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Analysis of Evaporating Mist Flow for Enhanced Convective Heat Transfer

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ABSTRACT

Enhancement of forced convective heat transport through the use of evaporating mist flow is investigated analytically and by numerical simulation. A two-phase mist, consisting of finely dispersed water droplets in an airstream, is introduced at the inlet of a longitudinally-finned heat sink. The latent heat absorbed by the evaporating droplets significantly reduces the sensible heating of the air inside the heat sink which translates into higher heat-dissipation capacities. The flow and heat transfer characteristics of mist flows are studied through a detailed numerical analysis of the mass, momentum and energy transport equations for the mist droplets and the airstream, which are treated as two separate phases. The coupling between the two phases is modeled through interaction terms in the transport equations. The effects of inlet mist droplet size and concentration on the thermal performance of the heat sink are analyzed parametrically. The results provide insight into the complex transport processes associated with mist flows. The simulations indicate that significantly higher heat transfer coefficients are obtained with mist flows as compared to air flows, highlighting the potential for the use of mist flows for enhanced thermal management applications.

Keywords: Mist flow, two-phase flow, thermal management, heat transfer enhancement

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NOMENCLATURE

c_p	specific heat capacity, $J/(kg \cdot K)$
dV	numerical cell volume, m^3
h	heat transfer coefficient, $W/(m^2 \cdot K)$
h_{fg}	latent heat of evaporation, J/kg
h_m	mass transfer coefficient, m/s
k	thermal conductivity, $W/(m \cdot K)$
m_p	mass of single droplet, kg
$\dot{m}_p$	mass flow rate of the droplets, kg/s
m_v	mass fraction of vapor
n	number of moles
p	pressure, Pa
q''	heat flux applied to heat sink base, W/m^2
$\dot{q}'''$	volumetric heat source, W/m^3
t	time, s
u	velocity, m/s
x	axial distance along heat sink, m
y	distance along channel height, m
z	distance along channel width, m
A_p	droplet surface area, m^2
C_D	drag coefficient
D_p	droplet diameter, m
D_{AB}	mass diffusion coefficient, m^2/s
F	force, N
H	heat sink base plate thickness, m
H_{ch}	height of air channel, m

L	length of heat sink, m
Nu	Nusselt number
Re_D	droplet relative Reynolds number
RH	relative humidity
Sc	Schmidt number
Sh	Sherwood number
S_m	volumetric mass source, $kg/m^3 \cdot s$
T	temperature, $^{\circ}C$
W_{ch}	width of air channel, m
We	Weber number
W_f	width of heat sink fin, m

Greek symbols

μ	dynamic viscosity, $Pa \cdot s$
ρ	mass density, kg/m^3
τ	shear stress, Pa

Subscripts

0	inlet of heat sink
1	outlet of heat sink
ch	channel
f	fin
l	misting liquid
s	droplet surface
v	evaporated vapor
P	discrete phase/droplet

INTRODUCTION

Convective heat transfer to air or liquid flows is the most widely utilized approach for heat removal from heat sinks and heat exchangers in a variety of applications. In particular, air-based cooling technologies have been widely employed for thermal management of electronics. Such air-based cooling solutions have been developed for the thermal management of components with length scales ranging from micrometers (*e.g.*, hot spots on microprocessors) to several meters (data centers) [1]. The waste heat densities at all length scales have been steadily rising in the past decade, a result of faster devices and more densely-packed heat-generation sources at all length scales. Air-cooling technologies in their current form are not adequate to manage the increasing heat loads in next-generation microelectronics systems. This has resulted in significant research in developing novel technologies for enhanced heat-dissipation capabilities.

The use of a liquid as a medium to dissipate waste heat offers significant potential for high-heat-flux thermal management owing to the superior thermal properties of liquids relative to air. Of particular importance are phase change-based liquid cooling techniques, which may dissipate even larger heat fluxes than single-phase liquid cooling approaches. For instance, liquid spray cooling has received significant research attention as a high-heat-flux thermal management technology [2-7]. Mist impingement cooling is a variant of spray cooling wherein a two-phase mixture of finely dispersed liquid droplets in air is sprayed onto a hot surface [8-14]. It should be noted that the high heat dissipation capacity of spray cooling-based technologies is primarily a result of direct contact of the cooling fluid with the hot surface and subsequent evaporation of the liquid at the surface. A practical consideration associated with the use of spray cooling is the cumbersome piping and plumbing needed to direct the fluid to the hot component (and the evaporated fluid back to the condenser) which may not always be feasible.

