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TULTECH

The Effective Techniques for Enhancing the Turbulent Flow Between Two Parallel Plates: A Comprehensive Review

Azza A. Abou EL-Hassan*[1], Mohamed M. Khairat Dawood[1], Mohamed S. EL Morse[2] , Sayed I. Abdel-Mageed[1], Osama A. Sharaf[1]

[1] Mechanical Engineering Department, Faculty of Engineering, Suez Canal University, Ismailia, Egypt

[2] Mechanical Power Engineering Department, Faculty of Engineering, Cairo American University, Egypt

***azzaa7575@gmail.com**

Abstract

The development of heat transfer devices that are used for heat conversion and recovery in several industrial and household applications has depended for many years on the study of improving the heat transfer between two parallel plates. Enhancing thermal performance is of crucial importance to improving the performance and economy of the system, and this has been studied in numerous papers. As the turbulent effects increase, Reynolds numbers improve heat transfer. Therefore, enhancing the turbulent flow between two parallel plates needs a comprehensive review of all enhancement techniques. This review explains various methods to improve the heat transfer between two parallel plates for various plate types (such as flat, corrugated, wavy, chevron, and folded), and the research study was divided into experimental and numerical parts. Furthermore, critical information regarding different enhancement techniques, such as nanoparticle size, particle diameter, type of plate, flow regime, pressure drop, surface techniques, chevron angles, and parameters, is displayed in each section's thorough table. The review indicates that the folded plate has a more turbulent effect on the airflow and gives a more uniform temperature distribution. This system's thermal performance is 35% higher than that of a flat plate in terms of temperature distribution regularity, and it takes half the time to reach thermal equilibrium. The combination of a folded plate and PCM can enhance heat transfer. Therefore, we need more studies of all aspects of this area in the future.

Keywords: *Heat transfer enhancement; Nusslet number; Flat plate; Corrugated plate; Chevron and Folded plates; Two parallel plates; Nano fluid.*

INTRODUCTION

Environmental worries about thermal, air, and water pollution have driven the demand for energy reductions and growing use of renewable energy sources during the past few decades. Because heat transfer devices have been utilized for the conversion and recovery of heat in many industrial and household applications, improving the performance of a heat transfer device is of great interest because it can result in energy, material, and cost savings. The heating and cooling of fluid in concentrated solar collectors, the condensation of steam in power plants, the sensitive heating and cooling of milk in the dairy industries are a few examples of thermal processes used in the pharmaceutical and chemical sectors.

The impact of nanofluids on the rate of heat transfer and thermal performance of numerous current applications has been further examined in several research. A thorough assessment of studies on M/MCs and nanofluids, including theories, parameter effects, difficulties, and future work, has been published by [1]. The analysed studies revealed that the entrance region is where nanofluid flow is most beneficial for enhancing convective heat transfer. This suggests that short channels are more effective for improving heat transfer coefficient HTC. In [2] is presented the background, summary, and use of nanofluids in plate heat exchangers in artistically. The review revealed that, up to a certain amount, increasing nanoparticle volume concentration increases heat transfer rate. Additionally, it increases viscosity and friction, which reduces pumping power. The corrugated type of PHE significantly improves the Nusselt number, and the introduction of nanoparticles indicates improved thermodynamic performance. A higher Reynolds number might result in faster heat transfer.

The requirement for high heat transmission stimulated the development of several methods that enhanced convective heat transfer by reducing thermal resistance at the heated surface. A rise in the rate of heat transfer and pressure decrease leads to a substantial demand for pumping power. There are methods that minimize pressure drop while accelerating heat transfer. The forced flow of fluids such as air, water, mineral oil, ethylene glycol, and other nanofluids on the heated surface is one of these strategies. Depending on the applications, the heated surface may be slick, rough, stationary, or moving. The two main categories of heat transfer augmentation technologies are active and passive. To increase the rate of heat transfer in the active approach, some external power input is required. Depending on the needs of the system, the external power may be applied to either heated surfaces or fluids. Active approaches are challenging because it is difficult to analyses flow structure owing to outside influences. Passive approaches use changed surfaces and/or the insertion of elements (turbulence promoters) into the flow without requiring any external power. By changing the flow treatment, this approach raises the convective heat transfer coefficient. The removal of the thermal barrier layer and the promotion of fluid mixing by turbulence promoters result in a high rate of heat transfer. In [3] is summarized the different techniques that deal with the applications of active and passive methods. The effect of helical coiled wire in heat exchanger tubes employing ferrofluid in the presence of a magnetic field has been suggested as the optimum combination of active and passive procedures. Helical wire would disrupt the boundary layer without disrupting the core flow, and magnetic fields would drive the core flow, lowering the pressure drop and raising thermo-hydraulic performance as a result.

Increasing the effective heat transfer surface area and turbulence in the fluid flowing through the device are two ways that, in general, improve heat transfer and lower thermal resistance. As shown in Figure 1, the plates divided into, flat, corrugated, wavy, chevron and folded.

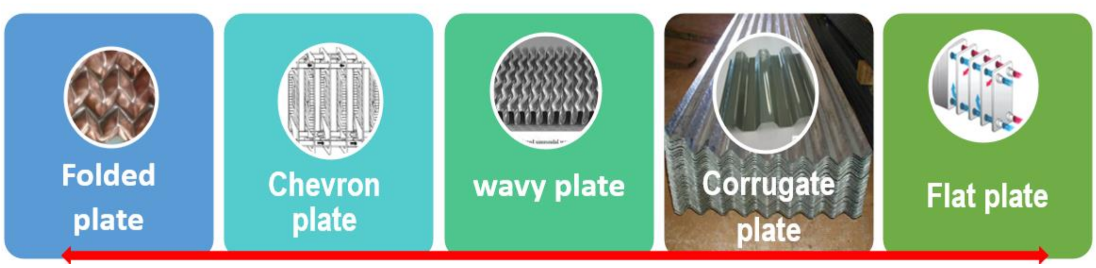


Figure. 1. Different Plate Types

Folded, chevron and corrugated plates are used for the purpose of increasing the effective surface area. The Thermal Performance Factor, or TPF, can be used to assess the efficacy of a heat transfer augmentation approach. The following equation provides its definition.

