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Performance Analysis of Porous Fins for Enhanced Heat Transfer

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Abstract:
The present study introduced a novel methodology for enhancing heat transfer performance from a specified surface through the application of porous fins. The thermal efficiency of porous fins was systematically evaluated and subsequently compared with that of conventional solid fins. The analysis demonstrated that the incorporation of porous fins with porosity ε (epsilon) significantly improved thermal efficiency when compared to equivalent solid fins. Furthermore, the adoption of porous fins contributed to a reduction in fin material consumption by approximately 100ε (100 epsilon) percent, indicating both thermal and material optimization.

In addition, the study investigated the influence of various design and operational parameters on the thermal efficiency of porous fins. These parameters included the Rayleigh number (Ra), Darcy number (Da), and thermal conductivity ratio. The findings revealed that enhanced thermal efficiency could be achieved at elevated Rayleigh numbers, particularly under conditions of larger Darcy numbers. Moreover, the results indicated the existence of a critical thermal conductivity ratio, beyond which no further improvement in fin efficiency was observed. These outcomes highlighted the significance of porous fin structures as an effective approach for heat transfer enhancement and material savings in thermal engineering applications.

Keywords: Porous fins, Heat transfer enhancement, Thermal efficiency, Rayleigh number, Darcy number, Thermal conductivity ratio, porous media, Convective heat transfer.

1. Introduction

Significant efforts have been made in the fin manufacturing industry to reduce both the size and production cost of fins. These efforts have primarily been motivated by the high cost of highly conductive metallic materials commonly used in fin fabrication, as well as the additional weight burden associated with fins, particularly in applications such as aircraft and motorcycles where weight optimization is critical. The reduction in fin size and manufacturing cost can be effectively achieved by enhancing the heat transfer performance of fins. Consequently, improving the thermal efficiency of fins has emerged as a major area of research and has attracted considerable attention from scholars and engineers.

Several approaches have been proposed to improve heat transfer from fins. These methods mainly include: (1) increasing the surface-area-to-volume ratio, (2) enhancing the thermal conductivity of the fin material, and (3) improving the convective heat transfer coefficient. Among these approaches, extensive investigations have been carried out on the optimization of fin geometry to maximize heat dissipation while maintaining a fixed volume of fin material. The fundamental concept behind this method involves optimally distributing one or more fin dimensions so that heat transfer performance is enhanced without increasing material consumption.

Numerous researchers have contributed to this field of fin optimization. For instance, Duffin [1] employed the variational calculus method to determine the optimum fin profile. Similarly, Jany and Bejan [2] investigated the optimum profile of fins considering temperature-dependent thermal conductivity. Bejan [2] further analyzed the optimum dimensions of plate fins under constant volume constraints and transversal laminar boundary layer conditions. Snider and Kraus [3] conducted an extensive study on fin shape optimization, while Poulikakos and Bejan [4] demonstrated that the problem of determining optimum fin shapes and dimensions could be solved through thermodynamic principles alone. Further optimization studies under the assumption of unidirectional heat transfer were carried out by various researchers [1], [2], [3], [4], [5], [6] whereas investigations under variable convective heat transfer coefficients were performed by others [7], [8].

The present study focuses on the application of porous fins as an effective technique for enhancing thermal performance. Owing to their larger effective surface area, porous fins exhibit superior heat transfer characteristics compared to conventional solid fins of equivalent weight. In recent years, porous substrates with high thermal conductivity have been increasingly utilized to improve the thermal efficiency of various thermal systems [9], [10], [11]. For example, porous substrates have been successfully implemented to enhance the performance of conventional as well as tubeless solar collectors [9], [10], and to improve the effectiveness of heat exchangers [9].

Porous fins can be fabricated using high thermal conductivity materials such as aluminum, copper, and silver, making them suitable for advanced thermal management applications. In this study, the thermal performance of porous fins was numerically investigated and compared with that of conventional solid fins. The analysis adopted the Brinkman–Forchheimer extended Darcy model to simulate fluid flow behavior within the porous structure. This model facilitated the evaluation of the effects of various operating and design parameters on the overall thermal performance of the porous fin system [12].

