

NUMERICAL MODELING OF OPEN-CELL METAL FOAMS THERMAL EFFICIENCY

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- Numerical techniques were used to calculate the extended surface efficiency and analyze the impact of the foam's thickness on its efficiency.
- It was considered copper and nickel foams with different thicknesses and two dielectric fluids, HFE-7100 and Ethanol.
- The foam efficiency increases as the thickness decreases.
- The Cu foam shows higher foam-finned efficiency than the Ni foam.
- Metal foam under pool boiling of HFE- 7100 has higher efficiency than pool boiling using Ethanol.
- The classical pin fin model with an adiabatic tip was the one that most suitably represents the efficiency of the foams.

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Abstract: This work used numerical techniques to calculate the extended surface efficiency and analyze the impact of the foam's thickness on its efficiency. Heat conduction was simulated in an open-cell metal foam of copper and nickel with three different thicknesses, 3 mm, 2 mm, and 1 mm, under convection boundary conditions obtained from pool boiling experiments with two dielectric fluids, HFE-7100 and Ethanol, at saturation conditions and atmospheric pressure. The geometries of the foams were segmented from μ CT images, converted to the stereolithography (STL) format, and used to build the computational meshes. The numerical simulations were carried out in foam-extend-4.0. The numerical extended surface efficiency was compared with classical analytical models and other correlations from the literature. The foam efficiency increased as the thickness decreased, *i.e.*, the thinnest foams showed better efficiency than the other ones. Metal foam under pool boiling of Ethanol had lower efficiency than HFE-7100. The pin fin model with adiabatic tip exhibited the lowest mean absolute error, lower than 10% for copper foams and 20% for nickel foams; the copper foam with 1 mm and HFE-7100 had the lowest error, approximately 1%. On the other hand, the models that consider a three-dimensional matrix presented errors higher than 30%.

Keywords: thermal efficiency; metal foam; numerical analysis; foam thickness; convective boiling.

1. Introduction

Direct-immersion cooling with two-phase change can increase power density and energy efficiency in electronic cooling [1,2]. The dielectric liquid cooling used to avoid a short-circuit has relatively poor thermophysical properties and an extremely small contact angle, requiring a large superheat to initiate the boiling process [3].

Heat transfer in immersion cooling begins with natural convection at low surface temperature. With further increase in the surface superheating, nucleate boiling starts up to the critical heat flux (CHF). Electronic devices should operate with temperatures below this superheat value to prevent reaching the CHF. In addition, reports show that this operating temperature must be lower than 85 °C [4]. Therefore, the use of an engineered surface has been studied to meet the cooling requirements of modern electronic devices [5,6].

Open-cell metal foams, a type of porous structure, were tested by Manetti et al. [7,8] and their findings showed that the metal foam surfaces provided a higher heat transfer coefficient (HTC) than plain surfaces and prevented thermal overshoot at the onset nucleate boiling. Moreover, the optimal foam thickness varies with the heat flux, and the thermal conductivity of the foam plays a role in the foam-finned efficiency.

Previous works tried simplifying models of the heat conduction in metal foams to obtain the foam-finned efficiency. Dukhan et al. [9] presented a one-dimensional heat transfer model for open-cell metal foams, assuming the geometry of the solid ligaments as a bank of cylinders (pin fin). Ghosh [10,11] modeled forced-convection heat transfer in an open-cell metal foam attached to an isothermal surface by assuming the foam matrix as a repetitive cubic structure. He considered the forced-convective heat transfer coupled with heat conduction through the foam fibers. Analytical expressions have been derived for the gas-solid temperature difference, the total heat transfer through the foam, and the efficiency of the foam as an extended surface in analogy with the traditional fin theory. Mancin et al. [12,13] developed a simplified scheme for overall foam-

finned surface efficiency, which considered the metal foam specific area and an equivalent length proposed by the authors.

An alternative approach to study the heat transfer in open-cell metal foams is to use computational simulations and several works in the literature employed them [14]. However, the majority of these works focused mainly on the flow through the foam structure. For example, Dixit and Ghosh [15] and Ranut et al. [16] presented a computational simulation of the flow through a three-dimensional foam structure reproduced using X-ray μ -CT. In contrast, Kopanidis et al. [17] modeled the metal foam geometry as a basic cell, *i.e.*, a matrix of polyhedral-shaped volumes with a triangular cross-section. Also, another group of studies investigated the effective thermal conductivity of metal foams by using computational simulation of conductive heat transfer [18–20]. Khaled [21] went further and, by reconstructing the foam geometries from μ -CT scans, studied the effect of assembly methods between the heat sink base and the metal foam fin. The authors employed a finite element model to perform their simulations and explored the effect of forced convection, fin orientation, and the number of fins on the efficiency, effectiveness, and thermal resistance.

Nevertheless, none of the mentioned previous works investigated the influence of the foam thickness on its operating parameters, such as efficiency. They exemplify, however, how flexible numerical techniques can be to study different physical aspects of the flow and heat transfer through metal foams if their geometries can be obtained. In this scenario, we performed numerical simulations of heat conduction in open-cell metal foams of two different materials, which geometries were obtained from μ -CT scans with different thicknesses. Our goal is to investigate the impact of varying thickness on the metal foam-finned efficiency and effectiveness and assess the analytical model (or models) of foam-finned efficiency that can be most suitably used to predict their thermal efficiency.

2. Materials and Methods

2.1 Foam geometries

The metal foams of copper (Cu foam) and nickel (Ni foam) with an open structure and original thickness of approximately 3 mm were used to obtain the geometries for the numerical simulations. First, the 3 mm-thick foams were scanned using micro computational tomography (μ CT) using a Skyscan 1272 at a resolution of 15 μ m (100 kV X-ray source voltage). Subsequently, the foams images were segmented by using the Vascular Modeling Toolkit (VMTK)[®] library [22] by using the level-sets segmentation technique and, then, converted to the stereolithography (STL) format using the marching cubes algorithm [23]. Although primarily used to segment vascular structures, Vascular Modeling Toolkit (VMTK)[®] library [22] has a suitable framework to deal with any medical-type images, like the ones generated by μ CT scans. The segmented surfaces were reduced to one-eighth of their original size to decrease the computational cost of the mesh generation (Figure 1a and Figure 1b show the resulting geometry for the Cu and Ni foams, respectively). This approach is reasonable because the structure of the foams is homogeneous. Finally, we cut the 3 mm-thick foams along a plane parallel to their base (indicated in Figure 1a) to obtain two geometries with 2 mm and 1 mm in thickness, as shown in Figure 1c (each new geometry is indicated and illustrated with different opacities). This procedure followed closely the one performed by Manetti et al. [8] in their experiments.