$$\text{PEC} = \frac{(^{NU}/_{NU\,0})}{(^{f}/_{f0})^{0.33}}, \qquad (1)$$

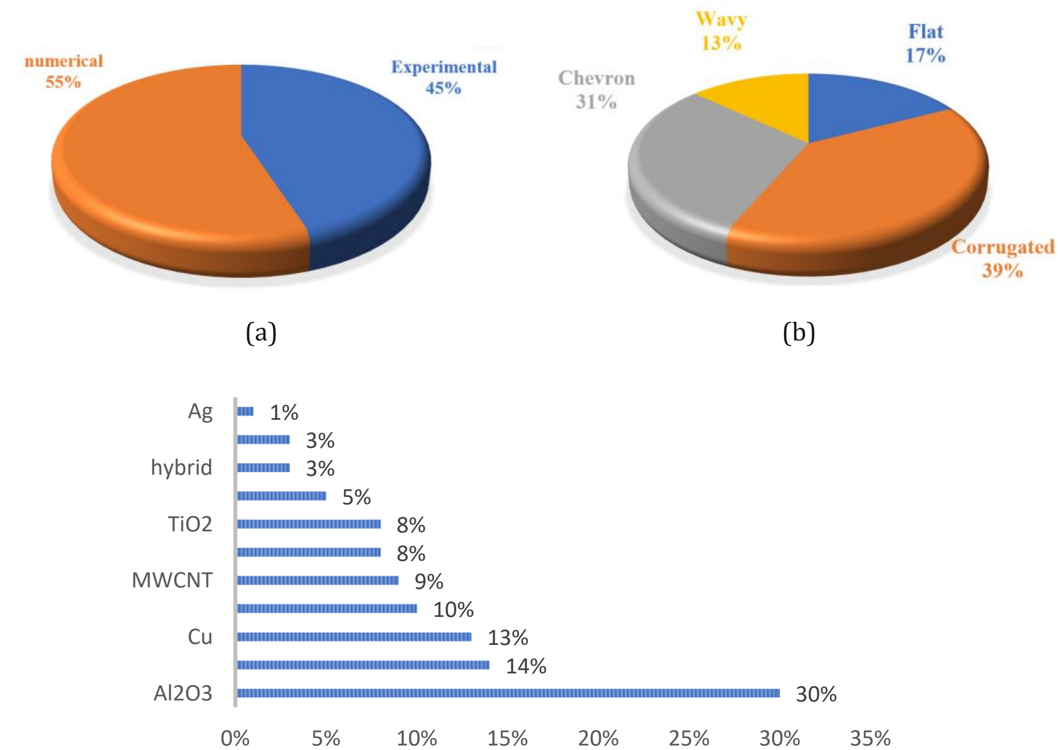
which shows the ratio of the relative effect of change in heat transfer rate to change in friction factor. Where: Nusselt number $NU = \frac{hl}{k}$ and friction factor $f = \frac{\Delta p}{(\frac{\rho u^2}{2})(\frac{L}{Dh})}$ [4].

In this paper, an attempt has been made to examine the assessments carried out by different researchers for the efficient methods for boosting the turbulence between two parallel plates with their Thermal Performance Factor. This essay is formatted as follows: In Section 1, the study on the use of enhancing strategies between two parallel plates, including Nano fluids, surface properties, and geometrical factors, is reviewed. The many studies, including a comparison of PECs and the mechanics of different heat transfer methods, are discussed in Section 2 of this article. Section 3 presents the paper's findings, while Section 4 suggests a few possible future research avenues. In order to help the reader, comprehend the content more easily, a table with the nomenclature and abbreviations is supplied after further studies.

EFFECT OF ENHANCEMENT TECHNIQUES

Effect of Nano Fluid

It is commonly known that nanofluids can enhance heat transmission. Oil, water, and ethylene glycol are examples of common liquids with low heat conductivity. To transfer superheating in various technical applications, nanoparticles (Al, Cu, Al_2O_3, CuO, etc.) have been utilized in conventional liquids. According to [5] the nanoparticles' tiny size and high surface to volume ratio boost thermal conductivity, making them the most ideal heating or cooling medium both now and in the future. The effects of Nano fluids on boosting the turbulent flow between two parallel plates and their enhancement ratio are summarized in Tables 1 and 2 based on experimental studies [6-16] and numerical studies [17-26]. These studies included a study These studies included the investigated the shape factor to carry out a hybrid nanofluid flow analysis between two parallel plates in the presence of a uniform magnetic field by [6]. The outcome shows that the Radial Basis Function approach and the numerical approach work very well together. Additionally, the effects of various form elements were investigated. On nondimensional velocity, temperature, and concentration profiles, it has been demonstrated that several active characteristics, including the viscosity coefficient, Brownian parameter, thermophoretic parameter, magnetic parameter and nanofluid form factor, have an effect. As the thermophoretic and Brownian parameters are raised, the Nusselt number decreases, but the viscosity coefficient tends to do the opposite. The findings demonstrate that increasing the form factor from 3 to 8.6 causes a maximum drop in the velocity, temperature, and concentration profiles of 26, 78, and 19%, respectively. Using (MWCNT-Ag) hybrid nanoparticles will improve the temperature profile by 13% at Ŋ = 0:43. When the shape of the nanoparticle is altered from spherical to blade, the velocity, temperature, and concentration profiles will all be decreased. The rate of heat transfer at [7], studied the effects of viscous dissipation and thermal radiation on an unsteady, incompressible squeezing flow that transports $CuO–Al_2O_3$–water hybrid nanoparticles between two aligned surfaces with changing viscosity. The findings demonstrate that $CuO-Al_2O_3$-water, Al_2O_3-water, and CuO-water depend on $\varphi 1$, $\varphi 2$, and λ, but that the rate of heat transfer at the bottom plate is negligible. In [8] is investigated the numerical and experimental effects of semicircle, trapezoidal, and straight channels on improving the heat transfer with SiO_2-water nanofluids. Studies range of particle volume fractions φ from 0.0 to 0.02 and Re from 10,000 to 30000. The outcomes demonstrate that the novel design of trapezoidal corrugated channel provides the best enhancement to heat transfer. As the volume percentage of SiO2 increases, the enhancement ratio rises. When φ rises from 1% to 2%, ER rises from 2.82 to 3.1 with the same Reynolds number. The trapezoidal corrugated channel (TCC) has a better enhancement ratio than the semicircle corrugated channel (SCC) and the straight channel (SC). The maximal ER of TCC is discovered to be 3.1 at Re = 10000 and φ= 2.0%. Our study covered about 21% of papers and conferences published from 2011 to 2022, as shown in Figure 2.



