JOURNAL OF NANO- AND ELECTRONIC PHYSICS
Vol. 14No 4, 04032(5pp) (2022)
ЖУРНАЛ НАНО- ТА ЕЛЕКТРОННОЇ ФІЗИКИ
Том 14 № 4, 04032(5cc) (2022)

Numerical Studies of Thermal Management of Multiple Electronic Devices Using Metal Foam Heat Sinks

N. Sid[1,*], S. Boulahrouz[1,2,†], A. Saoudi[1,3], O. Chahaoui[1,3], K. Mansouri[1,3]

[1] *Department of Mechanical Engineering, Abbes Laghrour University, 40000 Khenchela, Algeria*
[2] *Electromechanical Engineering Laboratory, 23000 Annaba, Algeria*
[3] *I.S.M.A. Laboratory, Abbes Laghrour University, 40000 Khenchela, Algeria*

(Received 05 May 2022; revised manuscript received 14 August 2022; published online 25 August 2022)

In this study, metal foam heat sinks (MFHS) are proposed for thermal management of electronic devices. Metal foams are excellent candidates for improving the heat transfer performance of heat sinks due to their unique characteristics such as the large surface area to volume ratio and their complex form, which favors mixing and convection. Numerical investigations of the transient thermal-hydraulic behavior and performance of the cooling process of electronic devices by MFHS are carried out. The physical model consists of a convective laminar air flow inside a channel equipped with multiple power electronic devices cooled by MFHS. MFHS consist of three plate fin heat sinks which are made of aluminum foam with a porosity of 0.95 and a permeability of 1.65×10^{-7} m^2, and the heat sink base is made of aluminum solid. Comsol software is used to solve the governing equations. Numerical results reveal that the thermal performance of MFHS is larger than that of a conventional heat sink and a clear channel under the same operating conditions, and the thermal behavior of electronic devices cooled by MFHS is stable and maintained at admissible temperatures. The validation of the numerical results shows perfect agreement with the experimental data with a maximum relative error of 3 %.

Keywords: Metal foam heat sink, Electronic devices, Cooling electronics, Numerical simulation, Comsol software.

DOI: 10.21272/jnep.14(4).04032 PACS numbers: 44.30. + v, 44.05. + e, 44.15. + a

1. INTRODUCTION

In recent decades, components and electronic devices have been faster, smaller, more reliable and efficient, and more productive than ever before. Additionally, the increase in heat generation becomes unfavorable, and this can reduce the performance of these devices and limit their durability [1-4]. Tuckerman and Pease [5] designed a microchannel heat sink that has proven to be one of the most promising cooling techniques for microelectronic devices.

A new technique of enhancing heat transfer from the surface of electronic devices is employing porous media as a heat sink. Metal foam is a porous material with a large surface area to volume ratio and low density, its complex form improves high surface contact and rises the local mixing between the metal foam and the coolant fluid, which provides good thermal performance [6, 7]. Several researchers experimentally and numerically investigated the thermal-hydraulic performance of metal foams used for cooling electronic components. Kim et al. [8] performed an experimental investigation from an aluminum foam heat sink placed on a heater in a channel. The aluminum-foam heat sink can improve the thermal performance by 28 % compared to a traditional parallel plate heat sink of the same dimensions. Mohanad A. Alfellag et al. [9] numerically investigated the Metal Foam Pin-Fin Heat Sink (MFPFHS) and compared it to the conventional Solid Pin-Fin Heat Sink (SPFHS) in turbulent regime. They found that the usage of MFPF, compared to SPF, provides a significant increase in heat transfer performance and a decrease in frictional losses. Hung et al. [10, 11] examined the thermal performance of microchannels (MCHS) with inserted porous media and showed that foam materials offer better heat-transfer performance. The numerical results of Kemerli and Kahveci [12] demonstrate that the addition of metal foams causes a small increase in the friction factor, while the heat transfer shows a significant increase until the addition of a certain number of fins. Bhattacharya et al. [13] experimentally studied the forced convective heat transfer in new finned metal foam heat sinks (MFHS). The results display that heat transfer is significantly improved when fins are placed in metal foam. Ghahremannezhad et al. [14] analyzed three-dimensional models of MCHS with different solid and porous fin thicknesses. They found that the optimized porous designation can improve the heat transfer and fluid flow performance compared with conventional heat sinks. Huang et Vafai [15, 16] numerically studied the fluid flow and heat transfer characteristics using multiple porous blocks arrangement under an external forced convective laminar flow. It is evident that the addition of porous blocks can significantly increase heat transfer. Lu et al. [17] analyzed the wavy microchannel heat sink with porous fins. They found that when compared to traditional wavy microchannel heat sinks with solid fins and the wavy microchannel heat sink with porous fins, both pressure drop and thermal resistance are reduced. Al-Athel et al. [18, 19] studied the influence of forced convection, number of fins, and fin direction on the performance and thermal resistance for the 3D porous media model which was