The μ CT images were also used to characterize the foam surface area, a_{sf} , and granulometry (porous and fiber diameters) using the iMorph software, as reported by Manetti et al. [24]. Table 1 shows the main characteristics of Cu foam and Ni foam.

Table 1. Metals foams characteristics from Manetti et al. [24]

Foam	PPI ^a [in ⁻¹]	ϵ [%]	a_{sf} [mm ⁻¹]	d_p [mm]	d_f [mm]
Cu	31.75	90.0	2166	0.46	0.13
Ni	62.72	98.4	5133	0.25	0.07

^aPores Per Inch

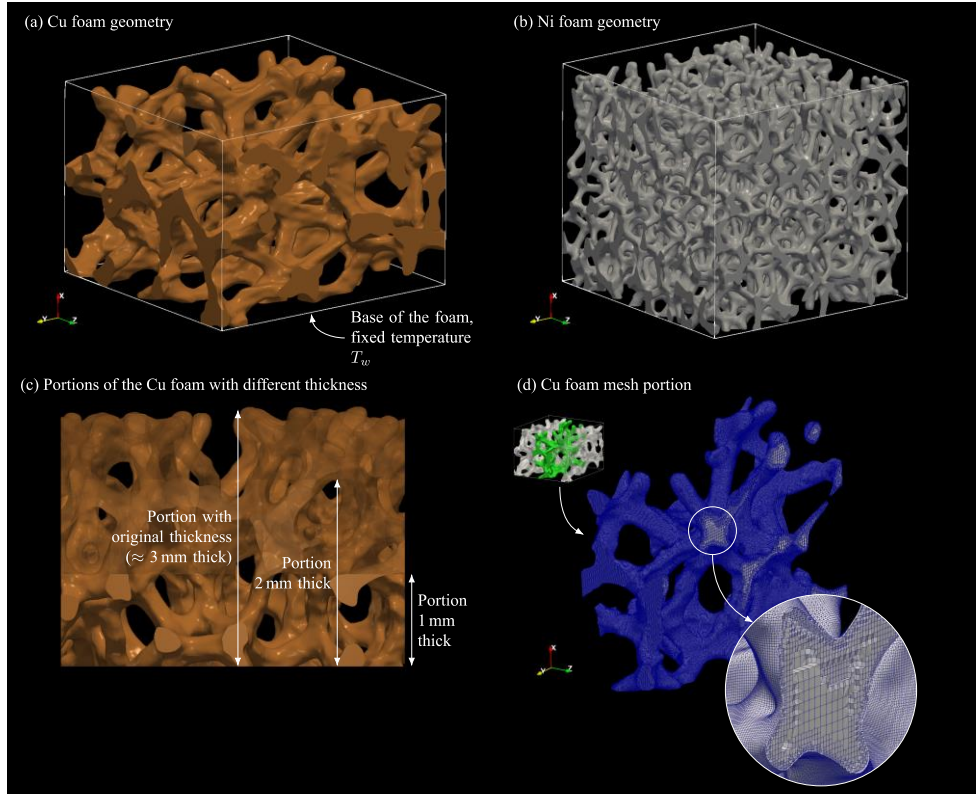


Figure 1. (a) Cu foam geometry with an indication to its base where a fixed temperature, T_w , was applied as the boundary condition for the numerical simulations; (b) Ni foam geometry; (c) indication of the plane cuts performed in the 3 mm-thick foam to obtain the geometries 1 and 2 mm thick; (d) details of the Cu foam computational mesh (green part in the thumbnail at the top left corner) showing its hexahedral-dominant cells structure.

2.2 Computational Meshes and Numerical Strategies

We used the open-source foam-extend library, version 4.0, to build the computational meshes and carry out the numerical simulations. This library implements the Finite Volume Method (FVM) to discretize partial differential equations, which guarantees second-order accuracy. The steady-state 3D conduction equation was solved for the temperature until its convergence with a normalized residual tolerance of 1×10^{-12} . The second-order accurate central differences scheme, with non-orthogonal correction, was used to discretize the Laplacian term of the conduction equation [25].

The computational meshes were created using the utility "cartesianMesh" of foam-extend (within the library cfMesh), which automatically builds polyhedral hexahedral-dominant meshes for better accuracy of the numerical solution. The computational meshes of the Cu foam for each thickness, as shown in Figure 1b, had approximately from 1 to 4 million cells to ensure a mesh-independent solution (the mesh-independence test was performed with the geometry of 3 mm thickness, yielding changes of less than 1% in the calculated heat flux) and with cells small enough to better capture the foam surface curvature. Figure 1d shows a portion of the Cu foam mesh with 3 mm (indicated in green in the thumbnail at the top left corner) with a detail of the mesh structure obtained with the cartesianMesh utility.

2.3 Boundary Conditions

The conduction problem's boundary conditions (BCs) were: fixed temperature on the base of the foam, T_w (indicated in Figure 1a), and the convective condition on the foam's surface. The base temperature, T_w , and the HTC used in the convective BC were obtained from Manetti et al. [24], who carried out pool boiling experiments with two dielectric fluids, HFE-7100 and Ethanol, at saturation conditions and atmospheric pressure. For each thickness, both fluids were tested. Table 2 shows the BCs of the cases simulated, each considering a surface-fluid combination for four heat fluxes (low, low-medium, medium, and high heat flux).

Table 2. Boundary conditions employed for the metal foams with different thicknesses and different fluids obtained from experiments performed by Manetti et al. [24]

Foam thickness	Fluid	Case	Cu foam			Ni foam		
			T_w	h_{exp}	q''	T_w	h_{exp}	q''
			[K]	[kW/m ² ·K]	[kW/m ²]	[K]	[kW/m ² ·K]	[kW/m ²]
3 mm	HFE-7100 ^a	A	337.36	4.87	20.44	338.64	3.41	18.81
		B	339.51	7.16	45.78	340.47	6.10	44.77
		C	345.99	11.32	146.02	343.18	9.38	94.60
		D	357.21	12.34	294.80	344.90	9.83	118.54
	Ethanol ^b	A	356.86	9.54	56.31	357.56	8.94	57.14
		B	358.14	14.49	104.47	358.97	12.99	101.47
		C	361.05	24.48	244.46	361.17	23.42	234.62
		D	364.50	27.39	367.98	363.92	25.00	318.68
2 mm	HFE-7100	A	337.11	5.19	20.73	337.79	3.85	18.34
		B	338.71	8.16	47.64	340.29	6.54	47.76
		C	342.85	15.11	149.51	342.34	10.11	94.58
		D	351.12	16.80	304.27	344.06	10.66	119.39
	Ethanol	A	356.72	9.61	54.42	357.09	9.95	58.27
		B	358.71	13.03	100.21	359.90	12.19	105.52
		C	360.95	27.74	276.51	362.13	21.57	236.73
		D	362.96	32.93	369.50	363.07	27.33	326.32
1 mm	HFE-7100	A	339.99	2.81	19.57	338.09	3.84	19.15
		B	340.05	6.57	46.15	340.05	6.81	47.81
		C	343.37	13.86	145.00	342.80	9.94	97.40
		D	348.47	18.94	293.96	343.83	11.53	124.96
	Ethanol	A	358.85	7.01	53.63	358.22	7.89	55.14
		B	358.99	13.02	104.01	360.19	11.66	104.65
		C	360.78	24.27	237.52	363.45	21.41	255.52
		D	362.56	32.82	379.65	365.21	25.64	350.92

