

A CFD study on turbulent forced convection flow of Al_2O_3 -water nanofluid in semi-circular corrugated channel

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Abstract. The performance of heat exchangers especially for single phase flows can be enhanced by many augmentation techniques. One of the most popular method used is a passive heat transfer technique. Researchers have been quite active in the search of novel ways on heat transfer augmentation techniques using various types of passive techniques to increase heat transfer performances of heat exchanger. Computational Fluid Dynamics (CFD) simulations of heat transfer and friction factor analysis in a turbulent flow regime in semi-circle corrugated channels with Al_2O_3 -water nanofluid is presented in this paper. Simulations are carried out at Reynolds number range of 10000-30000, with nanoparticle volume fractions 0-6% and constant heat flux condition. The results for corrugated channels are examined and compared to those for straight channels. Results show that the Nusselt number increased with the increase of nanoparticle volume fraction and Reynolds number. The Nusselt number was found to increase as the nanoparticle diameter decreased. Maximum Nusselt number enhancement ratio 2.07 at Reynolds number 30,000 and volume fraction 6%.

1. Introduction

The enhancement of convection heat transfer is a very interesting topic for different kinds of industrial and engineering applications and can be improved passively by changing the flow geometry and boundary conditions or by enhancing the thermophysical properties of the fluid. The ability of a fluid medium to transfer a large amount of heat across a small temperature gradient enhances the efficiency of energy conversion, as well as improves the design and performance of heat exchangers. Therefore, research on enhancement technique in such channels has become very prominent. For this purpose, using nanofluids as a cooling fluids in corrugated channels instead of traditional fluids can enhance thermal conductivity of the base fluids and thereby a further improvement in thermal performance of heat exchangers with a more compact design.

Various numerical and experimental studies on the forced convection flow of conventional fluids or nanofluids in channels exist in the literature [1-12]. Sunden and Skoldheden [1] experimentally studied the heat transfer and pressure drop in corrugated channels and smooth tubes. Sawyers [2] numerically and experimentally studied the effect of three dimensional hydrodynamics on the enhancement of heat transfer in corrugated channels for Reynolds numbers in the range of 9–149. Fabbri [3] studied laminar convective heat transfer in a channel composed of smooth and corrugated walls. Vasudevaiah and Balamurugan [4] studied theoretically the convective heat transfer in a corrugated microchannel. The transport equations of continuum theory were adopted under



incompressible flow conditions. Wang and Chen [5] applied a simple coordinate transformation method and the spline alternating direction implicit method for determining the heat transfer rates for flow through a sinusoidally curved converging–diverging channel. Gradeck et al. [6] experimentally studied the effects of the hydrodynamic conditions on the enhancement of heat transfer for single phase flow in corrugated channels. Naphon [7–10] conducted numerical and experimental study of forced convection heat transfer and flow developments in a channel with V-corrugated upper and lower plates in which all configuration peaks lie in an in-phase arrangement. The results show that wavy angle and channel height have significant effect on the temperature distribution and flow development. It was found that the sharp edge of the wavy plate (V-shaped) has a significant effect on the enhancement of heat transfer. The turbulent flow of nanofluids with different volume fractions of nanoparticles flowing through a two-dimensional duct under constant heat flux condition was simulated by Rostamani et al. [11]. Ahmed et al. [12] investigated numerically of the turbulent forced convection of nanofluid flow in triangular-corrugated channel over Reynolds number ranges of 1000–5000. It is found that the average Nusselt number, pressure drop, heat transfer enhancement, thermal–hydraulic performance increase with increasing in the volume fraction of nanoparticles and with decreasing in the diameter of nanoparticles.

In this paper, forced convection of nanofluids in semi-circular corrugated channels is numerically studied using finite volume method. The effects of Reynolds number, nanoparticles volume fraction (ϕ) and nanoparticle diameter (d_p) on heat transfer and friction factor are presented and analysed. The aim of this research is to determine the optimum corrugation profile for heat transfer augmentation and to serve as reference for experimental work in the future.