* noureddine.sid@univ-khenchela.dz
† boulahrouz.salim@univ-khenchela.dz

obtained by a micro-computed tomography scan. They found that the heat transfer and fluid flow characteristics are largely improved by the use of MFHS for the thermal management of electronic devices.

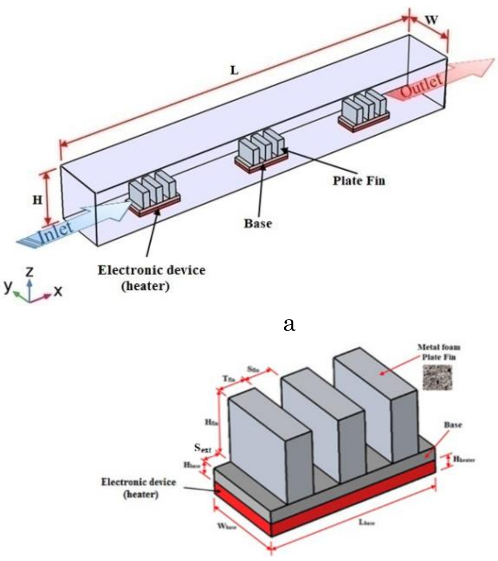
Table 1 – Geometrical dimensions of the MFHS

Parameter	Value
Heat sink's fin length ($T_{\rm fin}$)	6.35 mm
Heat sink's fin width ($W_{\rm fin}$)	20 mm
Heat sink's fin height ($H_{\rm fin}$)	15 mm
Heat sink's space fin ($S_{\rm fin}$)	6.35 mm
Heat sink's base length ($T_{\rm base}$)	39.75 mm
Heat sink's base width ($W_{\rm base}$)	20 mm
Heat sink's base height ($H_{\rm base}$)	3 mm
Extra Width on both sides of fin ($S_{\rm ext}$)	4 mm
Aluminum foam porosity (ε)	0.95
Air velocity ($U_{\rm in}$)	3.6 m/s
Heat flux (q_w)	5 W

In this study we present transient 3D numerical investigations of the thermal-hydraulic behavior of thermal management of multiple electronic devices in a horizontal channel through which air flows.

2. PHYSICAL DOMAIN AND ASSUMPTIONS

The physical domain and geometrical parameters, which are based on the experimental study of Al-Athel et al. [18, 19], are illustrated in Fig. 1. The dimensions, geometrical parameters, and morphological parameters of the metal foam which are based on the experimental study of Al-Athel et al [18, 19], are given in Table 1. The channel length (L), width (W), and height (H) are 360 mm, 50 mm, and 50 mm, respectively.



a

b

Fig. 1 – Schematic diagrams of (a) horizontal channel with electronic devices and MFHS, (b) MFHS

MFHS consists of three plate fin heat sinks made of aluminum foam and the heat sink base made of aluminum solid. Air is employed as working fluid with constant thermo-physical properties and $\rho_f = 1.177$kg/m^3, $c_p = 1006$J/kg·K, $k_f = 0.0262$W/m·K, and $\mu_f = 15.6 \times 10^{-6}$Pa·s.

A heater with a constant heat flux is placed at the bottom wall of the base to simulate an electronic device.