^a $T_{\infty, HFE-7100} = 333.3$ K; ^b $T_{\infty, Ethanol} = 351.1$ K.

2.4 Data Analysis

With the results of the numerical experiments, we calculated the total heat transfer rate from the surface of the foam to the fluid as follow:

$$q_{foam} = \int_{A_{foam}} h_{exp} [T(x) - T_{\infty}] dA_{foam} \quad (1)$$

which allow us to calculate the foam-finned efficiency,

$$\eta_{foam} = \frac{q_{foam}}{q_{max}} \quad (2)$$

and the foam effectiveness,

$$\epsilon_{foam} = \frac{q_{foam}}{hA_{c,b}(T_w - T_{\infty})} \quad (3)$$

where

$$q_{max} = h_{exp} A_{foam} (T_w - T_{\infty}) \quad (4)$$

is the maximum heat transfer rate expected, *i.e.*, if the foam was entirely at the base temperature; and $A_{c,b}$ is the cross-sectional area at the base of each foam.

The foam efficiency calculated with Eq. (2) was compared with analytical models from the literature (three classical ones presented in a heat transfer technical book [26]), consisting of a circular pin fin of uniform cross-section and one-dimensional heat transfer. As Dukhan et al. [9], the metal foams were simplified as a bank of one-dimensional circular cylinders with a diameter equals to the foam fiber diameter, d_f .

The first classical analytical model used for comparison was the well-known model of the fin with an adiabatic tip. The foam-finned efficiency can be calculated by using:

$$\eta_{foam,pin-fin-1} = \frac{q_{pin-fin-1}}{q_{max}} = \frac{\tanh(mL)}{mL} \quad (5)$$

where

$$m^2 = \frac{hP}{k_s A_c} = \frac{4h}{k_s d_f} \quad (6)$$

and L is the thickness of the foam.

The second classical analytical model was the fin with convection heat transfer on the tip in which the foam-finned efficiency can be calculated as follows:

$$\eta_{foam,pin-fin-2} = \frac{q_{pin-fin-2}}{q_{max,pin-fin}} \quad (7)$$

where

$$q_{pin-fin-2} = M \frac{\sinh mL + \left(\frac{h}{mk}\right) \cosh mL}{\cosh mL + \left(\frac{h}{mk}\right) \sinh mL} \quad (8)$$

and

$$M = \sqrt{hPA_c}(T_w - T_\infty) \quad (9)$$

Finally, the third classical analytical model was the infinite fin with foam-finned efficiency calculated as:

$$\eta_{foam,pin-fin-3} = \frac{q_{pin-fin-3}}{q_{max,pin-fin}} = \frac{M}{q_{max,pin-fin}} \quad (10)$$

where the maximum heat transfer rate, $q_{max,pin-fin}$, for the last two models were calculated as:

$$q_{max,pin-fin} = hA_{pin-fin}(T_w - T_\infty) \quad (11)$$

and $A_{pin-fin}$ is the convective area of the pin fin.

Furthermore, the results were also compared with specific models for open-cell foams. The first one was from Ghosh [10], based on the traditional fin theory. The model considers the random geometry of the solid ligaments as a repetitive cubic structure. The authors calculated the foam-finned efficiency as follows:

$$\eta_{foam,Ghosh} = \frac{\tanh ML}{ML} \quad (12)$$

where

$$M = \sqrt{\frac{hP}{k_s A_c}(1 + 4\eta_{1/2})} = m\sqrt{1 + 4\eta_{1/2}} \quad (13)$$

and

$$\eta_{1/2} = \frac{\tanh(m d_p/2)}{(m d_p/2)} \quad (14)$$

due to cross-connections in the foam filaments.

The other two specific models were obtained from Mancin et al. [12,13], who developed a simplified scheme for overall foam-finned surface efficiency that takes into account the foam surface area,

$$\eta_{foam,Mancin} = \frac{1 + \Omega \cdot a_{sf} \cdot L}{1 + a_{sf} \cdot L} \quad (15)$$

where

$$\Omega = \frac{\tanh(m_{eq} \cdot L_{eq})}{(m_{eq} \cdot L_{eq})} \quad (16)$$

with m_{eq} and L_{eq} being empirical factors adjusted to the authors' experimental data. According to Mancin [12]:

$$m_{eq} = \sqrt{\frac{4h}{k_s d_f}} \quad (17)$$

$$L_{eq} = 6.6 \cdot L \cdot PPI^{0.99} (0.0254 - d_f \cdot PPI) \quad (18)$$

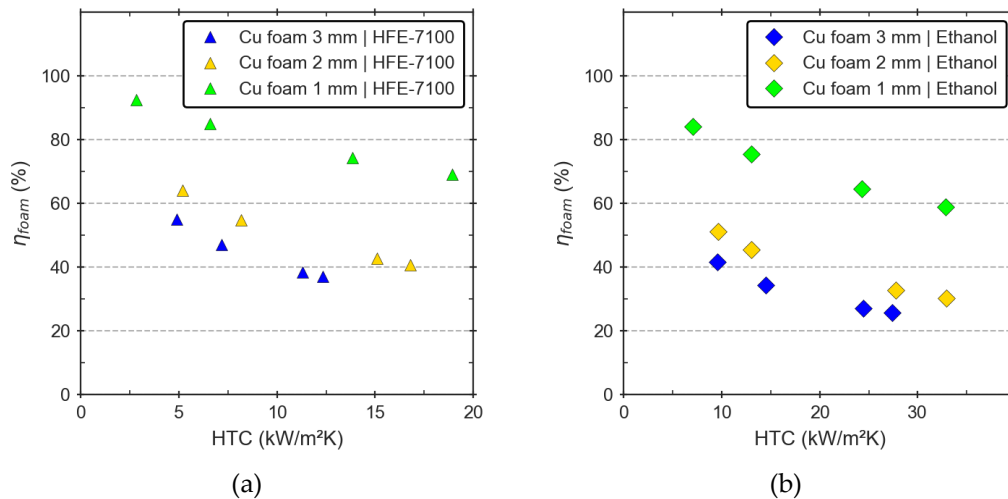
Moreover, according to Mancin [13]:

$$m_{eq} = \sqrt{\frac{4h}{k_s d_f}} \left(\frac{k_s}{k_l} \right)^{-0.52} \quad (19)$$

$$L_{eq} = 1055 \cdot L^{1.18} \cdot PPI (0.0254 - d_f \cdot PPI)^{0.66} \quad (20)$$

