

Enhancement of the cooling by mixed convection of a CPU using a rotating heat sink: Numerical study

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Proc IMechE Part E:

J Process Mechanical Engineering

2024, Vol. 238(1) 144–157

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DOI: 10.1177/09544089221139110

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Abstract

The technological progress made in recent years has driven electronic apparatus to become not only more efficient and work faster but also considerably smaller in weight and size. Furthermore, the power densities of these devices have known an impressive increase. However, the challenge for the electronic industry is the lifetime device improvement by controlling the adequate removal of their excess heat. The use of more efficient cooling systems is, therefore, crucial in order to ensure durable device functionality. In this work, computational simulation was carried out to study the enhancement cooling process mixed convection of an electronic component (CPU mounted in a motherboard) using a new design of a heat exchanger by combining the heat sink (the finned surface) and the fan in a single component, this allows more heat to be moved faster and with less energy than a conventional cooler. Three radial heat sinks (HS18, HS24 and HS36) are considered depending on circumferential fin numbers ($n = 18, 24$ and 36). The effect of Reynolds number, heat flux and rotational velocity is investigated, and the optimum comprehensive performance was determined. The results reveal the cooling performance turns out to be better for heat sink with $n = 36$. The rotational velocity operates a significant effect on the temperature field only for values below than 900 rpm. We also found that the improvement in the Nusselt number and its percentage enhancement is intensified with increased rotational velocity and decreased with heat flux. A bigger Ω and Re_Ω meant a more obvious heat transfer enhancement (Nu_Ω/Nu_0) in the case of smaller Q , but (Nu_Ω/Nu_0) decreased with increasing Q .

Keywords

Cooling of a CPU, mixed convection, rotating heat sink, fins, computational fluid dynamic

Date received: 24 June 2022; accepted: 28 October 2022

Introduction

With the fast technological progress made in recent years, computers have become increasingly omnipresent in all aspects of modern life. Worldwide, the number of personal computers is over 4 billion PCs and the number of computers sold each year, is nearly 410 million or more than 13 every second.¹ The economic market demands ever higher clock speeds and simultaneously smaller in weight and size. Since computer chip heat fluxes increase with increasing clock speeds and decreasing chip sizes. The heat dissipation in central processing units (CPUs) has known an impressive increase from 30 W in the early 1990s² to 130 W in 2011 for an Intel Core i7 CPU.³

However, the great challenge of the electronic industry is to remove the heat generated by the chip efficiently. The use of consistent heat dissipation systems is, therefore, crucial in order to ensure durable device functionality. In this framework, the scientific community worldwide is active and undergoes a continuing development of optimal and new electronic cooling systems. Nowadays,

several techniques based on classical and new cooling processes, in such systems, have been adopted in scientific works such as the usage of extended surfaces or fins with different shapes offering greater surface contact with the environment.

Numerous experimental and numerical studies of the design of the heat sinks have been carried out by considering different cross-sections of pin fin; rectangular, square, circular, elliptic and perforated fins.^{4–10} The results showed that

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the elliptical fin gives the highest heat transfer coefficient and improves the thermal efficiency with a moderate increase in pressure drop. The use of perforated fins has been considered as one of the promising and useful methods in fins optimisation.

The second technique for improving cooling processes is to use the high thermal conductivity of solid metals. The use of nanofluids based on metal particles has recently been considered among the most promising solutions in the field of improving thermal transfer.

Bakhti and Si-Ameur,¹¹ Ghasemi et al.,^{12–14} Chen and Tso¹⁵ and Ambreen et al.¹⁶ presented numerical and experimental investigations to study the convection heat transfer and laminar flow of Cu, Al₂O₃, TiO₂ and CuO nanoparticles dispersed in distilled water in a microchannel heat sink with different fin geometries (square, triangular and circular). Their results indicate that the addition of nanoparticles to the water leads to an improvement in the thermal performance of the heat sink with a slight increase in pumping power.

Jet impingement is another technique employed to eliminate heat that occurred on the die surface of electronic devices. This cooling method has been widely used in many high-power industrial systems and many jet impingement studies have been aimed directly at electronics cooling. For instance, Greco et al.,¹⁷ Markal¹⁸ and Nakharinr et al.¹⁹ conducted a numerical and experimental investigation to study a single and multiple water and air jet impingement of a pin fin heat sink. They showed that the intensity of convective heat transfer enhances significantly with decreasing nozzle-to-plate distance.

To achieve an optimum design of heat sink for effective cooling of microprocessors, especially in Central Processing Unit (CPU). Wang et al.^{20,21} and Staats and Brisson²² conducted an experimental investigation to study the enhancement of the cooling using a new design of the heat sink with a heat pipe and several flat plate condensers and a centrifugal fan mounted on the heat sink. The role of the fan is to lift the air around the fins to facilitate the movement of heat outward. They found that the fan enhances the convective heat transfer between the heat sink surfaces and the air flowing through the device due to turbulent flow structures induced by the fan. The added heat pipes permit the heat from the CPU to be distributed over the heat sink, but this solution is more complex and expensive, unfortunately, not all computers have space for such a device.

The conventional design of air-cooled heat exchanger has led to other some limiting problems: for example, power hungry, the electricity used by computing centres represents 1.1–1.5% of global electricity, about half used for cooling.²³ More powerful computer chips cannot be used to their full potential as they may overheat. In addition, it is heavy in ineffectiveness, the most glaring perhaps being the fact that only about 5% of the energy produced by the fan produces a cooling effect. A layer of air stagnant tends to cling to the fins of the heat sink, thus isolating them from the airflow surrounding them and retaining heat. Drive the fan at higher velocities can

help, but it becomes noisy, which makes our devices and computers more bulky and boring. To overcome these problems, some researchers have proposed new designs for a heat exchanger by combining the heat sink (the finned surface) and the fan in a single component; this allows more heat to be moved faster and with less energy than a conventional cooler. Among these researchers, Sparrow and Preston²⁴ who conducted an experimental study of convective heat transfer in arrays of the annular fins attached to a rotating shaft. They found that the fins heat transfer coefficient decreases when the space between the fins reduces. The results showed that the closely spaced fins can be used at a high rotational velocity without a significant decrease in the transfer coefficient linked to spacing, but at the low velocity, the fins must be sufficiently separated to avoid low values of the heat transfer coefficient. It also decreases when the rotational velocity decreases.

An experimental study of the convective heat transfer from the central fin of a rotating finned tube was undertaken by Watel et al.²⁵ The Nusselt number of the fins is obtained, as a function of the rotational Reynolds number ranging from 400 to 30,000, for different fin spacing values. They found that the Nusselt number increases with increasing fin spacing and rotational Reynolds. Moreover, fin spacing has a high effect on heat transfer at low rotational velocities. Hung et al.²⁶ studied experimentally the heat transfer in a rotating rectangular channel with arrays of fins of different sizes. The results show that the big fins produced a higher heat transfer and local flow velocity in the spaced fin heat sink. The rotation improved the heat transfer at the rear and front surfaces.

Jeng et al.^{27,28} and Yang et al.²⁹ studied experimentally and numerically a turbulent flow and the heat transfer enhancement of air jet impingement onto rotating and stationary heat sink with square fins. Their results indicate that for the stationary heat sink the Nusselt number increases with the Reynolds number and the distance of the nozzle to fin top. But for rotating heat sink, they noticed that the increase of the rotational Reynolds number improved the Nusselt number for low values of the Reynolds number and it decreased with the increase of Re. An experimental analysis of heat transfer by convection in a rotating heat sink was presented by Hassan and Harmand.³⁰ Their results revealed that when the radius of rotation increases four times, the temperature of the heat sink can be reduced by about 50% at low rotation velocity and by about 20% at high rotation velocity.