(a) (b)

(c)

Figure 2. The proportions of studies that focused on improving heat transfer in (a) the research method (b) plate type and (c) type of Nano fluid

Performance comparison among the Nano fluids for all plate

Figure 3 plots the PECs of heat transfer enhancement strategies against the kind of plates, with a focus on Nano fluids. According to Figure 3, the effect of Nano fluids on all plates (chevron, wavy, corrugated, and flat) of the available PEC values is larger than one, indicating that employing these enhancement strategies, the heat transfer performance may be improved while using the same amount of pumping power. PEC decreases as Reynolds numbers rise, showing that rapid pressure drops increases outweigh rising heat transfer coefficients as Reynolds numbers rise. For Figure 3, the Nano fluids (Al_2O_3-, Cu-, CeO_2-, TiO_2water and MWCNT) exhibit greater promise for improving thermal hydraulic performance for all types of plates than other Nano fluids, particularly MWCNT/water, although it is more expensive. When compared to other materials, CeO_2 nanoparticles are less necessary for optimal operation, which reduces cost and creates problems with sedimentation and agglomeration.

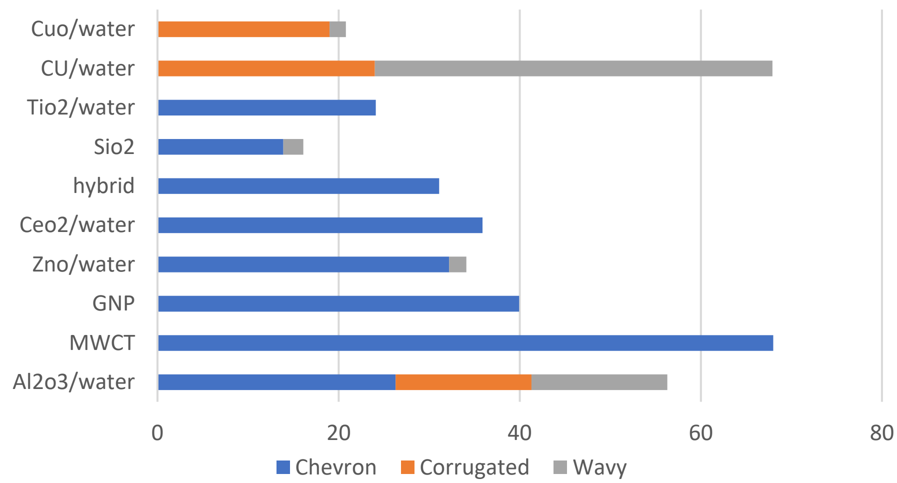


Figure 3. PEC for nanofluids used between two parallel plates from the literature [9-17, 22-26].

Table 1 and 2 depicts experimental and numerical studies for nanofluids between two parallel plates.

Table 1. Experimental studies using nanofluids between two parallel plates

Authors	Method	Type of plate	Re	Particle	φ %	d_p nm	Findings HTE %	Findings Δp (pa)	Key findings
Pandey& NemaK [9]	EXP	Corrugated	800≤ Re ≤ 3000	Al_2O_3	2-4&	40–50	11	6500	that max HTE s is 11% of that for water at 2 vol and should be avoided the use of Nano fluids with φ higher than 2 vol., since they'll cost more and have more higher Δp , due to their greater viscosity, without a single advantage for heat transfer
Kabeel et al [10]	EXP	chevron	800≤Re≤2000	Al_2O_3	1-4	10–100	13	1800	The max increase in Δp is recorded 45% above the base fluid value at 4% vol.& pumping power is recorded 90%.
Tiwariet et al [11]	EXP	Chevron		CeO_2, Al_2O_3, TiO_2 &SiO_2	0-3	10-45	16		The CeO_2/water yields best performance, Al_2O_3/water is next, then TiO_2/water, and finally SiO_2/water.
Huanget et al [12]	EXP	Chevron	100≤ Re ≤ 700	Al_2O_3 & MWCNT	0.56-1.5	40 &9.5	-	6500	MWCNT/water nanofluids have a higher HTE than Al_2O_3/water nanofluids do.
Kumar et al [13]	EXP	chevron	-	CeO_2, Al_2O_3, TiO_2, ZnO, hybrid, GNP, MWCNT	0.5-2	20–80	53	539	For spacing of X = 5 mm at = 0.75 vol%, MWCNT measured the largest climb (53%). (optimum).
Kumar et al [14]	EXP	Chevron		Zno	0.5–2	20–80	32.2	390	The maximum HTE at ϕ (1.0 vol.%.) for β = 60°/60°

Sarafraz & Hormozi [15]	EXP	Chevron	700<Re<25000	MWCNT	0.5-1.5	100	68	500	The thermal conductivity of the water is increased by up to 68% when MWCNT nanotubes are present.
Elias et al [16]	EXP	Chevron	200<Re<5800	Al_2O_3	0-0.5		15.14	600	Al_2O3 has a max heat transfer rate and pressure drop that are respectively 15.14% and 17.3% higher than water at = 0.5%. For β =60°