3. Results and Discussion

In general, for all surfaces-fluid combinations, the first case of BCs presented the best foam-finned efficiency, as shown in Figure 2. According to the experimental work of Manetti et al. [7], at low heat flux, there are just a few bubbles on the surface, which causes a lower HTC and the temperature field is more uniform in the foam. Hence the foam heat transfer rate, q_{foam} , is close to the maximum heat transfer rate, q_{max} . As the experimental heat flux increases, the wall temperature (foam base) increases, more nucleation sites are activated and more bubbles rise from the surface, increasing the HTC. On the other hand, the foam-finned efficiency decreases because the higher HTC dissipates the heat faster, and therefore, there is no need to use the entire foam thickness. Moreover, as the thickness decreases, the foam-finned efficiency increases; so, the thicker foams have a significant part of their length useless, as can be seen in Figure 3 to Figure 6. Therefore, there is a balance between thickness, efficiency, and heat transfer rate.



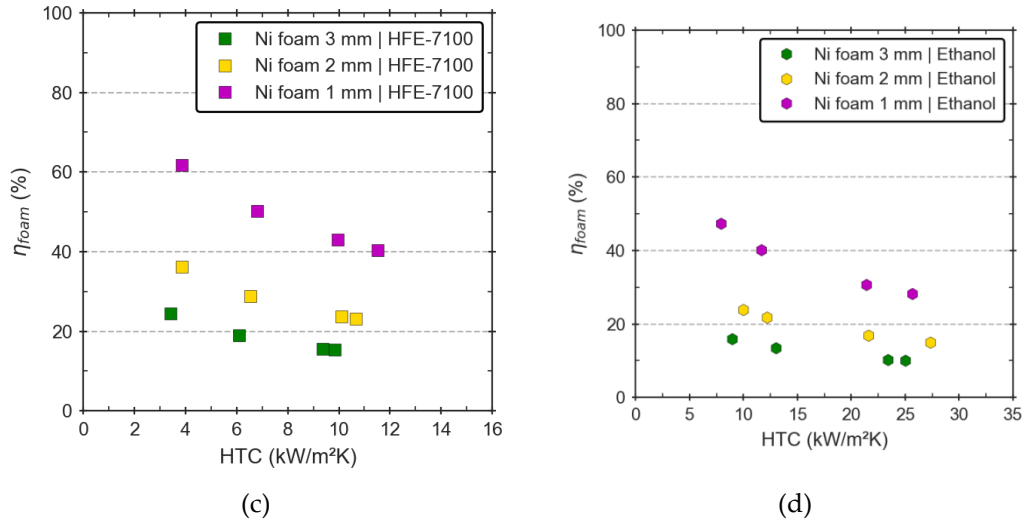


Figure 2. Foam-finned efficiency as a function of the experimental HTC for different foam thicknesses. (a) Cu foam in HFE-7100; (b) Cu foam in Ethanol; (c) Ni foam in HFE-7100; and (d) Ni foam in Ethanol.

Furthermore, by comparing the same thickness and fluid, but different materials, as shown in Figure 3 and Figure 4, or Figure 5 and Figure 6, it is reasonable to expect that the efficiency of the Cu foam to be higher than the Ni one, as suggested by the plots in Figure 2, because the thermal conductivity, k_s , and the foam fiber diameter are higher for Cu foam than Ni foam.

For different fluids, the foam-finned efficiency is higher for HFE-7100 (approximately 40%) than for Ethanol. Figure 3 and Figure 5 show the temperature fields of the numerical results of the extreme cases (A and D) for all three Cu foams thicknesses and both fluids, HFE-7100 and Ethanol, respectively. The Cu foam of 1 mm presents a temperature field relatively uniform, mainly for Case A, while 2 mm and 3 mm present a temperature gradient. This behavior is related to the HTC, which is higher for Ethanol, mainly due to the thermo-physical properties, as explained by Manetti et al. [24].

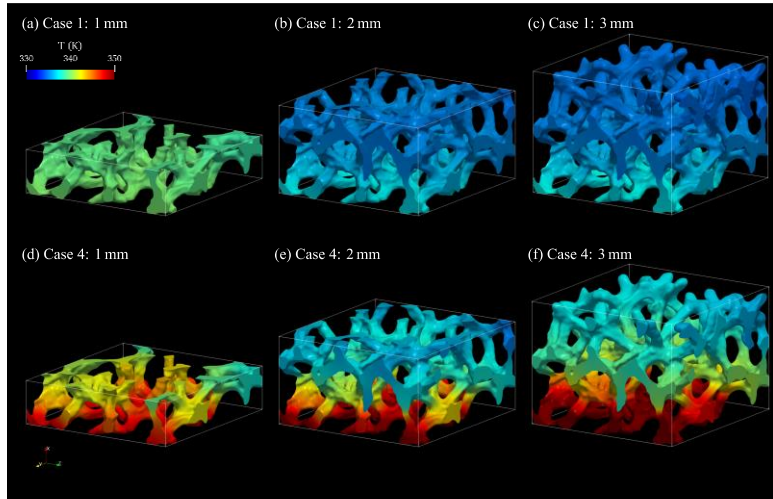


Figure 3. Temperature distribution in the Cu foam with different thicknesses and Cases A and D for HFE-7100.