2. Computational Model

Figure 1 shows the geometry of the corrugated channel which will be denoted here as the test section while Figure 2 shows the computational model. The total length of the channel is $L_{\text{total}}=700\text{mm}$. The length of the test section is $L_2=200\text{mm}$, with an upstream rectangular section of $L_1=400\text{mm}$ upstream to ensure a fully developed flow at the leading edge. The downstream section has a length of $L_3=100\text{mm}$ which is used to prevent the occurrence of adverse pressure effects caused by reversed flow which might at the trailing section. The channel height (H) is 10mm while the channel width (W) is 50mm. The semi-circular corrugated height $h=2.5\text{mm}$ with fixed pitch ($p=1.5H$). The geometry configuration was achieved by using SolidWorks. The fluid medium is Al_2O_3 -water nanofluid. It is assumed that the flow is steady and incompressible.

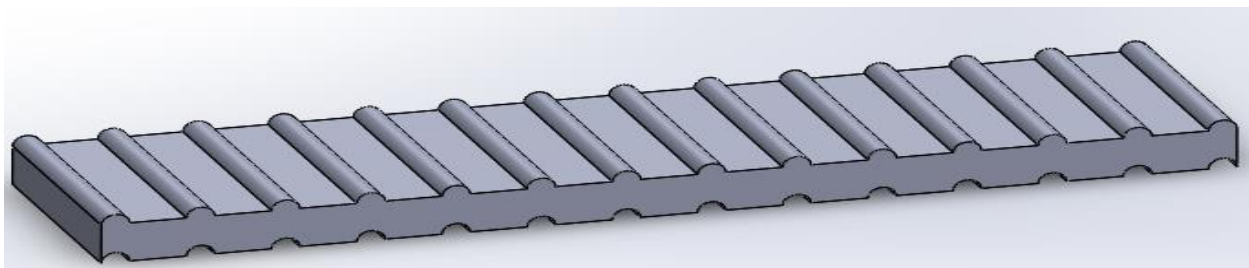


Figure 1. Semi-circular corrugated channel (test section).

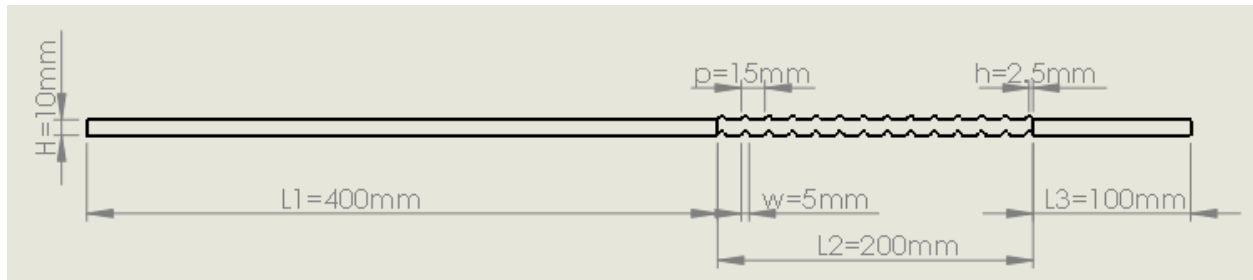


Figure 2. Physical model of the present study.

Five sets of grid distributions are tested on the semi-circular corrugated channel using water as working fluid to analyze the effects of grid size on the results. They are 162632, 267696, 329668, 413520 and 544235 elements, respectively. By comparing the third, fourth and fifth mesh configurations, in terms of average Nusselt number, the corresponding percentage relative errors are 0.021%. Therefore, the fourth grid case has been adopted to obtain an acceptable compromise between the computational time and the results accuracy.

3. Numerical procedure

This simulation utilizes FLUENT (V16.1) as the solver to the governing equations for turbulent three-dimensional incompressible steady-state Reynolds-averaged Navier-Stokes equations shown below:

Conservation of mass:

$$\frac{\partial}{\partial x_i} (\rho u_i) = 0 \quad (1)$$

Conservation of momentum:

$$\frac{\partial}{\partial x_j} (\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left[\mu \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \rho \overline{u_i u_j} \right] + \rho g_i \quad (2)$$

Conservation of energy:

$$\frac{\partial}{\partial x_j} (\rho u_j T) = \frac{\partial}{\partial x_j} \left[\left(\frac{\mu}{Pr} + \frac{\mu_t}{\sigma_t} \right) \frac{\partial T}{\partial x_j} \right] \quad (3)$$

The governing equations are discretized by finite volume method and solved in steady-state implicit format. The SIMPLE algorithm is used to couple the velocity and pressure fields. The second-order upwind scheme is applied and standard $k-\varepsilon$ turbulent model with standard wall function is selected.