Bahadur *et al.* [15] introduced a novel concept of a recirculating, two-phase, evaporating-condensing mist flow for high-heat-flux thermal management. The cooling scheme introduces finely dispersed liquid droplets into the airstream (subsequently referred to as *mist*) entering a channel populated with electronic components as shown in Figure 1. Alternatively, the mist could be selectively introduced close to a heat-generating component instead of, or in addition to, at the channel inlet. In any case, droplets evaporate as they flow downstream and absorb heat from the surrounding air. The heat-absorption capacity of the mist stream is increased relative to that of the (pure) air-stream which reduces the streamwise temperature gradient. At the downstream end of the channel, the hot air and the vapor (resulting from droplet evaporation) pass through a heat exchanger (*e.g.*, a finned bank of heat pipes coupled to external cooling water) which causes the vapor to condense. This condensate and the cooled air are recirculated back to

the channel inlet. It is important to note that this mist cooling scheme does not rely on heat removal due to direct contact of the liquid droplets with the heated surface. Mist cooling is thus fundamentally different from conventional spray cooling-based technology, since the latter relies upon impingement and evaporation of the liquid droplets on the heated surface.

Modeling mist flow and heat transfer presents several challenges owing to the coupling between the mass, momentum and energy transport equations for the continuous (gas + vapor) and discrete (droplets) phases. Bahadur *et al.* [15] developed first-principles-based analytical models to arrive at estimates of key operational and performance parameters for mist cooling systems. A major focus of that work was to develop first-order models to simulate the physics governing droplet evaporation in such systems. The optimum droplet diameter to prevent dryout, the maximum allowable inlet mist concentration (to prevent suppression of droplet evaporation due to saturation of the airstream), and the heat transfer and pressure drop associated with mist flow were calculated.

Limited studies are available in the literature on numerical modeling of mist-based systems. Terekhov and Pakhomov [16] used a finite-difference scheme to simulate heat transfer in laminar mist flow over an isothermal flat plate. The droplets were modeled as internal sources of mass, momentum and energy in the gas phase by using a ‘particle source in cell’ model. A similar modeling effort by Yao and Rane [13] studied the effect of droplet evaporation in two-phase flows by considering the droplets as heat sinks in the energy equation. A preliminary numerical study of mist flows was conducted by Kumari *et al.* [17] in which the liquid droplets and the air were treated as two separate phases. The two phases were modeled separately, with additional terms in conservation equations accounting for the phase-interactions. This approach of modeling a mist flow using two separate phases has also been utilized by Li and Wang [18] and Adiga *et al.* [19]. An alternative approach, referred to as the Euler/Euler approach, treats both phases as two inter-penetrating continua and was used by, for example, Groll *et al.* [20], to examine droplet evaporation in a turbulent mist flow through a duct.

The present work further develops and refines the numerical study of Kumari *et al.* [17] to achieve a detailed understanding of the flow and thermal performance of mist cooling systems. The ‘discrete-phase’ model in the commercially-available CFD software package FLUENT [21] is used to analyze mist flow through a copper-finned heat sink. Parametric analyses are carried out to study the effect of the inlet droplet diameter and the inlet concentration of the misting fluid on the heat dissipation capacity of the heat sink. The thermal characteristics of mist cooling are quantified in terms of the thermal resistance and heat transfer coefficients associated with mist flow; the corresponding performance under air flow (no mist) is also presented for comparison. The results from the modeling effort provide significant insight into the complex transport phenomena associated with mist flows. The heat dissipation estimates

from the simulations show that mist cooling offers substantial promise for enhanced heat dissipation as compared to conventional air cooling.

Although the present work analyzes mist flow through a heat sink, the modeling approach and framework developed in the present work can be used for the analysis of a wide range of mist-based systems such as telecommunications cabinets and data center cooling systems [17], gas turbine cooling [18], and fire suppression systems [19].

DROPLET EVAPORATION MODEL

This section briefly describes an elementary droplet evaporation model for flow through a finned heat sink, which serves as a starting point for the analysis of mist-based systems. The heat sink considered is shown in Figure 2 and has a footprint of 32 mm by 32 mm. The single-droplet Lagrangian model as developed in [15] analyzes the evaporation of droplets as they flow through the heat sink. It assumes that the Sherwood number characterizing mass transfer between the droplet and continuous phase is constant and equal to 2, its value for steady-state, diffusive mass transfer in an infinite medium. Additionally, the mass transfer driving force is assumed constant and equal to the initial water vapor concentration difference between the droplet surface and the ambient. The domain of interest is the unit cell of air associated with a single droplet, with the droplet vapor assumed to remain in the unit cell after evaporation. The evaporation-induced diameter decrease of a mist droplet in this diffusion-limited mass transfer regime is modeled as:

$$D_p^2 = D_{p,0}^2 - \frac{8D_{AB}(\rho_{v,s} - \rho_{v,\infty})}{\rho_l} t \quad (1)$$