Latour et al.³¹ presented an experimental investigation of the convective heat transfer around the annular fins mounted on a rotating cylinder subjected to airflow parallel to the fins. The influence of fin spacing and Reynolds numbers Re and Rec rotational was analyzed. These authors proposed a correlation giving the value of the Nusselt number as a function of the spacing between fins and the two Reynolds numbers. The convective heat transfer characteristics of fins extend around the circumference of a rotating shaft has been investigated

experimentally by Tawfik.³² He studied the effect of the length and the number of fins and the orientation angle of the fin on the Nusselt Number. The results indicated that the effect of the orientation angle decreases with increasing rotational velocity. At high rotational velocities, the Nusselt number is independent of the number of fins and depends weakly on the low Reynolds number.

Murthy³³ studied the flow and heat transfer from a cylinder with circumferential fins rotating inside a stationary shroud. The local and overall heat-transfer coefficients are presented as a function of the Grashof number and geometrical parameters such as the fin clearance and fin spacing. The greatest augmentation of the heat transfer is found for relatively short fins placed close together. The flow between two co-rotating discs within a fixed envelope has been investigated numerically by Herrero.³⁴ The author shows that the reduction in inter-fin spacing generates boundary layer interactions, which reduce air renewal at the fins, leading to a significant reduction in convective exchanges. The works of Lallave et al.³⁵ and Rahman³⁶ aim to study the confined liquid jet hitting a solid rotating disc and uniformly heated. The results show that increasing the rotational velocity improves the values of the local Nusselt number over the entire solid–fluid interface, but for a high rotational velocity, the thermal boundary layer separates from the wall and generates inefficient cooling.

According to the literature review, it can be noticed that the convective heat transfer from stationary heat sink has been quite thoroughly investigated. On the other hand, is still a great lack of numerical and experimental work concerning the convective heat transfer in rotating heat sinks used in the cooling of electronic components with high heat dissipation despite its importance. Thus, the objective of the present research is to continue our research work (Bakhti and Si-Ameur^{7,10,11}) by realising a new design of the rotating heat sink in order to increase the convective heat transfer, and therefore, increase the cooling of electronic systems. In this paper, we consider the cooling by mixed convection of a CPU mounted on a motherboard using both stationary (HS) and rotating heat sink (RHS). We also aim to analyse the influence of the heat flux, Reynolds number and rotational velocity of the heat sink on the Nusselt number. Numerical simulations were carried out for the two studied cases; stationary and rotating heat sink for a heat flow $Q = 50\text{--}200\text{ W}$, and Reynolds number $100 \leq Re \leq 400$ which corresponds to a Richardson number $2 \leq Ri \leq 40$. For the rotating heat sink, numerical tests were carried out for a rotational velocity $\Omega = 300\text{--}1000\text{ rpm}$, which corresponds to a rotational Reynolds number $Re_\Omega = 38.31\text{--}766.20$.

Mathematical formulation

Physical models

Figure 1 shows isometric views of the heat sink physical model. It consists mainly of a circular base with

circumferential fins arranged radially at regular intervals. The fins width varied in the radial direction and is defined by a pair of intersection arcs. The significant fin geometrical parameters are illustrated in Figure 1(a). Three configurations of the heat sink (HS18, HS24, HS36) are considered depending on the fins number $n = 18, 24$ and 36 , and therefore, fin to fin spacing angle $\varphi = 20^\circ, 15^\circ$ and 10° , respectively. Aluminium is the principal material of the heat sink and the circular base is subjected to a constant heat flux Q . Because of the high number of grids used to mesh the entire domain of the 3D heat sink, and in order to reduce the computation time, only a single domain of fin was explicitly considered in the computational. Furthermore, a periodic boundary condition on both sides of the computation domain is applied as it is shown in Figure 1(b).

Governing equations and boundary conditions

The governing equations for the incompressible three-dimensional airflow under a steady-state condition can be written as follows:

- *Mass conservation equation:*

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0 \quad (1)$$

- *Momentum equations:*

$$u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} + w \frac{\partial u}{\partial z} = -\frac{1}{\rho_0} \frac{\partial P}{\partial x} + \nu \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2} \right) \quad (2)$$

$$u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} + w \frac{\partial v}{\partial z} = -\frac{1}{\rho_0} \frac{\partial P}{\partial y} + \nu \left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2} \right) \quad (3)$$

$$u \frac{\partial w}{\partial x} + v \frac{\partial w}{\partial y} + w \frac{\partial w}{\partial z} = -\frac{1}{\rho_0} \frac{\partial P}{\partial z} + \nu \left(\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2} \right) + g\beta(T - T_0) \quad (4)$$

- *Energy equation for fluid:*

$$u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} + w \frac{\partial T}{\partial z} = \frac{k_f}{\rho_0 C_p} \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right) \quad (5)$$

- *Energy equation for solid of heat sink:*

$$k_s \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right) = 0 \quad (6)$$

The thermal physical properties for both coolant and heat sink solid material considered in this research are regarded as constant. However, the density varies

