



Second-Law Analysis of Transient Heat Transfer Enhancement in Tubes Equipped with Elliptical-Cut Twisted Tape Using Hybrid Nanofluids under Mixed Convection

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Abstract

This study numerically investigates entropy generation during transient heat transfer in a vertical receiver tube fitted with elliptical-cut twisted tape inserts, with particular emphasis on the thermodynamic impact of forced and mixed convection conditions. Simulations are performed using STAR-CCM+, employing an Al₂O₃-Cu/water hybrid nanofluid as the working medium to enhance thermal transport and capture realistic solar receiver behavior. The analysis focuses on the distribution of entropy production arising from both heat transfer and viscous dissipation under unsteady flow conditions influenced by buoyancy effects. Results indicate that entropy generation is strongly dependent on the convection regime, where the inclusion of buoyancy in mixed convection reduces thermal irreversibility by improving temperature uniformity and weakening local gradients along the tube surface. In contrast, forced convection exhibits higher entropy levels due to stronger thermal non-uniformity. Although the twisted tape induces localized increases in frictional entropy generation due to intensified velocity gradients, its overall contribution remains secondary compared with thermal effects. A comparison between the two regimes shows that mixed convection achieves a notable reduction in total entropy generation, reaching approximately 58.2%, highlighting its superior thermodynamic performance. The findings demonstrate that coupling hybrid nanofluids with swirl-inducing inserts under buoyancy-assisted conditions is an effective strategy for reducing irreversibility and improving the efficiency of solar thermal receiver systems.

Keywords: Hybrid nanosuspension; Entropy generation (local/global); Bejan parameter; Time-dependent heat transfer; Elliptical-cut swirl tape; Thermodynamic irreversibility.

Nomenclature

a	Diameter-cut width (m)
b	Diameter-cut long (m)
Be	Bejan number (-)
C_1, C_2, C_μ	Model constants (-)
C_p	Specific heat (J Kg ⁻¹ C ⁻¹)
D	Diameter of pipe (m)
f	Friction Factor (-)
G_k	Turbulent kinetic energy generation (J Kg ⁻¹)
h	Heat transfer coefficient (W m ⁻² C ⁻¹)
k	Turbulent kinetic energy (J Kg ⁻¹)
k_c	Thermal conductivity of fluid (W m ⁻¹ C ⁻¹)
L	Pipe length (m)
S_{ij}	Linear deformation rate (-)
P	Pressure (Pa)
t	Time (s)

T	Fluid temperature (C)
u	Velocity (m s ⁻¹)
w	Twisted tape width (m)
y	Twisted tape pitch (m)

Greek Symbols

ρ	Density (Kg m ⁻³)
ϕ	Solid volume fraction (-)
μ	Fluid dynamic viscosity (Pa-S)
τ_w	Wall shear stress (Pa)
η	Thermal efficiency factor (-)
δ	Thickness (m)
σ_ε	Model constant (-)
Δp	Pressure drops (Pa)

Acronyms

TECT	Twisted pipe with elliptical cut
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1. INTRODUCTION

Solar receiver tubes are fundamental components in parabolic trough collector (PTC) systems, as they are responsible for converting incident solar radiation into useful thermal energy. These tubes account for a significant portion of the total system cost and play a crucial role in determining overall efficiency and reliability [1]. Solar thermal energy has been widely utilized in various applications, including domestic and industrial water heating, solar-driven chemical processes, desalination systems, and concentrated solar power plants [2, 3]. In recent years, research efforts have increasingly focused on improving the thermal performance of parabolic trough systems to meet growing industrial energy demands.

One of the primary challenges in these systems is the non-uniform distribution of heat flux around the circumference of the absorber tube, which leads to substantial temperature gradients. These gradients can induce thermal stresses, potentially causing structural damage to the receiver tube and its protective glass envelope [4]. Furthermore, elevated absorber temperatures contribute to increased thermal losses, thereby reducing system efficiency [5]. High concentration ratios and optical efficiencies, while desirable, often intensify these issues by generating large heat fluxes and non-uniform temperature fields [6]. Therefore, enhancing heat transfer within the receiver tube is essential to minimize temperature variations and improve both performance and durability.

Among various heat transfer enhancement techniques, twisted tape inserts have been widely studied due to their ability to induce swirl flow and promote fluid mixing. This results in a thinner thermal boundary layer and higher convective heat transfer rates [7]. However, most previous studies have focused on steady-state conditions with uniform heat flux, which do not accurately represent real operating conditions in solar thermal systems. In practice, solar radiation varies with time, leading to transient heat transfer behavior that must be considered for accurate system analysis [8].

In addition to geometric modifications, improving the thermophysical properties of the working fluid has proven to be an effective approach for enhancing heat transfer. Nanofluids, which are formed by dispersing nanoparticles in a base fluid, exhibit improved thermal conductivity and heat transfer characteristics compared to conventional fluids [9]. More recently, hybrid nanofluids, consisting of multiple types of nanoparticles, have demonstrated even greater potential due to synergistic effects between different particles [10]. When combined with flow-enhancing devices such as twisted tapes, these fluids can significantly improve thermal performance.

Mixed convection, which occurs when both forced and natural convection mechanisms are present, plays an important role in many thermal systems, including solar receivers. The interaction between buoyancy-driven flow and externally



imposed flow can significantly influence heat transfer behavior, depending on operating conditions and system geometry [11]. Despite its importance, relatively few studies have examined mixed convection in conjunction with advanced enhancement techniques such as twisted tape inserts and hybrid nanofluids, particularly under transient conditions.

While most studies on heat transfer enhancement rely on the first law of thermodynamics, this approach does not provide insight into the quality of energy or the extent of irreversibility within the system. The second law of thermodynamics introduces entropy generation as a measure of energy degradation and system inefficiency [12]. Entropy generation occurs due to heat transfer across finite temperature differences and viscous effects associated with fluid flow [13]. Therefore, entropy generation analysis has become a valuable tool for evaluating and optimizing thermal systems.