2. Analysis

With reference to Fig. 1, an array of porous fins was mounted on a heated surface with the objective of enhancing heat transfer from the higher temperature region to the lower temperature region. For the formulation and solution of the governing equations, several simplifying assumptions were adopted to establish a feasible analytical framework [13].

These assumptions were as follows: (1) the fins were considered to possess infinite depth, thereby neglecting the end effects of the fin array; (2) the fluid flow was assumed to remain laminar, with no internal heat generation within the fluid medium; (3) viscous dissipation effects were ignored; (4) the fluid was treated as a homogeneous, isotropic, and fully saturated single-phase medium with constant thermophysical properties; (5) the porous medium was similarly assumed to be homogeneous, isotropic, and saturated, with invariant physical properties; (6) thermal radiation exchange at the surface was neglected; and (7) the solid matrix and the fluid phase were assumed to be in local thermodynamic equilibrium [14].

The interaction between the porous medium and the adjoining clear fluid region was modeled through the Darcy–Brinkman–Forchheimer formulation, ensuring continuity of both velocity and stress across the interface as reported by previous studies [15]. By employing the dimensionless parameters defined in the nomenclature, the governing continuity, momentum, and energy equations for both the porous and fluid domains were transformed into their corresponding non-dimensional forms for subsequent analysis [16].

$$\frac{\partial U_1}{\partial X} + \frac{\partial V_1}{\partial Y} = 0 \tag{1}$$

$$U_1 \frac{\partial U_1}{\partial X} + V_1 \frac{\partial V_1}{\partial Y} = -\frac{1}{2}\frac{\partial P_1}{\partial X} + \frac{1}{Pr^2 F}\left(\frac{\partial^2 U_1}{\partial X^2} + \frac{\partial^2 U_1}{\partial Y^2}\right) - G\left(2 + \frac{1}{Pr^2}\right) Da U_1^2 - \frac{1}{2} A U_1 (U_1^2 + V_1^2) A U_1 \tag{2}$$

$$U_1 \frac{\partial V_1}{\partial X} + V_1 \frac{\partial V_1}{\partial Y} = -\frac{1}{2}\frac{\partial P_1}{\partial Y} + \frac{1}{Pr^2 F}\left(\frac{\partial^2 V_1}{\partial X^2} + \frac{\partial^2 V_1}{\partial Y^2}\right) - G\left(2 + \frac{1}{Pr^2}\right) Da V_1^2 - \frac{1}{2} A V_1 (V_1^2 + V_1^2) A U_1 \tag{3}$$

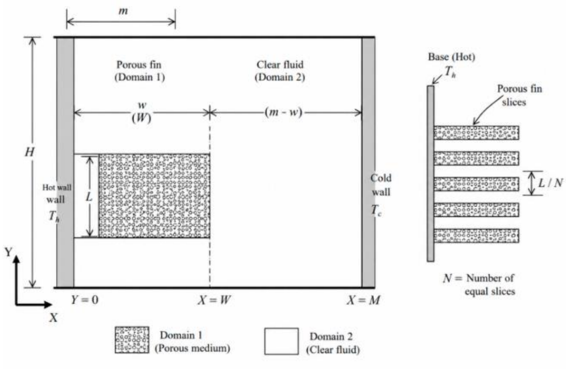

Fig. 1 Schematic Diagram for The Fins Configuration

$$\frac{\partial u_1}{\partial X} + \frac{\partial v_1}{\partial Y} = aR_F\left(\frac{\partial^2 U_1}{\partial X^2} + \frac{\partial^2 U_1}{\partial Y^2}\right) + G \tag{4}$$

$$\frac{\partial u_2}{\partial X} + \frac{\partial v_2}{\partial Y} = 0 \tag{5}$$

$$U_2 \frac{\partial U_2}{\partial X} + V_2 \frac{\partial U_2}{\partial Y} = -\frac{\partial P_2}{\partial X} + Pr^2 F\left(\frac{\partial^2 U_2}{\partial X^2} + \frac{\partial^2 U_2}{\partial Y^2}\right) + G + Ra_2 P_{r_2 u_2} \tag{6}$$

