In this paper, a numerical study is performed to investigate the effect of Reynolds number and nanoparticle concentration on the thermal performance of single-walled carbon nanotubes (SWCNTs) nanofluids flowing through a straight pipe with constant heat flux in a turbulent flow regime. The governing equations (continuity, momentum, energy, rate of turbulent production, and rate of turbulent dissipation) are solved using a computational fluid dynamic approach (i.e., finite volume method). In the sensitivity analysis, the Reynolds number is varied between \((10\,000-200\,000)\) and nanoparticle concentration between \((0\%-0.25\%)\). The simulation results show that convective heat transfer (average Nusselt number) increases by \(7.48\%\) while the pressure drop and pumping power increase by \(119\%\) and \(199\%\), respectively. In addition, at low Reynolds number \((10^4)\), increase in nanoparticle concentration \((0\%-0.25\%)\) decreases entropy production rate by \(16.95\%\). However, at higher Reynolds number \((2\times10^5)\), the entropy production rate increases by \(149.77\%\). Furthermore, other results such as entropy generation number, Bejan number, second thermodynamic law efficiency, and thermal performance factor, which combine the first and second law of thermodynamics, are presented as functions of Reynolds number and nanoparticle's concentration. The overall result shows that SWCNT nanofluid will only be beneficial at low Reynolds number of \((10\,000-50\,000)\).

**KEYWORDS**
nanofluid, single-walled carbon nanotube, thermal performance, turbulent flow

1 INTRODUCTION

The development of energy-efficient heat transfer fluids has numerous limitations due to their low thermal conductivity. To overcome this challenge, a new kind of fluid has been proposed; the so-called nanofluid.\(^1\)\(^2\) Nanofluids are a class of heat transfer fluids created by the suspension of small amounts of nanoscale metallic, nonmetallic, or polymetallic particles in a base fluid (water, glycerine oil, ethylene glycol (EG)) having high thermal conductivity. They find applications in various engineering fields such as transportation, biomedical, nuclear energy, and electronic cooling.\(^3\)\(^-\)\(^5\)
The thermal performance of common nanofluids such as Al₂O₃/H₂O, TiO₂/H₂O, MgO₂/H₂O, Cu/H₂O, Al/H₂O, and CuO/EG among others have been investigated by various researchers and were found to have better convective heat transfer properties as compared to base fluids. However, in recent times, a new type of nanoparticle called carbon-based nanostructures (single-walled nanotube, double walled nanotube, and multi-walled nanotube) has attracted great attention due to their improved properties such as high thermal conductivity, large specific surface area, high aspect ratio, and low specific.

A number of authors have investigated the heat transfer performance of carbon nanotubes. Among these, Estellé et al experimentally studied the thermal performance of multi-walled carbon nanotube nanofluids (MWCNT/H₂O and MWCNT/H₂O-EG) flowing through a coaxial heat exchanger tube in laminar flow regime. The results obtained show that the heat transfer performance increases as the concentration of the carbon nanotube increases. Furthermore, the convective heat transfer coefficient of MWCNT/H₂O-EG (EG) is higher than that of MWCNT/H₂O-EG. Baskar et al experimentally studied the effect of nanoparticle concentration and Reynolds number on heat transfer property of multi-walled carbon nanotube (MWCNT) dispersed in a mixture of water and EG. In the cited work, various concentrations of nanofluids were passed through a horizontal copper test rig under laminar flow conditions. The authors reported 30% and 34% enhancement in convective heat transfer for 0.15% and 0.2% volume ratios, respectively. Shanbedi et al experimentally studied the effect of three input variables namely: nanoparticles concentration, flow rate, and magnetic field on the heat transfer performance of MWCNTs dispersed in water and flowing through a straight horizontal circular pipe under the influence of a magnetic field. The experiment was optimized using the response surface methodology (RSM). The optimized results for the three variables (volume fraction, flow rate, and magnetic flux) are given as 0.09%, 301.3 mL/min, and 400 G, respectively. Experiment was conducted with the aforementioned values and the results obtained confirmed the ones predicted by the RSM.

Nikkhah et al numerically studied the effect of nanoparticle volume fraction, slip coefficient, and Reynolds number on flow field and temperature field of functionalized multi-walled carbon nanofluid (FMWCNT)/H₂O flowing through a microchannel under oscillating heat flux and slip boundary conditions. They reported an increment in the average Nusselt number as nanoparticle concentration and slip coefficient increase. Bahiraei et al numerically studied the effect of nanoparticle concentration and Reynolds number on thermal and hydraulic properties of eco-friendly graphene nanofluid flowing in a counter-current spiral heat exchanger. They showed that the convective heat transfer increases as nanoparticle concentration increases but is accompanied with a high-pressure drop. Mamourian et al investigated the effect of Reynolds number, nanoparticle volume fraction, and nanoparticle size on the average Nusselt number using numerical approach and RSM. They reported that all the three input parameters affect the average Nusselt number. Bahmani et al studied the effect of nanoparticle concentration, flow direction, and Reynolds number on Al₂O₃/H₂O nanofluid flowing in a parallel and counter-flow double-pipe heat exchanger. They observed that the convective heat transfer coefficient increases while outlet temperature decreases as the concentration of the nanoparticle increases. Furthermore, the outlet temperature of the counter flow was found to be lower than that of the parallel flow. Rahimi et al numerically studied the effect of nanoparticle size and concentration of Carboxymethyl cellulose (CMC)/copper oxide (CuO) through a microtube in turbulent flow regime. The results show enhancement in convective heat transfer as nanoparticle size decreases while nanoparticle concentration increases. Safaei et al numerically studied the effect of Raleigh Number and emissivity value of Al₂O₃/H₂O nanofluid on heat transfer composed of radiation and free convection in two-dimensional (2D) shallow cavity using Boltzmann method. The result showed that both parameters (Rayleigh number and emissivity) enhance the overall heat transfer. Alrashed et al numerically studied the heat transfer of FMWCNT/H₂O in a 2D backwards-facing abrupt contacting channel. The study considers the effect of nanoparticle concentration and Reynolds number on the overall flow and thermal fields. The results show that an increase in Reynolds number and nanoparticle concentration reduces the surface temperature of the channel and enhances the convective heat transfer.