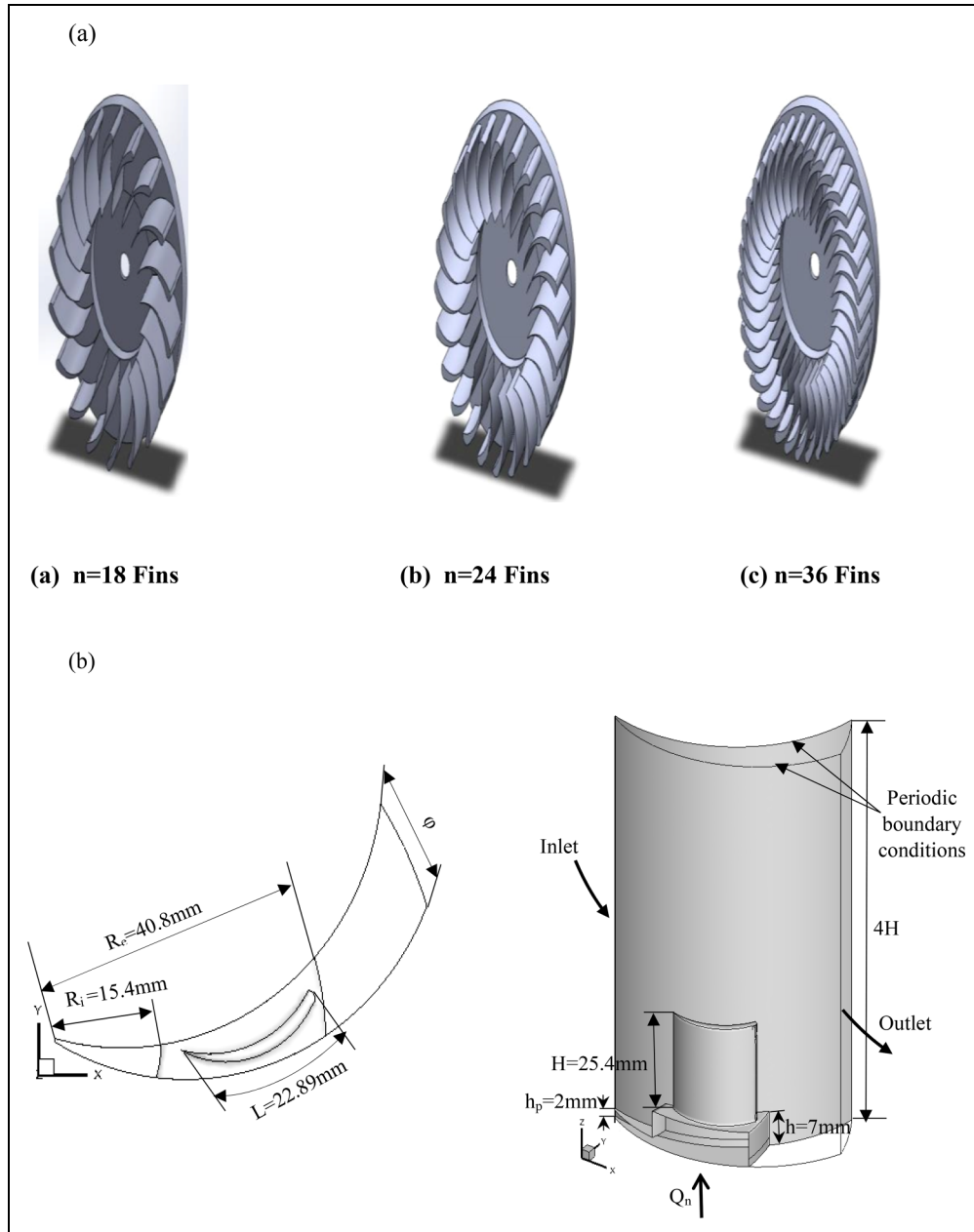


Figure 1. Views of heat sink configurations and computational domain. (a) Isometric views of heat sink configurations HS18, HS24 and HS36. (b) Schematic views of the computational domain.

according to Boussinesq's approximation. The conjugate heat transfer is explicitly taken into account in heat sink solid material.

- The following boundary conditions were used to fulfill the physical problem formulation:

✓ Inlet boundary:

$$T = T_{\text{inlet}} = 20^\circ\text{C}, u = u_0, v = 0, w = 0.$$

✓ Periodic boundary condition is imposed on the side walls of the computational domain.

✓ Outlet condition:

$$\frac{\partial u}{\partial x} = \frac{\partial v}{\partial x} = \frac{\partial w}{\partial x} = \frac{\partial T}{\partial x} = 0, P = P_{\text{atm}}$$

- Symmetry boundary condition is imposed on the upper wall of the computational domain.

$$\frac{\partial u}{\partial z} = \frac{\partial v}{\partial z} = \frac{\partial T}{\partial z} = 0, w = 0$$

✓ Heat flux applied to the bottom of the heat sink

$$Q_n = \frac{Q}{n}$$

- ✓ At the solid/fluid interface, the following quantities evaluated in both the solid and fluid regions are matched
- $$T_s = T_f$$

$$k_s \frac{\partial T}{\partial n} \bigg|_{\text{wall}} = k_f \frac{\partial T}{\partial n} \bigg|_{\text{wall}} \quad (7)$$

For stationary heat sink

$$u = v = w = 0$$

For rotational disk

$$u = -r.\Omega. \sin \theta, \quad v = r.\Omega. \cos \theta, \quad w = 0$$

where

$$r = \sqrt{x^2 + y^2}, \quad \theta = \tan^{-1} \left(\frac{y}{x} \right)$$

- The Reynolds number at the inlet of the heat sink is defined as follows:

$$Re = \frac{u_0 D_h}{\nu}$$

The hydraulic diameter D_h is defined as follows:

$$D_h = \frac{4.A}{P}$$

where A is the inlet section and P is the perimeter

- The rotational Reynolds number Re_Ω is calculated by

$$Re_\Omega = \frac{\pi \Omega L^2}{120 \nu}$$

where Ω is the rotational velocity of the heat sink and L is the length of the heat sink.

- The average Nusselt number is calculated according to the following expression:

$$\overline{Nu} = \frac{\bar{h}.L}{k_f} = \frac{q_c}{A.(\bar{T}_w - \bar{T}_m)} \cdot \frac{L}{k_f} \quad (8)$$

where $\bar{T}_w$ is the average temperature of the walls of the fins heat sink.

$\bar{T}_m$ is the average temperature of the air.

- The improvement of the heat transfer in the rotating heat sink RHS36 compared to the stationary heat sink HS36 is estimated by percentage enhancement of the Nusselt number as follows:

$$\text{Enhancement } \overline{Nu}_\Omega \% = \frac{\overline{Nu}_\Omega - \overline{Nu}_0}{\overline{Nu}_0} \cdot 100 \quad (9)$$

- The thermal resistance R_{th} is defined as follows:

$$R_{th} = \frac{(\bar{T}_b - \bar{T}_{inlet})}{q_c} \quad (10)$$

where $\bar{T}_b$ is the mean temperature of the heat sink base

- The friction factor is calculated as follows:

$$f = \frac{2D_h.\Delta p}{\rho.u^2.L} \quad (11)$$

where Δp is the pressure drop across the heat sink

$$\Delta p = p_{out} - p_{inlet}$$

Numerical procedure

In this study, the mixed convection in the target thermal system heat sink is controlled not only by the Reynolds number Re and heat density Q but also by the rotational velocity and the rotational Reynolds number Re_Ω . The numerical computations are carried out by solving the governing equations (1)–(6) and specific boundary conditions by means of the finite volume method. The SIMPLE algorithm is used for velocity–pressure coupling. In order to improve the numerical computation accuracy, a power law scheme is applied to discretize the diffusive and convective terms. The iterative procedure is considered converged on the basis of scaled residuals, for continuity, momentum and energy equations. For more precision, a solution convergence is set to 10^{-6} .

Grid independence study

For the three models of the heat sink, tetrahedral cells were generated to mesh the computation domain as shown in Figure 2, denser grids are used in regions near the solid and fins walls to capture the boundary layer effects correctly.

In order to have a reasonable compromise between the computation time and the precision of the results, the choice of an adequate mesh is necessary. This requires preliminary simulations to test the effect of the mesh on the sensitivity of the solutions. In this study, we have considered different meshes for the three configurations studied. The results obtained for a Reynolds number $Re = 200$ in terms of the average Nusselt number and the deviation of Nusselt number $\Delta \overline{Nu}$ calculated by the relation (12) are presented according to the mesh in Table 1:

$$\Delta \overline{Nu} = \left| \frac{\overline{Nu} - \overline{Nu}_{ref}}{\overline{Nu}_{ref}} \right| \cdot 100 \quad (12)$$

This table shows that a good compromise between the computation time and the precision of the results was obtained with meshes of:

- 136318 nodes for the HS36.
- 150813 nodes for the HS24.
- 189746 nodes for the HS18.

So, we definitely adopted them to produce all the results of both stationary and rotating heat sinks.