The thermophysical properties of the nanofluid used in present study are the ones used by Ahmed et al. [12] as shown in Table [1]. At the inlet, the water velocity can be set for different values according to the Reynolds number. The flow is simulated at five Reynolds number values which are 10000, 15000, 20000, 25000 and 30000. Top and bottom walls for the test section are subjected to a constant heat flux of 10 kW/m^2 while adiabatic boundary condition is applied to the remaining walls. The inlet temperature of the working fluid is $T_{in}=300\text{K}$. Al_2O_3 -water nanofluid is set to flow steadily over the test section at Reynolds numbers stated above.

Table 1. Thermophysical properties of nanoparticle and base fluid at T=300 K

Material	Density ρ (kg/m ³)	Dynamic viscosity μ (N s/m ²)	Thermal conductivity K (W/m.k)	Specific heat C_p (J/kg.k)
Water	998.2	1.00E-03	0.6	4182
Al ₂ O ₃	3600	0	36	765

4. Results and discussion

The effect of Reynolds number, nanoparticles volume fraction and nanoparticles diameter on heat transfer and friction factor are analysed and discussed in this section.

4.1 Validation of straight channel simulation results

For validation, the results obtained by simulation for straight channel which included Nusselt number (Nu) and friction factor (f) are compared with empirical correlations of Dittus–Boelter and Petukov, respectively [13]:

$$Nu = 0.023Re^{0.8}pr^{0.4} \quad (4)$$

$$f = (0.79 \ln(Re) - 1.64)^{-2} \quad (5)$$

Comparisons of Nusselt number and friction factor which shows a very good agreement demonstrated in figure 3 and 4.