Rezaei et al studied the effect of Reynolds number, geometry, number of ribs, and nanoparticle concentration on hydrodynamic and heat transfer of Al₂O₃-water flowing through a three-dimensional microchannel with triangular section in laminar flow regime. The result revealed that both nanoparticle concentration and presence of the ribs enhance convective heat transfer and the maximum number of ribs for optimum performance is five. Shamsi et al simulated heat transfer in non-Newtonian fluid (water and 0.5 wt% CMC) flowing through a microchannel with triangular ribs. The study examined the effect of angle of attack (30°, 45°, and 60°) of the triangular rib, nanoparticle concentration, nanoparticle size, and Reynolds number on average Nusselt number. The result showed that increase in the concentration of the nanofluid enhances the average Nusselt and the triangular rib with the angle of attack equal to 30° gave the highest heat transfer enhancement and lowest pressure drop among the three. Kaska et al studied the hydrodynamic and heat transfer of hybrid nanofluid (AlN-Al₂O₃ –water) flowing through a straight pipe in the laminar flow regime.
The effect of Reynolds number and nanoparticle volume fraction were investigated on average Nusselt number, pressure drop, and performance evaluation factor. The result revealed that hybrid nanofluid enhances heat transfer significantly. Sözen et al.\textsuperscript{32} examined the effect of nanoparticle concentration and Reynolds number on the heat transfer performance of TiO\textsubscript{2}-water nanofluid flowing through a flat plate heat exchanger using both experimental and numerical approaches. The results obtained from the two were compared and found to be in agreement. Furthermore, the author reported that the thermal performance of nanofluid increases as nanoparticle concentration increases.

Salari et al.\textsuperscript{25} experimentally studied flow boiling heat transfer (combined effect of forced convection and nucleate boiling heat transfer) of Al\textsubscript{2}O\textsubscript{3}-water nanofluid. The study considered the effect of heat flux, nanoparticle size, and time on the boiling heat transfer coefficient. The author reported that for short-time study (0-60 min), the presence of nanoparticle enhances boiling heat transfer coefficient. However, for the longer time (60-1000 min), the heat transfer coefficient began to decrease as time increases and this was attributed to surface roughness, static contact angle, and thermal fouling resistance parameter. Arya et al.\textsuperscript{36} experimentally studied the effect of Reynolds number and nanoparticle concentration on the thermal performance of MgO/EG nanofluid flowing through a countercurrent corrugated plate heat exchanger in both laminar and turbulent flow regime. The author reported an increase in convective heat transfer and pressure drop of 35\% and 85\%, respectively. Furthermore, heat performance index factor shows higher value in the turbulent flow regime than laminar. Kouloulias et al.\textsuperscript{27} experimentally studied hydrodynamic and heat transfer of Al\textsubscript{2}O\textsubscript{3}/H\textsubscript{2}O nanofluid under subcooled pool boiling condition. The effect of nanoparticle on bubble dynamic and nucleation site on the heated surface was presented. The author reported that nanoparticle alters the nucleation site density, bubble size, and the frequency of the bubble generation from the heated surface. Gobinath and Venugopal\textsuperscript{28} experimentally studied the effect of heat flux and nanoparticle volume fraction on nucleate pool boiling of CuO-isobutane refrigerant nanofluid. They reported that an increase in nanoparticle and heat flux enhances the nucleate boiling heat transfer. Ji et al.\textsuperscript{29} experimentally investigated pool boiling heat transfer. They considered the effect of surface roughness (3.9-6 μm), contact angle (47°-157°), and heat flux (30-300 kW) on boiling heat transfer coefficient of SiO\textsubscript{2}/H\textsubscript{2}O nanofluid. The result revealed that at higher roughness, hydrophilicity reduces the boiling heat transfer coefficient while hydrophobic enhances it. Fadodun et al.\textsuperscript{30} investigated the thermal efficiency and pumping power of nanofluid flowing through a straight pipe in turbulent flow regime using computational fluid dynamic (CFD) and RSM. The study considered the effect of Reynolds number, nanoparticle volume fraction, nanoparticle size, and entrance temperature on thermodynamic second law efficiency and pumping power. The authors reported that all the four input variables are statistically significant to both thermal efficiency and pumping power. In addition, the results show that more interactions were significant in pumping power than thermal efficiency. Furthermore, the sensitivity analysis carried out on the regression model show that Reynolds number is the most sensitive parameter to both the thermal efficiency and the loss in pumping power. Garoosi and Rashidi\textsuperscript{31} studied the mixed convection heat transfer of nanofluid in a 2D square enclosure using mixture model. The study considered the effect of Richardson number, nanoparticle volume ratio geometry (presence of conductive obstacle) and nanoparticle size on the flow field, and heat transfer of various nanofluids (Cu, Al\textsubscript{2}O\textsubscript{3}, and TiO\textsubscript{2}). The authors reported that at low Richardson number, addition of nanoparticle has a minor effect on the heat transferred. Furthermore, the presence of conductive wall reduces the heat transfer rate significantly. Umadevi and Nithyadevi\textsuperscript{32} studied the effect of thermal boundary conditions and nanoparticle volume ratio on the flow field and heat transfer of Ag/H\textsubscript{2}O nanofluid enclosed in a 2D square enclosure. The study showed that an increase in nanoparticle volume enhances convective heat transfer. Toghraie et al.\textsuperscript{33} investigated the flow field and heat transfer in water and nanofluid (CuO/H\textsubscript{2}O) flowing through various shapes (smooth, sinusoidal, and zigzag shape) of microchannels in laminar flow regime. They reported that an increase in nanoparticle volume ratio enhances heat transfer. However, the enhancement due to modification of geometry (use of sinusoidal and zigzag microchannels) was found to be greater than that of nanofluid.