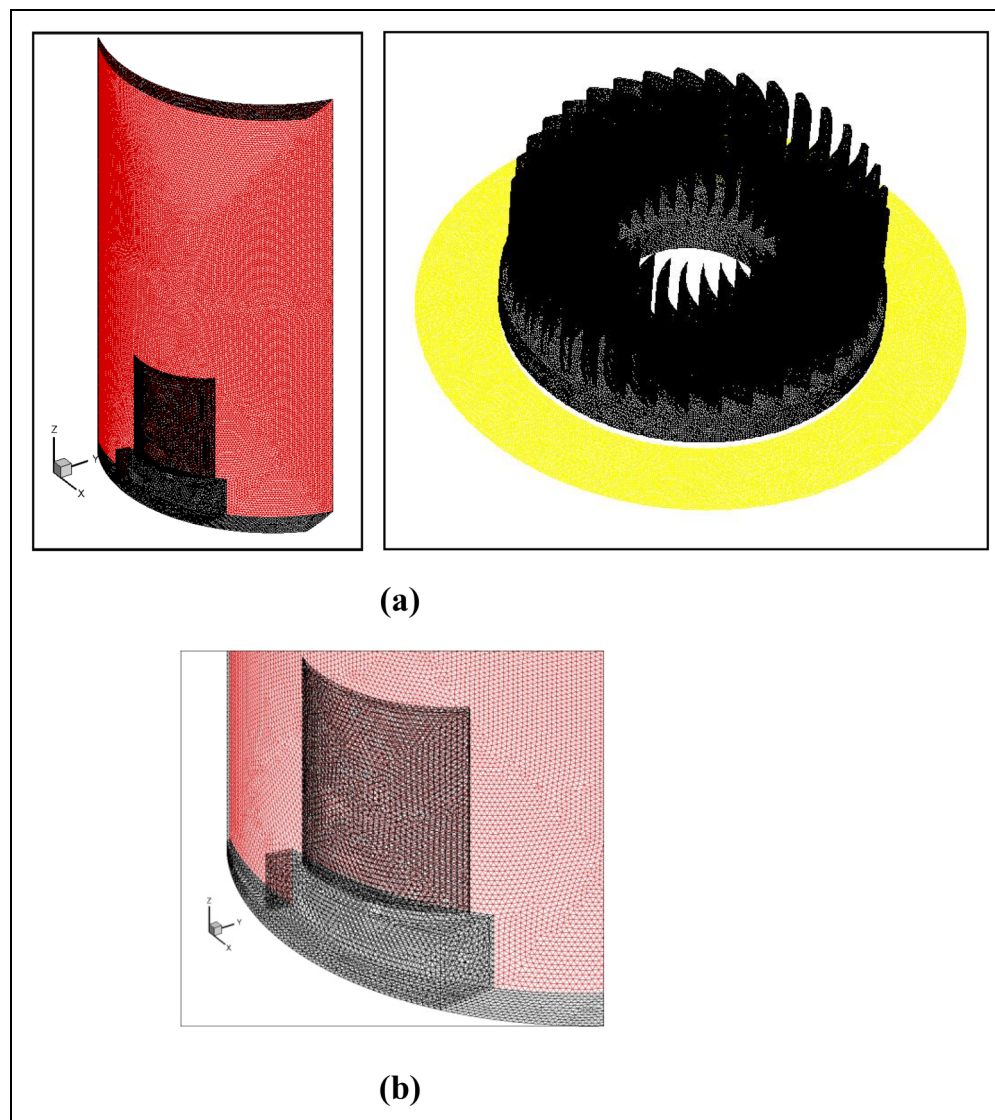


Figure 2. (a) Views of grid. (b) enlarged view of the mesh.

Validation of the numerical simulation

To validate the current simulation results, the thermal resistance is computed and compared with the experimental data of Johnson et al.,³⁷ under the same operating and design conditions. Figure 3 shows the comparison of numerical and experimental results, a good agreement is obtained with the Sandia Laboratory measurements. A maximum deviation of less than 9% is clearly observable. The approach used in the present numerical study is then feasible and the numerical results are reliable.

Results and discussion

Numerical simulations are carried out to analyse the effect of several key parameters on thermal-fluid fields in this promising type of heat sink. Realistic heat flux values are considered which ranged from 50 to 200 W. The HS18, HS24 and HS36 are examined in cases of Reynolds number values $Re = 100$ –400. However, the RHS36 study is undertaken for rotational velocity Ω

values 300–1000 rpm, which corresponds to rotational Reynolds numbers $Re_{\Omega} = 38.31$ –766.20.

Thermal characteristics

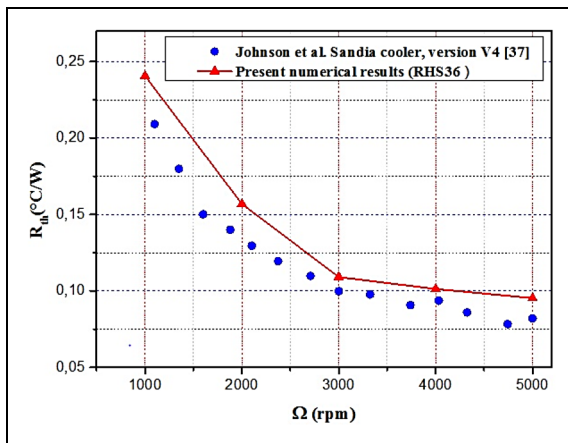
Figure 4(a) depicts air and walls temperature contours for HS18, HS24 and HS36 in the case of $Re = 250$ and $Q = 100$ W. It can be observed that the heat sink walls keep the highest temperature values, while the air temperature is relatively low, except in the region around the fin, where the air has a height value at the base plate and rose by convection upwards in the upper regions of the heat sink, because the density of the air became lower than that of the surrounding air.

Furthermore, it is noticed that the heat sink temperature rises with the decreasing of fins number or the increasing of the angle between them. Thus, the cooling performance turns out to be better for HS36 comparatively to HS24 and HS18. Evidently, with the increase in the number of fins the area exposed to thermal convection increased and

Table 1. Grid independence study ($Re = 200$, $Q = 200$ W).

HS36 ($n = 36$)			HS24 ($n = 24$)			HS18 ($n = 18$)		
N^{br} of Nods	$\overline{Nu}$	$\Delta\overline{Nu}$ (%)	N^{br} of Nods	$\overline{Nu}$	$\Delta\overline{Nu}$ (%)	N^{br} of Nods	$\overline{Nu}$	$\Delta\overline{Nu}$ (%)
36067	21.9726	1.6715	42865	12.9133	2.4428	31788	7.1530	2.8951
56893	21.8694	1.1940	67029	12.8443	1.8950	61081	7.1362	2.6527
79963	21.8396	1.0563	92355	12.7498	1.1456	81832	7.0443	1.3311
101338	21.6216	0.0475	120249	12.7028	0.7725	129808	6.9938	0.6041
136318	21.6181	0.0315	150813	12.6546	0.3900	189746	6.9568	0.0720
164017	21.6113	0	177449	12.6054	0	223306	6.9518	0

Bold values indicates different meshes tested for the three configurations and the meshes adopted to produce all the results.

**Figure 3.** Comparison of experimental and numerical results of average thermal resistance.

the interaction between the fin and the cooling fluid was better, which increases the convective heat transfer.

The evolution of the RHS's temperature with the rotational velocity in the case of $Q = 100$ W indicates a decreasing tendency with the rotational velocity increases due to the rise of the flow velocity and so the increase in the mass flow rate, which hit the fins walls, as it is shown in Figure 4(b). Indeed, the RHS initiates an air velocity augmentation and consequently an improvement of the local heat transfer coefficient, because the temperature difference between fin walls and the stream of air decreases, which leads to an enhancement of the convective heat transfer efficiency.

In addition, the temperature decreases sharply with a rotational velocity increase in the neighbourhood 300 rpm. It is interesting to note that the rotational velocity operates a significant effect on the temperature field only for values below than 900 rpm. Indicatively, rotational velocity values of 400 rpm, 700 rpm and 1000 rpm induce a temperature diminution of about 8.04°C, 11.59°C and 12.48°C, respectively. This can be explained as follows; when the rotational velocity of the heat sink is above 900 rpm, the high rotational velocity stops the flow from entering from the upper region and passing through the fins, and therefore, reduces the heat transfer by convection. If the heat transfer enhancement by the rotation of the heat sink cannot compensate for the heat transfer reduction due to the high velocity of rotation, the

temperature diminution at the high rotational velocity will be less than that at a low rotational velocity.