Table 2. Numerical studies using nanofluids between two parallel plates

Authors	Method	Type of plate	Re	Particle	φ %	d_p nm	Findings		Key findings
							HTE%	Δp (pa)	
Ahmed et al [17]	NUM	Corrugated	100 ≤ Re ≤ 1000	Cu	0-5		43.9	-	The largest boost in heat transfer was found at Re= 200 and =5%, with an improvement of 43.9% over the basic fluid.
Ahmed et al [18]	NUM	Wavy	100 ≤ Re ≤800	Cu	0-5				the improvement in heat transfer increases with, a, and Re.
Abdolbaqi [19]	NUM	Flat	10^4–10^6	TiO2, Al2O3&CuO	1 -4	20	-	-	Nu higher for CuO followed by TiO_2, Al_2O_3.
Ahmed et al [20]	NUM	trapezoidal corrugated	100 ≤ Re ≤ 700	Cu	0-5		-	-	Cu-water in a trapezoidal corrugated channel improved heat transmission even further.
RASHIDI et al [21]	NUM	Wavy	200 ≤ Re ≤1800	Cu	0-3				For single-phase models with high Re, the improvement in heat transfer improves with an increase in, whereas for two-phase models with low Re, the improvement in heat transfer increases in the front and middle of the wavy channel but gradually declines along the wavy channel.
ahmed et al [22]	NUM	different shapes of corrugated	100≤ Re ≤800	CuO	0-5		146	-	The maximum performance factor of 1.46 at Re =800 and ϕ = 0.05.
Yang et al [23]	NUM	Wavy	250≤Re ≤1000	Cu, Al_2O_3 ,Cuo	0-5		15-24	-	With = 3% and 5% of Cu/water Nano fluids, the thermal enhancement can reach 15% and 24% in the wavy channel flow compared to pure fluid.
Hassanzadeh&Tokgoz[24]	NUM	Corrugated	500≤Re ≤2000	Al_2O_3	1-8	25	15.8	-	The largest efficiency index improvements are 15.8% when Re = 500 and an 8% volume fraction of nanoparticles is used.
Ajeel et al [25]	NUM	Semicircle-zigzag corrugate	1000≤Re≤30000	ZnO, Al_2O_3, CuO, SiO_2	0-8	20-80	222	5500	Al2O3, ZnO, and CuO-water nanofluids recorded the next-best heat transfer improvement after SiO2-water. With an average Nusselt number increase of 170%, semicircle corrugated channel exhibits improvements in heat transport that are 1.5–2.7 times better than flat channel.
Ajeel et al [26]	NUM	Trapezoidal-corrugate	1000≤Re≤30000	ZnO, Al_2O_3, CuO, SiO_2	0-8	20-80	250	5500	Al_2O_3, ZnO, and CuO-water nanofluids recorded the next-best heat transfer improvement, followed by SiO_2-water. Corrugated channels with trapezoidal symmetry can advance heat transfer enhancement 1.6-2.88 times faster than straight channels.

Effect of passive surface characteristics on heat transfer

Every particle on an object's surface participates in the process of thermal conduction, which is greatly facilitated by surface formations. Surface heat-conducting particles are present on larger surfaces. As a result, the surface area through which heat is transported determines how quickly heat is transferred. These methods typically indicate that the plate surface falls into one of the following groups: either a surface that is corrugated or roughened. Tables 3 and 4 examine the impact of passive surface approaches for boosting the turbulent flow between two parallel plates and their enhancement ratio, provide an experimental [28-30] and numerical summary [31, 32] of the pertinent study work. These studies included a study the effect of non-uniform arrangements of delta-winglet tapes (DWTs) inside a heated channel to enhance the thermal performance is Both experimental and numerical techniques by [27]. Eight non-uniform DWTs are analyses in total, and their Nusselt number and friction factor are compared to the uniform case. Additionally, the combination of these criteria, which creates an overall performance index, allows for the measurement of overall performance (OPI). the results show that, Nusselt number and friction factor improvements range from 6.3% to 62.2% and 1.9% to 154.3%, respectively. Our study covered about 6% of papers and conferences published from 2010 to 2022, as shown in Figure 4.

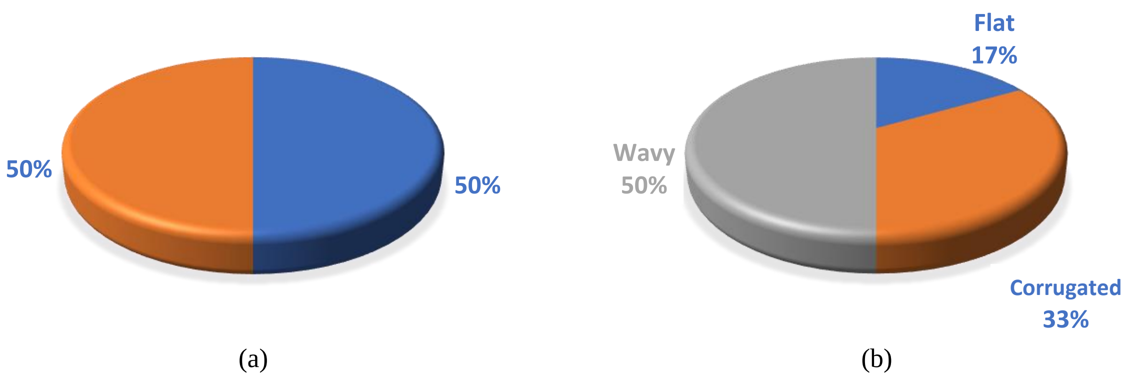


Figure 4. The percentages of the studies related to the heat transfer enhancement in (a) the research method (b) plate type

Table 3. Experimental studies for passive surface techniques on heat transfer between two parallel plates

Authors	Research type	Type of plate	Re	surface techniques	Findings		Key findings
					HTE%	Δp (pa)	
Li Xiao et al [28]	EXP	Flat	500 ≤ Re ≤1500	inclined discrete rib plates	310	-	The inclined discrete rib plate can increase heat transmission by 5–15% while decreasing Δp by 30% when compared to an inclined continuous rib plate channel. In comparison to an inclined continuous rib plate channel, there is a 20–25% increase in heat transmission at the same pumping power.
Kim et al [29]	EXP	Wavy	1000 ≤ Re ≤4000	Single wave and double wave surface	137	-	double-wave PHE demonstrated air-side heat transfer performance that was around 50% higher than that of

							single-wave PHE, while requiring 30% more Δp
Sui et al [30]	EXP	Wavy	300 ≤ Re ≤800	Sinusoidal Micro channels	211	-	In comparison to those using straight channels, micro channel heat sinks based on wavy micro channels have advantages. The heat transfer enhancement factor Enu and the Δp penalty factor Ef rise to 76% and 211%, respectively.

