Numerical Study on Thermal and Fluid Dynamic Behavior of Confined Impinging Slot Jets with Nanofluids in Partially Filled Configuration of Metal Foam

B Buonomo¹, A di Pasqua¹, O Manca¹*, S Nappo¹

¹ Dipartimento di Ingegneria, Università degli Studi della Campania "Luigi Vanvitelli", Via Roma 29, Aversa (CE) 81031, Italy

Corresponding author e-mail: oronzio.manca@unicampania.it

Abstract. In this work a numerical analysis on confined slot impinging jets constituted by a partially filled configuration of metal foam in mixed convection is illustrated. The employed working fluids are pure water or Al₂O₃/water based nanofluids. A two-dimensional model is constructed and several Peclet numbers (equal to 100, 200, 350, 750, 1500) are assumed. Rayleigh numbers is fixed equal to 30000. The nanoparticle volume concentrations are in a range from 0% to 4% and the nanoparticle diameter is considered equal to 30nm. The target surface is characterized by a constant temperature, evaluated according to the value of Rayleigh number. The distance of the target surface is five times greater than the slot jet width. Three different values of ratio between the total system and the metal foam length have been assumed. The thermal and fluid-dynamic properties of nanofluids are estimated by a single-phase model approach. The model of local thermal non equilibrium (LTNE) is assumed in order to replicate the behavior of the metal foam characterized by a number of pores per inch (PPI) equal to 20 and a porosity of 0.9005. Results show increasing values of the convective heat transfer coefficients for increasing values of Peclet number and nanoparticle concentration. This behaviour is more evident at low Peclet number values. In conclusion, the ratio between the thermal and pumping power is calculated in order to find a trade-off between the increase of heat transfer and pressure drop.

1. Introduction

Nowadays the heat transfer industry is characterized by the aim to develop and to build systems with a higher efficiency in many applications in several fields, such as automotive, aerospace, electronics and process industry. For obtaining efficient cooling systems, the use of confined impinging jets represents a good solving. Furthermore, the employment of working fluids with the addition of nanoparticles results a promising solution in order to enhance the thermal behaviour of the base fluids. For their high cooling effectiveness, uniformity and controllability, the slot jets are widely employed. These characteristics are particularly convenient in the cooling system of modern electronic devices because they require an increase in terms of heat fluxes to be removed and a decrease of dimensions. The application of a cover of porous medium on the impinged surfaces is becoming increasingly frequent because the metal foam is characterized by high conductivity and porosity that represent a further improvement of heat transfer performances. The combination of nanoparticles and porous media with high thermal conductivity is an interesting and promising solution.

Paulaj and Sahu [1] accomplished a numerical study on conjugate heat transfer improvement of slot jets with several nanofluids on an array of protruding hot sources. The results present that the dimensions of the primary and secondary vortices increase gradually with the increase in the Reynolds
number. The Nusselt number increases with increasing Reynolds number and nanoparticle volume concentration. Furthermore, the highest Nusselt number value is obtained for a diameter of nanoparticles equal to 10 nm. Lamraoui et al. [2] carried out a numerical investigation on thermal and fluid dynamic behavior of Al₂O₃–water nanofluids in a confined slot impinging jet. The nanofluid are assumed both Newtonian and non-Newtonian for a nanoparticle volume fraction between 0% and 5%. The results present that the heat transfer increases with Reynolds number and nanoparticle volume concentration $\phi$. The local Nusselt number is higher for non-Newtonian nanofluid respect to the Newtonian fluid especially near to the jet axis where it is present an increase of 9.8% for $\phi = 5\%$ and $Re = 300$. An experimental investigation on nanofluids in impinging jets with different distance between jet and plate is carried out by Barewar et al. [3]. Several distances are considered (2–7.5) and distinct nanoparticle volume concentration ($\phi = 0.02\%$ to 0.1%). The results show that the increase of nanoparticles concentration causes an increase in terms of the heat transfer coefficient that presents an improvement equal to 51% for $\phi = 1\%$ respect to the pure water. Furthermore, the maximum value of the heat transfer coefficient is obtained for a ratio between the jet to plate surface distance and the jet diameter equal to 0.1%. Alauddin et al. carried out [4] a numerical analysis on turbulent flow field and thermal behavior of an impinging jet with Al₂O₃/water nanofluid. The jet is round. Results show that the heat transfer coefficient increases with increasing nanoparticle concentration. Furthermore, it is possible to observe as the turbulent kinetic energy is constant with the presence of nanoparticles. For $\phi = 0.1\%$, the Nusselt number is higher of 28%. A numerical investigation on laminar flow of Al₂O₃/water nanofluids in a confined slot impinging jet is accomplished by Yousefi-Lafouraki et al. [5]. The Eulerian–Lagrangian model is used to simulate the behavior of liquid–solid laminar flow with a two-way coupling approach. The results present that the trajectory of nanoparticles is more stable from the jet in corresponding to the increase of Reynolds number and the ratio between the jet and the impingement surface. Furthermore, it is possible to note that the heat transfer coefficient of the mixture model is higher respect to that of the Eulerian–Lagrangian model.