Several researchers have applied entropy generation analysis to systems involving nanofluids and heat transfer enhancement techniques. For example, studies have shown that while enhanced mixing improves heat transfer, it may also increase fluid friction, leading to higher entropy generation [14]. However, most of these investigations are limited to steady-state conditions and simplified geometries. The combined effects of transient heat transfer, mixed convection, hybrid nanofluids, and advanced twisted tape configurations, such as elliptical-cut designs, have not been comprehensively explored.

Accordingly, the present study aims to investigate entropy generation and

transient heat transfer enhancement in a tube equipped with elliptical-cut twisted tape inserts under mixed convection conditions. The analysis considers turbulent flow using hybrid nanofluids as working fluid. The outcomes of this study are expected to provide deeper insights into the trade-off between heat transfer enhancement and thermodynamic irreversibility, thereby contributing to the design of more efficient and reliable thermal systems.

Numerical methods

Model geometry and boundary definitions of TECT

A three-dimensional computational model was established and implemented within the STAR-CCM+ environment to simulate the transient behavior of hybrid nanofluid flow inside a circular tube equipped with an elliptically-cut twisted tape insert. The governing equations, including the continuity, momentum, and energy equations, were formulated based on the Reynolds-averaged Navier–Stokes (RANS) approach and solved using an implicit unsteady scheme to ensure numerical stability under time-dependent conditions. The turbulence characteristics of the flow were captured using the realizable $k-\epsilon$ model, which is recognized for its improved capability in predicting rotational and swirling flows induced by internal inserts. The presence of the twisted tape generates strong secondary flow structures and enhances fluid mixing; therefore, an appropriate turbulence closure is essential for accurately resolving these complex interactions. To properly account for near-wall effects, a two-layer all y^+ wall treatment available in STAR-CCM+ was



employed. This approach enables seamless integration between the viscous sublayer and the fully turbulent region without requiring excessively fine mesh resolution near the wall, thereby improving computational efficiency while maintaining high accuracy in the prediction of wall shear stress and heat transfer rates. A transient simulation framework was adopted to

capture the temporal evolution of flow and thermal fields. The time discretization was performed using a constant time step of $\Delta t = 0.001$ s, which was selected based on stability and convergence considerations to ensure adequate temporal resolution of the unsteady flow features induced by the twisted tape geometry.

Table 1 Geometric details of the analyzed parabolic trough receiver.

Feature	D	L	w	y
Value	0.0195 m	1 m	0.019 m	0.057 m
Feature	δ	a/w	b/a	
Value	0.0008 m	0.7	2	

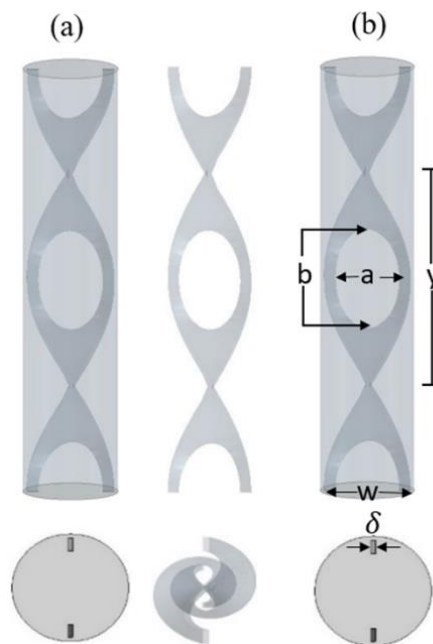


Figure 1 Illustration of (a) the twisted elliptical-cut tape tube (TECT) configuration and (b) the precise geometric dimensions considered in the present analysis.

In this investigation, a vertical tube carrying an $\text{Al}_2\text{O}_3\text{-Cu}$ /water hybrid nanofluid under mixed convection conditions was examined. The tube contains twisted tapes with elliptical cuts,

and the wall is subjected to a constant heat flux. The physical representation of the system was constructed and simulated using the STAR-CCM+ computational platform. The model



considers a three-dimensional, time-dependent flow field with gravity acting along the flow direction.

Turbulence effects were modeled using the realizable k-ε approach. Near-wall behavior was handled through the two-layer formulation available in STAR-CCM+ together with the all-y+ wall treatment, which helps improve the representation of viscous effects close to the surface. To increase temporal accuracy, transient calculations were performed with a time increment of $\Delta t = 0.001$ s. The boundary conditions applied in the present analysis are summarized below.

(1) The system configuration is illustrated in Figure 1 and summarized in Table 1, representing a circular tube equipped with twisted tape inserts that contain elliptical openings.

(2) Regarding the thermal conditions, the tube wall was subjected to a uniform heat flux of $4000 \text{ W} \cdot \text{m}^{-2}$ [15].

(3) The working fluid employed in the simulations was an Al_2O_3 -Cu/water hybrid nanofluid with a particle volume fraction of 2%.

Thermophysical properties of hybrid nanofluid (Al_2O_3 -Cu/water hybrid nanofluid)

In the present research, an Al_2O_3 -Cu/water hybrid nanofluid is employed as the heat transfer medium. The thermophysical characteristics that vary with temperature are expressed in Equations (1–6) [15]. Table 2 lists the thermophysical parameters of pure water, Al_2O_3 , and Cu, which were determined following the approach described by [16].