$$U_2 \frac{\partial V_2}{\partial X} + V_2 \frac{\partial V_2}{\partial Y} = -\frac{\partial P_2}{\partial Y} = Pr^2 F\left(\frac{\partial^2 V_2}{\partial X^2} + \frac{\partial^2 V_2}{\partial Y^2}\right) + G \tag{7}$$

$$U_2 \frac{\partial u_2}{\partial X} + V_2 \frac{\partial u_2}{\partial Y} = F\left(\frac{\partial^2 u_2}{\partial X^2} + \frac{\partial^2 u_2}{\partial Y^2}\right) + G \tag{8}$$

The governing equations for the conventional solid fin are therefore simplified to the following conduction equation within the solid:

$$\frac{\partial^2 u}{\partial X^2} + \frac{\partial^2 u}{\partial Y^2} = 50 \tag{9}$$

Where:

$$Da = \frac{K_1}{h_2},$$
$$A = \frac{F_1 h}{\sqrt{K_1}}$$
$$Ra_2 = \frac{gB_2 \Delta T h^3}{a^2 n^2}$$
$$Pr_2 = \frac{v_2}{a_2}$$
$$\Delta T = T_h - T_c$$

$a_R = \frac{a_1}{a_2}$

In Eqs. (1)– (9), subscripts 1 and 2 denote the porous medium and the clear-fluid region, respectively. For the problem under consideration, the momentum and energy equations are subjected to the following boundary conditions:

$$\text{At } Y = 0\text{: } W < X < M\text{:}$$
$$U_2 = 0,\ V_2 = 0,\ \theta_2 = 1,$$
$$U_1 = 0,\ V_1 = 0, \qquad \theta_1 = 1.$$
$$\text{At } Y = H \text{ and for any } X\text{:}$$
$$U_2 = 0, \quad V_2 = 0, \qquad \theta_2 = 0.$$

At $X = 0$ and for any Y, and at $X = M$ and for any Y:

$$\frac{\partial U_1}{\partial X} = \frac{\partial U_2}{\partial X} = \frac{\partial V_1}{\partial X} = \frac{\partial V_2}{\partial X} = \frac{\partial \theta_1}{\partial X} = \frac{\partial \theta_2}{\partial X} = 0.$$

At the interface $X = W$ and $0 < Y < L$:

$$U_1 = U_2, \qquad V_1 = V_2, \qquad \theta_1 = \theta_2,$$

$\frac{\partial U_1}{\partial X} = \frac{\partial U_2}{\partial X}, \frac{\partial V_1}{\partial X} = \frac{\partial V_2}{\partial X}, k_R \frac{\partial \theta_1}{a X} = \frac{\partial \theta_2}{a X}$

$$\text{At } Y = L \text{ and } 0 < X < W\text{:}$$

These conditions represent the no-slip and isothermal constraints at the horizontal boundaries, symmetry conditions at the vertical boundaries, and continuity of velocity, temperature, shear stress, and heat flux across the porous-fluid interface.

At $Y = L$ and $0 < X < W$, the continuity conditions at the porous-fluid interface are given by

$$U_1 = U_2,\ V_1 = V_2,\ \theta_1 = \theta_2,$$
$$\frac{\partial U_1}{\partial Y} = \frac{\partial U_2}{\partial Y}, \qquad \frac{\partial V_1}{\partial Y} = \frac{\partial V_2}{\partial Y}, \qquad k_R \frac{\partial \theta_1}{\partial Y} = \frac{\partial \theta_2}{\partial Y}.$$

For solid convectional fins, all boundary conditions remain identical to those described above, except for the following modifications.