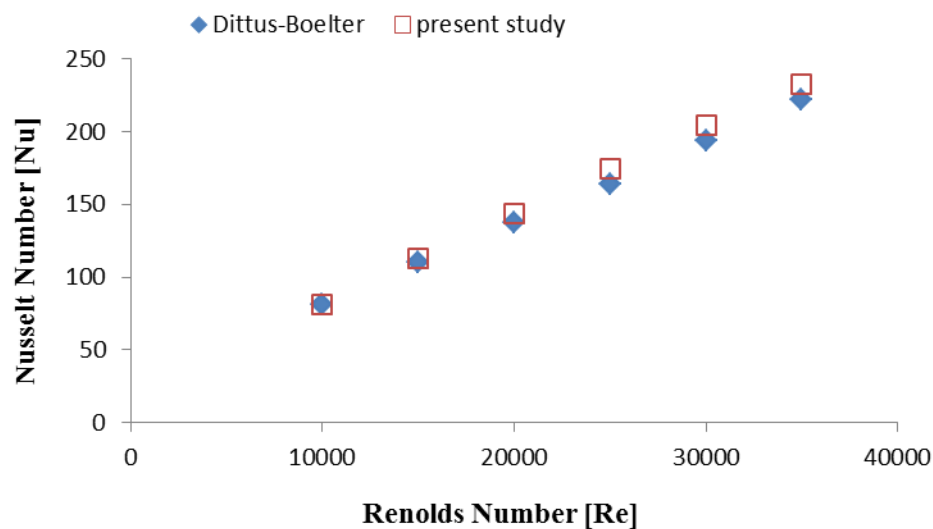


Figure 3. Smooth duct simulated Nusselt number vs. Reynolds number

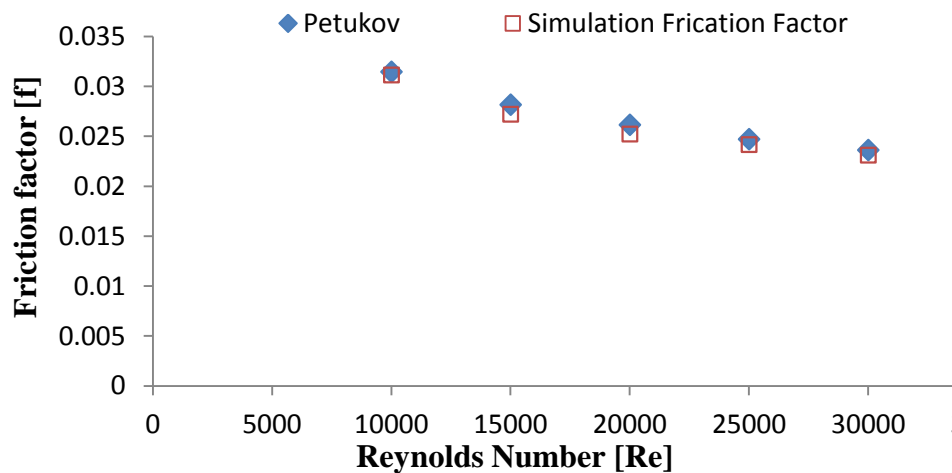


Figure 4. Smooth duct simulated friction factor vs. Reynolds number

4.2 Effect of nanoparticles volume fraction on heat transfer and friction factor

The effect of nanoparticles volume fractions on Nu at $dp=20$ nm is displayed in Figure 5. As expected, Nu increases with ϕ due to improve thermal conductivity of nanofluid. This enhancement is due to the high thermal conductivity of nanoparticles and the role of Brownian motion of nanoparticles on the enhancement of thermal conductivity, which is due to the large surface area of nanoparticles for molecular collisions. Previous studies [11] and [12] reported a similar result. Figure 6 displays the friction factor for different values of ϕ . It is observed that the friction factor increases due to the increase in the viscosity of the nanofluid. Also, with an increase in nanoparticles volume fraction, the possibility and number of contacts between walls and corrugated surfaces increase, which leads to increase in friction factor. The result shows the higher Nusselt number enhancement ratio is 2.07 at Reynolds number 30000 and volume fraction 6%.