Density (ρ)_{hnf}:

$$\rho_{hnf} = \rho_f + \phi_A(\rho_A - \rho_f) + \phi_C(\rho_C - \rho_f) \quad (1)$$

In this formulation, the symbols A and C correspond to aluminum oxide (Al_2O_3) and copper (Cu), respectively. The parameter ϕ_{hnf}

indicates the combined volume fraction of the two nanoparticle species distributed within the hybrid nanofluid, and it is determined as follows:

$$\phi_{hnf} = \phi_A + \phi_C \quad (2)$$

Specific heat (C_p)_{hnf}:

$$(\rho C_p)_{hnf} = (\rho C_p)_f + \phi_A[(\rho C_p)_A - (\rho C_p)_f] + \phi_C[(\rho C_p)_C - (\rho C_p)_f] \quad (3)$$

Buoyancy (β)_{hnf}:

The Buoyancy for hybrid nanofluids can be determined through the following relation, which has been referenced in numerous previous studies.

$$(\rho\beta)_{hnf} = (\rho\beta)_f + \phi_A[(\rho\beta)_A - (\rho\beta)_f] + \phi_C[(\rho\beta)_C - (\rho\beta)_f] \quad (4)$$

Dynamic viscosity (μ)_{hnf}:

$$\mu_{hnf} = \mu_f \cdot (1 - \phi_A)^{-2.5} (1 - \phi_C)^{-2.5} \quad (5)$$

Thermal conductivity (k_c)_{hnf}:

$$k_{hnf} = k_f \frac{(\frac{k_p}{\phi_{hnf}} + 2(k_f + k_p) - 2\phi k_f)}{(\frac{k_p}{\phi_{hnf}} + 2k_f + \phi k_f - k_p)} \quad (6)$$

In these formulas, ϕ_A and ϕ_C refer to the volumetric proportions of solid aluminum oxide particles and copper nanoparticles, respectively. The symbols f and hnf correspond to the base liquid and the hybrid nanofluid. Moreover, k_{hnf} represents the thermal conductivity of the hybrid nanofluid, while k_f , k_A , and k_C indicate the thermal conductivities of the base fluid, aluminum oxide, and copper, respectively.



Table 2 Thermal and material properties of the working fluid and dispersed nanoparticles.

Substances	Water	Al ₂ O ₃	Cu
$\beta \times 10^{-5} (K^{-1})$	21	0.85	1.67
$\rho (Kg m^{-3})$	997.1	3970	8933
$C_p (J Kg^{-1} K^{-1})$	4179	765	385
$k_c (Wm^{-1} K^{-1})$	0.613	40	400

Governing equation

For buoyancy-influenced convective heat transfer (mixed convection), fluid properties are constant; however, the density depends on the difference in local and reference temperatures [17].

$$(\rho)_{hnf} = \rho_o [1 - \beta_{hnf}(T - T_o)] \quad (7)$$

The fundamental equations describing the behavior of a hybrid nanofluid can be formulated based on the following assumptions:

Continuity equation:

$$\frac{\partial}{\partial t}(\rho) + \frac{\partial}{\partial x_i}(\rho u_i) = 0 \quad (8)$$

Momentum equation:

$$\frac{\partial(\rho u_i)}{\partial t} + \frac{\partial(\rho u_i u_j)}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\partial \tau_{ij}}{\partial x_j} + \rho_{hnf} \beta_{hnf} (T - T_o) g \quad (9)$$

The buoyancy term appearing in Eq. (9) is evaluated by applying the Boussinesq Approximation, as described in the ANSYS Fluent theory documentation [18]:

$$(\rho_{hnf} - (\rho_{hnf})_o) g \approx -\rho_{hnf} \beta_{hnf} (T - T_o) g$$

Energy equation:

$$\frac{\partial}{\partial t}(\rho u_i) + \frac{\partial}{\partial x_i}(\rho u_j C_p T) = \frac{\partial}{\partial x_i} \left(k_{eff} \frac{\partial T}{\partial x_i} + \frac{\mu_t}{\sigma_{h,t}} \frac{\partial (C_p T)}{\partial x_j} \right) u_j \frac{\partial p}{\partial x_i} + [\mu_{eff} \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \frac{2}{3} \mu_{eff} \frac{\partial u_i}{\partial x_i} \delta_{ij}] \quad (10)$$

$$\mu_{eff} = \mu + \mu_t \quad (11)$$

According to established turbulence modeling approaches [19], the k - ϵ model expresses the effective turbulent viscosity (μ_t) as a function of the local turbulent kinetic energy (k) and its dissipation rate (ϵ). This formulation provides a closure for the Reynolds-averaged momentum

equations by accounting for the enhanced mixing effects induced by turbulence.

$$\mu_t = \rho C_\mu \frac{k^2}{\epsilon}$$

Equation of kinetic energy (k) of turbulence:



$$\frac{\partial(\rho k)}{\partial t} + \frac{\partial(\rho k u_i)}{\partial x_i} = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] - \rho \varepsilon + G_k \quad (12)$$

Equation of turbulence dissipation rate (ε)

$$\frac{\partial(\rho \varepsilon)}{\partial t} + \frac{\partial(\rho \varepsilon u_i)}{\partial x_i} = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} \right] + \rho C_1 S_\varepsilon - \rho C_2 \frac{\varepsilon^2}{k + \sqrt{(\mu/\rho) \varepsilon}} \quad (13)$$

The quantity G_k denotes the rate at which turbulent kinetic energy is generated due to mean velocity gradients, and its mathematical

expression follows the standard formulation adopted in k - ε turbulence closures, given by:

$$G_k = \frac{\partial u_j}{\partial x_i} \mu_t \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \frac{2}{3} \left(\rho k + \mu_t \frac{\partial u_i}{\partial x_i} \right) \delta_{ij} \quad (14)$$

In the realizable k - ε turbulence formulation, the model constants C_1 and C_μ are not treated as fixed values; instead, they are evaluated through specific functional relationships derived within the model framework, as expressed by the following equations [20].

$$C_1 = \max \left[0.43, \frac{\eta}{\xi + 5} \right], \quad \xi = S \frac{k}{\varepsilon}, \quad S \equiv \sqrt{2 S_{ij} S_{ij}}$$

