Advances in Energy Research, *Vol. 8, No. 3 (2022) 125-136*
https://doi.org/10.12989/eri.2022.8.3.125

Second law thermodynamic analysis of nanofluid turbulent flow in heat exchanger

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(*Received July 15, 2022, Revised September 4, 2022, Accepted September 7, 2022*)

Abstract. Entropy generation along with exergetic analysis is carried out using turbulent nanofluid flow in the heat exchanger. To obtain the optimized percentage constituent of nanofluid, the nanofluid volume concentrations is varied for the given input conditions. For different Reynolds number of the fluid and heat capacity rate ratio between the streams, the heat transfer improvements are studied in terms of nano particles diameter. Parametric analysis is carried out for a counterflow heat exchanger using turbulent nanofluid flow with exergetic efficiency along with entropy generation number as performance parameters. The exergetic efficiency provides realistic approach in the design of nanofluid applications in heat exchanger leading to conservation of energy.

Keywords: entropy generation; exergetic efficiency; heat-transfer enhancement; optimum concentration fractions; particle diameter

1. Introduction

Energy conservation is an important aspect in the today's energy sector due to rise in fuel costs and to reduce greenhouse gas emissions. Heats exchangers is having opportunities to reduce losses and to conserve energy. So, it is becoming inevitable to enhance the heat transfer and to look into ways for different techniques for heat dissipation. One of the methods used to enhance heat transfer in heat exchangers was extended surfaces or increase in surface area for given volume. Another method which becoming popular in these days is to use nanofluids to increase heat exchangers heat transfer. There comes the use of nano particles in usual base fluids leading to increase in nanofluids properties namely thermal conductivity, heat transfer coefficient, etc.

Using nanofluids in heat exchangers is having disadvantages along with so many advantages like increase in pressure drop because of enhanced values of viscosity, density, etc of fluids. First law analysis consider energy as same form and it is conserved. But second law analysis considers the quality of energy conversion and takes into account the irreversibilities in the system. Exergy and entropy generation analysis considers all the losses due to irreversibilities because of heat transfer, pressure drop, imbalance between fluids, etc (Bejan 1996). Second law thermodynamic method of analyses are reviewed in (Manjunath and Kaushik 2004) along with constructal theory.

Some of the literature reviews are carried out based on the analysis of this manuscript to

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http://www.techno-press.org/?journal=eri&subpage=7 ISSN: 2287-6316 (Print), 2287-6324 (Online)

understand the nanofluids research in heat exchangers. Theoretical study of the entropy-generation to analyse nanofluids heat transfer and fluid flow is investigated by Singh *et al.* (2010). Second law analysis of the nanofluids flow in a pipe is theoretically analysed in Moghaddami *et al.* (2011). Entropy generation of Al_2O_3–water nanofluid in terms of constant wall heat flux in turbulent forced convection flow performed in Bianco *et al.* (2014).

Exergetic performance of electronic devices by using exergy-entropy analysis applied to nanofluids by environment effect was investigated by Khaleduzzaman *et al*. (2016). Numerical study of the second-law for the flow of the water– Al_2O_3 nanofluid in a mini channel of triangular shape in terms of constant wall heat flux is performed in Bahiraei and Majd (2016). Second law performance of counter-flow plate heat exchanger having corrugation with the use of hybrid nanofluid in a milk chilling plant is applied in Bhattad *et al.* (2019).

Numerical simulations have been performed by Nakhchi and Rahmati (2021) on analysis of turbulent nanofluid flow in heat exchanger having twisted tapes. Energetic and exergetic analysis in a corrugated counterflow plate type heat exchanger by hybrid nanofluid with different particle volume ratios were carried out experimentally by Bhattad *et al.* (2020). A novel configuration of turbulator was analysed by Shafee *et al*. (2021) to model the nanomaterial turbulent flow to improve the performance by irreversibility analysis. Recently the hybrid nanofluid laminar flow analysis is carried out for double pipe heat exchanger by Manjunath (2021) incorporating entropy generation and exergy analysis.