where D_p is the droplet diameter, $D_{p,0}$ is the initial droplet diameter, D_{AB} is mass diffusion coefficient between the air and water vapor, ρ_l is the water density, and $\rho_{v,s}$ and $\rho_{v,\infty}$ are the vapor densities at the droplet surface and at infinity, respectively. This expression predicts that the square of the droplet diameter decays linearly with time and is subsequently compared with the numerical results of the present study. Equation (1) can be used to estimate the time required by a droplet to completely evaporate. For example, in a zero humidity environment ($\rho_{v,\infty} = 0$), t_{evap} is given by:

$$t_{evap} = \frac{D_{p,0}^2 \rho_l}{8D_{AB} \rho_{v,s}} \quad (2)$$

Equation (2) estimates the critical inlet droplet diameter which will ensure complete evaporation of the droplet at the downstream end of the heat sink. This critical diameter depends on the residence time of the droplet inside the heat sink, which is determined by the length of the heat sink and the droplet velocity. It is important to size the droplets at the inlet appropriately; droplets smaller than the critical

diameter will evaporate completely before reaching the downstream end of the heat sink, leading to dryout and loss of evaporative cooling capacity.

NUMERICAL MODELING OF MIST FLOWS

The modeling of the fluid and thermal phenomena in mist cooling involves analysis of the complex transport processes associated with a two-phase, two-component system. The flow stream at any location (upstream of the inlet to the domain) consists of three mass species, *i.e.*, air, liquid droplets, and the vapor phase corresponding to the evaporated droplets. Furthermore, the composition of the flow stream changes as the droplets evaporate, which also results in flow acceleration. The mass, momentum, energy and species equations are coupled due to the evaporation process and temperature-dependent physical properties such as viscosity along with interaction terms between the continuous and discrete phases.

In the present work, the transport equations for mist flows are solved using the ‘discrete-phase’ model formulation in FLUENT [21]. This modeling approach treats the air-water vapor mixture as a continuous phase and the droplets as a separate and discrete phase. The computational scheme involves the solution of the time-averaged transport equations for the continuous phase (air); the transport equations contain additional volumetric mass source and heat sink terms due to the existence of the discrete phase. The trajectory of the droplets (discrete phase) is calculated using a Lagrangian formulation. The interaction between the continuous phase and the discrete phase is captured through mass, momentum and energy exchange terms.

The transport equations for mass, momentum, energy, and water vapor transport, respectively, in the continuous phase can be represented as:

$$\frac{\partial}{\partial x_i}(\rho u_i) = S_m \quad (3)$$

$$\frac{\partial}{\partial x_i}(\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial \tau_{ij}}{\partial x_j} + F_i \quad (4)$$

$$\frac{\partial}{\partial x_i}(\rho c_p u_i T) = \frac{\partial}{\partial x_i} \left(k_i \frac{\partial T}{\partial x_i} \right) + \dot{q}''' \quad (5)$$

$$\frac{\partial}{\partial x_i}(\rho u_i m_v) = \frac{\partial}{\partial x_i} \left(\rho D_{AB} \frac{\partial m_v}{\partial x_i} \right) + S_m \quad (6)$$

where ρ is the density, u is the velocity, p is the pressure, τ is the extra stress tensor, c_p is the heat capacity, k is the thermal conductivity, T is the temperature, m_v is the water vapor mass fraction, and S_m , F

and $\dot{q}'''$ are source terms of mass, momentum and energy that account for interactions between the continuous and the discrete phases; these terms are discussed further below. The governing equations for the droplets are:

$$\frac{dm_P}{dt} = -h_m(\rho_{v,s} - \rho_v)A_P \quad (7)$$

$$\frac{du_P}{dt} = \frac{18\mu}{\rho_P D_P^2} \frac{C_D Re_D}{24} (u - u_P) \quad (8)$$

$$m_P c_{p,P} \frac{dT_P}{dt} = h A_P (T - T_P) + h_{fg} \frac{dm_P}{dt} \quad (9)$$

where subscript P represents the droplet, ρ_v is the average vapor density in the control volume associated with the particle, A_P is the surface area of the droplet, D_P is the droplet diameter, C_D is the drag coefficient and T is the local continuous-phase temperature. The momentum of the droplet is affected by the drag force acting on the droplet due to the relative motion between the droplet and the air. The energy equation of the droplet assumes no spatial temperature distribution within the droplet; the droplet energy is affected by the convective heat transfer and the evaporation processes. The relative Reynolds number Re_D in Equation (8) is defined as:

$$Re_D = \frac{\rho D_P |u_P - u|}{\mu} \quad (10)$$

The following correlations [22-24] are used to calculate the mass transfer coefficient h_m and the heat transfer coefficient h :

$$Sh = \frac{h_m D_P}{D_{AB}} = 2.0 + 0.6 Re_D^{0.5} Sc^{1/3} \quad (11)$$

$$Nu = \frac{h D_P}{k} = 2.0 + 0.6 Re_D^{0.5} Pr^{1/3} \quad (12)$$

The above equations for the Sherwood and Nusselt numbers (developed for steady-state conditions for a droplet surrounded by an infinite medium) account for diffusive as well as convective processes, which govern the mass and heat transfer between the droplet and the continuous phase. In the present work, the transport process is predominantly diffusive, and the second terms on the right hand side of Equations (11) and (12) have only a small contribution.