To simplify the problem formulation, several assumptions are established as part of the numerical modeling process:

- Thermal properties of the aluminum foam and the fluid are constant;
- The flow is incompressible, steady-state and laminar;
- Aluminum foam is isotropic and homogenous;
- The fluid phase and solid phase are in the local non-equilibrium state.

3. GOVERNING EQUATIONS AND BOUNDARY CONDITIONS

The fluid flow in MFHS is governed by the Forchheimer-Brinkman extended Darcy model to numerically solve the physical model based on the previous assumptions. The governing equations for continuity, momentum, and energy are as follows:

Equation of continuity:

$$\nabla \cdot (\rho V) = 0 . \tag{3.1}$$

Momentum conservation equation is

$$\begin{aligned}&\rho_f / \varepsilon^2 (V \cdot \nabla) V = -\nabla P + \mu_f \nabla^2 V - \\ &-\left[(\mu_f / K) + \left(\rho_f C_F / \sqrt{K}\right) V \right] V,\end{aligned} \tag{3.2}$$

where ρ_f and ρ_s are the densities of the fluid phase and solid phase, respectively; V is the air velocity; P is the pressure; ε is the aluminum foam porosity; μ_f is the fluid viscosity; K is the permeability, C_F is the inertial coefficient.

Energy conservation equations:

fluid phase:

$$(\rho c_p)_f \left(V \cdot \nabla T_f^f\right) = \nabla \cdot \left(k_{fe} \nabla T_f^f\right) + h_{sf} a_{sf} \left(T_s^s - T_f^f\right), \tag{3.3}$$

solid phase:

$$\nabla \cdot \left(K_{se} \nabla T_s^s\right) - h_{sf} a_{sf} \left(T_s^s - T_f^f\right) = 0 , \tag{3.4}$$

where T_f^f and T_s^s represent the temperatures of the fluid and solid phases, respectively; c_p is the heat capacity, k_{fe} is the effective fluid thermal conductivity; k_{se} is the effective solid thermal conductivity; a_{sf} is the thermal diffusivity solid/fluid; h_{sf} is the convective coefficient.

The parameters calculated in this study are as follows:

the friction factor

$$f = 2 (\Delta p / L) D_h / \left(\rho_f u_{in}^2\right), \tag{3.5}$$

the average heat transfer coefficient

$$h_m = q_w / \left(T_{w,m} - T_{f,m}\right), \tag{3.6}$$

the average Nusselt number

$$Nu_m = h_m D_h / k_f , \tag{3.7}$$

thermal performance ratio

$$pf = \left(Nu_{m,mf} / Nu_{m,pf}\right) / \left(f_{mf} / f_{pf}\right)^{1/3} . \qquad (3.8)$$

As mentioned in the boundary conditions section below and in order to mimic the real behavior of power electronic devices, we consider that in the first moments the air flows within the channel by free convection, the operating electronic devices heat up by a constant heat flux q_w, and their temperatures increase rapidly, and as soon as one of these temperatures reaches 60 °C (corresponding to $t = 1080$ s), the process of forced convection starts with the inlet velocity value $V = U_{in}$.

The channel walls are considered adiabatic, except the base wall, which contains electronic devices. The no-slip condition is applied to the air velocity at the surfaces in contact with these walls. At the exit of the channel, the pressure is equal to atmospheric pressure. At the base of each MFHS, the heat flux is equal to the flow dissipated by the electronic device q_w.

The boundary conditions used in the numerical simulation are described as follows:

(1) Inlet ($x = 0$): $u = u_{in}, T = T_{in}, v = w = 0$,

(2) Bottom wall of heat sinks $(y = 0) : q = q_w$,

(3) Outlet $(x = L) : \left(\partial T_f^f / \partial x\right) = \left(\partial T_s^s / \partial x\right) = 0,$

$\left(\partial u / \partial x\right) = \left(\partial v / \partial y\right) = \left(\partial w / \partial z\right) = 0$.