In order to estimate the heat sink thermal performance, the inlet and the outlet temperature difference $\Delta T = T_{out} - T_{inlet}$ is computed for the different configurations (HS18, HS24, HS36 and RHS36) considered in this work and according to relevant key parameters.

In this context, Figure 5(a) illustrates the air ΔT for stationary HS18, HS24 and HS36 in the case of several heat flux values. It is evidently observable that ΔT increases with the Q rise. Moreover, it can be seen that the convective transfer rate is directly influenced by fins number. Indeed, HS36 lowers significantly the air outlet temperature comparatively to HS18 and HS24. The importance of HS36 is noteworthy at high values of Q .

The ΔT for stationary HS36 and RHS36 (in the case of rotational velocity $\Omega = 1000$ rpm) is depicted in Figure 5(b). The RHS36 gives a lower ΔT comparatively to the stationary HS36. Obviously, the rotation favors the hot and cold air layers mixing, which leads to a decrease in the outlet temperature. Indeed, the effect of the rotational velocity on the air outlet temperature is shown in Figure 5(c). It can be observed that $\Delta T = T_{out} - T_{in}$ decreases with increasing the rotational velocity and then the forced heat convection. This tendency is more pronounced for a rotational velocity of about 300 rpm. However, the rise of the rotational velocity of more than 900 rpm is not significant on the temperature field.

The flow streamlines

Figure 6 indicates the flow streamlines for the three stationary heat sinks at $Re = 250$ and heat flux $Q = 100$ W. It is noteworthy that the direction of the fluid motion is from the inlet to the outlet of the heat sink, and the streamlined color is in terms of velocity. It is found that fluid comes out from the upper right part of the heat sink with maximum velocity because of the increase of the thermal convection in this region.

It can be seen a large vortice occurs on the upper left corner of the heat sink due to the turning of flow and a small vortice is formed at the lower of the heat sink and moves upwards by increasing the number of fins. Besides, the comparison of the three heat sinks reveals that the HS36 leads to the lowest air velocity while in the HS18 the air has the highest velocity.

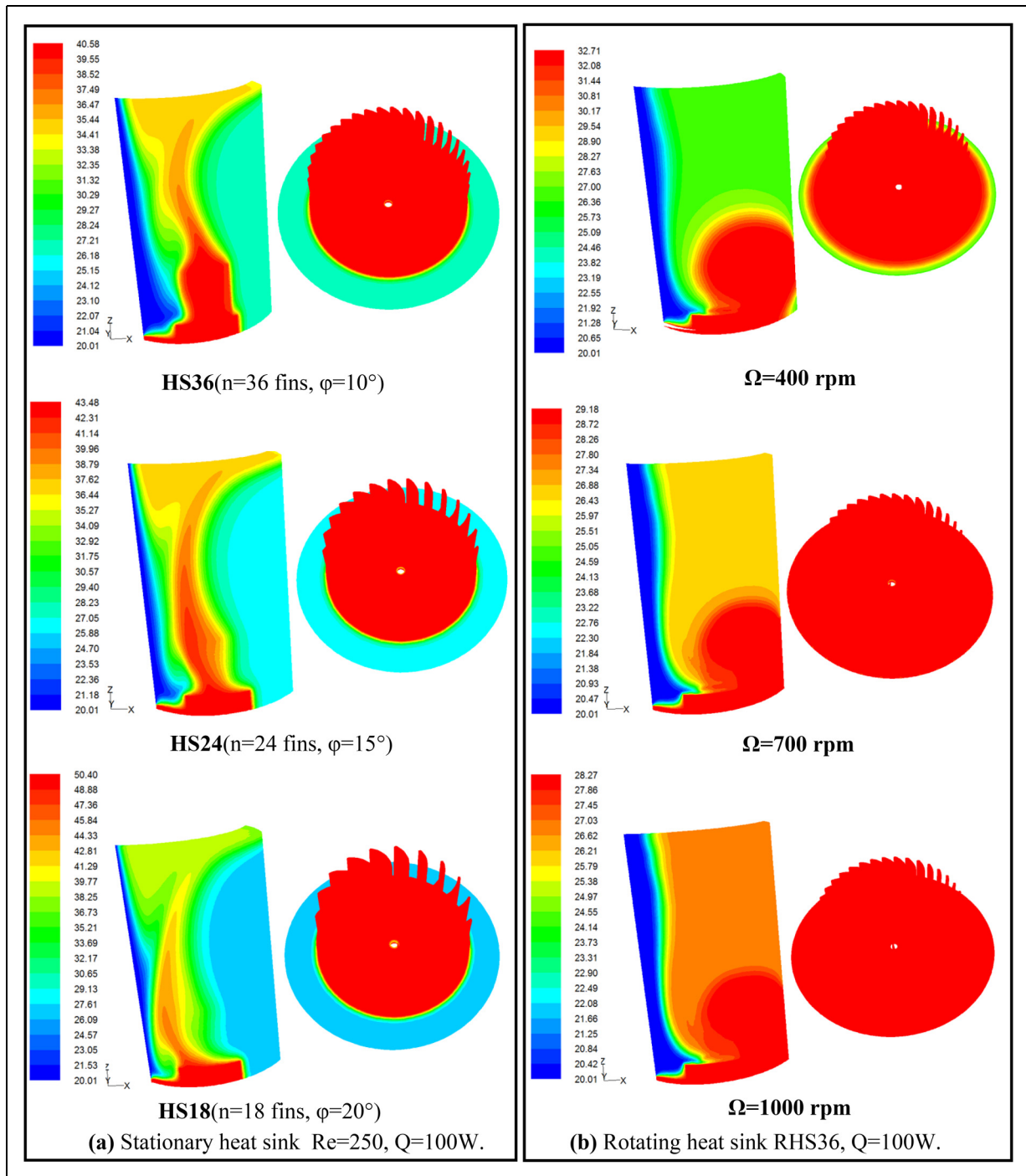


Figure 4. Contours of temperatures. (a) Stationary heat sink $Re = 250, Q = 100 W$. (b) Rotating heat sink RHS36, $Q = 100 W$.

Figure 7 shows the streamlines in the rotating heat sink according to the rotational velocity, it is noted that the fluid turns around the fins with maximum velocity because of the rotation of the heat sink. It is also found that the velocity of the fluid increases with the rotational velocity of the heat sink.

Variations of average Nusselt number

For further evaluating the intensity of heat transfer, Figure 8(a) shows the comparison of the average Nusselt number against the heat flux for various types

of heat sinks. Nusselt numbers have the same trend for the three heat sinks and decrease with the increase of Q .

We also note that $\overline{Nu}_0$ increases with the number of fins, a maximum improvement was recorded at $Q = 50 W$ for the HS36 of approximately 42.14% and 70.03%, respectively, compared to the HS24 and HS18.