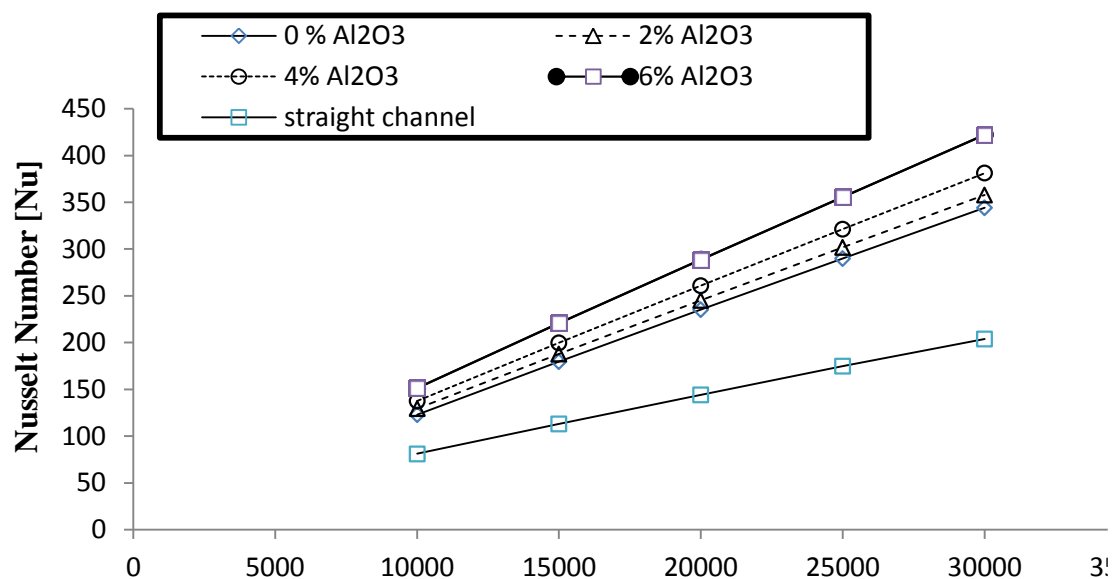


Figure 5. Nusselt number vs. Reynolds number for different ϕ at $dp=20$ nm

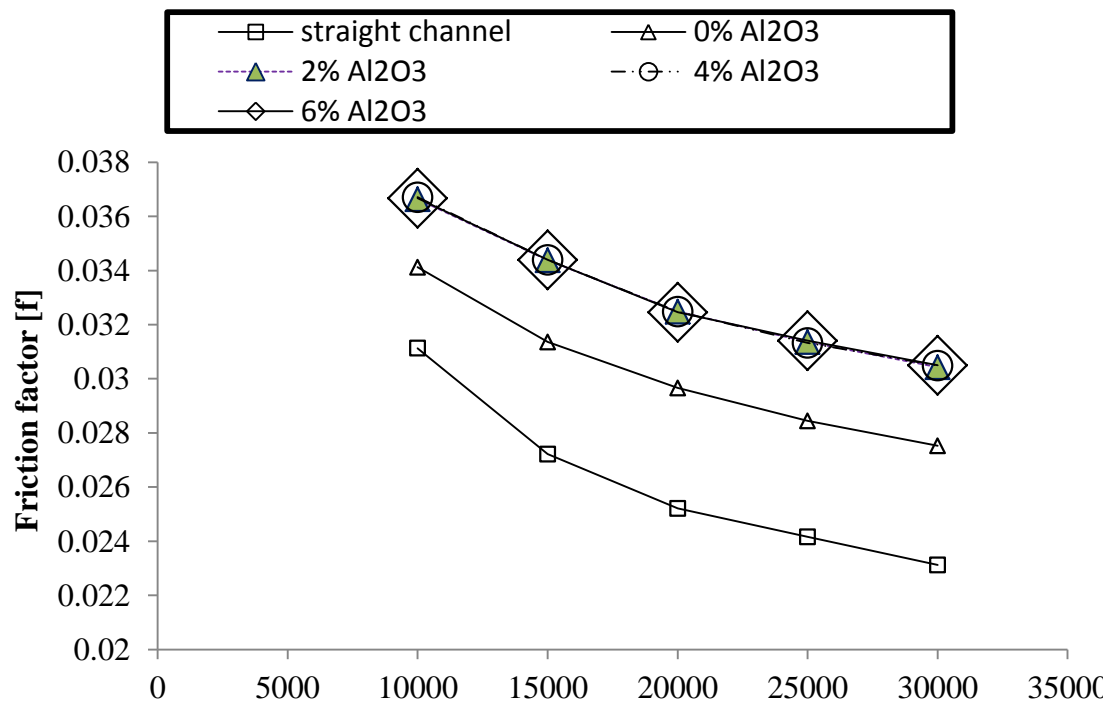


Figure 6. Nusselt number vs. Friction factor for different ϕ at $d_p=20$ nm

4.3 Effect of nanoparticles diameters on heat transfer and friction factor

The effect of nanoparticle diameter on Nusselt number in the range of 20 to 60 nm with constant volume fraction of 6% is shown in figure 7. The results reveals that the nanofluid with smaller particle diameter gives higher Nusselt number as seen for Al₂O₃–water nanofluid. This is due to stronger Brownian motion at smaller nanoparticle diameters, which leads to higher thermal conductivity of nanofluid. The effect of particle size may be attributed mainly to two reasons which are the high specific surface area of the nanoparticles and the Brownian motion. As the particle size reduces, the surface area per unit volume increases. Since the heat transfer is dependent on the surface area, the effectiveness of nanoparticles in transferring heat to the base liquid increases. However, reducing the particle size means increasing the Brownian motion velocity, which again adds up to the contribution by the nanoparticles to the total heat transfer by continuously creating additional paths for the heat flow in the fluid. As presented in this figure the nanofluid with 20 nm nanoparticle diameter has the highest Nusselt number, whereas, the nanoparticle with a diameter of 60 nm has the lowest Nusselt number.

This is consistent with the numerical studies of Ahmed et al. [12]. The comparison of friction factor for Al₂O₃–water nanofluid with different nanoparticle diameters is presented in Figure 8 at various Reynolds numbers. It is noted that there is a slight change in the friction factor when nanoparticle diameters of Al₂O₃ nanofluid are varied. So, for high heat transfer enhancement, the Al₂O₃-water with 6% volume fraction and 20nm particle can be the best choice.