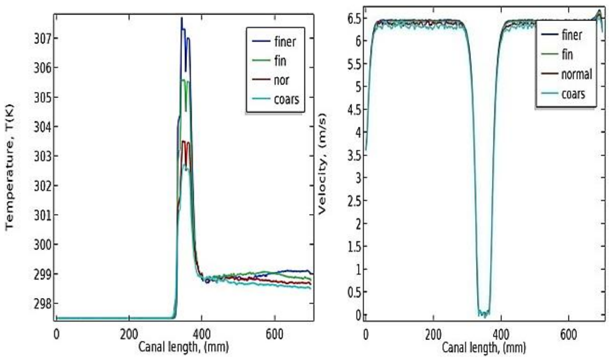
The initial conditions are:

$$t = 0, u = 0, v = 0, w = 0, T = T_{atm}, p = p_{atm} .$$

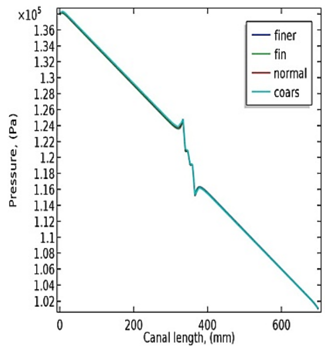
$$t \geq 1080 s, u = u_{in}, v = 0, w = 0.$$

4. GRID INDEPENDENCE AND MESH CONSISTENCY

In this study, Comsol software is used for the numerical simulation of the transient thermal-hydraulic behavior during thermal management of electronic devices by using aluminum foam heat sinks. In order to study the grid independence and mesh consistency, four meshes with triangular non-uniform elements were studied and tested: coarse (153958 elements), normal (204231elements), fin (638743 elements), and finer (1813971 elements). The use of the four meshes in the numerical simulation gives results concerning the temperature, the air velocity and the pressure with a maximum difference of 1.19 %, which allows us to choose the fine mesh in the calculation suite for a saving of memory space and time.



a b



c

Fig. 2 – Study of the grid independence and mesh consistency: (a) temperature, (b) velocity and (c) pressure

5. VALIDATION OF NUMERICAL RESULTS

Fig. 5 shows plots of validation of the average temperatures of the electronic device (red color), the base heat sink (black color), and the metal foam fin (green color) versus the experimental ones presented by Al-Athel et al. [18, 19]. As can be seen in the figure of validation, the numerical results are in perfect agreement with the experimental data with a maximum relative error of 3 %.

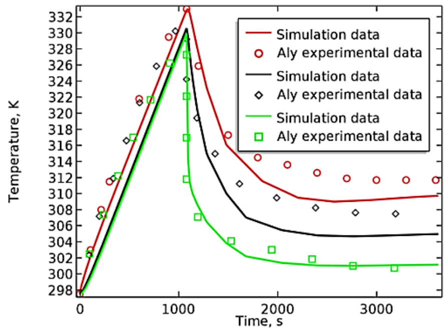


Fig. 3 – Validation of numerical simulation results by experimental data [31] for aluminum foam heat sink

5.1 Comparison of Thermal Performances

The cooling performance of the air-cooled MFHS is compared with that of conventional aluminum heat sinks and with a free channel without heat sinks under the same conditions of heat flux $q_w = 5$ W, inlet air velocity $U_{\text{in}} = 3.6$ m/s, and time $t = 2700$ s, as shown in Fig. 4. As can be seen in the figure, the temperatures of the electronic devices in the free channel without heat sinks are very high in the order of 370 K, while for the case of conventional aluminum heat sinks, the maximum temperatures are in the order of 350 K, and for the case of aluminum foam heat sinks, the maximum temperatures are in the order of 340 K. This shows the cooling performance of aluminum foam heatsinks for the thermal management of electronic devices.