Besides the enhancement in convective heat transfer obtained with the use of nanofluids, there was an accompanying high-pressure drop penalty due to an increase in viscosity. The increase in thermal conductivity leads to reduction in thermal entropy production whereas increase in viscosity increases viscous entropy production rate. The enhancement in convective heat transfer was due to increase in thermal conductivity and Reynolds number that reduces the entropy production due to thermal effect but on the other hand, an increase in entropy production due to viscous flow occurs. Therefore, it is imperative to optimize the thermal system to obtain a minimum entropy generation rate as proposed by Bejan in entropy generation minimization theorem.\textsuperscript{34,35}

Several works exist in literature on the evaluation of the performance of nanofluids from the perspective of thermodynamics second law (entropy production). For instance, Ji et al.\textsuperscript{36} studied the effect of nanoparticle volume ratio, Reynolds number and heat flux on entropy generation, and Bejan number of Al\textsubscript{2}O\textsubscript{3}/H\textsubscript{2}O nanofluid flowing through a circular pipe in turbulent flow regime under constant heat flux boundary conditions. Their results showed that the Bejan
number decreases as the Reynolds number and nanoparticle volume ratio increase. Furthermore, a decrease in nanoparticle volume ratio and an increase in heat flux enhance Reynolds number. Khairul et al studied the entropy production in Al$_2$O$_3$/H$_2$O and Al$_2$O$_3$/EG nanofluids. Their results showed that the thermal performance of the system increases when entropy production due to heat transfer is greater than entropy production due to fluid flow. Konchada et al studied the effect of entrance temperature and mass flow rate on the entropy generation of Al$_2$O$_3$/H$_2$O nanofluid flowing through a longitudinal thin heat exchanger using CFDs. Entropy analysis was carried out using Taguchi technique and the results showed that both inlet temperature and mass flow rate are statistically significant to the entropy generation but the effect of inlet temperature is the most profound between the two. Nakhchi and Esfahani numerically investigated the effect of Reynolds number, geometry, number of holes, and nanoparticle concentration on entropy production rate in Cu/H$_2$O nanofluid flowing through a heat exchanger equipped with a perforated conical ring. The results revealed that the thermal entropy production is dominant for the range of Reynolds number considered. However, it decreases as nanoparticle concentration increases. Furthermore, the viscous entropy production decreases as the number of holes in the conical ring increases. Berrehal and Maougal analytically studied the effect of nanoparticle concentration and Reynolds number on entropy production in Al$_2$O$_3$/H$_2$O and Al$_2$O$_3$/EG nanofluid in both laminar and turbulent flow regime. The results obtained showed that the addition of nanoparticle decreases the entropy production of Al$_2$O$_3$/H$_2$O nanofluid for Reynolds number less than 40,000. Lastly, Amirahmadi et al studied the effect of design parameter on lamina convective heat transfer and exergy analysis of fluid flowing through a trapezoidal duct using a numerical approach. The effects of the beveled corner and surface roughness on the entropy generation are compared. The results show that both designs reduce entropy production in the trapezoidal duct but vortex generator (surface roughness) performs better as it reduces entropy production by 26.6% whereas the beveled corner reduced by 8.1%.

A single-walled carbon nanotube (SWCNT) is hollow, cylindrical in shape, and made of one atomic sheet of carbon arranged in a honeycomb lattice. It has a high aspect ratio like others (DWCNT and MWCNT). However, its thermal conductivity and optical properties are higher compared to DWCNT and MWCNT, and this justifies its choice as the potential nanoparticle of study. There are limited works in literature that studied heat transfer performance of SWCNT nanofluid. The ones available mainly focus on investigating the enhancement of convective heat transfer. However, to the best of our knowledge, there is no work that studied the rate of entropy production in SWCNT nanofluid, which is known to affect the efficiency of the thermal system. Furthermore, majority of the works on heat transfer performance in SWCNT nanofluid use empirical correlation to estimate the thermophysical properties (density, thermal conductivity, specific heat capacity, and viscosity) of the nanofluid, which either underestimate or overestimate the actual value. Thus, this work carried out a comprehensive study of thermal performance of SWCNT nanofluid flowing through a straight pipe in the turbulent flow regime. We investigated the effect of Reynolds number and nanoparticle concentration on average Nusselt number, pumping power (PP), Bejan number, entropy production rate, second thermodynamic law efficiency, entropy generation number, and thermal performance factor derived from the first and second laws of thermodynamics using CFD approach and experimental data of thermophysical properties of SWCNT nanofluid. This work presents variation of average Nusselt number, PP, entropy production rate, Bejan number, entropy production rate and second thermodynamic law efficiency as a function of Reynolds number and nanoparticle concentration. Furthermore, the optimum flow condition for the thermal performance of SWCNT nanofluid is determined.

The rest of this paper is organized as follows. Section 2 describes the statement of the problem and the theoretical background of the study. Section 3 provides the numerical simulation and validation approach, whereas Section 4 presents the results and discussion. Finally, the paper is concluded in Section 5.

## 2 | PROBLEM STATEMENT

The problem under investigation involves the steady state turbulent forced convection of SWCNT/H$_2$O nanofluid flowing through a horizontal pipe as shown in Figure 1. The geometry consist of a cylinder of a diameter of 0.02 m and a length of 2 m. The fluid enters the pipe with uniform velocity and temperature. A constant heat flux of 1000 W is applied to the wall of the pipe and zero pressure gauge is imposed at the outlet. The flow and temperature fields are assumed to be symmetrical about the tube’s main axis. Thus, a 2D axisymmetric formulation is employed to simplify the problem.

### 2.1 | Single-phase model

This model assumes that nanoparticles dispersed in the base fluid are fluidized and thermal equilibrium exists between the nanoparticle and the base fluid. Thus, the resulting mixture is treated as a single-phase fluid. As a result of this,
the governing equations (continuity, momentum, and energy) applied to the classical Newtonian fluid are also valid for this model. Assuming the nanofluid is incompressible and neglecting both viscous and compressive work in the energy equation. The hydrodynamic and heat transfer equations take the forms of Equations (1), (2), (3), and (4).44

\[ \nabla \bar{u} = \frac{\partial u_x}{\partial x} + \frac{\partial u_r}{\partial r} + \frac{u_r}{r} = 0 \]  

(1)

\[ u_x \frac{\partial}{\partial x} (u_x) + u_r \frac{\partial}{\partial r} (u_x) = -\frac{1}{\rho_{nf}} \frac{\partial p}{\partial x} + \frac{1}{\rho_{nf}} \left( \frac{1}{r} \frac{\partial}{\partial r} \left( r \left( \mu_{nf} + \mu_t \right) \frac{\partial u_x}{\partial r} \right) \right) + \frac{1}{\rho_{nf}} \frac{\partial}{\partial x} \left( \left( \mu_{nf} + \mu_t \right) \frac{\partial u_x}{\partial x} \right) \]  