Izadi et al. [6] accomplished a numerical investigation on the thermal performance of a heat sink characterized by metal foam and impinging jet. The used working fluids are Hydrogen, air and Cu-water nanofluid. The results demonstrate that the velocity and temperature values decrease with increasing porosity. Moreover, an increase in terms of heat transfer is due to the increment of Darcy number. The temperature profile, instead, shows an increment when the distance from the symmetry axis increases. In conclusion, the better configuration in terms of enhancement of heat transfer rate is represented by the impinging jet with a non-uniform velocity profile and a decreasing slope. A numerical analysis on a cylindrical heat sink characterized by porous media and air and nanofluid imping jet is carried out by Siavashi et al. [7]. Several dimensionless parameters such as form drag coefficient, Reynolds number of round jet, Darcy number, porosity and nanofluid volume fraction are employed in this analysis. The results show that the increase of Reynolds number leads to an increase of heat transfer rate. Moreover, the increment of porosity diminishes the shear stress near to the wall and, for this reason, decreases the temperature profile. In conclusion, the increase of nanoparticles concentration improves the value of the average Nusselt number.

In this paper, a numerical study is executed on a mixed convective heat transfer problem in a confined slot impinging jet partially filled with aluminium foam. Al₂O₃/water based nanofluids characterized by particle volume concentrations from 0% to 4% and the particle diameter equal to 30 nm are employed. A two-dimensional model is studied with different Peclet number values. Rayleigh numbers is considered equal to 30000. Three different values of ratio between the total system and the metal foam length (equal to 1, 3.5 and 7) have been assumed. The metal foam is characterized by a number of pores per inch (PPI) equal to 20 and a porosity equal to 0.9005. Results are presented in terms of convective heat transfer coefficient, pressure drop and ratio between the thermal and pumping power.

2. Governing equations and physical domain

The physical domain is made up of a confined slot impinging jet in a metal foam with nanofluids. The 2D sketch of the system under investigation is illustrated in the Figure 1. The impinging slot jet system is characterized by a channel with a distance between the heated plate and slot jet inlet $H$ equal to 30
mm and a length $2L$ of 420 mm. The length of heated zone of the wall opposite the jet is $2L_{hw}$ is 60 mm and the width of jet, indicated as $2W$, is equal to 6 mm. The microscopic flow equations are written using the local volume averaging process [8-9] to describe the thermal and fluid dynamic behavior into the domain. The metal foam is considered homogeneous and isotropic; moreover, the properties of the fluid phase and solid matrix are assumed constant. The Darcy–Forchheimer-Brinkman model and LTNE (Local Thermal Non-Equilibrium) hypotheses are assumed to consider the presence of the aluminum foam in the impinging slot jet. Using the hypothesis above indicated, the governing equations are:

- **Conservation of mass**

  \[
  \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \tag{1}
  \]

- **x- momentum equation**

  \[
  \rho_{nf} \left( \frac{u}{\varepsilon^2} \frac{\partial u}{\partial x} + \frac{v}{\varepsilon^2} \frac{\partial u}{\partial y} \right) = -\frac{\partial p}{\partial x} + \frac{\mu_{nf}}{\varepsilon} \left( \frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} \right) - \eta \left( \frac{\mu_{nf}}{K} u + \frac{C_F}{K/U^2} \rho_{nf} \sqrt{u'^2 + v'^2} u \right) + \rho_{nf} g \beta_{nf} (T_{nf} - T_0) \tag{2}
  \]

