Three-dimensional Numerical Study of Laminar Confined Slot Jet Impingement Cooling using Slurry of Nano-encapsulated Phase Change Material (Postprint)

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Abstract

The pintle injector used for a liquid rocket engine is a newly re-attracted injection system famous for its wide throttle ability with high efficiency. The pintle injector has many variations with complex inner structures due to its moving parts. In order to study the rotating flow near the injector tip, which was observed from the cold flow experiment using water and air, a numerical simulation was adopted and a verification of the numerical model was later conducted. For the verification process, three types of experimental data including velocity distributions of gas flows, spray angles and liquid distribution were all compared using simulated results. The numerical simulation was performed using a commercial simulation program with the Eulerian multiphase model and axisymmetric two dimensional grids. The maximum and minimum velocities of gas were within the acceptable range of agreement, however, the spray angles experienced up to 25% error when the momentum ratios were increased. The spray density distributions were quantitatively measured and had good agreement. As a result of this study, it was concluded that the simulation method was properly constructed to study specific flow characteristics of the pintle injector despite having the limitations of two dimensional and coarse grids.

Full Text

Preamble

Three-dimensional Numerical Study of Laminar Confined Slot Jet Impingement Cooling using Slurry of Nano-encapsulated Phase Change Material

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This article presents a three-dimensional numerical model investigating the thermal performance and hydrodynamic features of confined slot jet impingement using slurry of nano-encapsulated phase change material (NEPCM) as a coolant. The slurry is composed of water as a base fluid with n-octadecane NEPCM particles of 100 nm mean diameter suspended in it. A single-phase fluid approach is employed to model the NEPCM slurry. The thermophysical properties of the NEPCM slurry are computed using modern approaches proposed recently, and the governing equations are solved with a commercial finite volume-based code. The effects of jet Reynolds number varying from 100 to 600 and particle volume fraction ranging from 0% to 28% are considered. The computed results are validated by comparing Nusselt number values at the stagnation point with previously published results using water as the working fluid. It was found that adding NEPCM to the base fluid results in considerable heat transfer enhancement, with the highest heat transfer coefficients observed at H/W = 4 and Cm = 0.28. However, due to the higher viscosity of the slurry compared with the base fluid, the slurry can produce a drastic increase in pressure drop that rises with NEPCM particle loading and jet Reynolds number.

Keywords: Nano-encapsulated phase change material, heat transfer enhancement, confined slot jet impingement

Introduction

Heat transfer enhancement in jet impingement processes represents one of the most reliable techniques for high heat flux removal from heated surfaces in numerous engineering and industrial applications, such as cooling of gas turbine blades and electronic chips, annealing of glass, and drying of textiles. A jet of working fluid leaving a slot or round nozzle is directed toward a targeted heated surface where it generates high heat transfer coefficients with relatively low pressure drop. The use of a slot nozzle provides a larger stagnation zone with uniform spreading of coolant after impingement [1]. In fact, the flow fields and heat transfer parameters in slot jets differ from those in circular jets [2]. The flow regime can vary from laminar to completely turbulent depending on the application, with laminar impinging jets frequently used for cooling electronic chips [3]. Moreover, several studies have been conducted on both laminar [4] and turbulent [5] flow regimes of confined jet impingement.

Jet impingement has two major design configurations: confined and unconfined. The presence of a confining top surface produces significant effects on fluid flow dynamics as well as heat transfer characteristics of the jet [6]. Limited space requirements for compact design make the confined configuration more suitable, while unconfined impinging jets are simpler in design [7]. The importance of confinement in jet impingement heat transfer has been investigated in [8]. Lin et al. [9] experimentally observed that heat transfer coefficients increase with jet Reynolds number. Park et al. [10] numerically reported that the nozzle-to-target height influences local and average Nusselt numbers for both laminar and turbulent cases.