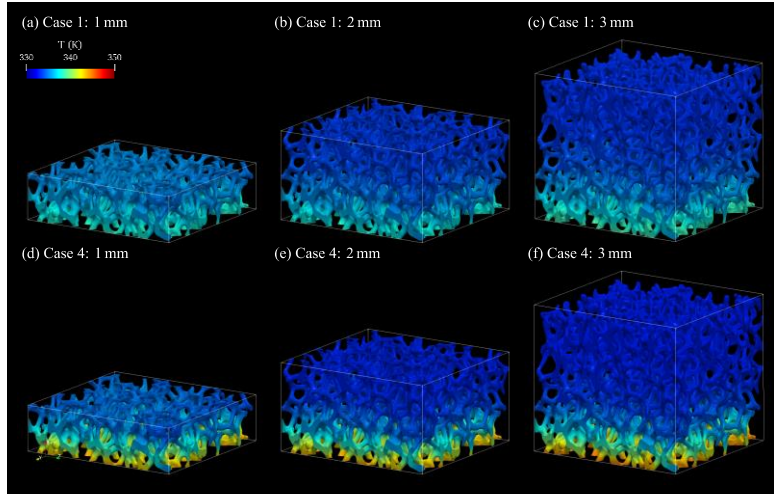


Figure 4. Temperature distribution in the Ni foam with different thicknesses and Cases A and D for HFE-7100.

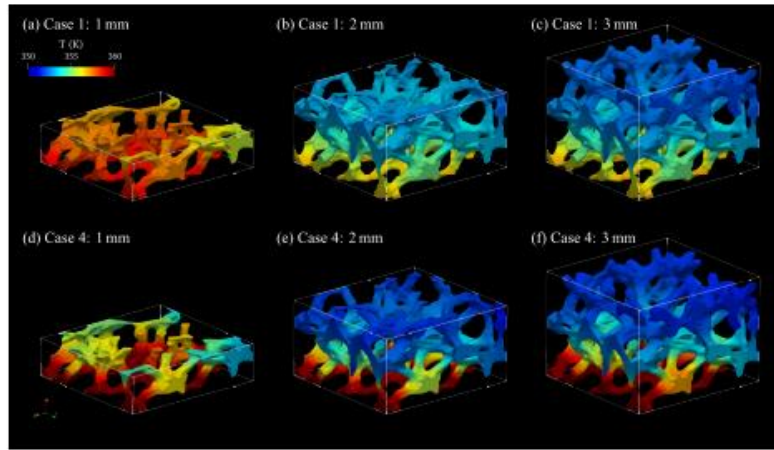


Figure 5. Temperature distribution in the Cu foam with different thicknesses and Cases A and D for Ethanol.

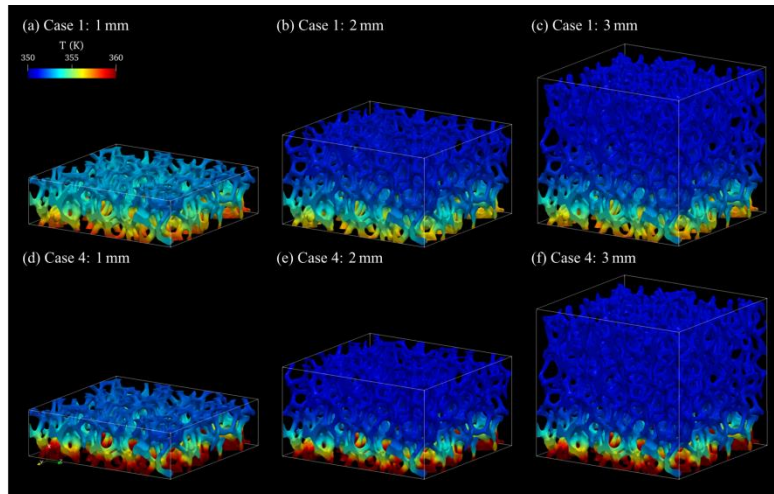


Figure 6. Temperature distribution in the Ni foam with different thicknesses and Cases A and D for Ethanol.

Similarly to the behavior shown in Figure 2 for the foam efficiency, the foam effectiveness with the first set of BCs also showed the best foam-finned effectiveness among all surfaces-fluid combinations, as shown in Figure 7 (however, the foam effectiveness does not have the unity as the upper limit, as the foam efficiency has). Moreover, the effectiveness also decreases as the HTC increases, whereas, as foam thickness decreases, the effectiveness exhibited a different behavior than the efficiency.

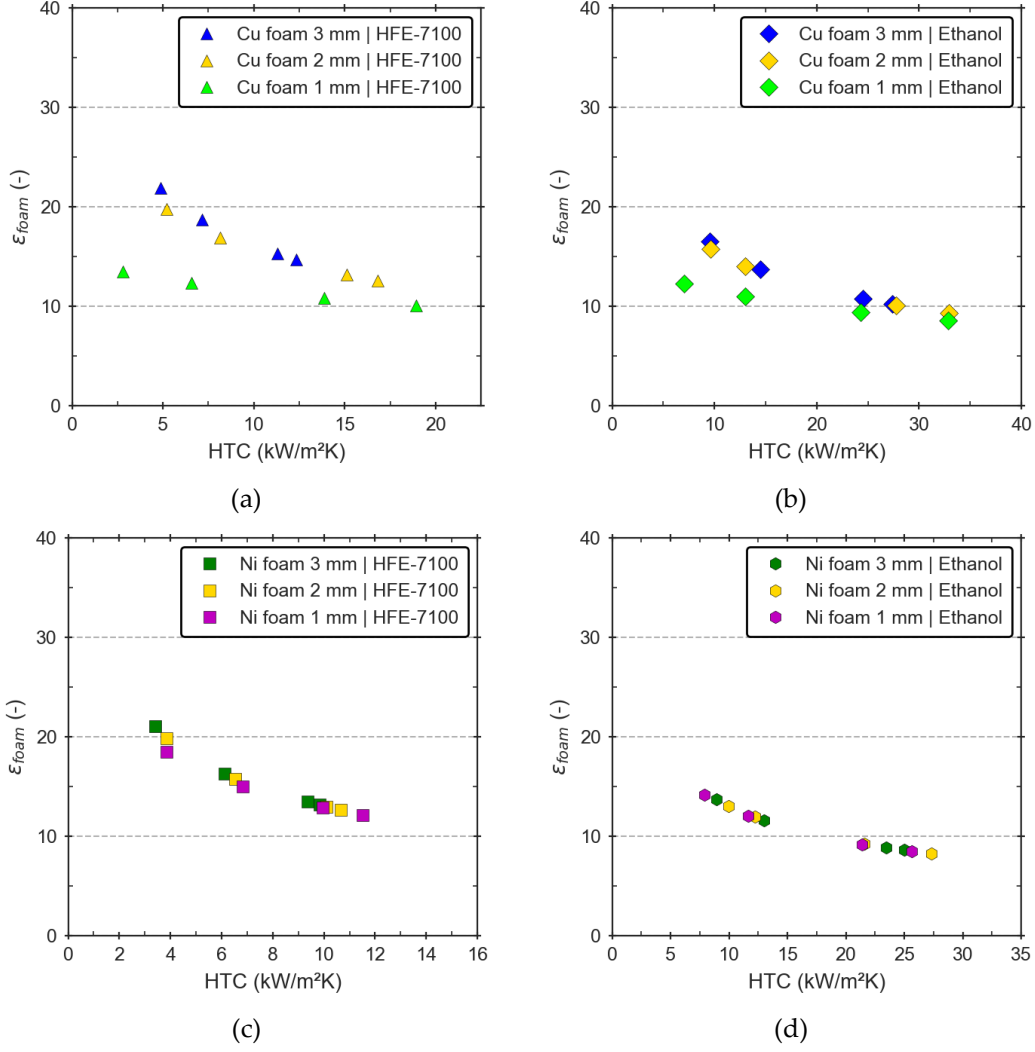


Figure 7. Foam-finned effectiveness as a function of experimental HTC for different foam thicknesses. (a) Cu foam in HFE-7100; (b) Cu foam in Ethanol; (c) Ni foam in HFE-7100; and (d) Ni foam in Ethanol.