Numerical investigation of irreversibilities characteristics of shell and tube heat exchanger having helical baffles is analysed by Bahiraei *et al*. (2021) having different shapes of nano particles in nanofluid. Improvement in performance by using hybrid nanofluid flow in microplate heat exchanger is reported by Garud *et al.* (2021) using first and second law analysis. A ternary hybrid nanofluid was used to analyse heat transfer improvements in radiator using exergetic method Sahoo (2021). Experimental comparison with numerical performance for nanofluid flow in mini channels was investigated by Wang *et al.* (2022) for heat transfer and pressure drop characteristics. The influence of nano particle shape considering spherical and non-spherical types on thermal performance of printed circuit heat exchanger is investigated by Nogueira (2022).

Considering the nano particle shape and fins effects of a mini shell and tube heat exchanger, the irreversibility analysis is performed by Bahiraei and Monavari (2022) to decide whether the fins provide improvements along with nanofluids. First and second law analysis was employed by Yang *et al*. (2022) to investigate heat transfer and pressure irreversibilities for the nanofluid flow in hair pin heat exchanger having annulus helical baffle. Novel nanofluid flow in heat exchanger was investigated experimentally and numerically by Irshad *et al*. (2022) and also analysis carried out using hybrid nanofluids. In the double pipe heat exchanger, the nanofluid parametric optimization is done by Manjunath (2022) incorporating exergetic and entropy generation performance techniques.

From the literature review, it can be found out that there exists very less manuscripts which uses both the performance methods namely exergetic efficiency analysis and entropy generation minimization. By using nanofluids to know whether there are advantages and disadvantages is not easily able to investigate by using first law analysis (heat transfer and pressure drop). But by using second law analysis the irreversibilities found in the heat exchangers can be investigated. The second law analysis of heat exchanger using nanofluid and considering the imbalance irreversibilities is not yet carried out in literatures. Optimum values of operating parameters of nanofluids particles present in the base fluids are not obtained for turbulent flow in heat exchanger by the earlier investigators.

In this manuscript, the important behaviour of nanofluid usage in heat exchanger is investigated to know the benefits and disadvantages. The second law performance known as exergetic efficiency are used to investigate the performance of a counterflow heat exchanger turbulent flow by the use of nanofluid. The present work is validated with the previous work and variations are investigated with the possible reasons. The results are compared and supported by another dimensionless performance parameter entropy generation number. Water-alumina (water-Al_2O_3) nanofluid is considered having volume fractions and particle diameter variations. Also, for turbulent fluid flow, the Reynolds number and heat capacity ratio are varied to study the response of nanofluid in different working conditions.

2. Analysis

In this analysis, same type of nanofluid is considered in both the streams of a counterflow heat exchanger. The assumptions used in this analysis are as follows: fully developed turbulent fluid flow in both the streams; perfect heat transfer exists between hot and cold streams; in the constant cross section of tubes, the pressure drop is due to friction loss and material resistance between the streams are neglected Manjunath and Kaushik (2015).

2.1 Thermophysical properties of nanofluid

Referring to Bianco *et al*. (2014), thermophysical properties considered in the investigation are as follows, Nanofluid density is calculated as

$$\rho_{nf} = (1-\varphi)\rho_{bf} + \varphi\rho_p \tag{1}$$

Specific heat of nanofluid is calculated as

$$c_{p,nf} = \frac{(1-\varphi)\rho_{bf}c_{p,bf} + \varphi\rho_p c_{p,p}}{\rho_{nf}} \tag{2}$$

Nanofluid thermal conductivity is calculated as

$$\frac{k_{nf}}{k_{bf}} = 1 + 4.4Re_d^{0.4}Pr_{bf}^{0.66}\left(\frac{T_{nf}}{T_{fr}}\right)^{10}\left(\frac{k_p}{k_{bf}}\right)^{0.03}\varphi^{0.66} \tag{3}$$