The behaviour of the average Nusselt number with the Reynolds number of different heat sinks is shown in Figure 8(b). $\overline{Nu}_0$ increases with the increase in Re for all three heat sinks. The largest improvement in $\overline{Nu}_0$ is obtained at $Re = 400$ for HS36, around 27.63% and 63.18% higher than HS24 and HS18, respectively. This

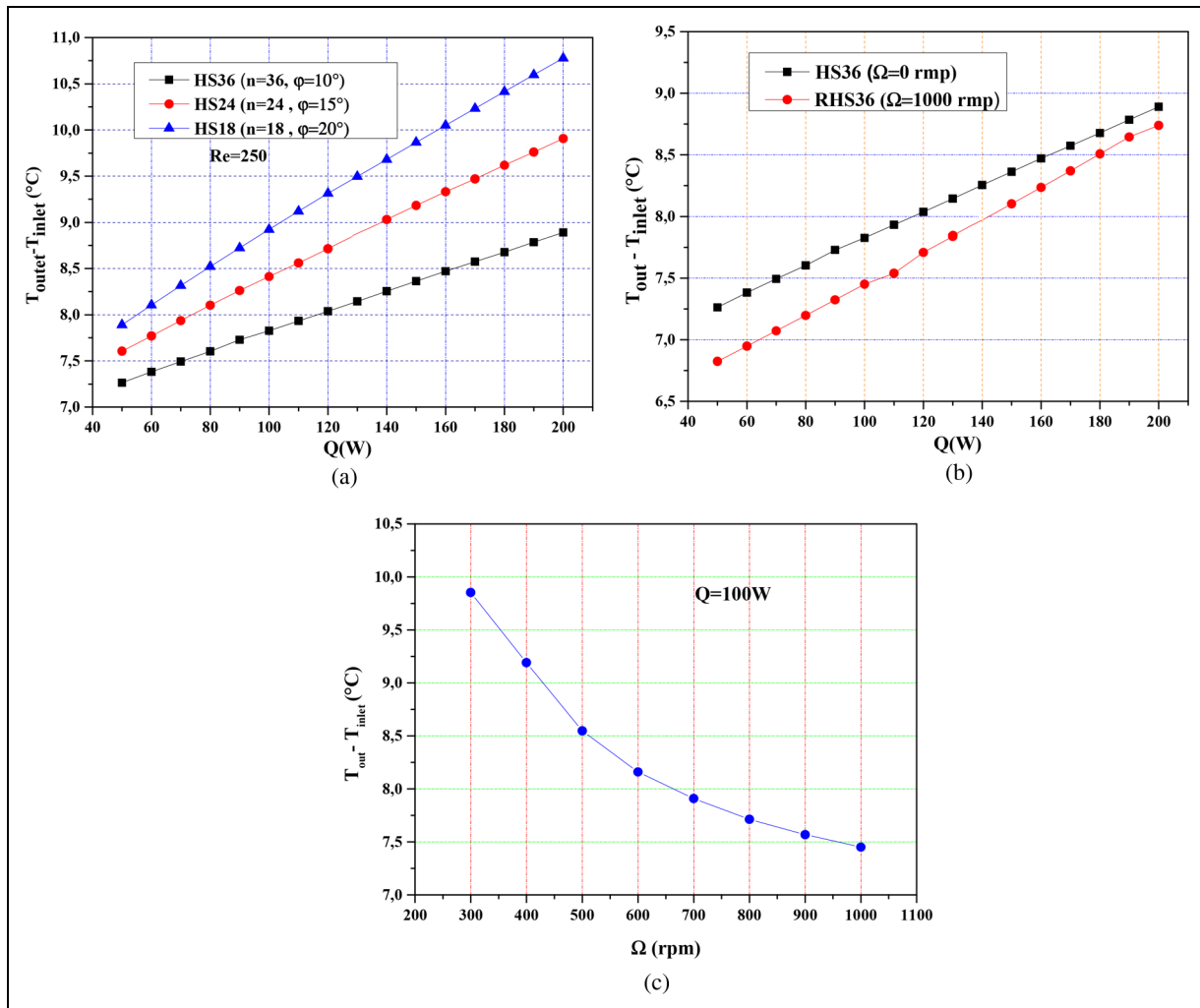


Figure 5. Variation in the difference temperature according to the heat flux and rotational velocity.

can be attributed to the increase in the exchange surface between the walls of the heat sink and the ambient air and consequently the improvement of the convective heat transfer and Nusselt number.

The effect of the rotating velocity on the Nusselt number and its percentage enhancement are depicted in Figure 8(c) and (d). It is noted that the Nusselt number increases sharply with the rotational velocity, this increase in the values of $\overline{Nu}_\Omega$ is due to the increase in the velocity of the flow, which leads to an increasing the convection heat transfer coefficient between the ambient and the heat sink surface and consequently an improvement of the convective transfer.

This figure also shows that the Nusselt number enhancement increases with the rotating velocity and decreases with the heat flux. The enhancement in the Nusselt number for $\Omega=1000$ rpm is higher than for $\Omega=700$ rpm and $\Omega=300$ rpm records the lowest values.

For example, the heat transfer enhancement values for $Q=100$ W are 50.26% for $\Omega=300$ rpm, 94.37% for $\Omega=700$ rpm and 95.71% for $\Omega=1000$ W.

Figure 8(e) shows that the Nusselt number ratio ($\frac{\overline{Nu}_\Omega}{\overline{Nu}_0}$) decreases dramatically with increasing of Q . In addition, when the rotating velocity increases a significant

enhancement is observed in the Nusselt number ratio. For $Q=50$ W and $\Omega=1000$ rpm, ($\frac{\overline{Nu}_\Omega}{\overline{Nu}_0}$) is higher around 22.45% and 91.88% than RHS with $\Omega=700$ rpm and $\Omega=300$ rpm, respectively. This thermal enhancement can be attributed to the rotation of the heat sink causing a rise of the fluid velocity and consequently more intensive mixing between the cold fluid at the upper of the heat sink and hot fluid near the heat sink walls.

Variation of friction factor

Figure 9 shows the variation of the friction factor versus heat flux for various types of heat sinks. As shown in this figure, the friction factor rise with the increase in heat flux Q .

The friction factor for the three kinds of heat sinks is almost the same trend. HS18 gives a smaller value than that HS24 and HS36 over the Q range studied. The decrease in the friction factor with the number of pins is due to a decrease in pressure drop. This is due to the larger inter-fin space which increases flow penetration into the heat sink in comparison to two other fin heat