The Cu foam's effectiveness at the low and medium heat fluxes is very similar between 3 mm and 2 mm, and higher than the foam's effectiveness for 1 mm. This indicates that the thicker Cu foams dissipate more heat than the thinner ones even with lower efficiency, supported by Manetti et al. [8]. As the heat flux increases, the HTC increases as well, and the effectiveness tends to be equal for the three thicknesses while the foam-finned efficiency of the thinner ones is higher (see Figure 3). Therefore, larger foam thicknesses improve the foam-finned effectiveness but negatively impact the foam-finned efficiency. According to Dixit and Ghosh [27], the selection of an optimum foam thickness becomes necessary due to these two opposing effects. In this way, Manetti et al. [8] proposed an inverse S-shaped curve correlating the optimum foam thickness and the heat flux.

For the Ni foam, with a thermal conductivity 4.5 times lower than the Cu foam, the effectiveness is almost independent of the thickness. This behavior was also noted by Dixit and Ghosh [27] in their parametric work by using the Ghosh [10] model, wherein foams materials with low thermal conductivity exhibited effectiveness almost independent of their length. Therefore, the heat dissipated in the thicker foam is almost the same dissipated in the thinner one.

3.1. Comparison between the numerical results and the efficiency models

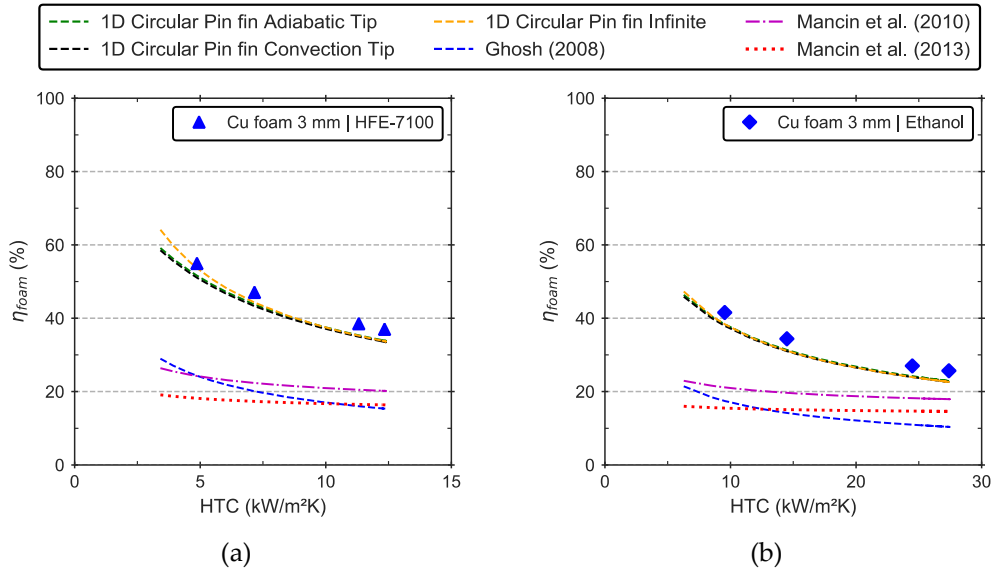
Figure 8 to Figure 10 compares the numerical results and the efficiency predicted by models presented in Section 2. The mean absolute percentual error (MAPE) for each analytical model relative to the numerical data

is shown in each plot. The classical models (circular pin fin models) presented lower errors than the foam-specific models for all foams.

For classical models, similar behavior was found for the thickness of 3 mm, regardless of foam material and working fluid (MAPE was close to 10%). Therefore, for 3 mm, the temperature at the fin tip is equal or very close to the fluid temperature; hence the infinite tip boundary conditions fit well with the numerical results. In addition, the tip cross-section is small and both the convective or adiabatic boundary conditions yield the same predictions. As the thickness decreases, the material of the foam plays a vital role in the comparison. For the Cu foam, the MAPE of the infinite tip model increased up to 56.5%, whereas the MAPE of the convective and adiabatic tips decreased. Therefore, for thicknesses lower than 3 mm, the temperature at the top of the foam is different from the fluid temperature; also, the thickness of 1 mm with HFE-7100 was the case in which the adiabatic tip model showed the lowest error, approximately 1%, probably due to the convection area, which is more significant in the radial area (lateral) than in the axial area (tip).

On the other hand, as the thickness of the Ni foam decreased, the infinite tip conditions showed a good agreement with the numerical results as well as the convective and adiabatic tips conditions. Therefore, the temperature at the top of the foam with a thickness of 1 mm is very close to the fluid temperature. Overall, the adiabatic tip model showed the lowest average from the MAPE of the three thicknesses of each surface-fluid combination.

The foam-specific models presented larger MAPE, higher than 30%, probably due to the particular modeling conditions used in Mancin et al. [12,13], who used empirical factors from experimental tests with forced convection of air. Ghosh's model [10] exhibited a similar trend as the 1D circular pin fin. However, the values are of the order of two times smaller. The lowest efficiency presented by Ghosh's model should be related to the term $\eta_{1/2}$, which in turn is related to the cross-connection in the simple cubic structure. Therefore, the one-dimensional pin fin model provides efficiencies closer to the numerical simulation results for the boiling process.