The remaining empirical parameters are assigned according to the values reported in the literature [20]:

$$\sigma_\varepsilon = 1.2, C_2 = 1.9.$$

The tensor S_{ij} characterizes the rate of strain associated with the deformation of a fluid element under motion, as defined in [20]:

$$S_{ij} = \frac{1}{2} \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \quad (15)$$

Entropy generation

Under all conditions examined, entropy production at the local scale was determined, with heat transfer and frictional effects treated as the only sources:

$$S_{g,t} = S_{HT} + S_{FF} \quad (16)$$

Entropy generation within a control volume is typically defined by the following relation [21]

$$S_{HT} = \frac{k_c}{T^2} \left(\frac{\partial T}{\partial x_i} \right)^2 \quad (17)$$

$$S_{FF} = \frac{\mu}{T} \left\{ 2 \left[\left(\frac{\partial u_i}{\partial x_i} \right)^2 \right] + \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right)^2 \right\} \quad (18)$$

The overall entropy generation $S_{g,t}$ is obtained by integrating the local entropy production terms throughout the pipe volume.



$$S_{g,t} = \{ \int S_{HT} + \int S_{FF} \} \partial V \quad (19)$$

The Bejan number is commonly used to quantify the relative contribution of individual irreversibility mechanisms to total entropy generation. It is defined as [14]:

$$Be = \frac{S_{HT}}{S_{g,t}} \quad (20)$$

Numerical analysis

The numerical analysis was performed using the commercial computational fluid dynamics platform STAR-CCM+, which was employed for both geometric modeling and mesh generation of the investigated tube equipped with an internal swirl-inducing insert. The governing conservation equations of mass, momentum, and energy, together with the transport equations of the k-ε turbulence model, were discretized using the finite volume method in accordance with the assumptions outlined in Section 2.1. Pressure-velocity coupling was handled through the Semi-Implicit Method for Pressure-Linked Equations (SIMPLE), originally introduced by Suhas V. Patankar and Brian Spalding [22]. A second-order temporal discretization scheme was adopted to ensure improved accuracy in capturing the transient evolution of the thermal field. Unlike purely forced convection analyses, the present investigation accounts for mixed convection conditions, where buoyancy effects arising from temperature-induced density variations interact with the imposed flow. This coupling introduces additional complexity in the velocity and temperature distributions, requiring careful numerical treatment under unsteady flow conditions. The flow inside the tube is considered turbulent to accurately resolve the enhanced mixing and heat transfer behavior induced by the internal insert. Model validation was carried out by comparing the predicted temperature distributions with previously published experimental and numerical results [15, 23-27], demonstrating good agreement and confirming the reliability of the adopted numerical approach. The detailed outcomes of

the simulation are discussed in the Results and Discussion section.

Mesh Independence Assessment and Numerical Model Validation

Grid Independence Assessment

A grid-independence (mesh sensitivity) study was conducted to confirm that the numerical results are not affected by the discretization of the computational domain. Simulations were performed using the Realizable k-ε turbulence model at a Reynolds number of 11,000 for turbulent flow in a vertical tube fitted with elliptical-cut twisted tape inserts. Five different meshes with approximately 4,661,961; 4,046,985; 3,834,556; 3,500,893; and 3,426,964 cells were generated to evaluate the effect of mesh density, with particular refinement applied near the tube walls and insert surfaces to accurately capture strong velocity and thermal gradients. The outlet temperature, averaged over the exit cross-section, was used as the primary comparison parameter, and its variation with time for each mesh is presented in Figure 2, showing very similar trends and convergence behavior across all cases. The mesh with 4,046,985 cells was selected as the reference, and the relative deviations in outlet temperature for the other meshes were found to be only 0.3%, 0.2%, 0.4%, and 0.1%, respectively, indicating negligible differences. These results demonstrate that further mesh refinement does not significantly influence the predicted thermal performance, confirming that the solution is effectively independent of the grid resolution; therefore, the 4,046,985-cell mesh was adopted for all subsequent simulations as it provides an optimal balance between computational accuracy and efficiency.

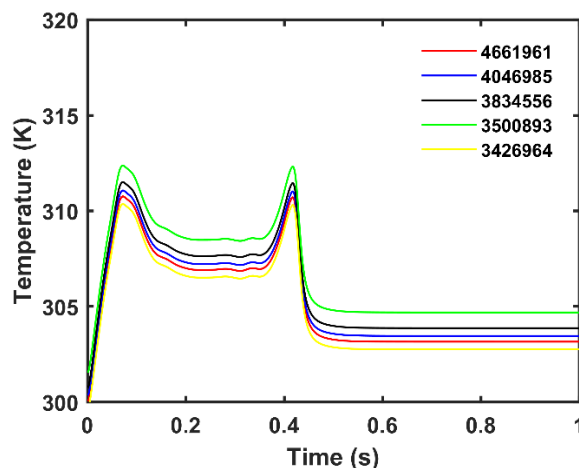


Figure 2 Mesh refinement verification

Numerical model verification

The numerical methodology adopted in this study was validated by comparing its predictions with previously published results reported by Ahmet Tandiroglu [28] under the same flow condition of Reynolds number $Re = 10,000$. This validation step was performed to ensure that the present computational model can reliably reproduce established thermal and flow characteristics for turbulent regimes. Figure 3 illustrates the variation of outlet temperature as a function of time for both the current simulation and the reference data, where a very close agreement in trend and magnitude can be observed. Both results exhibit similar transient behaviour and converge toward nearly identical

steady-state values, indicating that the essential heat transfer mechanisms, including convection and turbulence effects, are accurately captured. A quantitative comparison shows that the deviation between the present results and the reference data remains within approximately $\pm 6.9\%$, which is considered acceptable for numerical simulations involving complex turbulent flows and enhanced heat transfer geometries. These minor differences may arise from variations in mesh structure, numerical discretization schemes, and boundary condition implementations; however, the overall consistency confirms that the numerical model employed in this study is valid and capable of producing accurate and reliable predictions for further analysis.