5.2 Transient Thermal Behavior

Fig. 5 shows the evolution of the cooling process of electronic devices by MFHS. The cooling process of air within the channel goes through two phases; in the

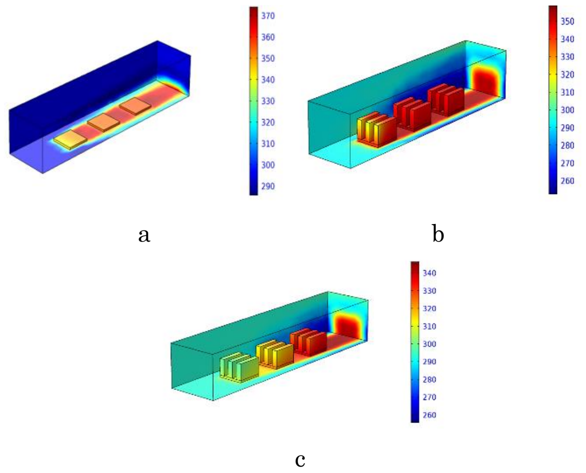


Fig. 4 – Comparison of the thermal performance of: (a) clear channel with electronic devices, (b) electronic devices with conventional heat sinks, and (c) electronic devices with MFHS

first, the convection is free, the electronic components heat up rapidly from the temperature of the ambient air to the temperature close to 330 K corresponding to the time $t = 1080$ s. From this time, the cooling process passes to the second phase, where the convection is forced, the air flows with an inlet velocity $U_0 = 3.6$ m/s, and the fall of the temperatures of the electronic devices is fast, and it can reach 303 K, which shows the good performance of aluminum foam heat sinks.

Fig. 6 shows the evolution of the average temperatures of five different points located at the level of each MFHS and at the level of the corresponding electronic device. These points are chosen as follows: one point at the base of the heat sink, one point at the first fin, one point at the second fin, one point at the third fin, and one point at the electronic device.

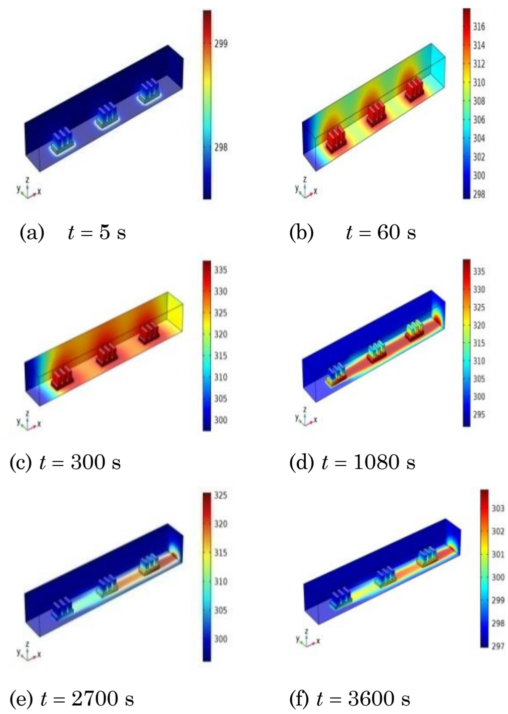


Fig. 5 – Transient thermal behavior of MFHS for $U_{in} = 3.6$ m/s and $q_w = 5$ W

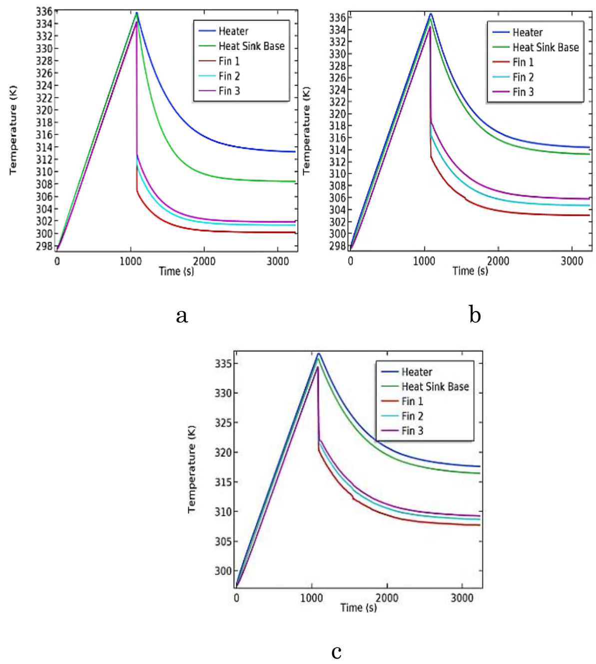


Fig. 6 – Transient temperatures of the MFHS and electronic devices: (a) heat sink 1, (b) heat sink 2 and (c) heat sink 3

As can be seen in Fig. 6, the evolution of the cooling process goes through two stages, in the first, corresponding to heating in free convection, the increase in temperatures is fast linear, while in the second phase, corresponding to the forced convection, a linear fall of temperatures is caused by air cooling. It is clear that the temperature levels increase progressively from the first to the last MFHS due to the cooling effect.