The mass, momentum and energy source/sink terms appearing in Equations (3)-(6) account for the interactions between the continuous and the discrete phases. These terms are calculated using the following equations:

$$S_m = \frac{\Delta m_p}{m_{p0}} \frac{\dot{m}_{p0}}{dV} \quad (13)$$

$$F_i = \sum \left(\frac{18\mu C_D \text{Re}_D}{24\rho_p D_p^2} (u_{p,i} - u_i) \right) \dot{m}_p \Delta t \quad (14)$$

$$\dot{q}''' = \left[\frac{\bar{m}_p}{m_{p0}} c_{p,p} \Delta T_p + \frac{\Delta m_p}{m_{p0}} \left(-h_{fg} + \int_{T_{ref}}^{T_p} c_{p,v} dT \right) \right] \frac{\dot{m}_{p0}}{dV} \quad (15)$$

where $\dot{m}_p$ is the mass flow rate of the droplets, Δm_p is the total evaporated water vapor mass in the control volume during one time step, $\bar{m}_p$ is the average droplet mass in the control volume, m_{p0} is the initial droplet mass (at the inlet), and dV is the volume of a numerical cell. The energy exchange term includes the sensible heating of the droplet, the latent heat required for the evaporation process and sensible heating of the water vapor generated in the evaporation process.

The flow problem is completed by specifying the computational domain and associated boundary conditions. Figure 3 shows the numerical domain for the mist-cooling simulations. The selection of the domain comprising half of a single channel and half of a heat sink fin as shown in Figure 3(a), is prompted by symmetry considerations. The dimensions of the entire heat sink are specified in Figure 2; the length L is 32 mm, the fin width W_f is 0.5 mm, the channel width W_{ch} is 2.65 mm, the thickness H of the heat sink base is 2 mm, and the channel height H_{ch} is 13 mm. The inlet of the channel is specified as plug flow with a velocity of 1 m/s and a temperature of 25 °C. The mist droplets are injected uniformly at the channel inlet, and the droplet diameter and mass flow rate are specified. As the flow in the channel is laminar ($\text{Re} \approx 300$), there is little mixing. Consequently, the droplets are expected to flow along the channel without impinging on the channel walls; the interaction between droplets and the channel walls is therefore not included in the model. Moreover, the maximum droplet Weber number We (based on the relative velocity of the droplet with respect to the air), which is a ratio of inertial to surface tension forces, is approximately 0.12; hence, droplet breakup is neglected [25]. The exit of the channel is set as a pressure outlet. The top of the channel is adiabatic with a no-slip velocity condition while the bottom wall of the heat sink corresponds to a uniform-heat-flux boundary condition. The left and right boundaries are assigned symmetry conditions as shown in Figure 3(b).

The equations and associated boundary conditions for the continuous and discrete phases are solved in a two-step iterative process. In the first step, Equations (3)-(6) for the continuous phase are solved to calculate the flow velocity and temperature, using the mass-weighted average material properties of air and water vapor. The material properties utilized for air are: density $\rho = 1.225 \text{ kg/m}^3$, specific heat $c_p =$

1006.4 J/kg-K, thermal conductivity $k = 0.0242$ (W/m-K) and viscosity $\mu = 1.79 \times 10^{-5}$ kg/m-s. The material properties for water vapor are: $\rho = 0.5542$ kg/m³, $c_p = 2014$ J/kg-K, $k = 0.0261$ W/m-K, and $\mu = 1.34 \times 10^{-5}$ kg/m-s. These material properties are assumed to be independent of temperature. In the second step, the results of the computations for the continuous phase are then utilized as inputs to solve Equations (7)-(9) for the discrete phase. The saturation vapor density in Equation (7) depends on the saturation vapor pressure and temperature. The saturation vapor pressure varies with temperature; in the present work, the data provided by Incropera and DeWitt [26] from 0°C – 100 °C is utilized. The mass, momentum and energy source/sink terms are then updated using Equations (13)-(15) and employed in the next iteration. These iterations are repeated until a steady-state solution is reached.