$$\text{At } X = 0 \text{ and for any } Y\text{:}$$
$$\frac{\partial U_2}{\partial X} = \frac{\partial V_2}{\partial X}, \qquad \frac{\partial \theta_1}{\partial X} = \frac{\partial \theta_2}{\partial X} = 0.$$
$$\text{At } X = W \text{ and } 0 < Y < L\text{:}$$

$$U_2 = 0, \qquad V_2 = 0,\ k_R \frac{\partial \theta_1}{\partial X} = \frac{\partial \theta_2}{\partial X}.$$

$$\text{At } Y = L \text{ and } 0 < X < W:$$

$$\theta_1 = 1.$$

The dimensionless parameters are defined as

$$k_R = \frac{k_1}{k_2}, \qquad W = \frac{w}{h}, \qquad H = \frac{h}{h} = 1, \qquad L = \frac{l}{h}, \qquad M = \frac{m}{h}.$$

Here, k_R denotes the thermal conductivity ratio of the porous fin material to the surrounding fluid, while $W, H, L,$ and M represent the dimensionless geometric parameters of the computational domain.

3. Solution Methodology

The governing equations along with the specified boundary conditions were solved numerically using computational analysis software based on the finite element method. The numerical model was employed to determine the velocity, pressure, and temperature distributions within both the porous fin region and the surrounding fluid domain. A mixed pressure-velocity formulation was adopted to achieve accurate coupling between the flow and thermal fields [17].

To ensure the reliability of the numerical results, a mesh independence test was carried out before the final simulations. The mesh refinement process was based on two important criteria: (i) negligible variation in the solution fields and (ii) insignificant change in the heat flux values. The acceptable variation limit was maintained below 1 percent to guarantee solution stability and accuracy [18].

The computational domain was discretized using graded mesh elements, where finer elements were concentrated near the fin boundaries, porous-fluid interfaces, and symmetry surfaces. This mesh grading was necessary to capture the sharp temperature and velocity gradients developed in these regions. The boundary elements were modeled using two-node linear elements, whereas the interior domain consisted of four-node quadrilateral elements for improved numerical precision [19].

Several mesh sizes ranging from 1000 to 4500 elements were examined during the simulation process. It was observed that increasing the mesh density beyond 4000 elements produced only negligible changes in the heat transfer calculations. Therefore, the mesh consisting of 4500 elements was selected for all final simulations, as it provided mesh-independent and stable numerical solutions [20].

To validate the developed numerical model, the simulation results were compared with benchmark solutions available for natural convection heat transfer in a square cavity. This validation case closely resembles the present thermal model without the porous fin arrangement. The comparison showed strong agreement between the numerical predictions and the benchmark results, with deviations remaining within 2 percent. This confirmed the accuracy and reliability of the adopted numerical procedure for investigating the thermal performance of porous fins [21].

4. Thermal Performance of the Porous Fins

The enhancement in fin thermal performance is evaluated using the heat transfer ratio q_p/q_s, where q_p represents the heat transfer rate from the heated surface equipped with porous fins and q_s denotes the corresponding heat transfer rate obtained using conventional solid fins. Thus,

$$\frac{q_p}{q_s}$$

is adopted as a measure of the effectiveness of porous fins relative to solid fins.

From the dimensionless governing equations, it is evident that the thermal performance of the porous fin is influenced by the following dimensionless parameters:

$$\varepsilon,\ Pr_2, Da, A, Ra_2, \alpha_R, k_R, W, M, L.$$

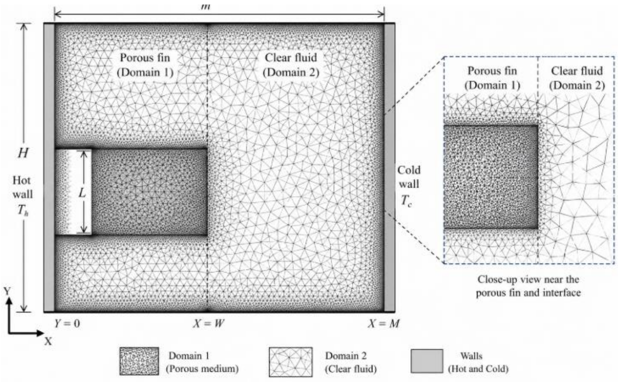

Fig. 2 Finite Element Mesh for The Flow Field

To investigate the effects of individual parameters, the remaining parameters are held constant at their reference values. Unless otherwise specified, the following values are employed throughout the analysis: Da=63×10^{-6}, $L = 0.125$, $Ra_2 = 3 \times 10^7$, $W = 0.0125$, $M = 0.25$, $k_R = 4400$, $Pr_2 = 0.71$, $\alpha_R = 2.5$, $A = 1.0$, $\varepsilon = 0.5$.