(2)

\[ u_r \frac{\partial}{\partial r} (u_r) + u_x \frac{\partial}{\partial x} (u_r) = -\frac{1}{\rho_{nf}} \frac{\partial p}{\partial r} + \frac{1}{\rho_{nf}} \left( \frac{1}{r} \frac{\partial}{\partial r} \left( r \left( \mu_{nf} + \mu_t \right) \frac{\partial u_r}{\partial r} \right) \right) \]  

(3)

\[ u_x \frac{\partial T_{nf}}{\partial x} + u_r \frac{\partial T_{nf}}{\partial r} = \left[ \frac{1}{r} \frac{\partial}{\partial r} \left( r \left( \alpha_{nf} + \alpha_t \right) \frac{\partial T_{nf}}{\partial r} \right) + \frac{\partial}{\partial x} \left( \alpha_{nf} + \alpha_t \right) \frac{\partial T_{nf}}{\partial x} \right] . \]  

(4)

where \( \mu_t, \mu_{nf}, \alpha_{nf}, \) and \( \alpha_t \) are the turbulence viscosity, the viscosity of the nanofluid, the thermal diffusivity of the nanofluid, and the turbulence diffusivity, respectively. The turbulence viscosity and diffusivity are given by Equations (5), (6), and (7).45

Using the Standard \( k - \varepsilon \) turbulent model, the equations for turbulent kinetic energy \( (k) \) and dissipation rate \( (\varepsilon) \) are given by Equation (8a) and (8b).46 \( \sigma_k, \sigma_\varepsilon \) are the turbulent Prandtl number for \( k \) and \( \varepsilon \), respectively, whereas \( G_k \) represents the rate of production of turbulent kinetic energy as given in Equation (9).47 Where \( S = \sqrt{2S_{ij}S_{ij}} \) is the modulus of the rate of strain tensor.

\[ \mu_t = C_\mu \frac{k^2}{\varepsilon} \]  

(5)

\[ C_\mu = \frac{1}{4.04A_s \left( k\bar{U}/\varepsilon \right)} \]  

(6)

\[ \sigma_k = \frac{\mu_t}{\rho_{nf} \sigma_t} \]  

(7)

\[ \frac{1}{r} u_r \frac{\partial}{\partial r} (rk) + u_x \frac{\partial k}{\partial x} = \frac{1}{\rho_{nf}} \left[ \left( \left[ \frac{1}{r} \frac{\partial}{\partial r} \left( r \left( \mu_{nf} + \mu_t \sigma_k \right) \frac{\partial k}{\partial r} \right) \right] \right) \right] + \frac{1}{\rho_{nf}} \frac{\partial}{\partial x} \left[ \left( \left( \mu_{nf} + \mu_t \sigma_k \right) \frac{\partial k}{\partial x} \right) \right] \]  

+ \( G_k \) \( \rho k \) \( \frac{\partial k}{\partial x} \( \) \( \frac{\partial k}{\partial x} \) \]  

(8a)

\[ \frac{1}{r} u_r \frac{\partial}{\partial r} (r\varepsilon) + u_x \frac{\partial \varepsilon}{\partial x} = \frac{1}{\rho} \left[ \left( \left[ \frac{1}{r} \frac{\partial}{\partial r} \left( r \left( \mu_{nf} + \mu_t \sigma_\varepsilon \right) \frac{\partial \varepsilon}{\partial r} \right) \right] \right) \right] + \frac{1}{\rho_{nf}} \frac{\partial}{\partial x} \left[ \left( \left( \mu_{nf} + \mu_t \sigma_\varepsilon \right) \frac{\partial \varepsilon}{\partial x} \right) \right] \]  

+ \( G_k \) \( C_\varepsilon \frac{\varepsilon^2}{k} \) \( \frac{\partial \varepsilon}{\partial x} \) \( \) \( \frac{\partial \varepsilon}{\partial x} \) \]  

(8b)

\[ G_k = \mu_t S^2 = -u_i' u_j' \frac{\partial u_i}{\partial x_j} \]  

(9)

\[ I = 0.16Re^{-1}. \]  

(10)

The values of the constants used in the turbulence model are presented in Table 1.35 The inlet turbulence intensity \( (I) \) is calculated using Equation (10).
### 2.2 Boundary conditions

The inlet and outlet boundary conditions are given as follows:\cite{48}:

**Inlet:**

\[ u_x = u_{in}, \quad u_r = 0, \quad T = 230K, \quad k = \frac{3}{2} \left( \frac{I_u}{u} \right)^2, \quad \varepsilon = \frac{c_{\mu}^{0.75} k^{1.5}}{L} \tag{11} \]

where \( \bar{u} \) is the velocity scale, which is equivalent to inlet velocity \( u_{in} \); \( L \) is the turbulence length scale, which is taken as the diameter of the pipe; and \( I \) is the turbulence intensity given by Equation (10).

**Outlet:**

\( (v_x, v_r, k, \varepsilon, T) \) are assumed to be fully developed while the atmospheric pressure is taken as 1 atm. The corresponding values of their derivative are given in Equations (12), (13), and (14).

Table 1 shows the values of constants used in \( k - \varepsilon \) turbulent model.

**Table 1 Constants and their corresponding values**

| Constant | \( C_{1x} \) | \( C_{2x} \) | \( C_{\mu} \) | \( \sigma_k \) | \( \sigma_\varepsilon \) | \( \sigma_t \) | Von Karman |
|----------|--------------|--------------|--------------|---------------|----------------|--------------|-------------|
| Value    | 1.44         | 1.92         | 0.09         | 1             | 1.3            | 0.9          | 0.4         |

**Wall:** no-slip condition constraint

\[ v_x = 0, \quad v_r = 0, \quad q = 1000W/m^2. \tag{13} \]

**Symmetry:**

\[ \frac{\partial v_r}{\partial r} = 0, \quad \frac{\partial T}{\partial r} = 0. \tag{14} \]