5.3 Thermal-Hydraulic Parameters

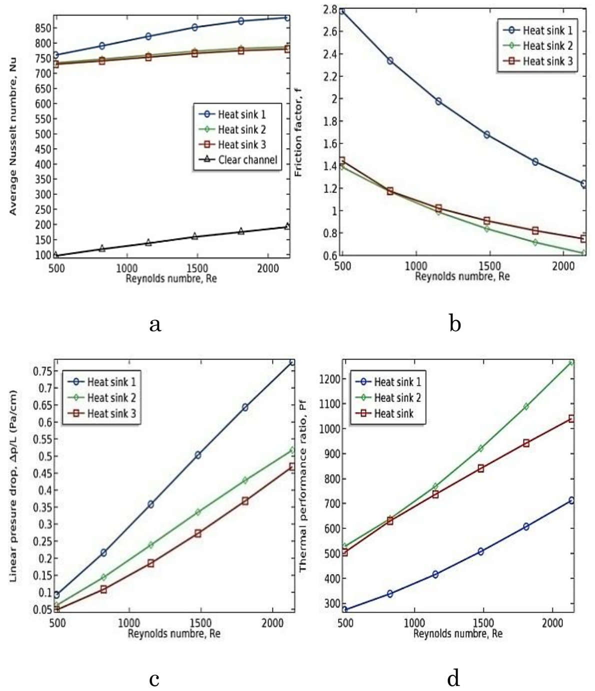


Fig. 7 – Thermal-hydraulic parameters for $q_w = 5$ W: (a) Nusselt number, (b) friction factor, (c) pressure drop, (d) thermal performance ratio

As can be seen in Fig. 7, the thermal-hydraulic parameters of the first MFHS are all high compared to the second and third heat sinks, with the difference that the second heat sink shows higher thermal performance than the others due to these high thermal parameters and its moderate pressure drop.

6. CONCLUSIONS

Transient numerical investigations of convective laminar flow of air in a horizontal channel equipped with multiple electronic devices cooled by MFHS were performed. Comsol software is used to solve the governing equations. The principal conclusions of these numerical investigations are cited as follows.

- The grid independence and mesh consistency studies showed that the use of four meshes in numerical simulations gives numerical results with a maximum difference of 1.19 %, which allows us to choose a fine mesh to save memory space and time.
- The numerical results are in perfect agreement with the experimental data of Al-Athel et al. [18, 19] with a maximum relative error of 3 %.
- A comparison of cooling performance of air-cooled MFHS with that of the conventional aluminum heat sinks and with that of the free channel without heat sinks under the same operating conditions shows that cooling performance of aluminum foam heat sinks is optimal for thermal management of electronic devices.
- The temperatures decrease significantly and linearly during the forced convection.
- The second heat sink demonstrates the best heat transfer performance and low pressure drop than the first and third heat sinks.