In recent years, researchers have focused mainly on different types of working fluids for impinging jet heat transfer. Numerous studies have been conducted using air as the working fluid [11]. However, impinging jets with liquid have attracted much more attention recently due to superior heat transfer performance [12]. Literature suggests that thermophysical properties of the working fluid greatly influence the cooling capability of jet impingement. Moreover, the specific heat capacity of traditional fluids like water is not sufficient to meet the requirements of high heat flux removal [13]. To address this issue, an innovative technique has been suggested recently: adding nano-encapsulated phase change material (NEPCM) particles to the base fluid to form a two-phase suspension [14]. Particle sizes vary from micro to nano depending on the application [15].

PCM has the capability to absorb and release heat simultaneously during melting and solidification, while encapsulation prevents PCM leakage [16]. In general, NEPCM particles consist of organic paraffin cores with surrounding shells of cross-linked polymer [17]. The latent heat of NEPCM capsules suspended in the base fluid drastically increases the thermal storage capacity of the slurry when particles undergo phase change [18]. Therefore, using these advanced fluids is beneficial for applications such as compact heat exchangers and heat sinks if cooling system parameters are well designed to maximize the advantage of NEPCM latent heat [19].

Numerous studies have reported heat transfer enhancement in forced convection by adding NEPCM particles to coolant. Wu [20] conducted an experimental study investigating the effect of NEPCM slurry on heat transfer enhancement in jet impingement cooling. Results suggested that the volume fraction of NEPCM in the base fluid greatly influences thermal performance, with slurry containing 28% volume fraction of NEPCM enhancing the heat transfer coefficient by 50% compared to the base fluid.

In the present article, a three-dimensional conjugate heat transfer model is used to study the cooling and hydrodynamic performance of a confined slot jet impingement in the laminar regime with NEPCM slurry as a coolant. To the best of the authors' knowledge, no published data exist in literature on experimental or numerical studies of confined slot jet impingement using NEPCM slurry. Therefore, this attempt is expected to provide a breakthrough for engineering applications such as electronic cooling and material processing.

Nomenclature

Roman Symbols
d - particle diameter (m)

cp - specific heat capacity of fluid (J/kg · K)
k - thermal conductivity (W/m · K)
ks - static thermal conductivity (W/m · K)
u, v, w - velocity components
x, y, z - spatial coordinates
H - nozzle to plate distance (m)
W - jet width (m)
L - length of copper plate (m)
P - pressure (Pa)
ΔP - pressure drop (Pa)
q" - heat flux of the hot wall (W/m^2)
Re - Reynolds number
Pe - Peclet number
Nu - Nusselt number
h - heat transfer coefficient (W/m^2 · K)
T - temperature (K)
T - lower melting temperature (K)
T - upper melting temperature (K)
TMr - melting range (K)
Tm - melting point (K)
hsl - latent heat of fusion of PCM (kJ/kg)

Greek Symbols
˙ - shear rate (1/s)
 - dynamic viscosity (mPa · s)
 - density (kg/m^3)
Φ - viscous dissipation
 - volume concentration of NEPCM
 - thermal diffusivity (m^2/s)

Subscripts
s - stagnation point
avg - average
p - particle
i - impingement
w - wall
pcm - phase change material
f - fluid
s - solid
eff - effective thermophysical properties of fluid
m - mass concentration of NEPCM

Geometrical Configuration and Model Description

The schematic diagram of the physical domain is presented in Figure 1 [Figure 1: see original paper], which illustrates the geometrical configuration of the confined slot jet impingement cooling system. The three-dimensional model has a nozzle width (W) of 6.2 mm and a plate length-to-width ratio (L/W) of 60, while the nozzle-to-plate distance (H/W) ranges from 4 to 6. A Cartesian coordinate system is employed in the present study, with its origin located at the center of the impingement surface. A constant heat flux of 10,000 W/m² is applied to the bottom surface of the copper plate. The confinement top is made of aluminum and is completely insulated to prevent heat transfer across it. Furthermore, uniform inlet velocities are considered with laminar and incompressible flow regime. Radiation and natural convection effects are small enough to be neglected in the model design.