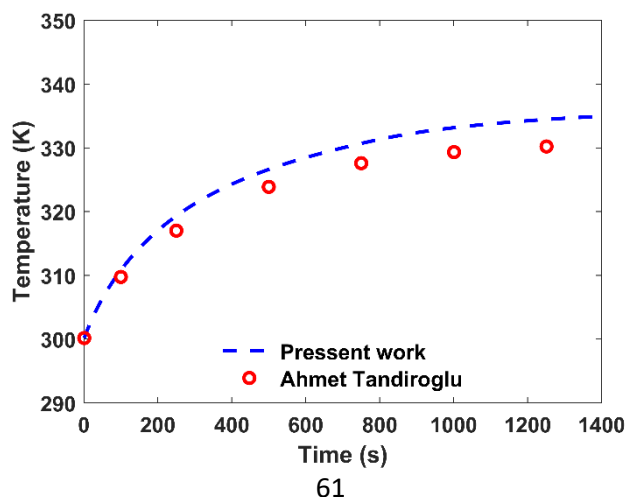




Figure 3 The variation of Temperature against time Ahmet Tandiroglu [28]

Analytical Assessment of the Findings

Temperature and turbulent kinetic energy contours

At a time, instance of $t = 5$ s, the thermal and turbulent characteristics of the flow are illustrated in Figures 4 and 5, where subfigures (a) and (b) correspond to mixed convection and forced convection conditions, respectively. The temperature fields shown in Figure 4 indicate a clear difference between the two cases, where the mixed convection configuration exhibits comparatively higher temperature levels throughout the tube cross-section. This behavior is mainly attributed to the presence of buoyancy forces generated due to temperature-induced density variations. As the fluid near the heated wall gains thermal energy, its density decreases, causing it to rise and interact with the main axial flow. This interaction enhances fluid motion and strengthens the overall heat transport process, resulting in a more pronounced thermal distribution when compared to the forced convection case where buoyancy effects are absent. The observed thermal enhancement is

consistent with previously reported studies in the literature [15].

A similar trend can be identified in the turbulent kinetic energy distributions presented in Figure 5. In both cases, the turbulent kinetic energy gradually increases in regions where velocity gradients are high, particularly near the wall and within the core flow affected by the twisted tape structure. However, the mixed convection case shows noticeably higher levels of turbulent kinetic energy compared to the forced convection scenario. This increase is caused by the additional momentum fluctuations introduced by buoyancy-driven motion, which intensifies flow instability and strengthens turbulent structures within the tube. The combined effect of forced flow and natural convection leads to more active mixing and enhanced turbulence generation. Consequently, both heat transfer and momentum transport are improved under mixed convection conditions. These findings are in good agreement with general observations reported in previous studies [15, 23, 29], confirming the physical consistency of the present results.

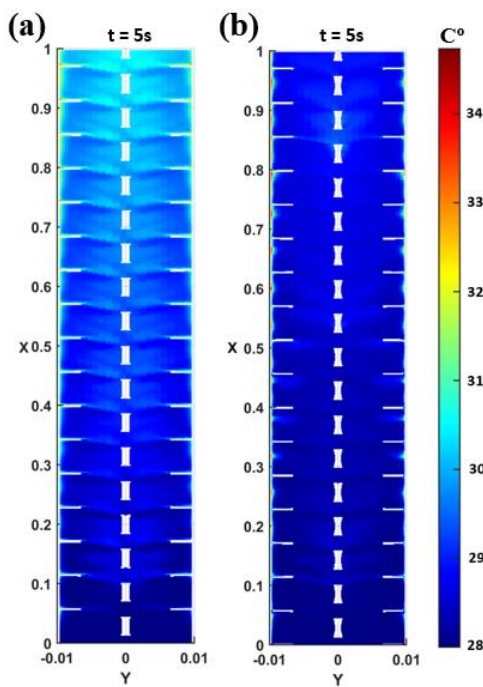


Figure 4 Temperature distribution of (a) mixed (b) mixed

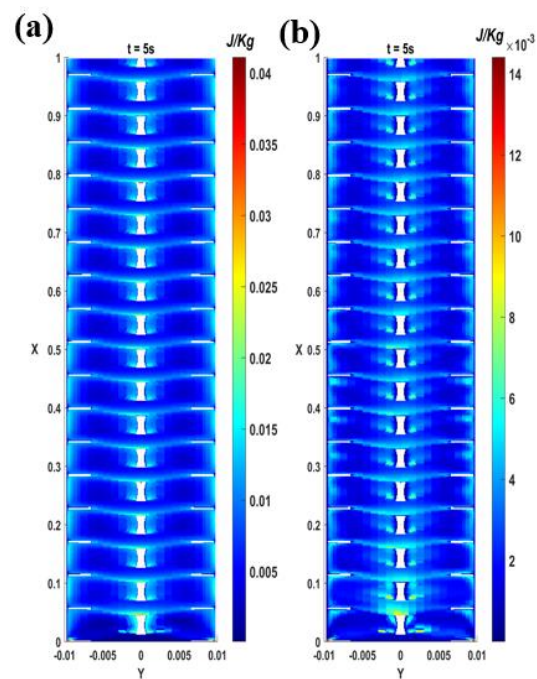


Figure 5 Turbulent kinetic energy distribution of (a) mixed convection (b) mixed

Outlet temperature and Transient heat transfer

Figure 6 presents a comparison between forced and combined (mixed) convection regimes in terms of the outlet temperature of a turbulent $\text{Al}_2\text{O}_3\text{-Cu}$ /water hybrid nanofluid. The results indicate that incorporating buoyancy-driven effects leads to a noticeable decrease in the exit temperature, with a reduction of approximately 9.3% relative to purely forced convection. This behavior can be explained by the interaction between inertial and buoyancy forces. Under constant mass

flow rate, increasing the residence time enhances buoyancy influences; however, the elevated dynamic viscosity of the hybrid suspension suppresses secondary flow structures. Consequently, the expected enhancement from natural convection is partially offset. These observations highlight that heat transfer performance cannot be interpreted solely based on effective thermal conductivity. Instead, additional thermophysical properties including density, specific heat capacity, and thermal expansion coefficient play a critical role in governing thermal transport under mixed convection conditions [15, 24].