Table 4. Numerical studies for passive surface techniques on heat transfer between two parallel plates

Authors	Research type	Type of plate	Re	Surface Techniques	Findings		Key findings
					HTE%	Δp (pa)	
Sui et al [31]	NUM	Wavy	100 ≤Re ≤800	Micro channels			The current wavy micro channels performed far better than straight ones at heat transmission. Microchannels have substantially smaller Δp values.
Zhenxing &Yangyan [32]	NUM	Corrugated	1000 ≤Re ≤6000	delta-shaped baffles			Nu is 2.1 to 4.3 and f grows by 3.3-106 times more when baffle height is equivalent to corrugation height, respectively, than when baffles are absent.

Performance comparison among the passive surface characteristics

Figure 5 plots the PECs of the heat transfer enhancement approaches against the kind of plates, with a focus on the surface passive techniques. According to Figure 5, all plates (chevron, wavy, corrugated, and flat) with accessible PEC values are more than one, implying that adopting these enhancement techniques will improve heat transfer performance while using the same amount of pumping power. PEC eventually falls out as Reynolds numbers rise, showing that rapid pressure drops increases outweigh rising heat transfer coefficients as Reynolds numbers rise. In Figure 5, surface passive approaches can improve heat transport by 110–310% as compared to smooth channels. 12–25 times as much pressure was lost as compared to smooth. In comparison to inclined (discrete & continuous) rib plate channels and wavy micro channels, sinusoidal micro channels with rectangular cross sections have a greater potential to increase thermal hydraulic performance.

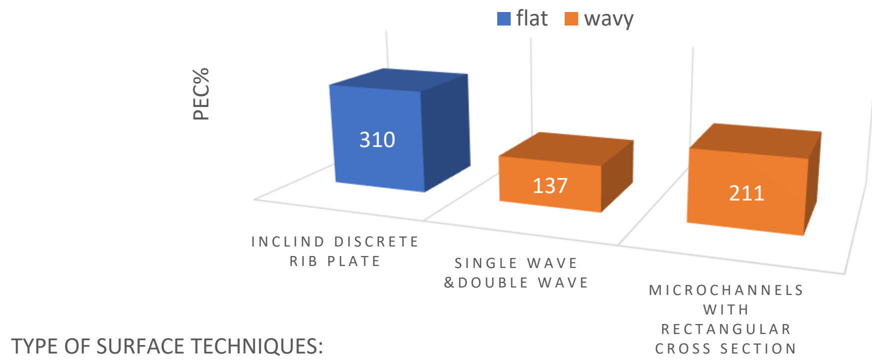
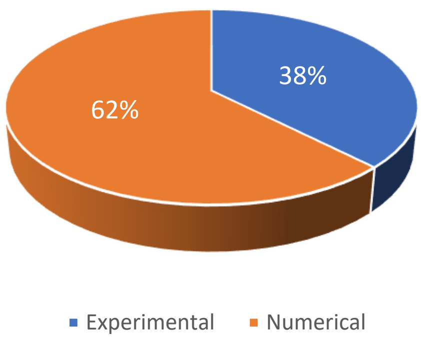


Figure 5. PEC for passive surface techniques used between two parallel plates from the literature [28-30]

Effect of geometrical parameters on heat transfer

Pressure drops and changes in plate geometry have an impact on heat transfer. Many studies have been conducted on this subject. The parameter β under study the most is. The researchers also looked at parameters that expressed the effectiveness of corrugation, such as η, γ, λ, H, n, b and others, The influence of plate shape on enhancing turbulent flow between two parallel plates and their enhancement ratio is summarized experimentally [35-48] and numerically [49-64] in Tables 5, 6. These studies included a study of enhancing the rate of heat transfer for industrial applications, which he did in [7] and is investigated the effects of viscous dissipation and thermal radiation on an unsteady, incompressible squeezing flow that transports $CuO–Al_2O_3$–water hybrid nanoparticles between two aligned surfaces with changing viscosity . Squeezing fluid parameter, Sq, is a heat transfer mechanism that reduces heat transfer. It is anticipated that adding squeezing fluid parameter reduces heat transfer rates by 1.69 and 1.47 percentage points, respectively, at the lower and higher plates. The rate of heat transfer at the lower plate was assessed to have decreased by 12.59% due to injection; this decrease was countered by an increase of 6.16% at the top plate. The rate of heat transfer was found to be increased by suction by 13.94% at the bottom plate but decreased by 6.08% at the higher plate. Temperature drops in accordance with m, Sq, Ec, R, and S. At the bottom and upper plates, respectively, a 0.05% and 0.06% increase and decrease in the heat transfer rate due to the viscosity fluctuation parameter were noted. The rate of orderly heat transfer is significantly increased and decreased by 765.32% and 259.39%, respectively, by the Eckert number at both the bottom and higher plates. The thermal radiation parameter causes a 36.02% drop in the rate of heat transfer at the lower plate, which causes an increase of 75.23% at the top plate. According to [33] is investigated the Casson dusty fluid's generalized natural convection movement. Between vertical plates, the flow is measured along with the fluctuating temperature. The graphical findings demonstrate that the modified Casson fluid model described a more realistic feature of the velocity profile than the traditional Casson fluid model. The enhancement of and Gr increases the fluid and particle velocity. With increasing Re, Pm, K, and M, the fluid and particle velocity-reducing behavior is seen. In relation to the high value of Pe, the temperature profile is rising. The advanced channels have been created and numerically examined by [34] employing forward triangular cross-section rods between parallel plates in the cross-stream plane. Based on the channel mean velocity and hydraulic diameter, numerical calculations using the finite volume approach have been carried out at different Reynolds numbers in the range of 400 to 1000. In the current study, several quantitative and qualitative findings are given. It is discovered that all the parameters taken into account, including S/Lh, G/H, and Re, have a significant impact on the advanced channel's flow behavior and rate of heat transfer. The advanced channel's maximum heat transfer improvement over a standard parallel plate is 2, which was achieved with 64.8% under the conditions of Re=1000, G/H=0.1, and S/Lh=1/5. Our study covered about 33% of papers and conferences published from 2003 to 2022, as shown in Figure 6.