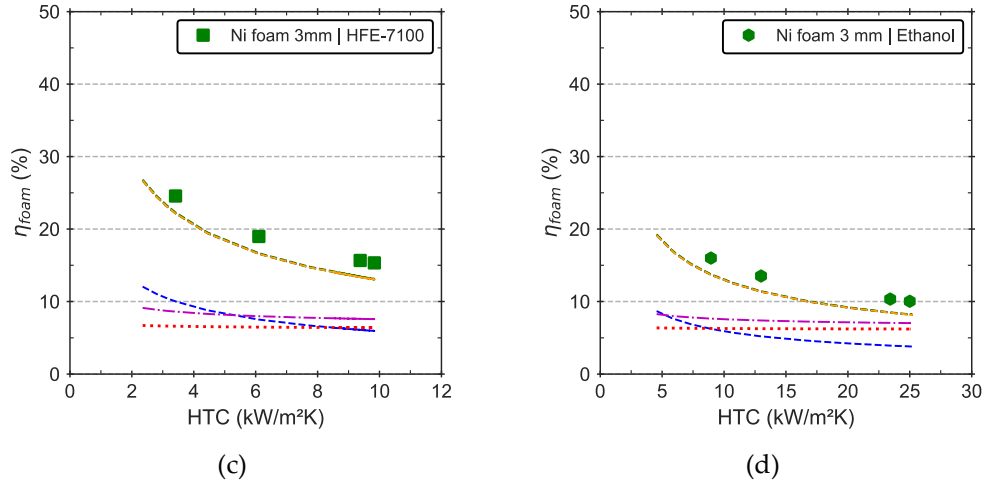


Figure 8. Comparison for foam efficiency between numerical results and analytical models for the 3 mm-thick foam: (a) Cu foam in HFE-7100; (b) Cu foam in Ethanol; (c) Ni foam in HFE-7100; and (d) Ni foam in Ethanol.

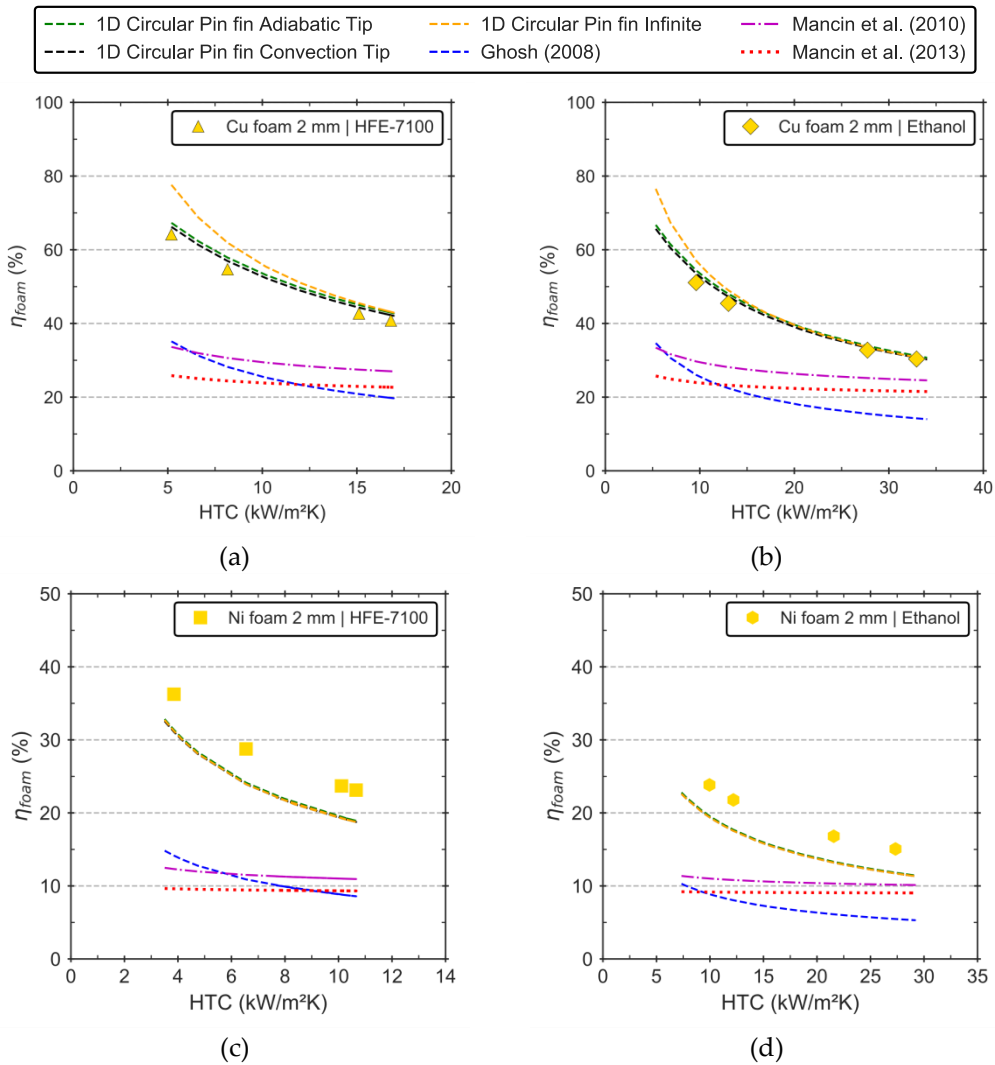


Figure 9. Comparison for foam efficiency between numerical results and analytical models for the 2 mm-thick foam: (a) Cu foam in HFE-7100; (b) Cu foam in Ethanol; (c) Ni foam in HFE-7100; and (d) Ni foam in Ethanol.