These baseline values are used as the reference conditions for evaluating the thermal behavior and heat transfer enhancement characteristics of porous fins.

5. Results And Discussion

The basic philosophy behind using porous fins is to increase the effective surface area through which heat is convected to the ambient fluid. As illustrated in Fig. 3, porous fins may be simulated by imagining that a single solid fin of fixed weight is divided into N slices of equal size, while maintaining the same total weight as the original fin. Such a configuration significantly increases the available surface area for heat transfer without increasing the amount of material used [10].

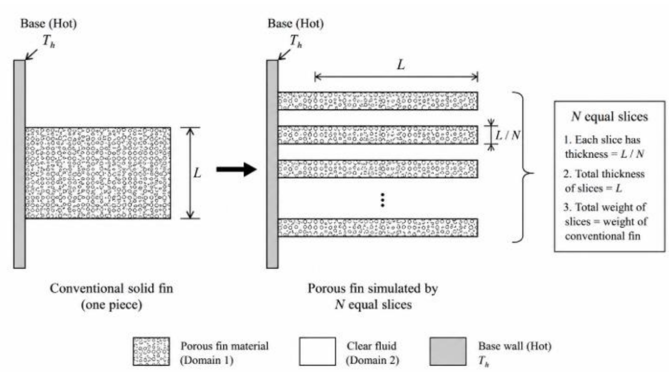

Fig. 3 Schematic Representation of Porous Fin Simulation Using N Equal Slices

Figure 4 presents the ratio of the heat transfer rate achieved by the N slices to that achieved by the original solid fin as a function of the number of slices, N. The total heat transfer from the sliced configuration is

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estimated by multiplying the heat transfer from a single slice by N. It is evident that dividing the fin into multiple slices substantially increases the total heat transfer rate. The same figure also illustrates the amount of material that can be saved while maintaining the same heat dissipation capability as the original solid fin. For example, when the original fin is divided into five slices, only approximately 30% of these slices are required to dissipate the same amount of heat as the original fin. Consequently, nearly 70% of the fin material can be saved [22].

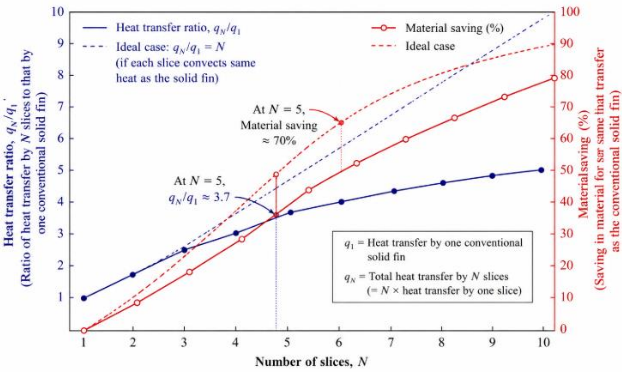

Fig. 4 Variation of heat transfer ratio and material saving with number of slices, N

The preceding simulation tends to overestimate the benefits obtained from porous fins because it assumes that the convective heat transfer coefficient within the porous medium is equal to that at the external surface of the solid fin. In practice, however, the convective heat transfer coefficient inside the porous structure is considerably lower due to flow resistance and reduced fluid mobility within the pores.
Figure 5 illustrates representative streamline and isotherm distributions within the computational domain. As shown in Fig. 5(a), the working fluid penetrates the porous fin, indicating active fluid motion through the porous matrix. The corresponding isotherm distribution demonstrates the influence of fluid penetration on the temperature field and heat transfer characteristics.