A uniform heat load of 10 W is applied to the base of the heat sink as shown in Figure 3(b), which corresponds to a heat flux q'' of 9766 W/m². A major objective of the simulations is to quantify the additional reduction in the heat sink temperature upon replacing airflow with a mist flow. The influence of the inlet droplet diameter and the inlet mist loading fraction on the thermal performance of the system is quantified through a parametric analysis. Four cases corresponding to different combinations of two inlet droplet sizes and two inlet mist loading fractions were simulated as shown in Table 1. The two inlet droplet sizes selected for this parametric analysis were 10 μm and 50 μm ; these correspond to typical droplet sizes which can be obtained from commercially available mist-generation systems. The two inlet mist loading fractions selected for the parametric analysis were 1% and 10% of the air mass flow rate. The corresponding mass flow rates of (air, water) through the heat sink are $(4.22 \times 10^{-4}, 4.22 \times 10^{-6})$ kg/s and $(4.22 \times 10^{-4}, 4.22 \times 10^{-5})$ kg/s, respectively. The cooling enhancement upon the use of mist cooling was quantified by comparison with a baseline case in which the heat sink is air-cooled (inlet air velocity = 1 m/s).

The computational domain used for all the simulations consisted of 150,000 cells. A mesh-independence study was conducted with three resolutions for both air-cooling and mist-cooling cases. For the air-cooled heat sink, the thermal resistance changed by approximately 3.5% and 2.1% when the number of cells were increased from 115,000 to 150,000, and from 150,000 to 260,000, respectively; the corresponding changes in resistance for the mist-cooled heat sink were approximately 4.3% and 3.0%. Consequently, all the simulation results presented in this work were obtained for a domain consisting of 150,000 cells.

RESULTS AND DISCUSSION

Figure 4 compares the velocity contours obtained with air flow (baseline case) and with mist flow consisting of $10\ \mu\text{m}$ diameter water droplets at an inlet mist loading fraction of 1% by weight (case I in Table 1). Figure 4(a) and (c) show the velocity contours on the channel symmetry and top planes for the air flow and mist flow cases, respectively. Figure 4(b) and (d) show the velocity contour on the yz plane at five equidistant axial locations for these cases. To facilitate quantitative comparison, velocity profiles along the channel height for two different transverse positions, $z = W_{ch}/4$ and $W_{ch}/2$, at axial locations of $L/2$ and L are shown in Figure 5 for the air and mist flow cases. It is seen that the airflow and mist flow velocity profiles correspond to the early stages of hydrodynamic development. The required channel length for air flow to reach fully developed conditions [26] is $66\ \text{mm}$; the length of the present channel is $32\ \text{mm}$ which confirms that developing flow should indeed be observed throughout the channel length. As would be expected of developing flows, the flow velocity close to the channel center increases along the channel length whereas the velocity close to the channel side wall decreases as seen in Figure 4(b) and (d). For both air and mist flows, the maximum velocity is not at the geometric center of the flow channel, but instead at two symmetric positions above and below the horizontal centerline of the channel as shown in Figure 5. This effect has also been observed by Hettiarachchi *et al.* [27] and is due to the presence of the top and bottom channel walls. At $x = L/2$ for instance, a maximum air flow velocity of $1.57\ \text{m/s}$ is obtained at $y = 4.28\ \text{mm}$ and $y = 12.72\ \text{mm}$; the corresponding maximum velocity for mist flow (case I) is $1.49\ \text{m/s}$ obtained at $y = 3.95\ \text{mm}$ and $y = 13.05\ \text{mm}$. It may also be noted that the average velocity at any yz plane for mist flow is higher than that for air flow. This is because of flow acceleration resulting from vapor generation from evaporation of the mist droplets. While the discrete droplets do act to slow down the continuous phase as can be seen at $x = L/2$ and $z = W_{ch}/2$ in Figure 5(a), flow acceleration is predominant by the time the exit of the channel is reached as indicated by the present simulations.

The temperature contours for the air flow (baseline case) and mist flow (case I) are shown in Figure 6. Figure 7 shows the temperature profile along the channel height for these two cases at two transverse positions, $z = W_{ch}/4$ and $z = W_{ch}/2$ and two axial locations, $x = L/2$ and $x = L$. As with the velocity profiles, it is again noted that the flow in both cases is thermally developing. The temperature is seen to be higher near the channel walls and lower at the channel center. The top wall of the channel is adiabatic and hence the temperature gradient is zero at $y = 15\ \text{mm}$ as shown in the inset of Figure 7. It is also observed from the temperature contour plot in Figure 6 that the thermal boundary layer from the fin of the heat sink continues to grow along the channel length (for both air-cooled and mist-cooled heat sinks); this heats the air close to the adiabatic top wall even though no heat is added from the top wall and

explains the temperature profile in Figure 7. Figure 7 also shows the locations of the minimum temperature for the cases of air and mist cooling; the locations of local minimum temperature for the mist flow correspond to regions of maximum evaporation and are discussed subsequently. The high thermal conductivity of the copper substrate results in negligible temperature gradients within the heat sink for air flow as well as mist flow cases. The simulations reveal an average heat sink temperature of 65.7 °C for air flow and 56.4 °C for mist flow. It is thus seen that replacing air flow with mist flow can lead to a substantial reduction in the heat sink temperature.