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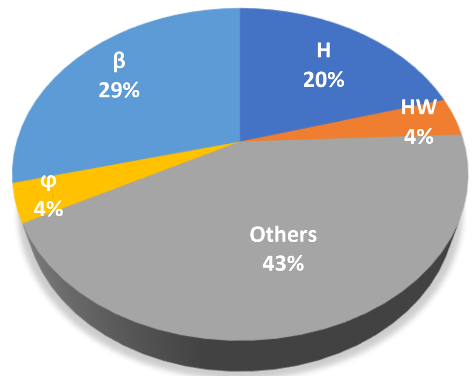

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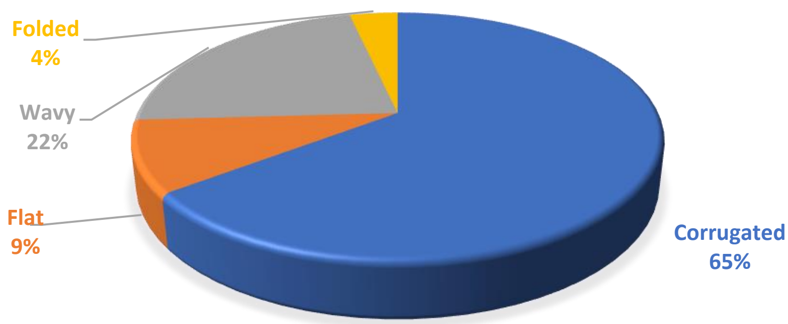

(c)

Figure 6. The percentages of the studies related to the heat transfer enhancement in (a) the research method (b) the geometrical parameters (c) plate type.

Table 5. Experimental works related to parametric studies/analysis of heat transfer enhancement between two parallel plates

Authors	Method	Type of plate	Re	parameter	Findings		Key findings
					HTE %	Δp (pa)	
Islamoglu & Parmaksiz ogl [35]	EXP	Corrugated	1200≤ Re≤4000	H=5&10mm	280		at H = 10 mm, Nu=100% higher than H = 5 mm f increase as H increase but Δp decreases.

Naphon [36]	EXP& NUM	Corrugated	400≤ Re ≤ 1600	β=20°, 40°&60°			Nu increases with increasing β.
Naphon [37] [38]	EXP& NUM	wavy	1500≤ Re ≤2500	in- out-phase			The wavy plate's arc-shaped and sharp edge had a large impact on improving heat transfer and P.
Khan et al [39]	EXP	chevron	500 ≤ Re ≤ 2500	β=30°/30°,30° /60°, 60°/60°			Nu is increases by 2.8 times as β rises from 30° to 60°.
Elshafei et al [40]	EXP	Corrugated	3220≤Re≤ 9420	S& ϕ	320	198	At high Re, the spacing changes on heat transfer and friction factor are more evident than those on phase shift variation.
Faizal& Ahmed [41]	EXP	corrugated	-	ΔX=6, 9, 12mm		45000	Overall, at the same flow rate, h and ΔP rise as ΔX declines.
Pehlivan et al [42]	EXP	Sinusoidal corrugated	1500≤Re≤ 8000	β&H	111		HTC for H = 10 mm is at least 100-280% greater than that of the plain surface, while for H = 5 mm it is between 10 and 50% higher. For a value of Reynolds number equaling 5000, the improvement in heat transfer is up to 4 times for H = 10 mm and 11 times for H = 5 mm over flat plate channels.
Pehlivan et al [43]	EXP	two sharp and two rounded	2000≤Re≤ 9000	β, H	150		Rounded shape better than sharp, Sharp ones is much better at H = 10 mm than they are at H = 5 mm.
Rao et al [44]	EXP	corrugated	500≤Re≤3 000	β = 30°, 40°,50°			As β increases, turbulence increases, increasing the rate of heat transfer
Tokgoz [45]	EXP& NUM	corrugated	3000≤ Re ≤6000	S/H=0.1to 0.3	131	1500	S/H improves the rate of heat transfer; S/H = 0.3 would result in improved heat transfer augmentation.
Azza et al [46]	EXP	folded	3969≤Re≤ 11384	H,q			The thermal performance for folded plate reaches to 35% than flat plat in regularity Temperatures distribution and it provides 50 % of the time required to reach thermal equilibrium.
Kumar et al. [47]	EXP	chevron	800≤ Re ≤ 2300	β=30°/30°, 30°/60° &60°/60°	220	12000	The increasing β from (60°/30°) to (60°/60°) is not recommended economically because the gain in efficiency does not match the rise in power needed.
Kurtulmuş et al [48]	EXP&N UM	sinusoidal - corrugated	4000≤Re≤ 10000	$M=H_{min}/H_{max}$	370		The highest TPF is 1.46 for the configuration of M = 0.5 at Re = 4000

Performance comparison among the geometrical parameters

As shown in Figures 7, the heat transfer enhancement methods' PECs, refers to the effect of different phase shifts and chevron angles, respectively. Most of the time, the PEC is greater than 1, suggesting that the overall thermal-hydraulic performance can be enhanced by expanding the chevron structure's geometrical characteristics (ϕ and β). In such a transitional flow zone as shown in Figure7, the heat transfer in parallel plate channels is boosted by a factor of 2.8, 2.6, and

3.2, respectively, compared to that in corrugated channels of 0°, 90°, and 180° phase shift. With a channel of ϕ = 180° for a relatively narrow channel (S = 4 mm), and of ϕ= 0° for a reasonably wide channel (S = 10 mm), good heat transfer enhancement was achieved [40]. While the wavy channel with 180° phase shift has the biggest pressure loss for Re<200, the 0° phase shift channel has the highest pressure drop for Re>200, followed by 45°, 180°, and 90° phase shift channels. However, among all channels over Reynolds number, the sinusoidal wavy channel with 0° phase shift has the best thermal hydraulic performance. As a result, the sinusoidal-wavy channel with 0° phase shift is theoretically suggested as the best structure for heat exchangers to attain the best performance and a more compact design [58]. The referenced authors' range of chevron angles (0°–180°) was used, and it was discovered that as β rises, thermal hydraulic performance improves more noticeably. For all values of Reynolds numbers, the thermal-hydraulic performance improves more noticeably as β =60°, leading to a high value of pumping power [51]. j is around 1.444 times greater in [53] from β=60° to β=30°.