### 2.3 Wall treatment

At the region near the tube wall, there is a high fluctuation of the turbulent quantities \( (\bar{u}, u', k, \varepsilon) \), which are not captured by the \( k - \varepsilon \) model. As a result of this, the empirical model’s wall function, which satisfies the physics of the flow, is applied to represent the condition near the wall. The law of the wall states that the mean velocity at a point \( p \) is proportional to the distance \( p \) from the wall.\cite{49} This is expressed mathematically as

\[ u^+ = \frac{1}{\kappa} \ln y^+ = \frac{1}{\kappa} \ln \left( \frac{\rho C_{\mu}^{0.25} k_p y_p}{\mu} \right). \tag{15} \]

where \( k_p \) is the turbulent kinetic energy at an arbitrary point \( p \), \( y_p \) is the distance from an arbitrary point \( p \) to the wall, \( \rho \) is the density of the fluid, \( \mu \) is the viscosity of the fluid, \( C_{\mu} \) is the turbulent constant, and \( \kappa \) is the Von Karman constant. For \( k - \varepsilon \) model, the range of \( y^+ \) is given as 30 \( \leq y^+ \leq 300. \)

### 2.4 Hydrodynamics

The Reynolds number, Darcy friction factor, and PP are defined as Equations (16), (17), and (19), respectively.

\[ Re = \frac{\rho D_h u_{\text{mean}}}{\mu}. \tag{16} \]

The Darcy friction factor for turbulent flow is given as

\[ f = \frac{2D_h \Delta p}{\rho u_{\text{mean}}^2 H}. \tag{17} \]

\( D_h \) in Equations (17) and (18) is the diameter of the pipe, whereas \( H \) is the length of the pipe.
The mean velocity $u_{\text{mean}}$ is defined as

$$u_{\text{mean}} = \frac{2}{R^2} \int_0^R u(x, r) r \, dr \quad (18)$$

Pumping power (PP) is given by

$$\frac{\dot{m} \Delta p}{\rho_{nf}} = \frac{\pi}{8} \frac{f}{D_h} \frac{\mu^3 \text{Re}^5}{\rho^2}. \quad (19)$$

### 2.5 Heat transfer

The convective heat transfer coefficient is given as

$$h(x) = \frac{q}{T_{\text{wall}} - T_{\text{bulk}}} \quad (20)$$

The bulk temperature is given as

$$T_{\text{bulk}}(x) = T(x_0) + \frac{\dot{Q}}{m C_p} x \quad (21)$$

where $T(x_0)$ is the mean temperature at the point where the flow is fully developed. $x_0$ is taken as 70 $D_h = 1.4$ m from the entrance point and $q(W/m^2)$ is the heat flux on the surface of the pipe.

The local Nusselt number is evaluated as

$$Nu(x) = \frac{h(x) D}{k} \quad (22)$$

The mean Nusselt number is evaluated using Simpson’s (1/3) rule given as

$$Nu_{\text{ave}} = \frac{1}{x_{\text{out}} - x_{\text{in}}} \int_{1.4}^{2} Nu(x) \, dx \quad (23)$$

The entropy generation equation in closed form is given as

$$S_{\text{gen}} = \frac{\mu \Phi}{T_{nf}} + \frac{\rho E}{T_{nf}} + \frac{k}{T_{nf}} (\nabla T)^2 \quad (24)$$

$$\Phi = 2 \left[ \left( \frac{\partial v_x}{\partial x} \right)^2 + \left( \frac{\partial v_z}{\partial x} \right)^2 \right] + \left( \frac{\partial v_y}{\partial r} + \frac{\partial v_z}{\partial r} \right)^2 \quad (25)$$

The first term of Equation (24) denotes the entropy production due to viscous dissipation, the second term denotes entropy production due to turbulence flow, and the third term denotes entropy production due to thermal dissipation. This can also be calculated using analytical approach. Herein this work, we use the Bejan formula expression as given by Equation (26).

$$\dot{S}_{\text{gen}} = \frac{\pi D_h^2 q^2}{k_{nf} Nu_{\text{ave}} T_{\text{ave}}^2} + \frac{8 \dot{m}^3 f}{\rho_{nf}^2 \pi^2 D_h^5 T_{\text{ave}}}, \quad (26)$$

where all the terms have their usual meaning, the $T_{\text{ave}}(K)$ is the average temperature calculated using Equation (27).

$$T_{\text{ave}} = \frac{(T_{\text{in}} - T_{\text{out}})}{\ln \left( \frac{T_{\text{in}}}{T_{\text{out}}} \right)}. \quad (27)$$

Exergy is defined as the maximum useful work that can be extracted from a system as it comes into equilibrium with its environment. The exergy destruction is the irreversibility that occurs within the control volume and this is proportional to the rate of entropy production. The irreversibility is given as

$$I = T_{\text{in}} S_{\text{gen}} \quad (28)$$

Exergy analysis can be written as

$$W_{\text{act}} = W_{\text{max}} - I \quad (29)$$
where $W_{\text{act}}$ and $W_{\text{max}}$ denoted the actual input exergy into the system and maximum work that can be recovered from the system, respectively. The entropy generation number ($N_s$) represents the ratio of exergy destruction (destr.) in the system to input exergy.\(^{50}\) This is express as

$$N_s = \frac{\text{exergy destr.}}{\text{input heat}} = \frac{S_{\text{gen}} T_{\text{in}}}{m C_p \Delta T}.$$  \hspace{1cm} (30)

The thermodynamic second law efficiency $\eta_{II}$ of a thermal system is given as

$$\eta_{II} = \frac{W_{\text{max}}}{W_{\text{act}}} = 1 - N_s.$$  \hspace{1cm} (31)

Optimum design of the thermal system from entropy generation minimization theory requires the entropy production to be minimized while convective heat transfer is maximized. To establish the best configuration of a thermal system, a dimensionless parameter ($E_p$) called thermal performance factor is defined as the ratio of average Nusselt number to entropy generation number given by Equation (32).\(^{51}\)

$$E_p = \frac{N_u}{N_s}.$$  \hspace{1cm} (32)

As stated above, there are two terms that contribute to entropy production in a thermal system: the entropy production due to thermal effect and fluid flow, but often these two vary inversely to each other. For instance, as entropy due to thermal effect increases, the entropy due to fluid flow decreases and vice versa. The ratio of entropy production due to thermal flow to total entropy production is called Bejan number ($Be$). This nondimensional number is essential in thermal design. The Bejan number is given by Equation (33).