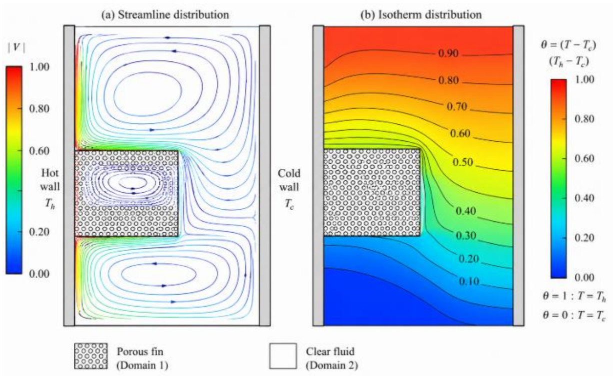

Fig. 5 Streamline and Isotherm Distributions in The Computational Domain

Figure 6 shows the effect of the thermal conductivity ratio (k_1/k_2) on the heat transfer ratio (q_p/q_s) for different fin lengths. As expected, increasing the thermal conductivity ratio increases the heat transfer ratio because the effective thermal conductivity of the porous fin becomes larger. Improved thermal conductivity facilitates heat conduction through the fin and raises the fin temperature, thereby enhancing heat dissipation to the surrounding fluid [23].

However, an optimum value of the thermal conductivity ratio exists beyond which no significant improvement in (q_p/q_s) is observed. Increasing the effective thermal conductivity raises the fin temperature until all regions of the fin approach the base temperature. At this condition, the fin attains its maximum heat transfer capability and additional increases in thermal conductivity no longer improve performance [23].

Furthermore, Fig. 6 demonstrates that porous fins having porosity ε improve the thermal performance of conventional fins by an amount that depends on fin length and operating conditions. Nevertheless, the principal advantage of porous fins is not merely the enhancement of heat transfer performance but also the significant reduction in fin weight. The percentage reduction in fin weight is approximately equal to [23]

$$\text{Weight Saving } (\%) = 100\varepsilon.$$

For example, if a porous fin with a porosity of 0.10.10.1 enhances the performance of a conventional fin by 10%, then the equivalent performance enhancement obtained by utilizing the same amount of material in the form of porous fins can be estimated as

$$\text{Performance Enhancement} = 100\left[\frac{(q_p/q_s)}{(1-\varepsilon)} - 1\right].$$

Substituting

$$\frac{q_s}{q_p} = 1.10, \varepsilon = 0.10,$$

Gives

$$100\left[\frac{1.10}{0.90} - 1\right] = 22\%.$$

Thus, using the same amount of material in the form of porous fins can increase the thermal performance by approximately 22%.

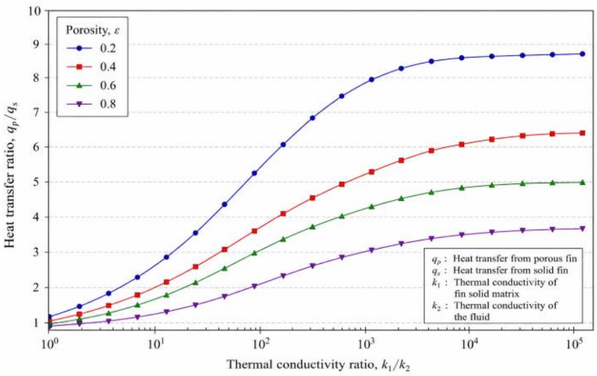

Fig. 6 Effect of thermal conductivity ratio (k_1/k_2) on heat transfer ratio (q_p/q_s)

The enhanced performance of porous fins is primarily attributed to the increase in effective surface area available for convection. Although the removal of solid material reduces the effective thermal conductivity of the fin, the resulting increase in heat transfer area compensates for this reduction and leads to a net improvement in thermal performance [24].

Figure 6 further reveals that increasing fin length reduces the heat transfer ratio. As the fin length increases, the temperature near the fin tip approaches the ambient fluid temperature. Consequently, the buoyancy force responsible for natural convection decreases, limiting the penetration of fluid into the porous structure. As a result, both porous and conventional fins derive less benefit from additional increases in fin length [24].