- **y- momentum equation**

  \[
  \rho_{nf} \left( \frac{u}{\varepsilon^2} \frac{\partial v}{\partial x} + \frac{v}{\varepsilon^2} \frac{\partial v}{\partial y} \right) = -\frac{\partial p}{\partial y} + \frac{\mu_{nf}}{\varepsilon} \left( \frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} \right) - \eta \left( \frac{\mu_{nf}}{K} v + \frac{C_F}{K/U^2} \rho_{nf} \sqrt{u'^2 + v'^2} v \right) \tag{3}
  \]

where $\varepsilon$ is the porosity, $\rho_{nf}$ and $\mu_{nf}$ are nanofluid density and viscosity, $u$ and $v$ are the velocity components in Cartesian coordinates, $K$ and $C_F$ are the permeability and inertial coefficient of the metal foam, respectively.

- **Fluid phase energy equation**

  \[
  \left( \rho_{cp, nf} c_{nf} \right) \left( \frac{\partial T_{nf}}{\partial x} + v \frac{\partial T_{nf}}{\partial y} \right) = \varepsilon k_{nf} \left( \frac{\partial^2 T_{nf}}{\partial x^2} + \frac{\partial^2 T_{nf}}{\partial y^2} \right) + \eta h_s \alpha_s (T_s - T_{nf}) \tag{4}
  \]

- **Solid phase energy equation**

  \[
  (1-\varepsilon) k_s \left( \frac{\partial^2 T_S}{\partial x^2} + \frac{\partial^2 T_S}{\partial y^2} \right) - \eta h_s \alpha_s (T_s - T_{nf}) = 0 \tag{5}
  \]

in which $c_{pf}$ is the nanofluid specific heat, $k_{nf}$ and $k_s$ are the fluid phase and solid matrix thermal conductivity, $T_{nf}$ and $T_s$ are the temperature of fluid phase and solid phases of metal foam, respectively. The terms $\alpha_{sf}$ and $h_{nf}$ are the specific surface area density and the interfacial heat transfer coefficient between the fluid phase and solid matrix, due to the LTNE model. Finally, the governing
equations (1)-(5) are used for both porous and clean region, where $\eta=1$ and $0<\varepsilon<1$ and for porous zone, and $\eta=0$ and $\varepsilon=1$ for clean zone. The evaluation of the parameters $K$ and $C_f$ was carried out with the correlations of Bhattacharyya et al. [10]. The relationships of the work [11] are used to estimate the values of $\alpha_{nf}$ and $h_{nf}$. The key parameters of the metal foam, employed in this study, are the pore density equal to 20 PPI, porosity, $\varepsilon=0.9005$, permeability, $K=9.0\times10^{-3}$ m$^2$ and the drag coefficient, $C_f=0.088$, [10, 11].

In the present paper the laminar flow and its convective heat transfers of pure water/ Al$_2$O$_3$ nanofluids under steady-state conditions are studied. The nanoparticle volume concentrations ranges from 0% to 4%. The diameter of nanoparticles ($d_{np}$) is equal to 30 nm. A single-phase model has been considered in order to illustrate the nanofluid presence. The density and the specific heat are estimated by the expressions of the work [12] by the following expressions

$$\rho_{nf} = \phi \rho_{np} + (1-\phi) \rho_{bf}$$

(6)

and

$$c_{p,nf} = \frac{\phi (\rho c_p)_{np} + (1-\phi) (\rho c_p)_{bf}}{\rho_{nf}}$$

(7)

where $\phi$ is the nanoparticle volume concentration and the subscripts $np$ and $bf$ indicate the nanoparticles and the base fluid, respectively.