Figure 8 compares the variation of the average heat transfer coefficient and the bulk mean temperature of the continuous phase along the channel length for air flow (baseline case) and mist flow (case I). The local heat transfer coefficient was calculated using the local heat sink wall temperature, the bulk mean temperature and local heat flux. The heat transfer coefficient was averaged across the entire perimeter at every axial location by means of a user defined subroutine. It is clear that the heat transfer coefficients for mist flow are only slightly higher than those for air flow. The bulk mean temperature of the continuous phase fluid is, however, significantly lower for mist flow as compared to air flow. This is a direct consequence of the evaporation of the mist droplets; the latent heat required for evaporation limits the air temperature rise for mist cooling as compared to air cooling. The resulting higher temperature difference (between the wall and the bulk mean temperature) available for heat transfer in the downstream regions results in significantly enhanced overall heat dissipation capacity of mist flow as compared to air flow. The simulations indicate that the bulk mean temperature and heat transfer coefficient at the outlet were (48.3 °C, 36.4 W/m²-K) and (36.3°C, 41.3 W/m²-K) for airflow and mist flow (case I), respectively.

Figure 9 shows the relative humidity contours for mist flow (case I) while Figure 10 shows the relative humidity profiles along the channel height at two transverse positions, $z = W_{ch}/4$ and $z = W_{ch}/2$, and two axial locations, $x = L/2$ and L . The relative humidity increases along the length of the channel as the mist droplets progressively evaporate. Interestingly, the maximum relative humidity location is not at the centerline of the symmetric plane of the channel in the downstream regions of the heat sink, as clearly seen in Figure 9. This is because of higher temperatures close to the channel wall, which increase local evaporation rates. Additionally, the slower velocities close to the wall result in increased evaporation rates as the residence time for the droplets in that region is higher. As the relative humidity builds up, further droplet evaporation is suppressed which leads to a lower rate of increase in the relative humidity towards the channel end as shown in Figure 9(b) (as seen by comparing the relative humidity contours at the $x = 3L/4$ and $x = L$ planes). The relative humidity profile close to the heat sink fin wall does not change significantly along the channel length (as seen by comparing relative humidity profiles for $z =$

$W_{ch}/4$ at $x = L/2$ and $x = L$ in Figure 10) as the droplets close to the wall evaporate completely within a small axial distance from the inlet. The maximum relative humidity for this case is approximately 15% which occurs at the channel outlet as shown in Figure 10. Such relative humidity contour plots can be used to establish an upper limit on the inlet mist-loading fraction beyond which droplet evaporation will be significantly suppressed.

Figure 11 shows the distribution of the droplet diameters along the yz -plane at four different axial locations for mist flow (case I). The droplet diameter distribution was obtained by employing a subroutine that tabulates the position of every droplet in the flow domain. All the droplets introduced at $x = 0$ have a uniform size. Along the channel length, the droplets close to the channel walls are seen to evaporate completely. Figure 11 also clearly shows the droplet sizes getting smaller along the channel length due to evaporation.

The key results from the four mist-cooling simulations for a total heat sink power dissipation of 10 W are summarized and compared with the air-cooled heat sink in Table 1. The droplets introduced at the inlet are of a uniform diameter; however, at the outlet a distribution of droplet sizes is obtained. Also, some of the droplets evaporate completely before reaching the exit. All these factors were considered while calculating the average size of the droplet from the numerical simulations as shown in the second row of Table 1. Some of the simulation results are compared with those from the analytical model described in Section 2. For comparison with the analytical model, the average droplet diameter shown in Table 1 is the arithmetic mean of all the droplets at the exit. A reasonable match is seen between the analytical and numerical estimates of droplet diameters at the exit for both cases of the 50 μm inlet diameter droplets. For the 10 μm inlet diameter droplets, however, the analytical model predicts complete evaporation before the exit is reached; the numerical model predicts a much smaller proportion of droplets evaporating completely. This discrepancy for the smaller droplets relates to neglecting the relative humidity increase with downstream distance in the analytical model. The maximum relative humidity values are seen to be less than 15% in all cases, except for the case of a 10% inlet mist loading fraction consisting of 10 μm diameter droplets. Relative-humidity distributions assume great significance in the design of mist-cooling systems since regions of high relative humidity will have suppressed evaporative cooling which may result in localized reduction in cooling capacity.