As the present simulation is conducted at low Reynolds number, effects caused by viscous dissipation are neglected [16]. The working fluid is either water or slurry with different mass concentrations of NEPCM particles in water. Fluid jet temperature is set at 298 K for all studied configurations to ensure that NEPCM particles experience phase change as soon as the slurry reaches the impingement surface. The particle mass concentration in the slurry is below 0.3; therefore, the fluid is assumed to be Newtonian [21]. Furthermore, it is assumed that the base fluid and NEPCM particles flow at the same velocity without any lag between the phases [22]. Additionally, no mass transfer occurs between the base fluid and NEPCM capsule to ensure that melted PCM inside the capsule will not disperse in the base fluid. NEPCM particles are considered spherical with a mean diameter of 100 nm [18]. Further, no temperature gradient exists inside the particle, and its melting range is between 293.15 K and 303.15 K. The effect of shell material on heat transfer is neglected as it is very thin relative to the core [19]. It is also important to mention that, due to homogeneous distribution of particles across the base fluid, the bulk properties of slurry are expected to be constant except for thermal conductivity and specific heat capacity, which depend on micro-convection and slurry operating temperature, respectively [17].

Governing Equations

The continuity, momentum, and energy equations constitute the governing equations for flow and heat transfer in our analysis. They are specified by the following expressions:

Continuity equation:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0$$

Momentum equations:

$$\rho_{\text{eff}} \left(u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} + w\frac{\partial u}{\partial z} \right) = -\frac{\partial P}{\partial x} + \mu_{\text{eff}} \nabla^2 u$$

$$\rho_{\text{eff}} \left(u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} + w\frac{\partial v}{\partial z} \right) = -\frac{\partial P}{\partial y} + \mu_{\text{eff}} \nabla^2 v$$

$$\rho_{\text{eff}} \left(u\frac{\partial w}{\partial x} + v\frac{\partial w}{\partial y} + w\frac{\partial w}{\partial z} \right) = -\frac{\partial P}{\partial z} + \mu_{\text{eff}} \nabla^2 w$$

Energy equation for the fluid domain:

$$\rho_{\text{eff}} c_{p,\text{eff}} \left(u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} + w\frac{\partial T}{\partial z} \right) = k_{\text{eff}} \nabla^2 T + \dot{\Phi}$$

Energy equation for the solid domain:

$$k_s \nabla^2 T = 0$$

The term $\dot{\Phi}$ in the energy equation predicts the rate at which viscous energy is dissipated per unit volume by virtue of fluid viscosity.

Thermophysical Properties of NEPCM Slurry

The bulk properties of slurry are functions of NEPCM concentration and operating temperature. The slurry considered in the present analysis is composed of water as a base fluid with NEPCM capsules suspended in it. The NEPCM capsules consist of an n-octadecane core as a phase change material with melting point (Tm) of 298.15 K and a surrounding cross-linked polymer shell. Table 1 illustrates the thermophysical properties of slurry components.

The effective properties of slurry are predicted by the following correlations derived from literature:

Density:

$$\rho_{\text{eff}} = (1 - c_m)\rho_{\text{water}} + c_m \rho_p$$

where c_m and ρ_p are the mass concentration of slurry and density of NEPCM particle, respectively.

Viscosity [23]:

$$\mu_{\text{eff}} = \mu_{\text{water}}(1 + 2.5\phi + 10.05\phi^2)$$

where ϕ is the volume fraction or particle loading, which is a function of temperature.