$$Be = \frac{S_{\text{Thermal}}}{S_{\text{Total}}}.$$  \hspace{1cm} (33)

3 | NUMERICAL SOLUTION AND VALIDATION

In this section, we present the numerical method used to solve and validate the model. The continuity, momentum, energy rate of production of turbulent kinetic energy ($k$), and rate of dissipation turbulent kinetic energy ($\varepsilon$) equations are solved with the boundary condition stated in Section 2.2 using finite volume software (ADINA CFD). The turbulent model is used since there is no flow separation, adverse pressure gradient, strong curvature, and jet flow in the model. The Semi-Implicit Method for Pressure Linked Equations (SIMPLE) algorithm was also used for pressure-velocity coupling and the second-order upwind scheme was applied to discretize convective term in the equations. Four node square meshes were used.\(^{52}\) The mesh size as shown in Figure 2 grows progressively from the wall along the radial direction to capture the rapid change in velocity and temperature at the boundary layer. The iterative procedure was terminated when the convergence criteria was satisfied (ie, the residual for continuity, momentum, energy, rate of turbulent production, and rate of turbulent dissipation are less than $10^{-6}$). Furthermore, grid sensitivity test was carried out as presented in Table 2 by varying the number of grids used in both axial and radial directions to subdivide the domains by using water at Reynolds number of 80 000 as a benchmark. Several combinations were used until a stable result ($N_r = 150$ by $N_c = 500$) was obtained: A further increase in the number of grids in the two direction would not change the value of average Nusselt number more than 1%. The accuracy of the grid combination chosen was further validated by estimating the

![FIGURE 2 Enlarged structure of the mesh grid](image-url)
average Nusselt number of pure water for a range of Reynolds number (10 000-200 000). The comparison of the result with Gnielinski correlation\(^5\) given in Equation (34) is shown in Figure 3. The maximum deviation obtained is 5.2\%. In addition, the friction factor was estimated using Darcy friction factor formula and compared with Blasius correction\(^5\) given by Equation (35). This result is shown in Figure 4 with a maximum deviation of 8.21%. Table 3 shows the thermophysical properties of single-walled carbon nanofluid.\(^5\)

\[ Nu = \frac{\xi}{\nu} \left( Re - 1000 \right) \frac{Pr}{1 + 12.7 \left( \frac{\xi}{\nu} \right)^{0.5} \left( Pr^{\frac{1}{3}} - 1 \right) \left( 1 + \left( \frac{d}{L} \right)^{\frac{2}{3}} \right) \left( 1 + \frac{d}{L} \right)^{\frac{2}{3}}}, \]  

(34)

where \( \xi = (0.79 \ln \ Re - 1.64)^{-2} \)

\[ f = \frac{0.316}{Re^{0.25}}, \]  

(35)

### Table 2

| \( N_r \times N_x \) | Nu  | % error |
|----------------------|-----|---------|
| 1 100 \times 200     | 495 |         |
| 2 100 \times 300     | 489 | 1.212121212 |
| 3 150 \times 400     | 484 | 1.022494888 |
| 4 150 \times 500     | 483.5 | 0.103305785 |

### Table 3

| Vol. | \( \mu \) | \( C_\rho \) | \( k \) | \( \rho \) |
|------|---------|---------|-------|--------|
| 0.00 | 0.0008  | 4.1428  | 0.6166| 997.369|
| 0.05 | 0.0009  | 3.9999  | 0.6250| 1007.37|
| 0.1  | 0.001   | 3.8571  | 0.6333| 1017.372|
| 0.25 | 0.0013  | 3.4285  | 0.6833| 1047.376|
4 | RESULTS AND DISCUSSION

4.1 | Nusselt number and hydrodynamics

This section deals with the result and discussion of the effect of Reynolds number and nanoparticle concentration on the thermal performance of SWCNT nanofluid passed through a straight horizontal pipe under constant heat flux in turbulent flow regime for a large range of Reynolds number (5000-200,000) and various nanoparticle concentration (0%-0.25%). Figure 5 shows the variation of the average Nusselt number of various concentrations of the SWCNT nanofluid against Reynolds number. The average Nusselt number increases as the Reynolds number and the concentration of the SWCNT increases. This result agrees with those obtained in the work of Mwesigye and Meyer, as an increase in Reynolds number and nanoparticle concentration will decrease the thickness of the thermal boundary layer and increase the thermal conductivity of the nanofluid respectively. Thus, more energy is transported from the pipe surface to the moving nanofluid.

Figures 6 and 7 show the variation of pressure drop and PP of SWCNT nanofluid as a function of Reynolds number and nanoparticle concentration. The graphs revealed that the two quantities (pressure drop and PP) increase as the concentration of the nanoparticles increase across the range of Reynolds number considered. The enhancement is due to an increase in viscosity of the SWCNT nanofluid, which increases in a nonlinearly manner as concentration increases. In addition, both quantities were also enhanced by an increase in Reynolds number for all nanoparticle concentrations. This is expected as high Reynolds number will increase the velocity gradient (shear stress), which in turn raises the pressure drop across the pipe.

Furthermore, Table 4 shows that the maximum cumulative heat transfer enhancement obtained for nanoparticle concentration of (0%-0.25%) is 9.09% and this occurs at \( \text{Re} = 200,000 \). At low Reynolds number (\( \text{Re} = 10,000 \)), the cumulative enhancement is 7.48%. In term of pressure drop, the maximum cumulative increase obtained is 119%. Finally,
maximum and minimum cumulative increase in PP are 199% and 181% and these occurred at Re = 30 000 and Re = 100 000, respectively. These values raise concern about the suitability of SWCNT nanofluid as a potential heat transfer fluid. The increment in convective heat transfer achieved required high PP, which is disadvantageous from the economic point of view.