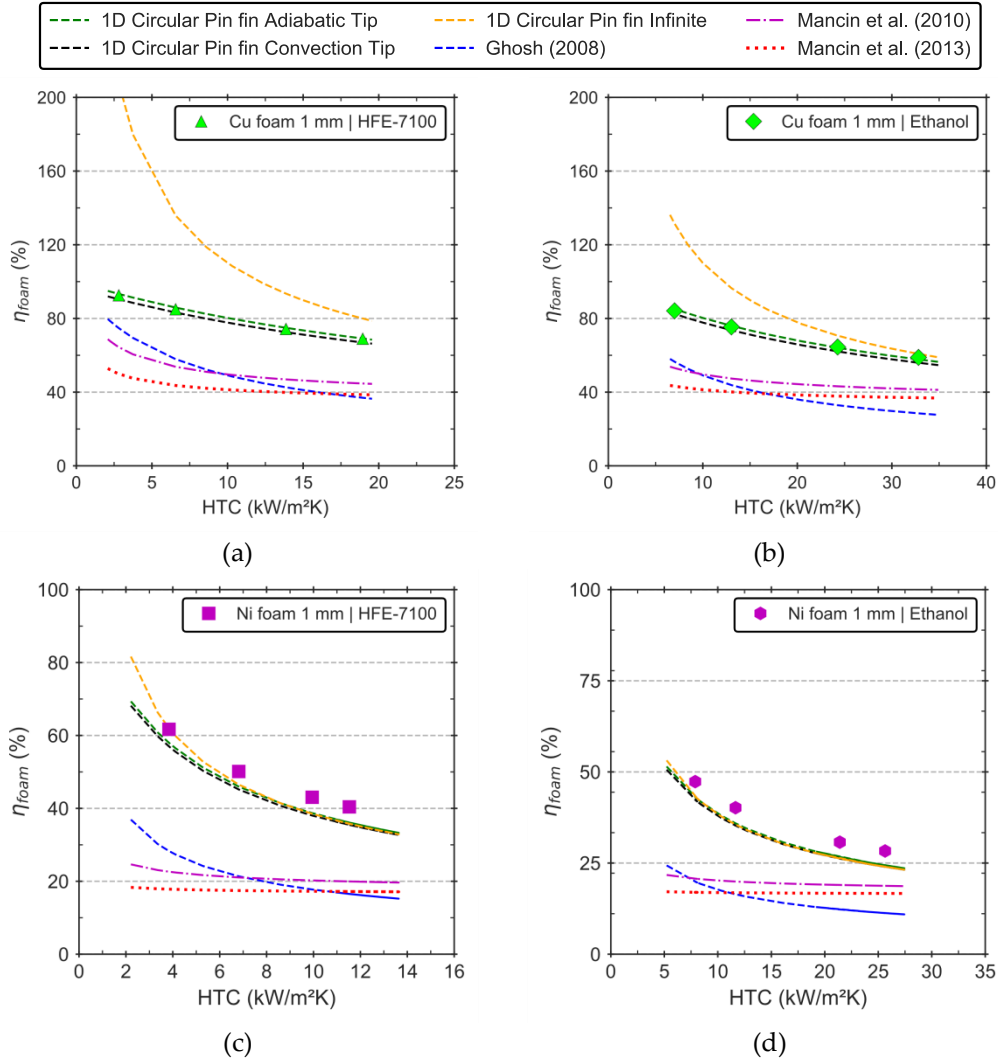


Figure 10. Comparison for foam efficiency between numerical results and analytical models for the 1 mm-thick foam: (a) Cu foam and HFE-7100; (b) Cu foam and Ethanol; (c) Ni foam and HFE-7100; and (d) Ni foam and Ethanol.

5. Conclusions

This work presented a numerical study of heat conduction in copper and nickel foams using the foam-extend library with BCs based on experimental data obtained by pool boiling of dielectric fluids to assess the impact of different foam thicknesses on its efficiency and effectiveness, while also comparing which foam-finned efficiency analytical models better predicted the numerical data.

The numerical results represented well the findings of our previous experimental works, where it was shown that the foam-finned efficiency increases as the thickness decreases, and both the foam-finned efficiency and effectiveness decrease as the heat flux increases.

In general, the thinnest the foam, the highest the foam-finned efficiency, since the temperature fields in the thinnest foams are more uniform and closer to the wall temperature. Moreover, pool boiling using HFE-7100 as the working fluid has higher efficiency than pool boiling using Ethanol due to the lowest heat transfer coefficient for HFE-7100, which allows the heat conduction in a larger part of the foam length compared to Ethanol.

More specifically, the Cu foam showed higher foam-finned efficiency than the Ni foam, mainly due to the higher thermal conductivity of the copper and larger fiber diameter of the Cu foam.

The behavior of foam effectiveness and foam efficiency was different. For the Cu foam, as the thickness decreases, the effectiveness decreases for lower and medium HTC; however, the effectiveness tends to be

equal at high HTC. On the other hand, for the Ni foam, the foam-finned effectiveness is quite the same independently of the HTC and thickness.

These results regarding the foam thicknesses can be helpful as a guide to select the optimum thickness for operational conditions of foams in applications such as direct-immersion cooling. In this case, our study is the first step to complete modeling the pool boiling in metal foams where the phase-change should be included in the simulations, solving for the heat interaction between the foam and the fluid. Nevertheless, this is substantially challenging due to the complex foam geometry.

Finally, the classical pin fin model with an adiabatic tip was the one that most suitably represented the efficiency of the foams, with the lowest MAPE compared to the numerical data, lower than 10% for Cu foams and 20% Ni foams, regardless of thickness or working fluid. Therefore, this model can be used as a guide during the design step to determine the metal foam characteristics of a cooling system.

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Nomenclature

A	Area	[m ²]
$A_{c,b}$	Cross-sectional area at the base of foam	[m ²]
$A_{c,b}$	Fin Cross-sectional area	[m ²]
a_{sf}	Metal foam specific area or density area	[m ² /m ³]
d	Diameter	[m]
h	Heat transfer coefficient	[W/m ² ·K]
k	Thermal conductivity	[W/m·K]
L	Foam thickness	[m]
m	Factor defined in Eq. (6)	[-]
M	Factor defined in Eq. (9)	[-]
MAPE	Mean absolute percentual error	[%]
PPI	Pores per inch	[in ⁻¹]
q	Heat transfer rate	[W]
q''	Heat flux	[W/m ²]
T	Temperature	[K]
T_{∞}	Temperature of the convective fluid	[K]
x	Cartesian axis direction	[m]

Special characters

ε	porosity	[-]
ϵ	Fin effectiveness	[-]
η	Fin Efficiency	[-]
$\eta_{1/2}$	Efficiency of the half struct from Ghosh [10]	[-]
Ω	Factor defined in Eq. (16)	[-]

Subscripts

Cu	Copper
eq	equivalent
exp	Experimental

<i>f</i>	fiber
<i>foam</i>	Metal foam
<i>pin-fin-1</i>	fin with an adiabatic tip
<i>pin-fin-2</i>	fin with convective tip
<i>pin-fin-3</i>	fin with infinite length
<i>Ghosh</i>	From Ghosh [10]
<i>l</i>	Saturated liquid
<i>Mancin</i>	from Mancin et al. [12,13]
<i>max</i>	Maximum
<i>Ni</i>	Nickel
<i>p</i>	pore
<i>s</i>	Metal foam solid material
<i>sat</i>	Saturation
<i>w</i>	Heating Wall

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Declaration of interests

☒The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

☐The authors declare the following financial interests/personal relationships which may be considered as potential competing interests:

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L.L. Manetti: Methodology, Validation, Investigation, Data curation, Formal analysis, Writing - Original Draft, Writing - Review & Editing

I.L. de Oliveira: Methodology, Software, Resources, Validation, Investigation, Data curation, Formal analysis, Writing - Original Draft, Writing - Review & Editing

E.M. Cardoso: Conceptualization, Funding acquisition, Project administration, Supervision, Formal analysis, Writing - Original Draft, Writing - Review & Editing