For the dynamic viscosity of the nanofluid, the used correlation is [13]:

$$\frac{\mu_{nf}}{\mu_{bf}} = 1 + 39.11 \phi + 533.9 \phi^2$$

(8)

The correlation of [14] is employed to estimate the thermal conductivity of the nanofluid:

$$\frac{k_{nf}}{k_{bf}} = \frac{\frac{k_{np}}{1 + 2\alpha} + 2k_{bf} - 2\phi(k_{nf} - k_{np} (1-\alpha))}{\frac{k_{np}}{1 + 2\alpha} + 2k_{bf} + \phi(k_{nf} - k_{np} (1-\alpha))}$$

(9)

where $\alpha = 2R_b k_{bf} d_{np}$ is the particle Biot number and $R_b = 0.77 \times 10^{-8}$ m$^2$KW$^{-1}$ is the interface thermal resistance.

The nanofluid thermal expansion coefficient $\beta$ is evaluated as [12]:

$$\frac{\beta_{nf}}{\beta_{bf}} = \left[ \frac{1}{1 + \frac{1}{\rho_{np}(1-\phi)} \frac{\beta_{np}}{\rho_{np}} + 1 + \frac{1}{\rho_{np}(1-\phi)} \frac{\beta_{np}}{\rho_{np}}} \right]$$

(10)

The properties of the alumina nanoparticles are $\rho_{np}=3880$ kg/m$^3$, $c_{p, np}=729$ J/kgK and $k_{np}=42$ W/mK. The thermophysical properties of the nanofluids employed in the analyses are listed in Table 1.

The boundary conditions of the problem are the following: the left edge is symmetric, the upper and bottom edges are adiabatic wall, and the heated wall is an isothermal edge. The impinging jet is characterized by a temperature $T_0$ equal to 293 K and a velocity equal to $u_0$. The impinging jets velocities $u_0$ are estimated from the definition of $Pe$. The outflow condition is imposed to the exit. The reference length is the width of the jet $2W$ for the dimensionless number.

The average heat transfer coefficient $h_{nf}$ is evaluated as:

$$\overline{h_{nf}} = \overline{h_{nf}^f} + \overline{h_s}$$

(11)

where $\overline{h_{nf}^f} = \frac{q_{nf}}{T_w - T_i}$ and $\overline{h_s} = \frac{q_s}{T_w - T_i}$ are the fluid phase and solid matrix heat transfer
Table 1. Parameters of the working fluids

| ϕ  | ρ (kg/m³) | μ (kg/ms) | cₚ (J/kgK) | k (W/mK) | β (1/K) |
|----|-----------|-----------|------------|----------|---------|
| 0% | 998.2     | 1.0x10⁻³  | 4182       | 0.60     | 2.1x10⁻³ |
| 1% | 1027      | 1.4x10⁻³  | 4052       | 0.61     | 2.0x10⁻³ |
| 4% | 1113      | 3.4x10⁻³  | 3701       | 0.63     | 1.8x10⁻³ |

coefficient of the aluminium foam, respectively. \( \overline{q}_{nf} \) and \( \overline{q}_s \) are the average wall heat flux in nanofluid and metal foam, respectively, and \( (T_w - T_j) \) is the temperature difference between the wall temperature on the heated zone and the inlet jet temperature.

Peclet and Rayleigh numbers are defined as:

\[
Pe = \frac{\rho_{nf}c_{p,nf}u_jW}{k_{nf}}; \quad Ra = \frac{\rho_{nf}^2c_{p,nf}β_{nf}ΔTW^3}{\mu_{nf}k_{nf}}
\]  

3. Numerical model

The finite volume method is employed to resolve the governing equations. The code Ansys-Fluent 15.0 is used to execute the numerical investigations. The SIMPLE algorithm is accomplished for the pressure-velocity coupling; the least square cell is used to evaluate the gradient evaluation for the spatial discretization. The PRESTO algorithm is employed to obtain the pressure calculation; the second order upwind scheme is applied for energy and momentum equations. Convergence criteria are equal to 10⁻⁵ for the continuity and the velocity components while for the energy equal to 10⁻⁸.

The numerical domain is showed in the Figure 1.