The reduction in effective thermal conductivity caused by the porous structure further limits the benefit obtained from increasing the length of porous fins compared with conventional solid fins. Another important observation from Fig. 6 is that the optimum thermal conductivity ratio [24],

$$\left(\frac{k_1}{k_2}\right)_{opt},$$

increases with increasing fin length. This behavior occurs because longer porous fins require greater thermal conductivity to compensate for the increased thermal resistance and maintain effective heat transfer [24].

A relationship between fin porosity and the thermal conductivity ratio can be derived for each fin length. The optimum thermal conductivity ratio corresponding to maximum fin performance is given by

$$\left(\frac{k_1}{k_2}\right)_{opt} = \varepsilon + (1-\varepsilon)\frac{k_s}{k_2}$$

or approximately

$$\left(\frac{k_1}{k_2}\right)_{opt} \approx (1-\varepsilon)\frac{k_s}{k_2}.$$

Hence,

Figure 7 presents the variation of porosity ε as a function of $\frac{k_s}{k_2}$. It is evident that larger values of $\frac{k_s}{k_2}$ permit greater savings in fin material while maintaining the same thermal performance. Nevertheless, the material saving approaches an asymptotic value beyond which further increases in $\frac{k_s}{k_2}$ yield negligible additional benefits.

As k_s increases, the fin temperature approaches the base temperature. In the limiting case,

$$\frac{k_s}{k_2} \to \infty,$$

the fin becomes nearly isothermal, and the heat transfer rate reaches its maximum possible value. Beyond this point, additional increases in thermal conductivity do not produce noticeable improvements in performance.

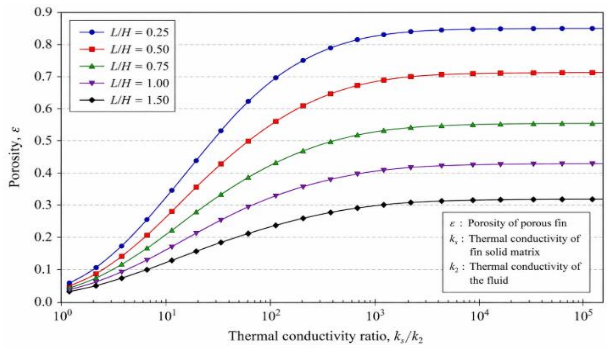

Fig. 7 Variation of porosity ε with thermal conductivity ratio $\frac{k_s}{k_2}$

Figure 8 illustrates the influence of the Rayleigh number (R_a) on the heat transfer ratio for different Darcy numbers (D_a). Increasing the Rayleigh number enhances the heat transfer ratio (q_p/q_s) because stronger buoyancy forces improve the convective heat transfer coefficient between the fin and the surrounding fluid [25].

In porous fins, convection occurs over a volumetric surface area that is substantially larger than the external surface area of conventional fins. Therefore, any increase in the convective heat transfer coefficient produces a greater enhancement in the performance of porous fins than in conventional fins [25].

The figure further indicates that the influence of the Rayleigh number becomes more significant at larger Darcy numbers. Increasing the Darcy number increases the permeability of the porous medium, thereby facilitating fluid penetration and heat transport within the porous structure. Conversely, porous fins with very small Darcy numbers behave similarly to conventional solid fins because of their low permeability. Under such conditions [25],

$$\frac{q_s}{q_p} \approx 1,$$

and the influence of the Rayleigh number becomes negligible.

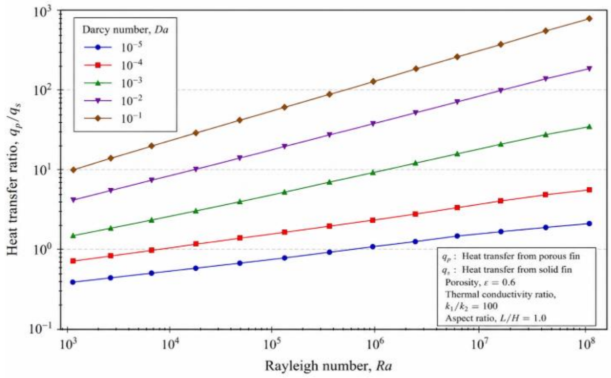

Fig. 8 Effect of Rayleigh number on heat transfer ratio for different Darcy numbers