Figure 7 shows the temporal evolution of temperature fields obtained using STAR-CCM+ for the same working fluid over a duration of 5s. Panel (a) illustrates temperature variations at selected axial locations along the tube centerline, while panel (b) reports the outlet temperature history. In both cases, a rapid temperature increase is observed during the initial stages, which is associated with intensified mixing induced by the twisted tape insert that disrupts the thermal boundary layer. The centerline temperature exhibits an approximately linear growth trend with time, accompanied by slight early-time fluctuations attributable to fine grid resolution effects [25].

The maximum centerline temperature reaches 32.8 °C for forced convection and 30.3 °C for mixed convection. A similar pattern is observed at the outlet, where peak values of 32.4 °C and 29.4 °C are recorded for forced and mixed convection, respectively. The lower temperatures in the mixed convection regime are linked to buoyancy-induced acceleration, which promotes fluid motion and enhances convective heat removal. Overall, the inclusion of buoyancy effects reduces the temperature rise by about 9.3% compared with the forced convection case, confirming its measurable influence on system thermal behaviour.

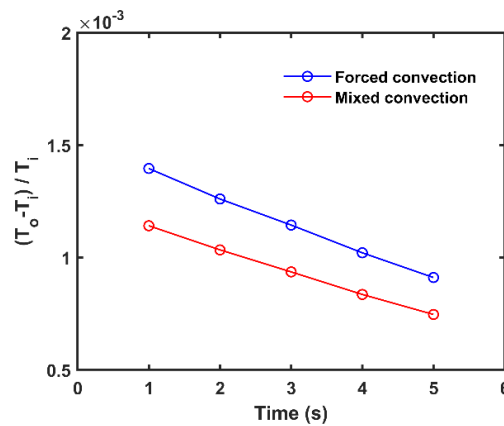


Figure 6 Average output temperature vs time

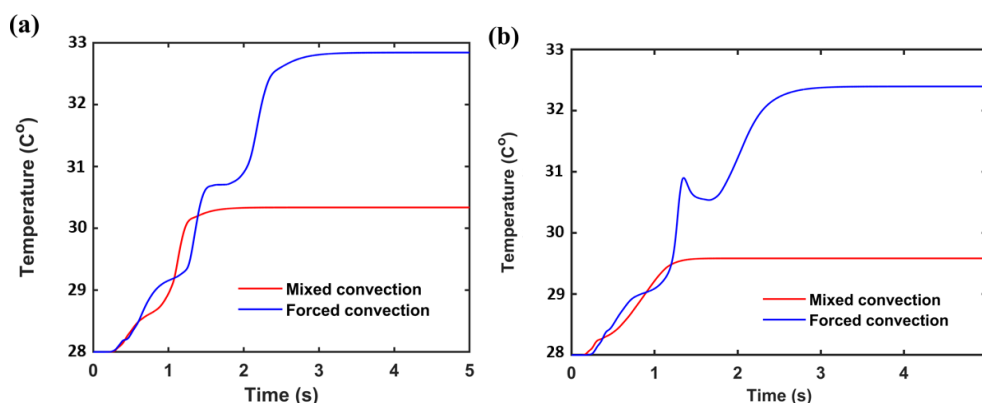


Figure 7 Curves of temperatures at (a) centre point (b) outlet.

Evaluation of Irreversibility and Entropy Generation

This section examines entropy generation during transient heat transfer in an Al_2O_3 –Cu/water hybrid nanofluid at a volume fraction of 2% within a TECT. Both total entropy production and the Bejan number are evaluated under forced and mixed convection conditions. Irreversibility arises from heat transfer and viscous effects [26], making entropy generation a key indicator of thermodynamic losses. Its spatial distribution provides insight into regions of dominant dissipation within the flow domain. The primary objective is to quantify the influence of mixed convection in a duct equipped with elliptical-cut and solid twisted tape inserts on entropy generation, considering only thermal and frictional contributions.

Local entropy production

Figures 8 and 9 present the spatial distribution of local entropy generation at $t=5$ s, separating the contributions arising from heat transfer and viscous effects under both mixed and forced convection conditions. A clear distinction is observed between the two flow regimes. The entropy generation associated with thermal transport is noticeably reduced when buoyancy effects are present. This behavior is linked

to the additional fluid motion induced by density variations, which promotes mixing and smooths temperature differences throughout the domain. As a result, the intensity of thermal gradients decreases, leading to a lower rate of thermally driven irreversibility compared with the purely forced case.

On the other hand, the entropy generation caused by fluid friction exhibits strong localization. The highest values are concentrated around the elliptical-cut twisted tape, particularly near its central region. This is due to the rotational motion imposed by the insert, which produces pronounced velocity variations and enhances shear within the fluid. These steep velocity gradients significantly increase viscous dissipation, making this zone the primary source of friction-related entropy generation. Although buoyancy modifies the global flow structure, its influence on viscous irreversibility remains secondary compared with the geometric effect of the twisted tape. Overall, the results indicate that mixed convection effectively suppresses entropy generation linked to heat transfer, while the insert-driven flow structure governs the distribution and magnitude of frictional losses [23].