Thermal conductivity:

Maxwell' s correlation is used to predict static thermal conductivity of NEPCM slurry at rest:

$$k_s = k_{\text{water}} \frac{k_p + 2k_{\text{water}} - 2\phi(k_{\text{water}} - k_p)}{k_p + 2k_{\text{water}} + \phi(k_{\text{water}} - k_p)}$$

The effects caused by particle-particle, particle-fluid, and particle-wall interactions are lumped together to compute the effective thermal conductivity of slurry, specified by the following correlation [24]:

$$k_{\text{eff}} = k_s + 0.18 k_{\text{water}} \phi \text{Pe}_p^{0.67}$$

where Pe_p is the particle Peclet number defined as:

$$\mathrm{Pe}_p = \frac{ed^2}{\alpha_{\mathrm{water}}}$$

where α_{water} is the thermal diffusivity of water and e is the shear rate, which is a function of spatial coordinates and corresponding velocities. The shear rate can be expressed as:

$$e = \sqrt{2\left[\left(\frac{\partial u}{\partial x}\right)^2 + \left(\frac{\partial v}{\partial y}\right)^2 + \left(\frac{\partial w}{\partial z}\right)^2\right] + \left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x}\right)^2 + \left(\frac{\partial u}{\partial z} + \frac{\partial w}{\partial x}\right)^2 + \left(\frac{\partial v}{\partial z} + \frac{\partial w}{\partial y}\right)^2}$$

Heat Capacity:

$$c_{p,\mathrm{eff}} = (1 - c_m)c_{p,\mathrm{water}} + c_m c_{p,p}$$

where $c_{p,\mathrm{water}}$ and $c_{p,p}$ are the specific heat capacities of water and NEPCM particle, respectively.

Alisetti and Roy [25] suggested that the difference between various profiles for calculating the specific heat capacity of NEPCM particles is less than 4%. Therefore, a sine profile is used in the present study to predict the specific heat capacity of NEPCM particles, as illustrated in Figure 2 [Figure 2: see original paper]. The melting process starts when the temperature of the NEPCM particle reaches T_1 (solidus temperature) and terminates at T_2 (liquidus temperature). Furthermore, T_{Mr} and T_m represent the melting range and melting point of the NEPCM particle. When the temperature is below or above the melting range, the specific heat capacity of the particle is given by $c_{p,\mathrm{pcm}}$. For temperatures within the melting range, the specific heat capacity is given by:

$$c_{p,\mathrm{pcm}} = c_{p,s} + \frac{\pi h_{\mathrm{sl}}}{2T_{\mathrm{Mr}}} \sin\left[\pi\left(\frac{T - T_1}{T_{\mathrm{Mr}}}\right)\right]$$

More details about correlation utilization in the present study are provided in the specified references.

Numerical Procedure and Boundary Conditions

A commercial computational fluid dynamics (CFD) code FLUENT [26] has been employed in the present study as the numerical solver. The three-dimensional governing equations of continuity, momentum, and energy are discretized using the finite volume method. The convection/diffusion terms present in the momentum and energy equations are discretized with the QUICK scheme. Furthermore, the SIMPLE algorithm is employed in the numerical solver to provide coupling between velocity and pressure.

The three-dimensional computational grid is displayed in Figure 3 [Figure 3: see original paper]. A commercial grid generation software Gambit is used to divide the computational domain into small hexahedral elements in both fluid and solid domains. A relatively fine grid is adopted in the local domain near the impingement surface where the solid-liquid interface occurs for the conjugate heat transfer boundary condition, while a coarse grid is used in the remainder of the domain.

The numerical solution is terminated when convergence is ensured at a particular iteration. Convergence is achieved when the summation of absolute values of relative errors (i.e., residuals) for pressure, velocity components, and temperature falls below 10^{-5}, 10^{-6}, and 10^{-8}, respectively. Furthermore, the Nusselt number and temperature values at the stagnation point are also monitored as a secondary convergence criterion.

The assigned boundary conditions are as follows:

Inlet boundary condition: Uniform velocity and temperature are specified at the nozzle inlet.

Outlet boundary condition: Pressure outlet boundary condition is applied at the outlet.