Here base fluid freezing point is T_{fr}. Reynolds number of nanoparticles is calculated as

$$Re_d = \frac{\rho_{bf}u_b d_p}{\mu_{bf}} \tag{4}$$

Where Brownian velocity is calculated as

$$u_b = \frac{2k_b T_{nf}}{\pi\mu_{bf}d_p^2} \tag{5}$$

Boltzmann's constant is $k_b = 1.38066 \times 10^{-23}$ J/K. Nanofluid viscosity is obtained by the following relation,

$$\frac{\mu_{nf}}{\mu_{bf}} = \frac{1}{1-0.10255d_p^{-0.264}\varphi^{1.028}} \tag{6}$$

The thermal conductivity and viscosity are based on empirical relations developed by large experimental results carried out by earlier works Bianco *et al.* (2014). But there will be some appreciable differences with the experimental results under certain range of operation.

2.2 Exergy and entropy analysis

The second-law efficiency is found for heat exchanger as Kotas (2013)

$$\psi = \frac{\dot{E}_c^T}{\dot{E}_c^T + \dot{I}} \tag{7}$$

The exergy of the cold side is given by the following expression Kotas (2013)

$$\dot{E}_c^T = C_c\left[\varepsilon\left(T_{h,in} - T_{c,in}\right) - T_o\, ln\left\{1 + \varepsilon\left(\frac{T_{h,in}}{T_{c,in}} - 1\right)\right\}\right] \tag{8}$$

Gouy-Stodola expression is calculated with respect to reference temperature and entropy generation rate as

$$\dot{I} = T_o\dot{S}_{gen} \tag{9}$$

The Eq. (9) is re-written in terms of entropy generation number and minimum of heat capacity.

$$\dot{I} = T_o(N_s C_{min}) \tag{10}$$

Where the N_s is calculated as follows Bejan (1996).

$$N_S = \frac{\dot{S}_{gen}}{(\dot{m}c_P)_{min}} \tag{11}$$

The S_{gen} is defined for heat exchanger as Manjunath and Kaushik (2014)

$$\dot{S}_{gen} = \left(\dot{m}c_p\right)_c ln\left(\frac{T_{c,out}}{T_{c,in}}\right) + \left(\dot{m}c_p\right)_h ln\left(\frac{T_{h,out}}{T_{h,in}}\right) + \left(\frac{\dot{m}}{\rho T_{ave}}\right)_c (\Delta P)_c + \left(\frac{\dot{m}}{\rho T_{ave}}\right)_h (\Delta P)_h \tag{12}$$

The S_{gen} is expressed as follows Bejan (1996)

$$\dot{S}_{gen,T} = \left(\dot{m}c_p\right)_c ln\left(\frac{T_{c,out}}{T_{c,in}}\right) + \left(\dot{m}c_p\right)_h ln\left(\frac{T_{h,out}}{T_{h,in}}\right) \tag{13}$$

$$\dot{S}_{gen,P} = \frac{2\dot{m}^3 fL}{\rho T_{ave} D A_c^2} \tag{14}$$

Heat capacity for hot and cold streams is given as

$\left(\dot{m}c_p\right)_c = C_c$ and $\left(\dot{m}c_p\right)_h = C_h$

Ratio of heat capacity rate is found as

$$C = \frac{C_{max}}{C_{min}} \tag{15}$$

Considering C_{max} as C_h and C_{min} as C_c

$$C = \frac{C_h}{C_c} \tag{16}$$

Using Eqs. (13) and (14) in Eq. (11), N_s for unbalanced heat exchanger is found as

$$N_S = ln\left[1 + \varepsilon\left(\frac{T_{h,in}}{T_{c,in}} - 1\right)\right] + C\, ln\left[1 - \frac{1}{C}\varepsilon\left(1 - \frac{T_{c,in}}{T_{h,in}}\right)\right] + \left[\frac{2\dot{m}^3 fL}{\rho T_{ave} D A_c^2}\right]_c + C\left[\frac{2\dot{m}^3 fL}{\rho T_{ave} D A_c^2}\right]_h \tag{17}$$

The effectiveness and *NTU* = heat transfer units expression for the counterflow configuration heat exchanger is as follows

$$\varepsilon = \frac{1-e^{-NTU(1-C_{min}/C_{\max})}}{1-(C_{min}/C_{max})e^{-NTU(1-C_{min}/C_{\max})}} \tag{18}$$