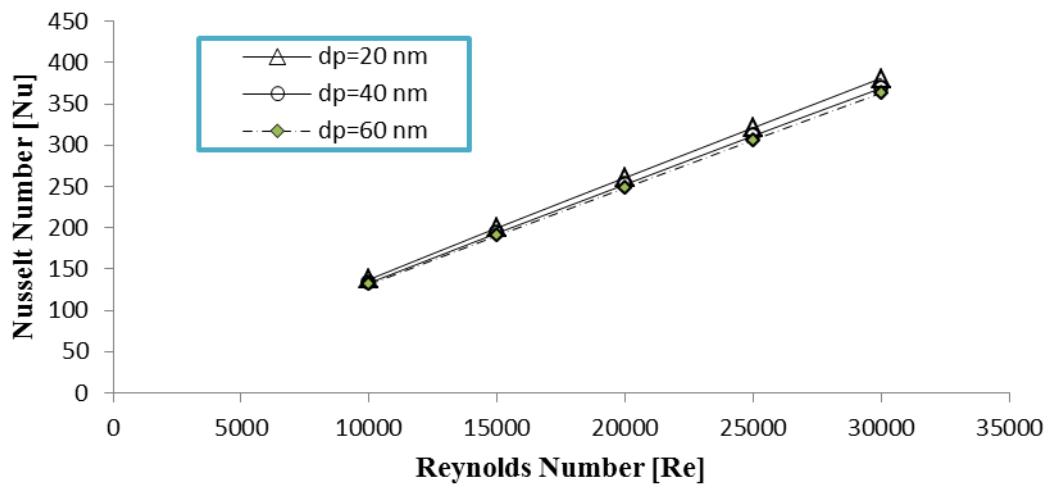


Figure 7. Nusselt number vs Reynolds number for different dp at $\phi=6\%$

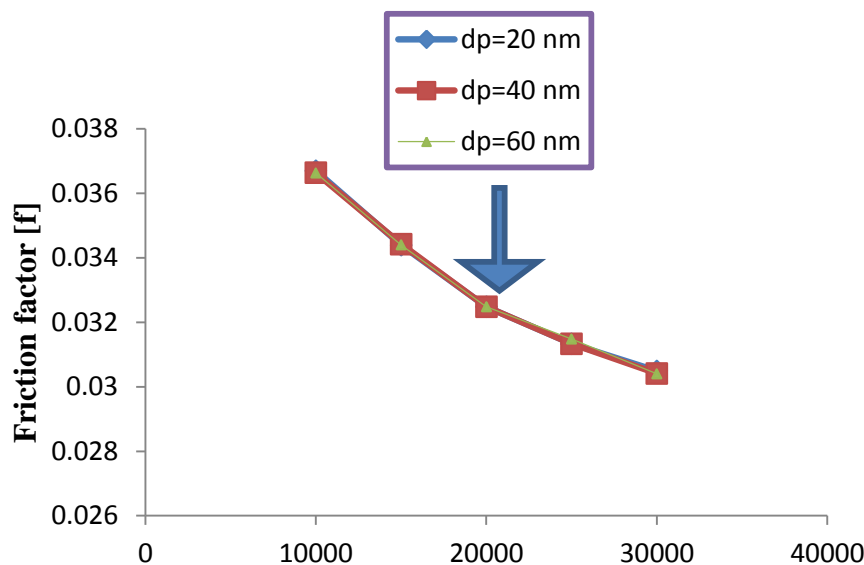


Figure 8. Friction factor vs Reynolds number at $\phi=6\%$

5. Conclusion

Numerical simulations of turbulent forced convection heat transfer in a semi-circular corrugated channel subjected to uniform heat flux were carried out. The computations were performed for a symmetrical semi-circular corrugated channel with varying Reynolds numbers ($10000 \leq Re \leq 30000$), volume fractions ($0\% \leq \phi \leq 6\%$) and nanoparticle diameters ($20\text{nm} \leq dp \leq 60\text{ nm}$). The results of numerical solution showed that Nu and f increase with increasing the Re , and nanoparticles volume fraction. In addition, by decrease of nanoparticle diameter, the Nusselt number increased. The results of the present study are consistent with the results presented by [7], [8], [9], [10], [11] and [12]. Finally, higher Nusselt number enhancement ratio which indicates the optimum configuration is 2.07 at Reynolds number 30000 and volume fraction 6%. Based on the above results, the use of nanofluids in semi-circular corrugated channel is a suitable method to achieve a good enhancement in the performance of many thermal devices as a passive method.

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