### 4.2 Thermodynamic performance

The optimum design of any thermal energy system requires irreversibility due to entropy production is usually minimal. A system with high entropy production will result in low available energy (Exergy), which is undesirable. Figure 8 shows the variation in the rate of entropy generation for various concentrations of nanofluids calculated using Equation (26) as a function of Reynolds number. This agrees with the work of Ji et al.\(^ {36}\) The graph revealed that at low Reynolds number (10 000-50 000), the rate of entropy production decreases as Reynolds number and nanoparticle concentration increases.
This is as a result of the dominance of thermal irreversibility. At this region, the thermal gradient, which contributes most to thermal irreversibility, decreases as Reynolds number and nanoparticle concentration rise. Further increase in Reynolds number (50 000 < Re < 200 000) results in enhancement of entropy production, which increases as the concentration of the nanoparticle increases. At this region, viscous irreversibility is dominant. High Reynolds number will increase both velocity gradient and turbulent dissipation (this occurs as a result of conversion of kinetic energy of the fluid particle to internal energy by viscous shear stress rate), which in turn, increases the viscous irreversibility. Furthermore, Table 5 shows that at low Reynolds number (Re = 1000), increase in nanoparticle concentration (0%-0.25%) reduces the entropy production rate by 16.95%. Similarly, it was found to increase the rate of entropy production by 149.77% at a high Reynolds number (Re = 200 000). Finally, for range of Reynolds number (1000 < Re < 50 000), the presence of nanoparticle decreases the entropy production while at Re > 50 000, it increases entropy production rate.

To have a better understanding of the variations of entropy production in SWCNT nanofluids, the graph of Bejan number, which is the ratio of entropy production due to thermal effect of the total entropy production against Reynolds number for various concentrations of nanofluids, is shown in Figure 9. The results obtained from this graph are slightly better than the results obtained in the works of Ji et al.36 and Mwesigye and Meyer.56 The results show that at low Reynolds number (10 000-50 000) for all nanoparticle concentrations, the Bejan number is very close to unity and decreases as nanoparticle concentration increases. This region indicates the dominance of thermal entropy production over viscous entropy production and the reduction is due to the increase in thermal conductivity of the nanofluid, which inversely decreases the temperature gradient within the fluids domain. As Reynolds number increases (50 000 < Re ≤ 200 000), a negative gradient is observed trending downward. This region indicates the dominance of viscous entropy production over thermal entropy production. In addition, Table 5 shows that at Reynolds number (Re = 10 000 and 200 000), increase in nanoparticle’s concentration from (0%-0.25%) reduces the Bejan number by 0.05% and 102.35%, respectively. Finally, at nanoparticle concentration of 0.25%, the Bejan number decreases by 92.03% across the range of Reynolds number.

Figure 10 shows the variation of second thermodynamic law efficiency $\eta_{II}$ for various nanoparticle concentrations of SWCNT nanofluids as a function of Reynolds number. This parameter is used to quantify the amount of available work (exergy) that can be obtained from the thermal system. The results showed that at low Reynolds number (10 000-50 000), the exergy increases and reaches a maximum value as the concentration of the SWCNT nanofluid increases. This is attributed to the dominance of thermal entropy production, which decreases as nanoparticle increases. After this regime, the exergy began to decrease as the concentration of nanoparticle increase. The decrease in exergy as a function of nanoparticle concentration became more pronounced at high Reynolds number. Table 5 shows that at low Reynolds number (Re = 10 000), increase in nanoparticle concentration (0%-0.25%) increases $\eta_{II}$ by 3.43%. However, at high Reynolds number (Re = 200 000), the value of $\eta_{II}$ was decrease by 15.55%. Finally, for Reynolds number (1000 < Re < 50 000), the presence of nanoparticle increases $\eta_{II}$ while at Re > 50 000, the presence of nanoparticle decreases the value of $\eta_{II}$. This result gave us insight into the flow regime in which SWCNT will be beneficial from the perspective of thermodynamics second law. Figure 11 displayed the effect of adding SWCNT to water in term of on entropy generation number for a range of Reynolds number. This term measures the amount of irreversibility per unit heat transfer in the thermal system. The result obtained is in agreement with the previous studies of Ji et al.36 and Behera.59 The graph showed that at low Reynolds
number (10000-50000), the entropy generation number decreases as the concentration of nanoparticle increases. As Reynolds number increases further, an intersection was observed and the entropy generation number began to rise with an increase in the concentration of nanoparticle. This result indicates that it will be inadvisable to use SWCNT nanofluid at high Reynolds number as high irreversibility is produced. Figure 12 shows the variation of thermal performance factor of SWCNT nanofluid as a function of Reynolds number and nanoparticle concentration. This result is consistent with the previous work of Ji et al.36 This term combines the first and second law of thermodynamics to examine thermal performance. It measures the average heat transfer per unit entropy generation number. The graph shows that at low Reynolds