The grid is made up of rectangular cells. Three different types of grids were analyzed to find an independent solution from the mesh. They are made up of 52800 cells, 105600 cells and 211200 cells, respectively. In corresponding of a \( ϕ = 4\% \), \( d_{np} = 30 \) nm, \( Pe = 1500 \), \( L_{mf}/L_{hw} = 3.5 \), the evaluation of \( \overline{h}_{nf} \) on the heated wall, as showed in Table 2, points out that the grid with 105600 cells had 0.2% error than the mesh with 211200 cells. The grid adopted for the computational was the one with 105600 elements because with it a compromise between solution accuracy and convergence time has been obtained. The grids for the other configurations have been built up with the same criteria of the construction of the mesh for the ratio \( L_{mf}/L_{hw} \) equal to 3.5.

The validation was accomplished comparing the present numerical model with the one presented in [15] for the confined impinging slot jet in metal foam and the one given in [16] for the confined impinging slot jet with nanofluids.

4. Results and Discussions

The thermal and fluid dynamic investigations are accomplished for five different Peclet values equal to 100, 200, 350, 750, 1500, and for Rayleigh number equal to 30000.

For brevity, the average heat transfer coefficient results, evaluated on the impinging heated wall, for different Peclet number values and for several ratio \( L_{mf}/L_{hw} \) are showed in the Figure 2 only for \( ϕ = 4\% \) because the behavior of the average heat transfer coefficient is the same for all nanoparticles volume concentrations.

From the Figure 2(a), one can observe as the average heat transfer coefficient increases with increasing of \( L_{mf}/L_{hw} \) for Peclet values equal to 750 and 1500 showing a partial overlapping for the results in corresponding to \( L_{mf}/L_{hw} = 3.5 \) and 7; while for \( Pe \) minor of 350 the heat transfer coefficient assumes a higher value for the ratio between metal foam length and heated wall length equal to 1.

From the Figure 2(b), it is possible to see as the \( \overline{h}_{nf} \) presents an almost constant value as a function of the ratio \( L_{mf}/L_{hw} \) for Peclet numbers equal to 350, 750 and 1500; instead, for \( Pe = 100 \) and 200 the average heat transfer coefficient presents a minimum value in corresponding to \( L_{mf}/L_{hw} = 3.5 \).
Table 2. Grid Independence

| Cells Number | $\bar{h}_{\text{tot}}$ (W/m²K) | % error |
|--------------|-------------------------------|---------|
| 52800        | 2836.92                       | 0.8%    |
| 105600       | 2854.08                       | 0.2%    |
| 211200       | 2859.80                       | --------|

Figure 2. Average heat transfer coefficient for $\phi = 4\%$ as a function of (a) $Pe$ number and (b) $L_{mf}/L_{hw}$

In the Figure 3, the pressure drop as a function of Peclet number values and $L_{mf}/L_{hw}$ is showed for nanoparticle volume concentration equal to 0%. It is possible to observe that the pressure drop increases both with the increase of Peclet number for each $L_{mf}/L_{hw}$ value (Figure 3(a)) and with the increase of the ratio between the metal foam length and heated wall length for each $Pe$ (Figure 3(b)).

In the Figure 4, the ratio between the thermal and pumping power $\frac{\dot{Q}}{\dot{W}}$ is represented for $\phi = 4\%$. From the data presented in the Figure 4(a), the configuration characterized by the ratio $L_{mf}/L_{hw}$ equal to 1 is characterized by a higher value of $\frac{\dot{Q}}{\dot{W}}$ for all values of Peclet number. This means that the increase of pressure drop is higher respect to the improvement in terms of heat transfer when the metal

Figure 3. $\Delta p$ for $\phi = 0\%$ as a function of (a) $Pe$ number and (b) $L_{mf}/L_{hw}$
foam length increases. Instead, from the Figure 4(b), it is possible to see as $Q/W$ is higher for $Pe = 100$ respect to the other Peclet number values. From the combination of these data, it is possible to notice that the configuration with the highest efficiency (highest $Q/W$) is the one characterized by the lowest value of $L_{mf}/L_{hw}$ and $Pe$ for fixed properties of metal foam and for fixed nanoparticle volume concentration.