Figure 9 shows the variation of the Nusselt number with the thermal conductivity ratio. The Nusselt number is defined as

$$\mathrm{Nu} = \frac{2q_p}{m' \Delta T k_2},$$

where m' is the characteristic enclosure dimension. The Nusselt number represents the dimensionless convective heat transfer coefficient between the heated surface, including the porous fins, and the cold surface.

As expected, the Nusselt number increases with increasing thermal conductivity ratio due to improved heat conduction through the porous fin. However, an optimum thermal conductivity ratio exists beyond which no significant increase in the Nusselt number is observed [26].

In addition, increasing fin length requires a larger optimum conductivity ratio,

$$\left(\frac{k_1}{k_2}\right)_{opt},$$

to achieve maximum thermal performance. Finally, Fig. 9 shows that the Nusselt number increases with fin length because a longer fin provides a larger heat transfer area, thereby increasing the total heat transfer rate and enhancing the overall convective heat transfer performance [26].

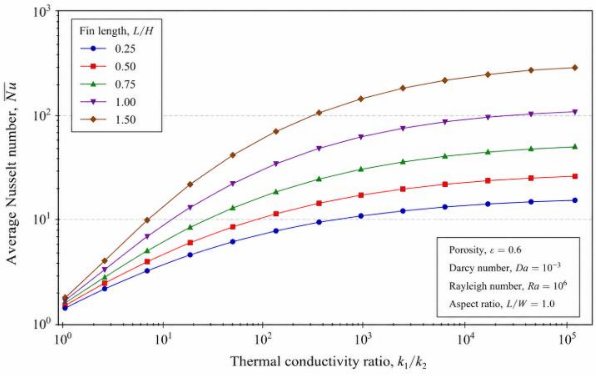

Fig. 9: Variation of Nusselt number with thermal conductivity ratio for different fin lengths

6. Conclusion

As is shown in the current research, the effectiveness of the novel porous-fin approach is proved to result in an efficient enhancement of the heat transfer rate due to an enlarged effective surface area for convection. Thermal behavior of porous fins was studied comprehensively and compared with conventional solid fins for different operational and design parameters.

Thus, the application of porous fins with porosity ε enables improving thermal efficiency while at the same time decreasing the amount of material used for the manufacture of fins. Hence, the considerable weight economy is obtained without losing heat transfer capabilities. It is evident that the advantages of porous fins are more pronounced for enhanced natural convection conditions.

The heat transfer performance of porous fins increases when the value of the Rayleigh number (Ra) is higher. The higher Ra values improve buoyancy-driven convection and the value of the convective heat transfer coefficient. As heat transfer in porous fins is realized over a considerably larger effective surface area compared to conventional solid fins, then improvements in the convective heat transfer coefficient have a more powerful effect on the performance of porous fins. Moreover, the effect of the Rayleigh number becomes more noticeable with the growth of Darcy number (Da).

Moreover, the research findings indicate that the Nusselt number (Nu) increases with an increase in the thermal conductivity ratio (k_1/k_2). The reason behind this is that the effective thermal conductivity increases as a result of the increase in the thermal conductivity ratio. However, there is an optimum thermal conductivity ratio above which any further increment has no impact on the heat transfer performance. This happens because, at this point, the fin attains the state of isothermality and the maximum possible heat transfer rate is attained.

Furthermore, it was observed that an increase in fin length leads to an increase in the Nusselt number due to the availability of the greater convection area. However, this increase becomes insignificant when the fins become too long since, at this stage, the temperature at the fin tip becomes close to the ambient fluid temperature.

Hence, it can be stated that the findings of this study support the concept that the use of porous fins represents a promising option as compared to the solid fins as they offer improved thermal performance, material conservation and heat transfer characteristics, especially at higher values of Rayleigh and Darcy numbers.

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