Bottom wall heat flux boundary condition: Constant heat flux of $q'' = 10,000$ W/m² is applied.

Solid-Liquid interface boundary condition: Continuity of temperature and heat flux is enforced.

Confinement top and other walls: Adiabatic boundary conditions are applied.

Grid Independence and Model Verification

The grid sensitivity of the numerical model was examined by considering four grid sizes: 134,872 (coarse), 369,700 (medium), 876,732 (fine), and 1,798,236 (very fine) elements. The grid independence test was conducted for the highest Reynolds number using water as the working fluid. The test results predicted that the numerical solution becomes grid-independent at the fine grid level, with maximum deviation of stagnation Nusselt numbers and pressure drops between fine and very fine grids below 3%. Therefore, the fine grid with 876,732 elements is sufficiently appropriate to capture heat and flow characteristics precisely.

To the best of the authors' knowledge, no published experimental data exist in the literature related to confined slot jet impingement cooling systems operating in the laminar regime. Therefore, to ensure code validity and solution reliability, the numerical simulation was conducted with the same geometrical and operating parameters as presented in Di Lorenzo [12]. In this validation test, water is considered as the coolant. Furthermore, a constant temperature of 313 K is maintained at the impingement surface while all other surfaces are insulated in the solid domain. The fluid jet temperature is kept at 292 K. Further details regarding the numerical models and procedures can be found in the original articles [9] and [12].

The validation of the present model is illustrated in Figure 4 [Figure 4: see original paper], which shows Nusselt numbers at the stagnation point. The maximum deviation found is below 20%, which demonstrates good agreement as Nusselt number calculations are extremely sensitive to grid size and design.

Results and Discussion

Numerical simulation is carried out to evaluate the thermal and hydrodynamic characteristics of confined slot jet impingement in the laminar regime using NEPCM slurry as a coolant. The effects of Reynolds numbers varying from 100 to 600 are considered. The ranges of NEPCM particle mass concentration (C_m) and nozzle-to-plate distance (H/W) are C_m = 0–0.28 and H/W = 4-6, respectively. The results are analyzed and reported in terms of Nusselt number, heat transfer coefficient, pressure drop, and bulk fluid temperature.

Figure 5 [Figure 5: see original paper] illustrates the effects of NEPCM particle mass concentration in slurry on bulk fluid temperature at different Reynolds numbers and constant nozzle-to-plate distance. As observed, using NEPCM slurry as a coolant reduces the bulk temperature of the fluid compared to water. Additionally, increasing the mass concentration of NEPCM in slurry reduces the bulk fluid temperature considerably. This occurs because adding NEPCM capsules to the base fluid increases the effective heat capacity of the coolant, which further enhances thermal energy storage capability with minimal rise in bulk fluid temperature. An additional observation is that with increasing Reynolds number, the fluid bulk temperature decreases and heat transfer enhancement occurs. This can be explained by the fundamental theory that at low Reynolds numbers, fluid flows slowly over the impingement surface, thus having more time to absorb heat. Therefore, diffusion becomes the dominant heat transfer mechanism, resulting in higher bulk fluid temperature. At higher Reynolds numbers, associated velocities increase and forced convection becomes the dominant factor for heat transfer. Consequently, more heat is transferred with minimal rise in bulk fluid temperature.

Figure 6 [Figure 6: see original paper] shows velocity streamlines of the impinging coolant in the three-dimensional domain. The fluid jet exhibits sudden velocity suppression just before hitting the heated surface. This velocity reduction tends to increase static pressure by converting the kinetic energy of partially stagnated coolant to potential energy. Therefore, the stagnation zone corresponds to high static pressure.