Table 1 also shows the heat sink temperature reductions obtained by replacing airflow with mist flow for each of the four cases. The temperature reductions and the heat transfer coefficients are seen to be very strongly dependent on the inlet droplet size and inlet mist loading fraction. This again underscores the need for detailed numerical simulations to arrive at a set of operating parameters that maximize the

thermal performance of the system. The simulations show that the heat sink temperature can be reduced by an additional $39.6\text{ }^{\circ}\text{C}$ by replacing air flow with water-based mist containing 10% water by weight and having inlet droplets sized at $10\text{ }\mu\text{m}$. This implies a heat sink thermal resistance of 0.11 K/W which is significantly lower than the heat sink resistance of the baseline air-cooled heat sink (4.07 K/W). (The temperature difference for this thermal resistance estimation is the difference between the heat sink temperature and the inlet air temperature). It is important to note that the mean temperature of air at the outlet is lower than the inlet air temperature for case II; this results because of substantial evaporation-induced cooling of the air stream. Table 1 shows that smaller-diameter droplets result in higher reductions in the thermal resistance of the heat sink compared to the larger-diameter droplets (for the same mist loading fraction). This is due to the larger surface area-to-volume ratio of the smaller droplets and the higher evaporation rate that results. Higher mist loading fractions lead to smaller thermal resistance reductions; however, there will be an upper limit on the loading fraction, above which relative humidity will suppress evaporation.

All the results presented thus far were for a heat sink dissipation of $q = 10\text{ W}$. In order to quantify the potential of mist cooling at higher heat fluxes, another set of simulations was carried out for a heat dissipation of 20 W ($q'' = 19532\text{ W/m}^2$), the results of which are summarized in Table 2. The inlet velocity of the mist is maintained at 1 m/s and the inlet droplet diameter was selected as $10\text{ }\mu\text{m}$ in the two simulations. Two simulations were carried out corresponding to inlet mist loading fractions of 1% and 10%, respectively, and compared to an air-cooled heat sink with $q = 20\text{ W}$. The results indicate that mist cooling can significantly reduce the heat sink temperatures and thermal resistances as compared to air cooling at higher heat fluxes as well. As an illustration Table 2 shows that the heat sink temperature can be reduced by an additional $55.5\text{ }^{\circ}\text{C}$ by replacing air flow with a water-based mist (10% water by weight, inlet droplet diameter = $10\text{ }\mu\text{m}$). The higher heat flux is sustained by higher droplet evaporation rates which are reflected in a higher percentage of droplets the evaporate completely in the channel (as compared to similar simulations for $q = 10\text{ W}$). These results confirm that mist cooling offers excellent potential for the development of enhanced heat transfer solutions.

CONCLUSIONS

The capabilities of mist flow and heat transfer in heat sinks is investigated through detailed numerical modeling. Two sets of coupled transport equations are solved: the first set of equations corresponds to the continuous phase (air), while the second set of equations corresponds to the discrete phase (individual droplets). The simulations reveal the complex nature of the fluid mechanics and heat

transfer associated with mist flows. It is seen that the inlet droplet diameter and the loading fraction of the misting fluid are key parameters which influence droplet evaporation and the thermal performance of mist flows. The reduction in heat-sink temperature upon replacing the air flow with mist flow is directly quantified; additionally, the thermal resistances and heat transfer coefficients for mist flows are also evaluated and compared to air flows. The present work underscores the importance of detailed numerical analyses for the performance prediction and optimization of mist-based systems. The heat transfer results from the present work highlight the promise of mist cooling for high-heat-flux thermal management; the present simulations indicate that the heat sink thermal resistance can be reduced by up to 97% upon replacing air flow with an evaporating mist flow.

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Table 1. Summary of simulation results for mist cooling ($q = 10 \text{ W}$).

	Air-cooled heat sink	Droplet diameter = $10 \text{ } \mu\text{m}$		Droplet diameter = $50 \text{ } \mu\text{m}$	
		Mist loading fraction: 1% by weight (Case I)	Mist loading fraction: 10% by weight (Case II)	Mist loading fraction: 1% by weight (Case III)	Mist loading fraction: 10% by weight (Case IV)
Average droplet diameter at exit from numerical simulations (μm) (diameter using Equation 1, μm)	-	4.79 (0)	7.55 (0)	48.81 (48.45)	48.95 (48.45)
Fraction of droplets which evaporate completely (%)	-	32	19	0	0
Maximum relative humidity (%)	-	15	99	1	11
Area-averaged heat transfer coefficient for ($\text{W}/\text{m}^2\text{-K}$)	36.4	41.3	86.8	37.9	45.4
Mean temperature of air at outlet for ($^{\circ}\text{C}$)	48.3	36.3	10.7	47.8	38.7
Heat sink temperature ($^{\circ}\text{C}$) (additional heat sink temperature drop over air cooling, $^{\circ}\text{C}^{\#}$)	65.7	56.4 (9.3)	26.1 (39.6)	64.9 (0.8)	56.5 (9.2)
Thermal resistance (K/W) (% reduction of thermal resistance compared to air-cooled heat sink)	4.07 (-)	3.14 (23 %)	0.1 (97%)	3.99 (2%)	3.15 (23%)

Additional heat sink temperature drop over air cooling = (Heat sink temperature with air cooling - Heat sink temperature with mist cooling)

Table 2. Summary of simulation results for mist cooling ($q = 20 \text{ W}$).