NTU is obtained as

$$NTU = \frac{UA}{C_{min}} \tag{19}$$

Heat exchanger thermal resistance is obtained by the addition of heat exchanger resistances of both sides Manjunath and Kaushik (2014)

$$\frac{1}{UA} = \frac{1}{(hA)_c} + \frac{1}{(hA)_h} \tag{20}$$

Here U = heat exchanger overall heat transfer coefficient and h = heat transfer coefficients for cold and hot streams. Here in the analysis, the material resistance of conduction separating the two streams is assumed very small and can be neglected. Using *NTU* expression from Eq. (19), the Eq. (20) becomes

$$\frac{1}{NTU} = \frac{1}{(NTU)_c} + \frac{1}{(NTU)_h} \tag{21}$$

Stanton number is defined as

$$St = \frac{h}{c_p G} \tag{22}$$

Here G = mass velocity. Considering the Stanton number and *NTU* expressions, the combination of *NTU* Eq. (21) used for cold and hot sides is obtained as

$$\frac{1}{NTU} = \left(\frac{A_c}{A}\right)_c \frac{1}{(St)_c} + \frac{C_c}{C_h}\left(\frac{A_c}{A}\right)_h \frac{1}{(St)_h} \tag{23}$$

Here the tube cross sectional area is represented as A_c and surface area is given as A. Considering that the same geometrical dimensions of both streams tubes and assuming same fluid is flowing in both tubes' sides, Eq. (23) reduces to

$$NTU = \frac{4L}{D} St\left[\frac{C}{C+1}\right] \tag{24}$$

For the assumptions done as above, the total N_s defined for unbalanced heat exchanger in respect of Eq. (17) is given as

$$N_S = ln\left[1 + \varepsilon\left(\frac{T_{h,in}}{T_{c,in}} - 1\right)\right] + C\, ln\left[1 - \frac{1}{C}\varepsilon\left(1 - \frac{T_{c,in}}{T_{h,in}}\right)\right] + \frac{2fL}{\rho D A_c^2}\left[\left(\frac{\dot{m}^3}{T_{ave}}\right)_c + C\left(\frac{\dot{m}^3}{T_{ave}}\right)_h\right]_c \tag{25}$$

In this analysis, the nanofluid flow is considered as turbulent flow (fully developed) the value of heat exchanger tube length to diameter ratio will be more than 60 ($L/D \geq 60$). Nusselt number value for a uniform heat flux circular duct is given as (Singh *et al*. 2010, Manjunath and Kaushik 2015, Manjunath and Kaushik 2014).

$$Nu = 0.023 Re^{0.8} Pr^{0.4} \tag{26}$$

Where *Re* is Reynolds number and *Pr* is Prandtl number. Expression for friction factor is given as

$$f = 0.316Re^{-0.25} \tag{27}$$

The Stanton number is given as

$$St = \frac{Nu}{Re.Pr} \tag{28}$$

Heat exchanger heat transfer rate is defined with effectiveness as

$$\dot{Q} = C_{min}\varepsilon\left(T_{h,in} - T_{c,in}\right) \tag{29}$$

Heat exchanger Pressure drop equation is obtained as

$$\Delta P = \left(\frac{fL\rho V^2}{2Dg}\right)_c + \left(\frac{fL\rho V^2}{2Dg}\right)_h \tag{30}$$

3. Results and discussion

In this analysis, the formulae considered are empirical which are obtained from open literature. These empirical relations are formulated and validated by experimental results and numerical procedure by earlier researchers.

For the investigation the input values considered as followed in (Bianco *et al.* 2014, Manjunath and Kaushik 2015, Manjunath 2014) are as follows: hot and cold streams temperature ratio at inlet ($T_{h,in}/T_{c,in}$) is 1.5; cold stream temperature at inlet ($T_{c,in}$) is 301 K; length of heat exchanger tube (L) is 1 m. The reference temperature considered as (T_o) is also equal to $T_{c,in}$. Water and alumina (Al_2O_3–water) nanofluid properties are obtained from (Singh *et al.* 2010). Water (base fluid): density, ρ_{bf} is 1000 kg/m^3; thermal-conductivity, k_{bf} is 0.6 W/m K; viscosity, μ_{bf} = 0.001 N s/m^2; specific-heat, $c_{p,bf}$ = 4180 J/kg K.