Temperature distributions in the fluid domain are shown in Figure 7 [Figure 7: see original paper] for water and NEPCM slurry with different mass concentrations. As expected, for all cases, the thermal boundary layer is relatively thinner at the stagnation zone compared to the area near the outlet. This is primarily due to strong temperature gradients existing at the stagnation zone, indicated by densely packed isotherms. Temperature gradients decrease away from the stagnation zone where a thick boundary layer is present, meaning that higher heat transfer rates occur at the stagnation zone. Figure 7 also demonstrates that the thermal boundary layer grows more slowly across the copper plate for NEPCM slurry compared to water. This is mainly attributed to the latent heat storage of NEPCM capsules suspended in water. Another important result shown in Figure 7 is that thermal boundary layer thickness reduces as the mass concentration of NEPCM increases in the base fluid at a fixed Reynolds number and nozzle-to-plate distance. The thermal layer of water is thicker than that of slurry with $C_m = 0.28$. Therefore, this reduction in boundary layer thickness enhances heat transfer by establishing strong temperature gradients across the copper plate in the thermal boundary layer region.

Figure 8 [Figure 8: see original paper] depicts the effects of nozzle-to-plate distance (H/W) on stagnation point Nusselt number with jet speed variation at constant NEPCM concentration. This trend also holds true for other mass concentrations of NEPCM particles in slurry. It is observed that the Nusselt number at the stagnation point increases with jet speed at a particular nozzle-to-plate distance. Additionally, as H/W increases slightly for a fixed jet speed, the stagnation Nusselt number decreases considerably. This effect is more prominent at high jet speeds, for example at 0.1 m/s. The common conclusion is that confined jet impingement cooling systems operating at low nozzle-to-plate distances are more beneficial for heat transfer enhancement.

Figure 9 [Figure 9: see original paper] plots the average heat transfer coefficient against Reynolds number for various slurry concentrations at $H/W = 4$. It shows that heat transfer from the impingement surface is highly affected by using NEPCM slurry instead of water as coolant. The predicted trend shows enhancement of the heat transfer coefficient with increasing jet Reynolds number and NEPCM mass concentration. This enhancement results from the high thermal heat capacity of the slurry. Significant heat transfer improvement is found at NEPCM $C_m = 0.28$ compared with water. Another important point is that the average heat transfer coefficient increases as Reynolds number increases for all considered cases.

The pressure drop between the nozzle exit and outlet of confined slot jet impingement systems for varying Reynolds numbers is illustrated in Figure 10 [Figure 10: see original paper] at $H/W = 4$. It is evident that at higher Reynolds numbers, the system exhibits significantly higher pressure drops. Moreover, pressure drop in the system is extremely sensitive to NEPCM concentration. This is because higher NEPCM concentrations in the base fluid result in higher effective viscosity of coolant, which consequently increases pressure drops and pumping power requirements. The pressure drop recorded for NEPCM $C_m = 0.28$ at $\mathrm{Re} = 600$ represents the worst scenario in the present model. Therefore, it is highly recommended that the higher pressure drops associated with high concentrations of NEPCM particles at elevated Reynolds numbers should be carefully considered when designing a confined slot jet impingement system with NEPCM slurry as coolant.

Conclusion

The cooling performance and hydrodynamic features of NEPCM slurry were compared with those of water used as coolants inside the confined slot jet impingement system. The effects of jet Reynolds number, jet-to-plate spacing, and mass concentration of NEPCM particles on heat transfer coefficient, pressure drop, and fluid bulk temperature were discussed. Good agreement was found between the present results and previous numerical works. Analysis predicted that adding NEPCM particles to the base fluid improves the Nusselt number and decreases the bulk temperature of the fluid. It was also found that the thermal performance of the system highly depends on the jet-to-target spacing. For the Reynolds numbers considered, increasing mass concentration above 0.2 results in very little heat transfer enhancement. However, the associated pressure drop increases drastically, which consequently reduces the total efficiency of the system. Therefore, the present findings are expected to provide guidelines for designing and optimizing confined slot jet impingement cooling systems using NEPCM slurry as a working fluid in the near future.

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Note: Figure translations are in progress. See original paper for figures.