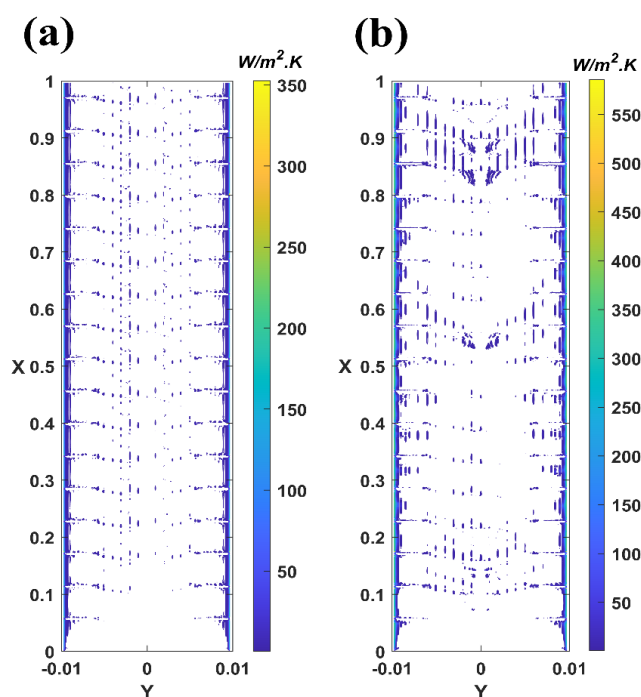


Figure 8 Thermal entropy generation of (a) mixed (b) forced convection at $t = 5s$.

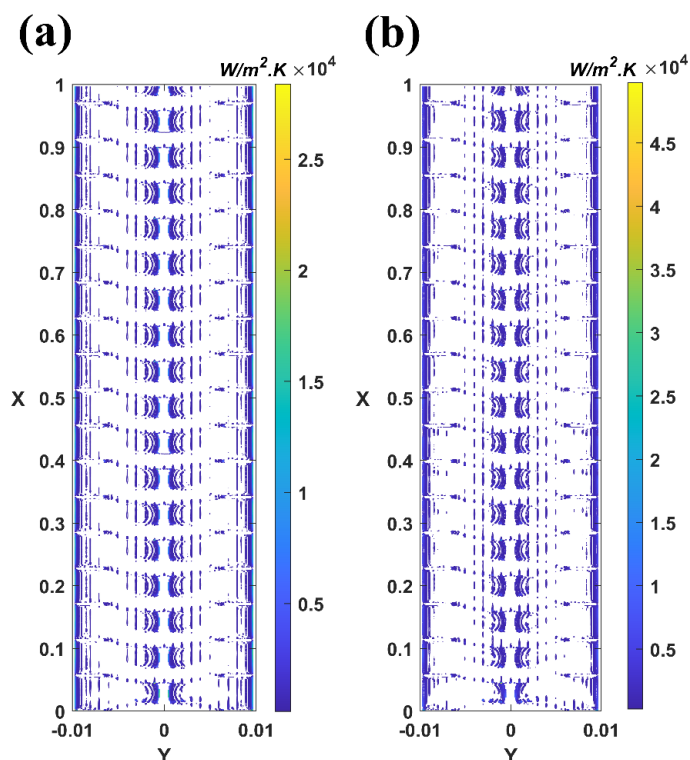


Figure 9 Frictional entropy generation of (a) mixed (b) forced convection at $t=5s$.



the modification of the flow structure caused by buoyancy forces. The additional motion induced by density variations tends to reduce temperature differences, thereby weakening thermal irreversibility. At the same time, this effect enhances flow complexity and intensifies local velocity gradients, particularly in the vicinity of the twisted tape. These stronger gradients lead to increased viscous dissipation, which shifts the balance of entropy generation toward frictional mechanisms and results in lower Bejan number values.

In contrast, the forced convection regime maintains higher Bejan numbers due to the persistence of stronger temperature gradients and comparatively lower viscous contributions. Overall, the observed trends indicate that the inclusion of buoyancy alters not only the magnitude but also the distribution of irreversibility, reducing the dominance of thermal effects and promoting a greater role of fluid friction throughout the domain [27].

Local Bejan number

Figure 10 presents the contours of the local Bejan number at $t=5$ s for (a) mixed convection and (b) forced convection conditions. This dimensionless parameter reflects the proportion of entropy generation attributed to heat transfer relative to the total irreversibility within the system. Values close to unity indicate that thermal effects are predominant, whereas lower values signify a stronger influence of viscous dissipation.

The results show that a significant portion of the flow field is characterized by Bejan numbers between 0.8 and 1.0, suggesting that heat transfer remains the main contributor to entropy production across most regions of the pipe. Nevertheless, a noticeable reduction in Bejan number is observed when buoyancy is present. In the mixed convection case, the values are consistently lower than those obtained under forced convection, indicating a relative increase in the contribution of frictional effects. This behavior can be attributed to

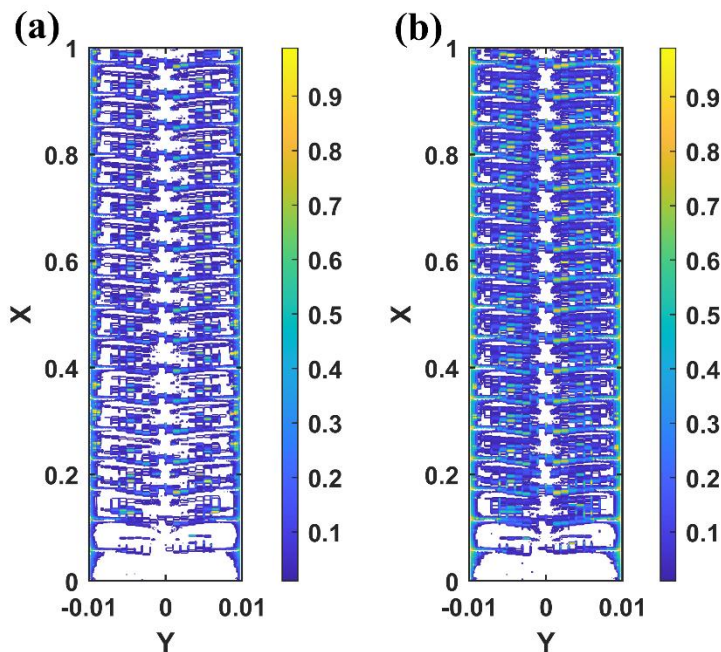


Figure 10 Bejan number of (a) mixed (b) forced convection at $t=5$ s.