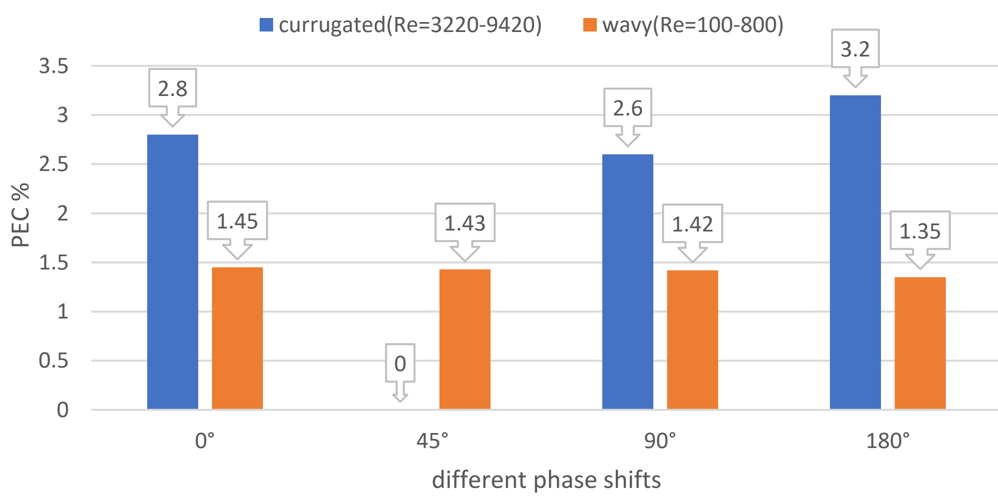

Figure 7. PEC of different phase shifts effect between two parallel plates from the literature [40, 58]

Table 6. Numerical works related to parametric studies/analysis of heat transfer enhancement between two parallel plates

Authors	Method	Type of plate	Re	Parameter	Findings		Key findings
					HTE %	Δp (pa)	
ZHANG et [49]	NUM	Corrugated	-	β (30°,40°,45°,60°,70°&80°)			With an increase in β, the flow pattern between the two plates transforms from "double cross-flow" to "zigzag flow."
Yang& Chen [50]	NUM	V corrugated	2000 ≤Re≤5500	β (20°,40°,60°)	630		The increase β gave rise in a heat transfer rate.
Ahmed &Abed [51]	NUM	corrugated	500≤ Re ≤2500	β=0° to 60°	402	220	the max enhancement in heat transfer occurs at β=60° for all Re.
Zhang and Che [52]	NUM	corrugated	1000≤Re ≤10000	cross-corrugated plates			Corrugated channels have a 1.1 to 3.1 times greater increase in heat transmission than smooth channels.
Han et al [53]	NUM	chevron	500 ≤ Re ≤ 1500	Β			The AAO approach is effective for producing PHE designs that are optimal.
Yin et al [54]	NUM	wavy	2000≤Re ≤10000	ϕ (0° to180°)			These channels of ϕ =0°to 90° perform better overall and have the best structures to obtain the best performance at ϕ = 0°.
Abhishek et al [55]	NUM	wavy	25≤ Re ≤600	H_{min}/H_{max} = 0.1-0.5	180		When the flow is stable, TPF decreases with increasing Re while showing an increasing trend with Reynolds number.
Wang et al [56]	NUM	chevron type (dimple corrugated)	500≤Re≤ 25000	β (30 to80°)	230		The dimple corrugated plate is about 59.2% and 58.7% higher than the typical one on heat transfer and friction factor, respectively, as the corrugation angle is 60° to that of 30° .
Mohammed et al [57]	NUM	corrugated channel with in &out phase	8000≤Re ≤20000	H, H_W, β	450	1000	The ideal values were β=60° and HW=2.5 mm with H= 17.5 mm, and they significantly improved heat transfer. ΔP increased as HW increased and β decreased as HW increased, respectively.
Ahmed et al [58]	NUM	wavy	100≤ Re ≤800	ϕ=0°,45°,90°, 180°	145		The max performance factor is achieved at ϕ= 0° channel at Re=800and ϕ =5%.
Harikrishnan&Tiwari [59]	NUM	skewed wavy	2000≤ Re≤ 4000	ϕ, a	120		The skewed channel with a= 0.2H and ϕ= 45° has the highest TPF.
Ajeel et al [60]	NUM	semicircle corrugated	10,000 ≤Re≤ 30,000.	H,p	255	5400	H = 2.5 mm with P= 15.0 mm are the optimum parameters in thermal performance factor
Dutta et al [61]	NUM	v corrugated	500≤ Re ≤ 1900	H= 12,14,16		3240	H = 12 mm with β =60°are the optimum parameters on

							the heat transfer enhancement
Ajeel et al [62]	NUM	semicircle corrugated	10000≤Re ≤30000	p/L, H/W	255	5400	The ideal settings, H/W = 0.05 and p/L = 0.07, significantly improved the thermal performance factor.
Hassanzadeh, et al [63]	NUM	wavy	100≤ Re ≤600	β=0°, 20°, 40°, 60°, 80°	108		the greatest PI for = 40° under Re = 600 was 1.44.
Islam et al [64]	NUM	chevron	500≤ Re ≤2500	β=30°/30°,60°/60°			The enhancement for Nu is up to 75% and for effectiveness was up to 42%.