Alumina (nano particle): density, ρ_p = 3900 kg/m^3; thermal-conductivity, k_p = 40 W/m K; specific heat, $c_{p,p}$ = 880 J/kg K. The analysis program is solved in EES software (Klein 2008). For turbulent flow which is fully developed the two values of Reynolds number considered are 1×10^4 and 5×10^3. The capacity ratio values are 1 and 2 considered to study the variations of imbalanced reversibility. The volume concentrations value as (φ = 0.03) and particle diameter value as (d_p = 20 nm) considered for analysis. The fluid properties are referred at average temperatures.

To validate the present work, comparison with previous work (Bianco *et al.* 2014) is carried out by considering input values as particle diameter = 46 nm, heat flux = 21 kW/m^2, Reynolds number ranging from 5×10^3 and 10×10^4, volume concentration of nano particles (0.009, 0.018, 0.036) and other input values referred from (Bianco *et al.* 2014). The results of validation and comparison is shown in the figure 1 where the performance parameter N_s is calculated. The deviation of results are about 2% to 5% is observed and it is having acceptable range. This is due to differences in thermophysical properties variations and the empirical relations of nanofluid properties used.

Fig. 2 gives the behaviour of ψ and N_s for different values of volume concentration of nano particles (φ), Reynolds number (Re) and heat capacity rate ratio (C). As φ increases, ψ will initially increases, establishes maximum value and further decreases. Corresponding to the attained higher value of ψ, it provides optimum value of φ for lower losses and increased performance for the considered input values. This is because of increase in heat transfer due to increase of φ which leads to ψ increase initially. There after certain φ value, there is decreasing trend because of pressure drop increase because of increase in nanofluid viscosity which decrease ψ after attending peak value. As

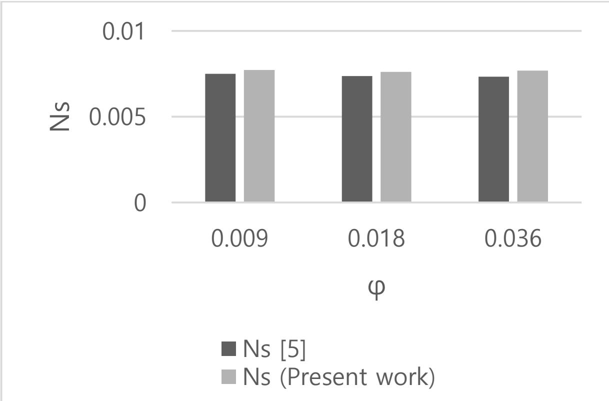

Fig. 1 Validation of present work with previous work (Bianco *et al.* 2014)

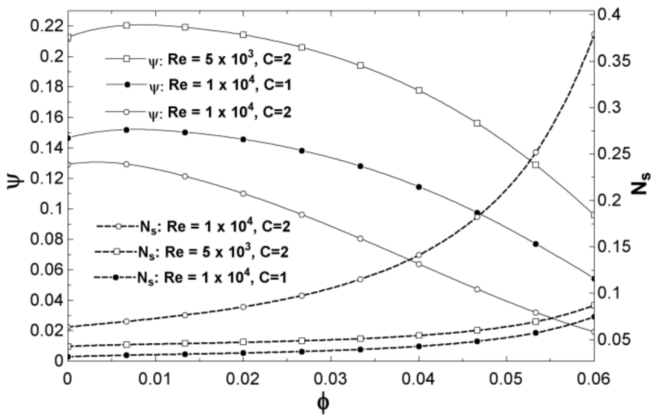

Fig. 2 Behavior of exergetic efficiency (ψ) along with entropy generation number (N_s) versus volume concentrations (φ)

Re decreases in the range of turbulent flow, ψ increases, indicating lowering of randomness and losses. There is increase in the ψ, as C decreased from 2 to 1. This is due to the absence of imbalance irreversibility between the fluid streams and the heat exchanger will become a balanced one.