REFERENCES

1. M.P. Vasilev, R.S. Abiev, R. Kumar, *Int. J. Thermal Sci.* **170**, 107177 (2021).
2. S. Feng, M. Shi, H. Yan, S. Sun, F. Li, T. Lu, *J. Appl. Thermal Eng.* **132**, 30 (2018).
3. M. Paknezhad, A.M. Rashidi, T. Yousefi, Z. Saghir, *Case Stud. Thermal Eng.* **10**, 199 (2017).
4. M. Bahiraei, N. Mazaheri, M. Rasool Daneshyar, *Appl. Thermal Eng.* **183**, 116 (2021).
5. D.B. Tuckerman, R. Pease, *IEEE Electron. Dev. Lett.* **2** No 5, 126 (1981).
6. H. Huisseune, S. De Schampheleire, B. Ameel, M. De Paepe, *Int. J. Heat Mass Transf.* **89**, 1 (2015).
7. C. Zhao, *Int. J. Heat Mass Transf.* **55**, 3618 (2012).
8. S.Y. Kim, J.W. Paek, B.H. Kang, *IEEE Trans. Compon. Packag. Technol.* **26** No 1, 262 (2003).
9. M.A. Alfellag, H.E. Ahmed, M.G. Jehad, M. Hameed, *Int. J. Thermal Sci.* **170**, 107109 (2021).
10. T.C. Hung, W.M. Yan, *Numerical Heat Transf. Part A* **63** No 9, 666 (2013).
11. T.C. Hung, Y.X. Huang, W.M. Yan, *Int. Commun. Heat Mass Transf.* **48**, 86 (2013).
12. U. Kemerli, K. Kahveci, *Int. J. Thermal Sci.* **161**, 106756 (2021).
13. A. Bhattacharya, R.L. Mahajan, *J. Electron. Packag.* **124** No 3, 155 (2002).
14. A. Ghahremannezhad, K. Vafai, *Int. J. Heat Mass Transf.* **122**, 1313 (2018).
15. P.C. Huang, K. Vafai, *Int. J. Heat Mass Transf.* **36** No 16, 4019 (1993).
16. P.C. Huang, K. Vafai, *J. Thermophys. Heat Transf.* **8** No 3, 563 (1994).
17. G. Lu, J. Zhao, L. Lin, X.-D. Wang, W.-M. Yan, *Int. J. Heat Mass Transf.* **111**, 1071 (2017).
18. S.P. Aly, A.F.M. Arif, K.S. Al-Athel, J. Mostaghimi, S.M. Zubair, *Appl. Thermal Eng.* **105**, 411 (2016).
19. Khaled S. Al-Athel, Shahzada P. Aly, Abul Fazal M. Arif, Javad Mostaghimi, *Int. J. Thermal Sci.* **116**, 199 (2017).

Чисельні дослідження теплового керування кількома електронними пристроями з використанням радіаторів із металевої піни

N. Sid[1], S. Boulahrouz[1,2], A. Saoudi[1,3], O. Chahaoui[1,3], K. Mansouri[1,3]

[1] *Department of Mechanical Engineering, Abbes Laghrour University, Khenchela 40000, Algeria*
[2] *Electromechanical Engineering Laboratory, Annaba 23000, Algeria*
[3] *I.S.M.A. Laboratory, Abbes Laghrour University, Khenchela 40000, Algeria*

У дослідженні запропоновано радіатори з металевої піни (MFHS) для теплового керування електронними пристроями. Металеві піни – це матеріали для покращення ефективності теплопередачі радіаторів завдяки їх унікальним характеристикам, таким як велике співвідношення площі поверхні до об'єму та їх складна форма, яка сприяє змішуванню та конвекції. Проведено чисельні дослідження перехідної теплогідравлічної поведінки та ефективності процесу охолодження електронних пристроїв за допомогою MFHS. Фізична модель складається з конвективного ламінарного потоку повітря всередині каналу, оснащеного кількома потужними електронними пристроями, що охолоджуються MFHS. MFHS складається з трьох пластинчастих радіаторів, які виготовлені з алюмінієвої піни з пористістю 0,95 і проникністю $1,65\times10^{-7}$ м^2. Основа радіатора виготовлена з твердого алюмінію. Для розв'язання базових рівнянь використовується програмне забезпечення Comsol. Численні результати показують, що теплова продуктивність MFHS більша, ніж у звичайного радіатора за тих самих умов експлуатації, а теплова поведінка електронних пристроїв, що охолоджуються MFHS, стабільно зберігається при допустимих температурах. Перевірка чисельних результатів вказує на високу узгодженість з експериментальними даними з максимальною відносною похибкою 3 %.

Ключові слова: Радіатор з металевої піни, Електронні пристрої, Електроніка охолодження, Чисельне моделювання, Програмне забезпечення Comsol.