| Sensitivity Analysis of Thermal Performance Factors |
|---------------------------------------------|
| **Entropy** |
| Re = 10^4 → Re = 3 \times 10^4 → Re = 5 \times 10^4 → Re = 10^5 → Re = 2 \times 10^5 → Re = (10^4−2 \times 10^5) |
| ↓ 0% | 61.92% | -34.20% | -30.16% | 111.52% | -62% |
| ↓ 0.05% | -3.91% | -3.99% | -3.19% | 7.84% | 26.17% |
| ↓ 0.1% | -3.38% | -3.29% | -2.42% | 9.09% | 25.60% |
| ↓ 0.25% | -9.66% | -3.29% | -4.38% | 29.28% | 98% |
| (0%-0.25%) | -16.13% | -10.57% | -9.66% | 52.09% | 213.78% |
| **Bejan** |
| ↓ 0% | -0.38% | -2.00% | -20.91% | -74.17% | -80.05% |
| ↓ 0.05% | -0.01% | -0.15% | -1.07% | -11.13% | -23.83% |
| ↓ 0.1% | -0.01% | -0.29% | -1.25% | -11.93% | -23.88% |
| ↓ 0.25% | -0.03% | -1.09% | -5.85% | -30.45% | -54.64% |
| (0%-0.25%) | -0.05% | -1.53% | -8.03% | -53.51% | -73.69% |
| **Ns** |
| ↓ 0% | -61.92% | -34.20% | -30.15% | 113.50% | -62.64% |
| ↓ 0.05% | -3.91% | -3.98% | -3.19% | 7.84% | 26.175 |
| ↓ 0.1% | -3.38% | -3.28% | -2.41% | 9.08% | 25.60% |
| ↓ 0.25% | -9.65% | -8.89% | -4.37% | 29.28% | 98.00% |
| (0%-0.25%) | -16.12% | -15.40% | -9.66% | 52.09% | 213.77% |
| ↓ 0% | -61.92% | -34.20% | -30.15% | 113.50% | -62.64% |
| **Ep** |
| ↓ 0% | 592.80% | 135.78% | 159.26% | -15.07% | 3497.49% |
| ↓ 0.05% | 6.87% | 7.02% | 6.43% | -4.34% | -18.14% |
| ↓ 0.1% | 5.73% | 5.81% | 4.95% | -6.03% | -18.30% |
| ↓ 0.25% | 13.61% | 12.89% | 7.66% | -20.27% | -47.88% |
| (0%-0.25%) | 28.37% | 28.06% | 20.25% | -28.33% | -65.15% |
| **η** |
| ↓ 0% | 13.31% | 2.47% | 1.39% | -3.63% | 13.46% |
| ↓ 0.05% | 0.84% | 0.29% | 0.15% | -0.25% | -1.85% |
| ↓ 0.1% | 0.69% | 0.23% | 0.11% | -0.31% | -2.23% |
| ↓ 0.25% | 1.90% | 0.59% | 0.19% | -1.10% | -11.47% |
| (0%-0.25%) | 3.46% | 1.11% | 0.448% | -1.66% | -15.13% |
number (10000-50000), the thermal performance of the nanofluid increases as the concentration of SWCNT increases. Furthermore, increase in Reynolds number shows intersection and thermal performance of nanofluid began to decrease as nanoparticle concentration increases. Table 5 shows that at low Reynolds number (Re = 10000), increase in nanoparticle concentration (0%-0.25%) increases $E_p$ by 26.21%. However, at high Reynolds number (Re = 20000), $E_p$ was reduced by 84.32%. The results also show that at 10000 $\leq$ Re $\leq$ 50000. The presence of nanoparticle enhances $E_p$, while at Re $\geq$ 50000, it suppresses it.

This further reinforces our earlier hypothesis that the use of SWCNT will only be beneficial at low Reynolds, that is, the usage of nanofluid will only be advisable at region before the intersection. Finally, Figure 13 shows the variation of thermal performance at various Reynolds number as a function of nanoparticle concentration. The graph showed that...
at low Reynolds number, the thermal performance gives a positive gradient as the concentration of the nanoparticle increases. However, at high Reynolds number, the gradient became negative. This shows that at higher Reynolds number, the thermal performance decreases with increase in concentration. Furthermore, intersections were observed between the graphs, which signify equal performance at that configuration. For instance, the thermal performance of the nanofluid of 0.05% at Re = 200 000 is the same as that at Re = 150 000.

5 | CONCLUSION

The thermal performance of SWCNT nanofluid flowing through a straight circular pipe has been investigated numerically using Reynolds-averaged Navier-Stokes ($k-\epsilon$) model. The effect of Reynolds number and concentration of nanoparticles on the average Nusselt number, pressure drop, PP entropy production, thermodynamic second law efficiency, entropy generation number, and thermal performance factor has been studied in great detail. The results show that although there is an enhancement of 6% in heat transfer due to the use of SWCNT nanofluid, there is also a high pressure drop penalty of 60%, which accompanied it. In addition, at low Reynolds number, the exergy (available energy) increases as the concentration of nanoparticle increases and reaches the maximum value at Reynolds number close to 50 000. After this point, the exergy began to decrease as concentration increases. The overall result shows that SWCNT will only be advantageous at a Reynolds number within a range of (10 000-50 000).

Other results obtained are summarized as follows:

1. At Reynolds number (Re = 10 000 and 200 000), the presence of nanoparticle decreases the Bejan number by 0.05% and 102.35%, respectively.
2. At low Reynolds number (Re = 10 000), the usage of SWCNTs nanofluid is beneficial as it reduces the entropy production rate by 16.9%. However, at high Reynolds number (Re = 200 000), the entropy production rate increases by 149.77%.
3. The increase in the concentration of SWCNT at low Reynolds number (Re = 10 000), increases the second law of Thermodynamics efficiency by 3.43%. However, the thermodynamics efficiency is decreased by 15.55% at high Reynolds number (Re = 200 000).
4. The thermal performance factor shows that SWCNT nanofluid will only be beneficial at low Reynolds number.
5. At Reynolds number 150 000 and 200 000, the thermal performance of SWCNT nanofluid with concentration 0.05% are the same.

**NOMENCLATURE**

| Symbol | Description |
|--------|-------------|
| Re     | Reynolds number. |
| T      | Temperature of base fluid (K). |
| Nu     | Nusselt number. |
| s      | Rate of entropy production (W/m.K). |
| cp     | Specific heat capacity at constant pressure (J/kg-K). |
| Pr     | Prandtl number of base fluid. |
| uᵢ, uₓ | Component velocity (m/s). |
| I      | Turbulence intensity. |
| h      | Coefficient of heat transfer (W/m²K). |
| f      | Friction factor. |
| k      | Turbulence kinetic energy (m²/s²). |
| Dₜ     | Diameter of the pipe (m). |
| S      | Modulus of the rate of strain tensor. |
| Gₖ     | Rate of production of turbulent kinetic energy (J/kg) |
| C₂ₑ    | Turbulent constant. |
| σᵣ    | Turbulent constant. |
| σₖ    | Turbulent Prandtl number for k. |
| σₑ    | Turbulent Prandtl number for ε. |
| Δy₁    | Hight of the first cell (m). |
| uᵣ    | Frictional velocity (m/s). |
| τₜ     | Wall shear stress (N/m²). |
| Cᵢf    | Skin friction coefficient. |

**GREEK SYMBOL**

| Symbol | Description |
|--------|-------------|
| ε      | Turbulence energy dissipation (m²/s²). |
| φ      | Nanoparticle volume fraction. |
| μ      | Dynamic viscosity (kg/ms). |
| ρ      | Density of base fluid (kg/m³). |
| α      | Thermal diffusivity (m²/s). |
| λ      | Thermal conductivity (W/m-k). |
| κ      | Von Karman constant. |

**SUBSCRIPTS**

| Symbol | Description |
|--------|-------------|
| nf     | Nanofluid. |
| p      | Nanoparticle. |
| f      | Base fluid. |
| fr     | Freezing point. |
| t      | Turbulent. |

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