The ψ results are validated from the relations of N_s with the φ in figure 2. As φ increases, there is parabolic increasing behaviour of N_s particularly when $C = 2$. N_s consisting of all the irreversibilities and provides trade-off between these losses for varying operating and geometrical parameters of heat exchanger. As the value of *Re* reduced in the turbulent range and also value of C is changed from 2 to 1 (balanced one), the value of N_s decreases for the given value of φ. After particular value of φ, the pressure drop irreversibility will become more prominent with respect to heat transfer irreversibility due to increase of nanofluid viscosity as compared to base fluid. This results in the optimum value for ψ. That is after certain value of φ, the pressure drop loss will be more than heat transfer loss. These aspects are displayed by N_s results.

Fig. 3 gives the behaviour of Q and ΔP for different values of nano particles volume-concentration (φ). It is observed that as φ increases, Q increases indicating the advantages of

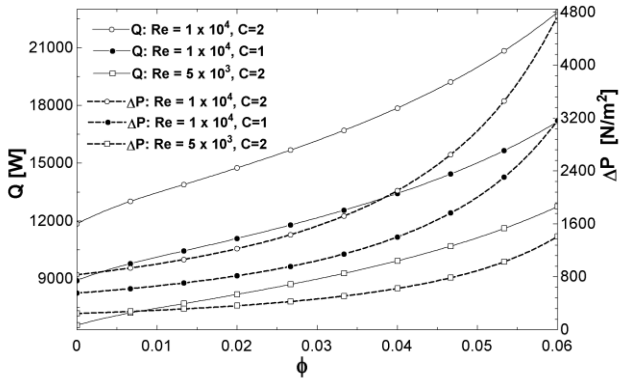

Fig. 3 Behaviour of Heat transfer rate (Q) along with Pressure-drop (ΔP) versus volume concentrations (φ)

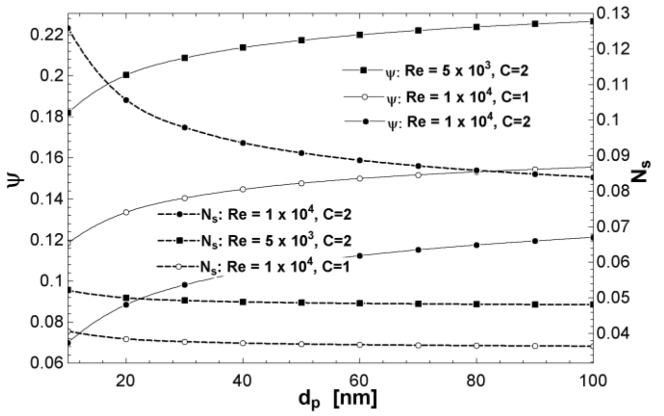

Fig. 4 Behaviour of second law efficiency (ψ) and entropy-generation-number (N_s) versus nano particle diameter (d_p)

nanofluids usage in heat exchangers leading in increase of heat transfer. when $\varphi = 0$, the value of Q will be lowest. This corresponds to the base fluid only, that is the conventional working fluid. For the given value of φ, the value of Q will increase as the value of Re increased in the given turbulent region. This is because of increase in randomness of nano particles. As C reduced from 2 to 1, Q reduces because of decrease in imbalance irreversibility. Also seen in the figure 3, the increase in ΔP for increase in φ because of increase in viscosity. As Re reduces, the ΔP reduces accordingly because of reduction in randomness of fluid. Also, as C increased, for the given inputs, ΔP increases. This is because of increase in the imbalance between the streams.

In Fig. 4, the relationship of ψ and N_s with the nano particle diameter (d_p) variation is presented. As d_p increases in nanofluid, the ψ increases with more slope initially and with less slope then afterwards. This indicates that after certain value of d_p, there will be no appreciable increase of ψ. As the values of Re and C are reduced, the value of ψ increase for a particular value of d_p. This is due to reduction of heat transfer irreversibility in turbulent flow and reduction of imbalance irreversibility respectively.