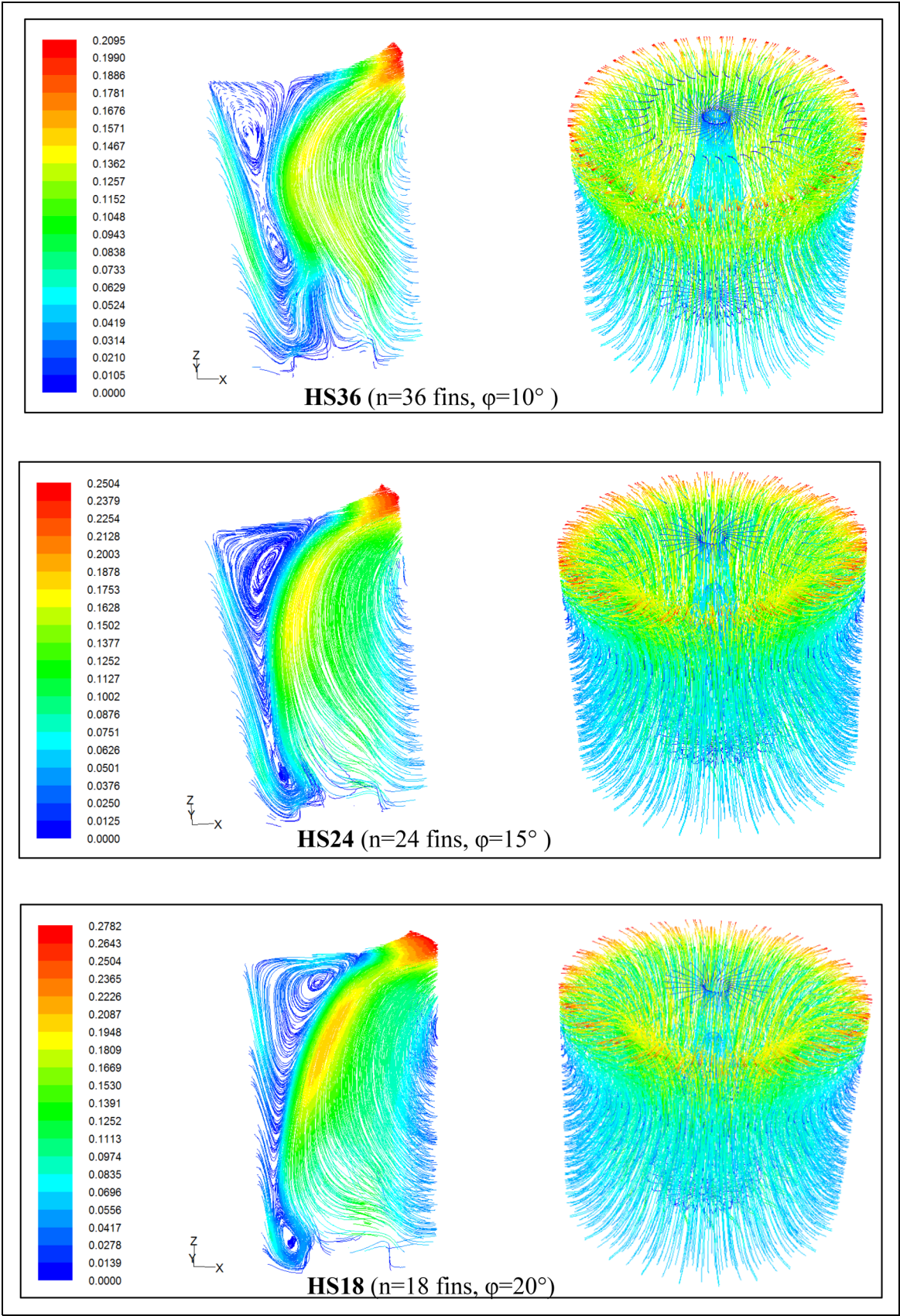


Figure 6. Streamlines for stationary heat sink, $Re = 250$, $Q = 100\text{ W}$.

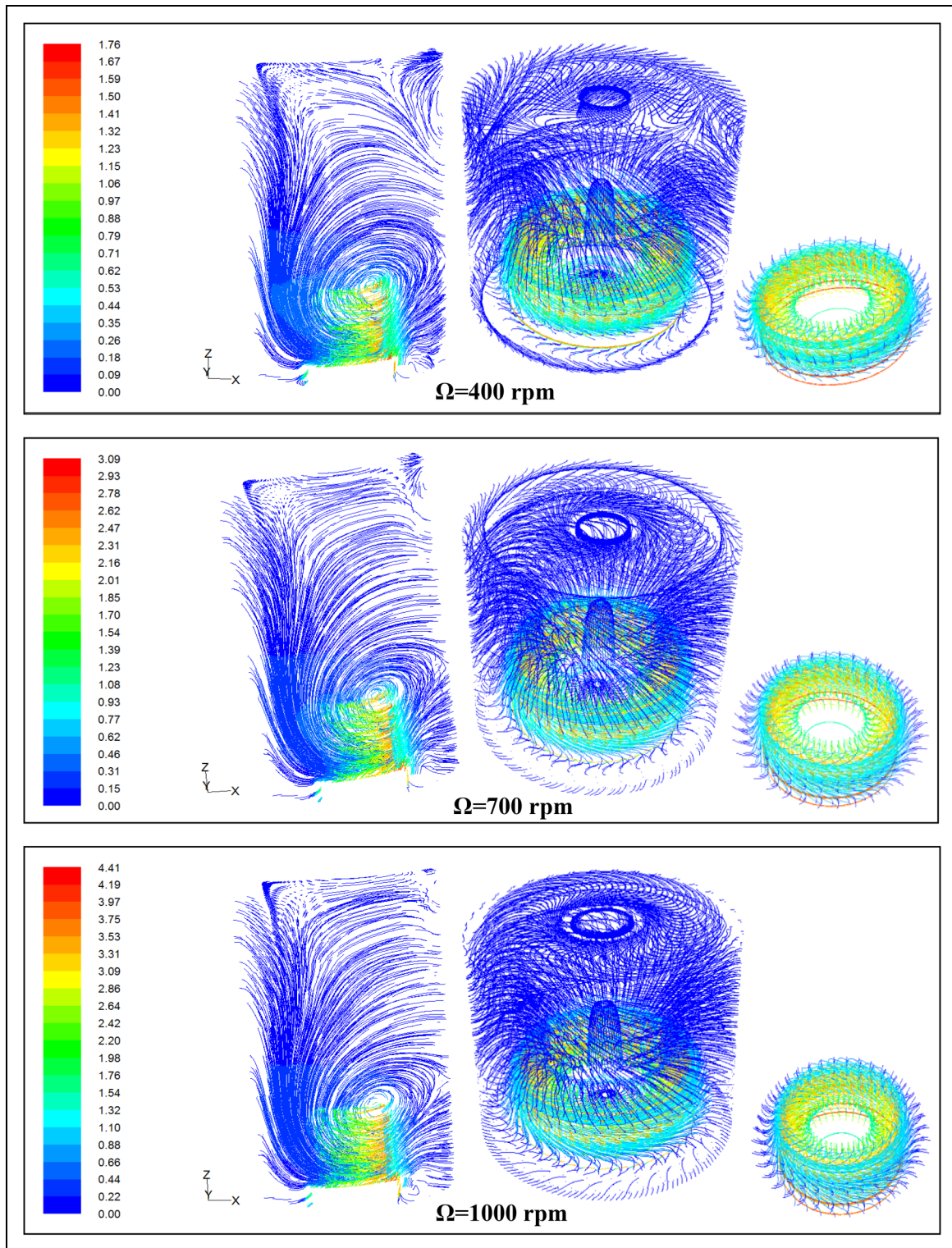


Figure 7. Streamlines for rotating heat sink RHS36, $Q = 100$ W.

sinks, and the velocity decreases, so the pressure drop decreases.

Conclusion

This study is based on a numerical investigation of the cooling process mixed convection in the case of an electronic component (CPU mounted in a motherboard).