Total entropy production

The total entropy generation in the system is determined by the combined effects of heat transfer and viscous dissipation, both of which represent sources of thermodynamic irreversibility. Figure 11 compares the temporal evolution of total entropy production for mixed and forced convection, providing insight into the role of buoyancy during transient heat transfer.

At very early times, the difference between the two convection modes is negligible. This is because the thermal and flow fields are still developing, and buoyancy forces have not yet become influential. Under these conditions, fluid motion is primarily governed by the imposed flow, resulting in similar temperature gradients and velocity distributions for both cases. Consequently, the total entropy generation remains nearly identical.

As time increases, buoyancy effects begin to influence the flow in the mixed convection regime. Density variations induced by temperature differences generate additional fluid motion, which enhances mixing and improves the redistribution of thermal energy. This process reduces temperature gradients, particularly near heated regions, leading to a decrease in entropy generation associated with heat transfer. In contrast,

forced convection lacks this mechanism, allowing stronger temperature gradients to persist and thereby increasing overall irreversibility.

Between approximately 2 s and 5 s, the entropy generation curves stabilize, indicating that the system has reached a quasi-steady condition. Although both cases exhibit similar trends during this period, the magnitude of entropy production remains consistently lower for mixed convection. This reduction is mainly attributed to the weakening of thermal gradients, which dominate entropy generation in such systems.

The most significant outcome is the substantial decrease in total entropy generation observed in the mixed convection case, with a maximum reduction of about 58.2% compared to forced convection. This behavior is consistent with previously reported findings in the literature [15], which also demonstrated that buoyancy-assisted flow can effectively reduce thermodynamic irreversibility by improving heat transfer distribution. Overall, the results confirm that incorporating buoyancy effects not only alters flow behavior but also significantly enhances system efficiency by limiting entropy production.

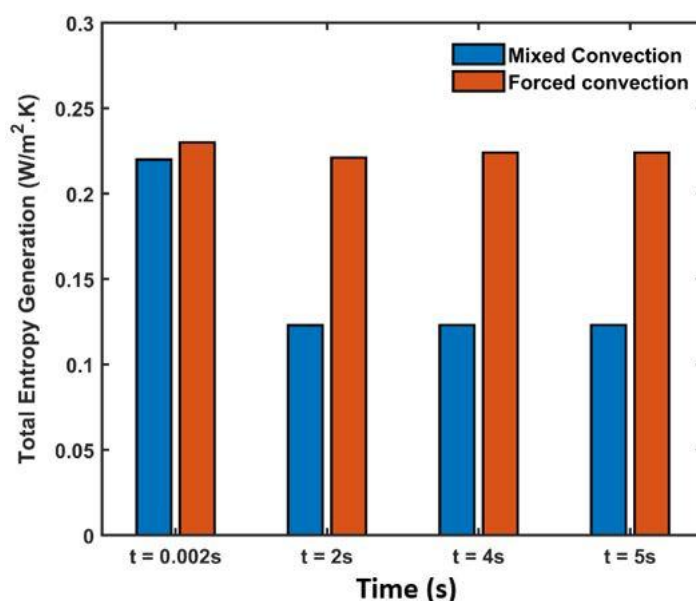


Figure 11 Total entropy generation of TECT at different times.

CONCLUSIONS

A computational investigation was conducted to examine the enhancement of transient thermal performance in a vertical tube fitted with elliptical-cut twisted tape (TECT) inserts under mixed convection conditions using an $\text{Al}_2\text{O}_3\text{-Cu}$ /water hybrid nanofluid. The analysis was performed using the STAR-CCM+ platform to evaluate the influence of flow structure development and buoyancy effects on the thermal behavior of the receiver tube. The principal findings are summarized as follows:

- The transient response of the system shows that both velocity and temperature fields gradually evolve toward a stable condition as simulation time progresses, indicating the development of fully established flow structures.
- The presence of the elliptical-cut twisted tape generates strong swirling motion and induces secondary flow patterns near the wall region. This behavior enhances

fluid mixing and increases the intensity of temperature gradients in the vicinity of the heated surface, which contributes to improved convective heat transfer.

- The incorporation of buoyancy forces under mixed convection conditions produces a measurable improvement in thermal performance. Compared with forced convection, the circumferential temperature variation is reduced by approximately 9.3%, indicating a more uniform thermal field and improved energy distribution.
- Entropy generation analysis reveals that thermal irreversibility is the dominant component across the system, while frictional effects remain secondary but locally significant near the twisted tape due to elevated velocity gradients.
- The evaluation of total entropy generation demonstrates that mixed convection consistently yields lower irreversibility than forced



convection, with a maximum reduction of about 58.2%. This confirms the beneficial role of buoyancy in enhancing thermodynamic efficiency by reducing temperature gradients.

- The Bejan number distribution further supports these findings, showing that heat transfer remains the primary source of entropy production across most of the domain, although localized regions of increased frictional influence appear near the swirl-inducing insert.

Overall, the results indicate that combining hybrid nanofluids with swirl-generating geometries significantly improves thermal performance in vertical tube systems. The inclusion of buoyancy effects not only enhances heat transfer but also reduces total entropy generation, leading to improved thermodynamic efficiency and more uniform temperature distribution in solar receiver applications operating under mixed convection conditions.

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Dr. Amir Khfagi completed his first degree in Mechanical Engineering in 2003 at the Higher Institute of Science and Technology, Ajdabiya, Libya, followed by a master's degree in advanced mechanical engineering at Sheffield Hallam University in 2015. In 2019, he joined the University of Glasgow to pursue a PhD in Mechanical Engineering, which he successfully completed in 2023 with a thesis entitled Investigation of

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