![Figure 4. $Q/W$ for $\phi = 4\%$ as a function of (a) $Pe$ number and (b) $L_{mf}/L_{hw}$ ratio](image)

5. Conclusions
The results in terms of average heat transfer coefficient evaluated on the heated wall of the system show that $\overline{h_{tot}}$ increases with increasing of $L_{mf}/L_{hw}$ for Peclet values higher than 350; while for $Pe$ minor of 350 the heat transfer coefficient assumes a higher value for $L_{mf}/L_{hw}$ equal to 1. Furthermore, $\overline{h_{tot}}$ is characterized by a constant trend for Peclet numbers equal to 350, 750 and 1500 as a function of $L_{mf}/L_{hw}$; instead, for $Pe = 100$ and 200 the average heat transfer coefficient has a minimum for a value of $L_{mf}/L_{hw}$ equal to 3.5. In conclusion, the efficiency of the system, estimated as the ratio $Q/W$, improves with decreasing both Peclet number and ratio $L_{mf}/L_{hw}$.

6. Nomenclature

| Symbol | Definition                  | Unit          |
|--------|-----------------------------|---------------|
| $C_d$  | drag coefficient            | m/s²          |
| $d_f$  | fiber diameter              | m             |
| $d_{np}$ | nanoparticle diameter     | m             |
| $d_p$  | pore diameter               | m             |
| $h_{sf}$ | interfacial heat transfer  | W/m² K       |
| $H$    | system height               | m             |
| $K$    | porous permeability         | m²            |
| $L$    | system lenght               | m             |
| PPI    | number of pores per inch   |               |
| $\dot{Q}$ | thermal power              | W             |
| $u, v$ | velocity components        | m/s           |
| $W$    | jet width                   | m             |
| $\dot{W}$ | pumping power              | W             |

Greek symbols

| Symbol | Definition                | Unit         |
|--------|---------------------------|--------------|
| $\alpha_{sf}$ | specific surface area    | m⁻¹          |
| $\varepsilon$ | porosity                 |              |
\( \phi \)  
nanoparticle volume concentration

**Subscripts**

bf  
base fluid

mf  
metal foam

nf  
nanofluid

np  
nanoparticle

s  
solid phase of metal foam

**References**

[1] Paulraj M P, Sahu S K 2019 *Num. Heat Transf., Part A: Applications* **76** (4) pp. 232-253.

[2] Lamraoui H, Mansouri K, Saci R *J. Non-Newtonian Fluid Mechanics* **265** pp. 11-27.

[3] Barewar S D, Tawri S, Chougule S S 2019 *Chem. Eng. Processing - Process Intensification* **136** pp. 1-10.

[4] Allauddin U, Mahrulk M, Rehman N U, Haque M E, Uddin N 2018 *Num. Heat Transf., Part A: Applications* **74** (8) pp. 1486-1502.

[5] Yousefi-Lafouraki B, Ramiar A, Ranjarbar A A 2018 *Particuology* **39** pp. 78-87.

[6] Izadi A, Siavashi M, Xiong Q 2019 *Int. J. Hydrogen En.* **44 (30)** pp. 15933-15948.

[7] Siavashi M, Rasam H, Izadi A 2019 *J. Therm. Analysis Calorimetry* **135 (2)** pp. 1399-1415.

[8] Nield D A, Bejan A, "Convection in Porous Media", 4th Edition, Springer, New York, 2013.

[9] Whitaker S, “The Method of Volume Averaging”, Springer, Netherlands, 1998.

[10] Bhattacharya A, Calmidi V V, Mahajan R L 2002 *Int. J. Heat Mass Transf.* **45** pp. 1017-1031.

[11] Calmeci V V, Mahajan R L 2000 *ASME J. Heat Transf.* **122** pp. 557-565.

[12] Khanafar K, Vafai K, Lightstone M 2003 *Int. J. Heat Mass Transf.* **46** (19) pp. 3639–3653.

[13] Pak B C, Cho Y I 1998 *Exp. Heat Transf.* **11** (2) pp. 151–170.

[14] Nan C W, Birringer R, Clarke D R, Gleiter H 1997 *J. Appl. Physics* **81** (10) pp. 6692–6699.

[15] Buonomo B, Lauriat G, Manca O, Nardini S 2016 *Int. J. Heat Mass Transf.* **98** pp. 484–492.

[16] Bondareva N S, Buonomo B, Manca O, Sherepet M A 2019 *Energies* **12** (11) 2074.