The equations governing a steady-state mixed convection in fluid volume and heat transfer conduction in a solid zone. The numerical scheme is based on the finite volume method. A parametric study is carried out to examine the effects of several key parameters linked to geometrical and fluid flow considerations. Indeed, for stationary heat sink, an extensive analysis regarding the effects of pins number, the heat flux Q and the

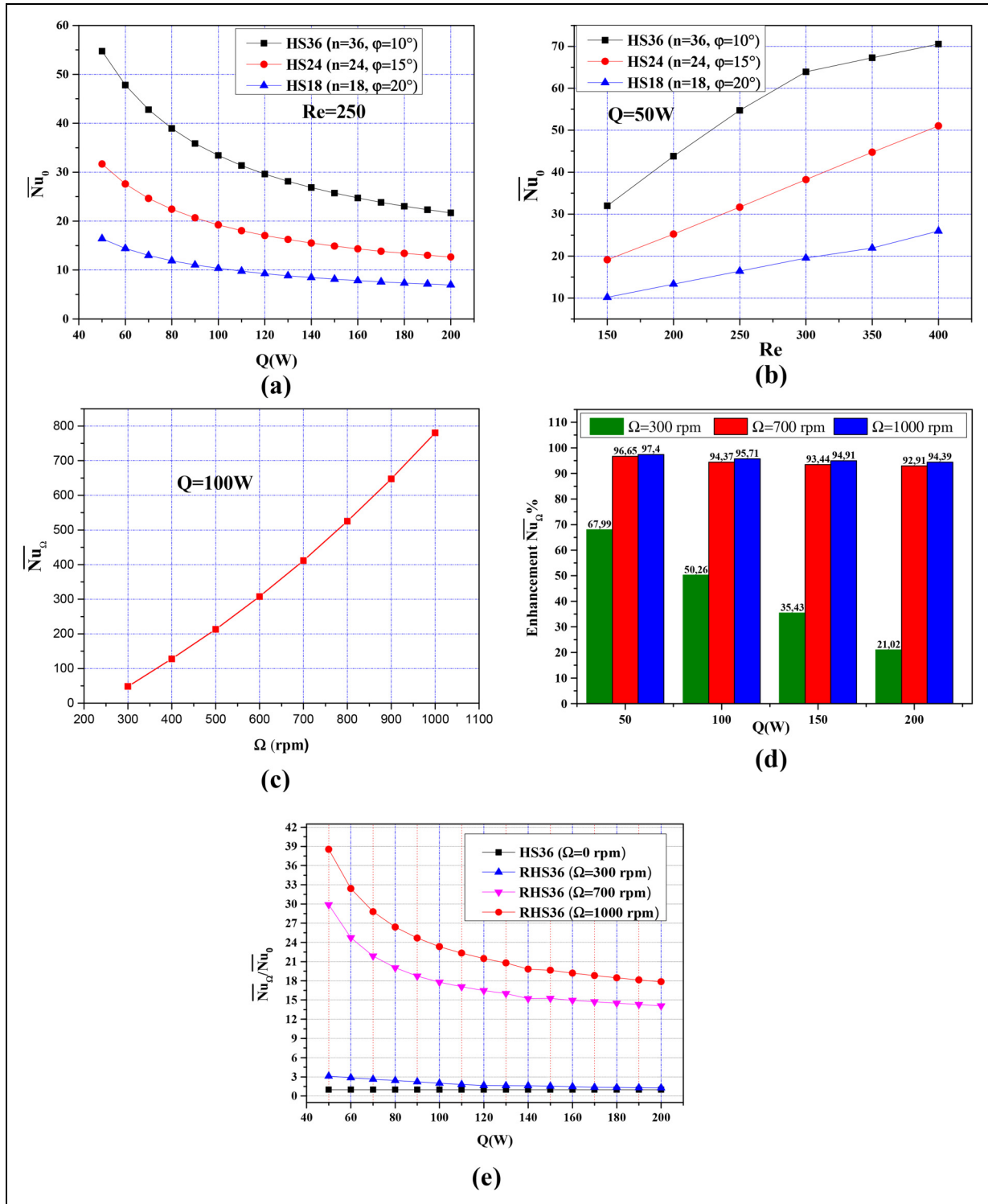


Figure 8. Variation of the average Nusselt number and its ratio number with heat flux, Reynolds number and rotational velocity.

Reynolds number Re on the average Nusselt number. Furthermore, the dynamical effect of the rotational velocity on the average Nusselt number in the case of a rotating heat sink is explicitly examined.

The main findings are summarised as follows:

- (1) It is noticed that the heat sink temperature rises with the decreasing of fins number or the increasing of

the angle between them. Thus, the cooling performance turns out to be better for HS36 comparatively to HS24 and HS18.

- (2) It is interesting to note that the rotational velocity operates a significant effect on the temperature field only for values below than 900 rpm.
- (3) It is found that a stationary heat sink $\overline{Nu}_0$ increases with Re and decreases with heat flux Q . The average

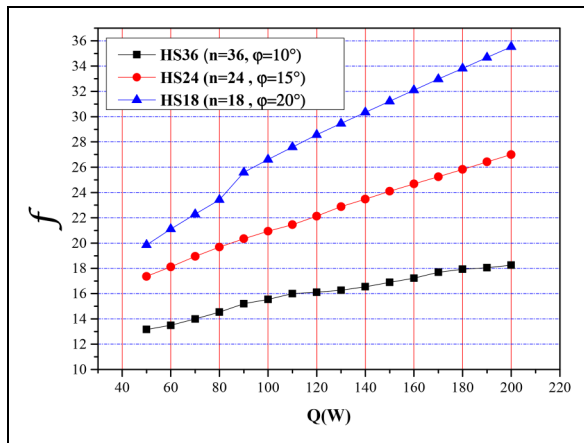


Figure 9. Variation of friction factor with the heat flux.

Nusselt number rises also with the increase of the number of pin fin and reaches maximum values for $n = 36$. An improvement of $\overline{Nu}_0$ was recorded at $Q = 50$ W for the HS36 of approximately 42.14% and 70.03%, respectively, compared to the HS24 and HS18.

- (4) The results reveal that the Nusselt number and its enhancement increase sharply with the rotational velocity and decrease with the heat flux. The percentage enhancement in the Nusselt number for $Q = 50$ W and $\Omega = 1000$ rpm ($\frac{\overline{Nu}_0}{Nu_0}$) is higher around 22.45% and 91.88% than RHS with $\Omega = 700$ rpm and $\Omega = 300$ rpm, respectively.
- (5) It is noted that the improvement in the Nusselt number ratio ($\frac{\overline{Nu}_0}{Nu_0}$) is intensified with increased rotational velocity and decrease with the heat flux.

Declaration of conflicting interests

The author(s) declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

Funding

The author(s) received no financial support for the research, authorship, and/or publication of this article.

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Nomenclature

A	heat transfer area
C_p	specific heat at constant pressure
D_h	hydraulic diameter
g	gravity acceleration
Gr	Grashof number
hp	thickness of the plate
h	thickness of the heat sink base
$\bar{h}$	Average heat transfer coefficient
H	height of pin fin
k_f	fluid thermal conductivity
k_s	solid thermal conductivity
L	length of the fin
n	number of fins
$\overline{Nu}_0$	average Nusselt number of stationary heat sink
$\overline{Nu}_\Omega$	average Nusselt number of rotating heat sink
P	pressure
q_c	convective heat flux
Q	wall heat flux
Q_n	heat flux applied to a single fin ($= Q/n$)
r	radial coordinate
R_e	outer ray
R_i	inner ray
Re	Reynolds number
Re_Ω	Rotational Reynolds number
T	temperature
$\bar{T}_b$	mean temperature of the heat sink base
$\bar{T}_m$	mean temperature of the fluid
T_o	inlet air temperature
$\bar{T}_{out}$	outlet air temperature
$\bar{T}_w$	temperature of the heat sink wall
u, v, w	velocity components
u_0	inlet air velocity
x, y, z	cartesian coordinates

Greek symbols

β	thermal expansion coefficient
θ	angular coordinate
μ	dynamic viscosity
ν	kinematic viscosity ($= \mu/\rho$)
ρ	the fluid density
ρ_0	the fluid density at T_0
φ	angle between two fins
Ω	angular velocity