	Air-cooled heat sink	Droplet diameter = $10 \text{ } \mu\text{m}$	
		Mist loading fraction: 1% by weight (Case I)	Mist loading fraction: 10% by weight (Case II)
Average droplet diameter at exit from numerical simulations (μm) (diameter using Equation 1, μm)	-	3.73 (0)	7.01 (0)
Fraction of droplets which evaporate completely (%)	-	44	22
Maximum relative humidity (%)	-	13.2	97
Area averaged heat transfer coefficient for ($\text{W}/\text{m}^2 \text{ } ^\circ\text{C}$)	35.9	39.5	64.0
Mean temperature of air at outlet for (mist flow, air flow) ($^\circ\text{C}$)	71.1	55.2	18.7
Heat sink temperature ($^\circ\text{C}$) (additional heat sink temperature drop over air cooling, $^\circ\text{C}^\#$)	106.5	94 (12.5)	51 (55.5)
Thermal resistance (K/W) (% reduction of thermal resistance compared to air-cooled heat sink)	4.08 (-)	3.45 (15)	1.3 (68)

Additional heat sink temperature drop over air cooling = (Heat sink temperature with air cooling - Heat sink temperature with mist cooling)

Figure 1. Schematic illustration of mist cooling in which mist is introduced at the inlet of the channel populated with electronic components.

Figure 2. Dimensions of the heat sink used in the numerical simulations.

Figure 3. Computational domain for the mist flow simulations: (a) isometric, and (b) front views.

Figure 4. Velocity magnitude contour plots at (a) the channel symmetry plane and channel top plane (air flow), (b) $x = 0, L/4, L/2, 3L/4$ and L planes (air flow), (c) the channel symmetry plane and channel top plane (mist flow case I), and (d) $x = 0, L/4, L/2, 3L/4$ and L planes (mist flow case I).

(a) $x = L/2$

(b) $x = L$

Figure 5. Velocity magnitude profiles along channel height ($2\text{ mm} \leq y \leq 15\text{ mm}$) for air flow and mist flow at $z = W_{ch}/4$ and $W_{ch}/2$ at axial locations of (a) $x = L/2$ and (b) $x = L$.

Figure 6. Temperature contour plots at (a) the channel symmetry plane and channel top plane (air flow), (b) $x = 0, L/4, L/2, 3L/4$ and L planes (air flow), (c) the channel symmetry plane and channel top plane (mist flow case I), and (d) $x = 0, L/4, L/2, 3L/4$ and L planes (mist flow case I).

(a) $x = L/2$

(b) $x = L$

Figure 7. Temperature profile along channel height ($2 \text{ mm} \leq y \leq 15 \text{ mm}$) for air flow and mist flow at $z = W_{ch}/4$ and $W_{ch}/2$ at (a) $x = L/2$ and (b) $x = L$. (The insets show the temperature profile near the top adiabatic wall, $14.95 \text{ mm} \leq y \leq 15 \text{ mm}$; the temperature gradient at the wall is confirmed to be zero).

Figure 8. Heat transfer coefficient and bulk mean temperature (T_{mean}) variation for the air-cooled baseline heat sink and the mist-cooled heat sink (case I).

Figure 9. Relative humidity contour plots for mist flow (case I) at (a) the channel symmetry plane and channel top plane, and (b) $x = 0, L/4, L/2, 3L/4$ and L planes.

Figure 10. Relative humidity profiles along channel height ($2\text{ mm} \leq y \leq 15\text{ mm}$) for mist flow (case I) at $z = W_{ch}/4$ and $W_{ch}/2$ and at $x = L/2$ and $x = L$.

(a) $x = L/4$

(b) $x = L/2$

(c) $x = 3L/4$

(d) $x = L$

Figure 11. Droplet diameter variation over the yz plane at (a) $x = L/4$, (b) $x = L/2$, (c) $x = 3L/4$, and (d) $x = L$ planes for the mist-cooled heat sink (case I). The vertical channel wall is at $z = 0.25 \text{ mm}$, the channel symmetry plane is at $z = 1.575 \text{ mm}$; the bottom wall of the channel is at $y = 2 \text{ mm}$, and top wall of the channel is at $y = 15 \text{ mm}$.