is adjusted to capture effectiveness of evaporation mechanism, yet, this condition is practically unachievable and neither temperature nor does heat flux remain constant radially. As indication of heating intensity heater temperature would be the second parameter of interest for current thermal analysis. Figure.4 compares thermal effectiveness of dry and mist jet impingement where heating intensity is varied. This indicates almost non-sensitive thermal response of dry jet cooling against heating intensity whereas in mist evaporation rate and according heat transfer coefficient is increased significantly as surface temperature increases.

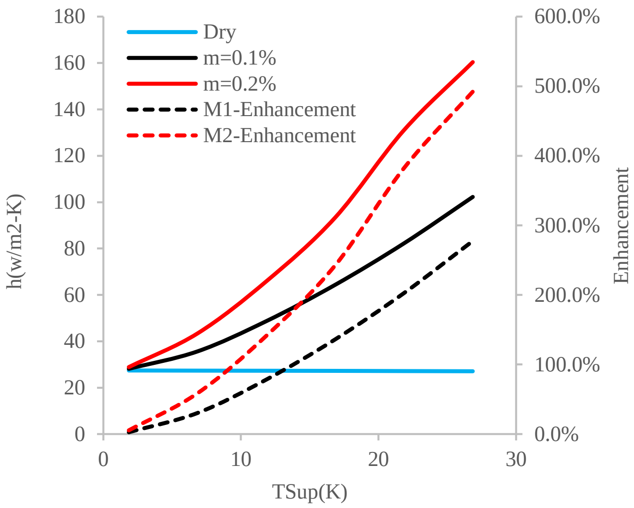


Figure 4. Heat transfer coefficient of dry and mist jet impingement at Re=400

This evaluation is carried out at highest laminar Reynolds number (Re=400) which has nearly identical thermal performance for dry and mist cases, just above saturation point as noted in Table.1 Nevertheless, effectiveness of additional latent heat removal owing to evaporation becomes more significant and an obvious gap between heat transfer coefficient is developed in higher superheat degrees. Such enhancement results in around 3 and 5 times higher heat transfer coefficients respectively for 1% and 2% of water droplet mass fraction, compared to dry jet values.

Conclusions

Numerical simulation of a laminar mist jet impingement is successfully developed and applied for investigation of thermal enhancement owing to latent heat based evaporation. Suggested CFD model successfully deploys species transport scheme for water micro-droplets, with dimensions in order of micron and mass fraction less than 1%. A mechanistic coupled heat and mass transfer scheme is incorporated for evaporation process, taking saturation properties, droplet characteristics and local thermal conditions into account. Results primarily reveal effectiveness of mist presence even for suggested small fraction of mist. The earlier reported experimental conclusion, is mechanistically investigated by evaluation of local evaporation rate and resulted augmentation factor. A brief and basic parameter analysis is also carried out to mark key influential parameters of phenomenon and highlights future optimisation approaches for mist thermal enhancement.

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