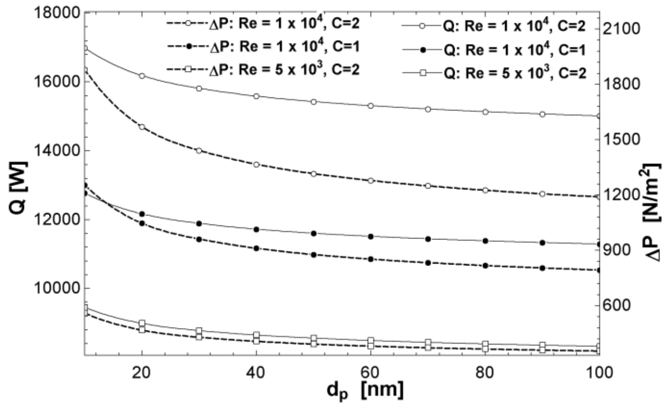

Fig. 5 Behaviour of heat transfer rate (Q) along with pressure drop (ΔP) with nano particle diameter (d_p)

The above results are validated from the reducing trend of N_s for increase of d_p in the Fig. 4. Initially the decrease in slope value is more, then after particular value of d_p, the decreasing slope is less. When *Re* value increases for a given value of d_p, the value of N_s value increases due to increase in irreversibilities. When *C* value increases for a given value of d_p and R_e, will results in increase of N_s because of increase in irreversibility due to imbalance between the fluid streams.

Fig. 5 provides the behaviour of Q and ΔP for changes in the value of nano particle diameter (d_p). There is decreasing trend for Q for increase in the value of d_p. This is due to that as nano particle diameter increases, the thermal conductivity decreases. This is because of decrease in Brownian velocity as the nano particle size increases which leads to decrease in heat transfer (Apmann *et al.* 2022 In the same trend, the value of ΔP decreases as the nano particle size increases due to increase in viscosity. As the values of *Re* and *C* reduced, the values of Q and ΔP reduces. Initial decrease in slope of curves is more and then afterwards becomes lesser.

4. Conclusions

For a counterflow heat exchanger second law analysis is performed with the use of nanofluid considering turbulent-flow. In this study, performance methods used are entropy generation and exergetic efficiency. To analyse heat exchangers operated by nanofluids, the irreversibilities considered are heat transfer, pressure drop and imbalance between the fluid streams in a single closed form equation. With the method of exergetic analysis, the optimum value of nano particles volume-concentration of nanofluid is obtained for higher thermal efficiency. This provides realistic operating parameters of nanofluid heat exchangers in heat transfer equipment with reduced losses leading to energy conservation. The present work is validated with the earlier work and the values variations is within 5%. As the value of Reynolds number is decreased under the range of turbulent fluid flow and also as the ratio of heat capacity rate is reduced to balanced fluid streams, the optimum value of volume concentration will increase for considered input parameters.

There is enhancement of heat transfer rate in heat exchangers considering nanofluid compared to conventional fluids. But also, it is observed that the pressure drop irreversibility increases in the nanofluids. Therefore, the second-law analysis provides trade-off between different irreversibilities to obtain optimum operating parameters. After some value of increase in nano particle diameter, there will no further increase of exergetic efficiency. This indicates that the effects of nano particle diameter are less in terms of thermal design and operation of heat exchangers. As the nano particle diameter increases, the heat transfer and pressure drop decreases because of decrease of Brownian motion in nanofluid.

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CC

Abbreviations

d_p	nano particle diameter (m)
$\dot{E}$	rate of exergy (W)
$\dot{I}$	irreversibilities (W)
N_s	dimensionless entropy generation
$\dot{S}_{gen}$	entropy generation rate (W/K)

Subscripts

c	cold-stream
h	hot-stream
in	fluid-inlet
out	fluid-outlet
bf	base-fluid
nf	nano-fluid
p	nano-particle

Greek symbols

ψ	rational or exergetic-efficiency
φ	volume concentrations