CONCLUSIONS

A review of this research work was offered on the effective techniques for enhancing the turbulence between two parallel plates with their Thermal Performance Factor, both for Flat, Corrugated, Wavy, Chevron and Folded plates. The various plate shapes were the main topic of the review's first section. The mechanisms underlying various heat transfer processes were examined in the second part of section 2, which also reviewed the state-of-the-art in heat transfer enhancement techniques between two parallel plates. A thorough evaluation and comparison of the thermal performances between two parallel plates with various geometrical parameters and enhancement techniques was also provided. Additionally, the performance evaluation criteria of several plates were determined using various geometrical parameters and various heat transfer augmentation approaches utilizing data from the open literature.

- It can be inferred from the present review that the enhancement of heat transfer happens in every case because of a reduction in the flow cross section area, an increase in turbulence intensity, and an increase in tangential flow created by different types of inserts. Geometrical parameters of inserts like the area of heat transfer surfaces, gap between plates, corrugated tilt angles, shapes of corrugated the boundary and different phase shifts between the upper and lower plates and the fluid type, etc. affect the heat transfer enhancement considerably.
- The two most widely used enhancement strategies are the use of passive surfaces and nano fluids. These techniques improve heat transfer efficiency while increasing pressure losses. The embossed surface exhibits better heat transfer performance than the roughened surface for passive surface approaches used in heat transfer. The heat transfer coefficient rises on the microstructure surface with the nano- and microporous layer. Surface passive methods can increase heat transmission by 110–315 %. Compared to smooth, a 12–25 times greater amount of pressure was lost. Sinusoidal micro channels with rectangular cross sections have a better potential to enhance thermal hydraulic performance than inclined (discrete & continuous) rib plate channels and wavy micro channels.
- For all types of plates, the Nano fluids (Al_2O_3-, Cu-, CeO_2-, TiO_2water and MWCNT), notably MWCNT/water, show superior potential for the enhancement of thermal hydraulic performance than other Nano fluids, although they are more expensive. When compared to other materials, CeO_2 nanoparticles are less necessary for optimal operation, which reduces cost and creates problems with sedimentation and agglomeration.
- Most influential geometrical parameter is the chevron angle. It has been seen that heat transfer and pressure drop increase with the increase of the chevron angle.

- Among the enhancement techniques, while lowering pressure drop by 30%, the inclined discrete rib plate can improve heat transmission by 5–15% [28]. However, the heat transmission coefficient rises by roughly 211% in the wavy microchannels. The best PEC was obtained by the SiO_2-water Nano fluid with 20 nm diameter, Re = 10000, and 8.0% volume fraction [30], respectively. It measured 222% and 250% for the semicircle-zigzag and trapezoidal corrugate channels [25, 26]. The corrugated channels had a heat transfer coefficient that was roughly 60-250% greater than that of the flat plate for H = 10 mm and 10-50% higher for H = 5 mm for a range of Reynolds Numbers (Re) from 2000 to 9000, H= 5 and 10 mm, and β=30 & 50. The experimental corrugated plates have a pressure drop that is almost six times greater than the flat plate [40]. The thermal performance is improved by non-uniform arrangements of delta-winglet tapes (DWTs) within a heated channel; the Nusselt number and friction factor improvements range from 6.3% to 62.2% and 1.9% to 154.3%, respectively [27].

RECOMMENDATIONS FOR FUTURE WORK

In this research work we have seen some important recommendations which are as follows:

- Thermal performance improvement of heat transfer between two plates, by changing the plate geometry with the aid of heat transfer enhancement methods. It is advised that more research be done to determine the ideal chevron angle and aspect ratio. The chevron angle significantly affects the convection-dominated heat transfer zones in the flow regime. Most relevant studies' experimental findings indicate that increasing the chevron angle enhances the heat transfer coefficient.
- More study on mechanisms of the Nano fluid flow and PCM and their effect on thermal performance and improvement on heat transfer between two plates. Despite the advanced performance shown by these common enhancement techniques, further research is necessary before commercialization to determine their technical and financial viability.
- Use a suitable numerical model to research the impact of a folded plate on heat transfer, you can enhance heat transfer by PCM with a folded plate and then design and production of the heat exchanger using the folded plate.
- Combination of types of plate with different geometrics, coating with high Thermal Conductive and nano fluid to increase improvement in thermal performance.

NOMENCLATURE

	List of Symbols in this study		**Greek symbols**
a	wavy amplitude, mm		
b	distance between two plates, mm	α	dimensionless wavy amplitude, (α=A/L)
de	equivalent diameter, m	S/H	aspect ratios
dp,	particle diameter, nm	λ	wavelength
D_h	Hydraulic diameter, m	*ϕ*	skewness angle
H	Channel height	φ	The nanoparticle volume fraction
Hw	height of wavy, mm	φ	Phase shift
h	height of corrugation (mm)	β	chevron or corrugation angle

h/W	height-to-width ratio
p	pitch of corrugation (mm)
p/L	pitch-to-length ratio
P_I	Thermal–hydraulic performance index
L	Corrugation length (mm)
Lw	wavelength of the wavy wall, mm
Gr	Grashof number
j/f	area goodness factor
M	magnetic parameter
n	Number of waves
Nu	Nusselt number, $NU = \frac{hl}{k}$
q	heat flux, kW/m^2
Re	Reynolds number
S	width of corrugated
T	time
W	Channel width
V	Air flow rate, liter/s

Abbreviations

AAO	approximation-assisted optimization
TPF	the thermal performance factor,
MWCNT	Multi-walled carbon nanotube
MHD	Magneto hydro dynamic
H. F	The heat flux W/m^2
HTC	heat transfer coefficient
EMHD	Electro magneto hydrodynamic
HTE	Heat transfer enhancement
PCM	Phase change material

CONFLICT OF INTERESTS

The authors confirm that there is no conflict of interests associated with this publication.

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