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Research Article

Evaluating the Deterioration of a Concentric Heat Exchanger using Energy and Exergy Degradation Factors

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ABSTRACT

The aim of the present study is to develop a new technique for assessing the sustainability of concentric heat exchanger through quantifying the deterioration caused by energy destruction and entropy generation. The study also addresses the interconnection between heat capacity rate, flow rate, exchange flow configuration, and the sources of energy destruction. The degradation of the exchanger performance is portrayed by the newly-developed coefficients. The outcome of the study shows that heat capacity rates of cold and hot fluids have remarkable impact on the effectiveness of the heat exchanger, particularly when the ratio of such capacity rate is close to unity. Quantitively speaking, by doubling the cold fluid flow rate, the effectiveness declines by ~13% since the minimum capacity ratio is also doubled. However, as the flowrate of the cold fluid further increases, the effectiveness ameliorates by ~ 6.5 % for parallel flow and 37.5% for counter flow, succeeding the reduction in the minimum capacity ratio by ~25% and 34 %, respectively. Experimental observation confirmed that deterioration due to irreversibility production generates eminent penalties in the exchanger performance, decreasing the exergy efficiency up to 52%.

1. Introduction

Applications of heat exchanger using conventional fluid are immense. Industries that deal with chemical treatments, food production, technical manufacturing, refrigeration and air-conditioning units, and thermal power establishments are but a few sectors that customarily use such thermal equipment, [1]. Heat exchangers are systems that interchange heat energy between two or more fluids at different temperatures, [2].

In recent decades, many studies have been conducted to increase the rate of heat transfer and enhance the performance of different types of heat exchangers using numerous methods. Jassim investigated the spatial and temporal behavior of spiral coil heat exchanger [3, 4]. Effectiveness and rate of heat transfer was found to relate directly to the coil geometry and the number of loops [5].

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Nomenclature			
A	surface area, m^2	*Greek symbols*	
c	Minimum heat capacity ratio	ρ	Density, kg/m^3
C_p	constant pressure specific heat, J/kg K	μ	Viscosity, Pa.s
d	Tube diameter, m	ε	Heat exchanger efficiency
D_h	hydraulic diameter, m	ψ	Exergy efficiency
$\dot{E}$	Rate of exergy, W	ϵ	Exergy degradation factor
f	Friction factor	τ	Energy degradation factor
h	Convection H.T coefficient, W/m^2 K	*Subscripts*	
$\dot{I}$	Irreversibility, W	b	Bulk average
k	Thermal conductivity, W/m K	c	Cold side
L	Length of the exchanger, m	ci	Cold inlet
$\dot{m}$	mass flow rate, kg/s	co	Cold outlet
$\dot{Q}$	Rate of heat transfer, W	h	Hot side
s	Entropy, J/kg K	hi	Hot inlet
T	temperature, ^oC	ho	Hot outlet
U	Overall H.T coefficient, W/m^2 K	lm	Log-mean
$\dot{V}$	Volume flowrate, m^3/s	NHL	value at the reference point

Recently, researchers concentrate their attention to enhance the performance of the heat exchanger by utilizing nano metallic particles with the conventional fluid. Choi and Eastman [6] reported that thermal conductivity of water improved significantly when nanoparticles of additive is homogenously mixed. Jassim and Ahmed [7] tested the enhancement in heat transfer in lubricant system of IC engine when the engine oil mixes with nano-sized copper. They concluded that heat transfer rate augments substantially resulting in an increase in the rate of heat transfer. Ghazvini et al [8] studied the convection heat transfer variation of nanofluid in laminar flow regime. The influence of the nanofluid type and volume fraction on the performance of different types of heat exchangers is experimentally investigated by [9] and [10]. Outcome of the assessment proved that increase in the nanofluid volume fraction does enhance the exchanger performance as well as the rate of transfer. However, optimization is required to avoid accretion of particles in the sublayer region.

Over the decades, many researchers have devoted substantial effort on optimizing the energy transfer in heat exchangers. Design and optimization of an efficient and reliable heat exchanger that could remove high heat flux is undoubtedly the prominent key research concern in advanced thermal engineering.

As climate change and its detrimental influence on the environment has been internationally prioritized, reducing or preventing energy dissipation in thermal equipment has become a major concern of recent studies.

Identifying and quantifying the source(s) of energy dissipation is crucial to diminish the energy interaction with the surrounding. However, energy analysis alone does not comprise all aspects of thermal losses. Exergetic analysis, on the other hand, allows optimal characterization of the behavior of thermal systems since it could limit the sources of energy destruction to the strictly inevitable.

First discussed in his article in 1980, Bejan [11] employed the first and second laws of thermodynamics to upgrade the performance of the heat exchanger by bridging the gap between the irreversibility concept and the thermodynamic optimization of the heat exchanger. His analysis brought up a new concept called 'System Availability' or Exergy. Exergy represents the degree of deviation of the state of a system relative to the environment [11]. Vilarinho et al., [12] mentioned that exergetic performance of heat exchanger is indispensable and could be used to enhance thermal plants. Jassim [13] reported that mixing nano-sized copper with engine lubricant could double the exergy potential and boost the exergy efficiency of the engine.

Exergy analysis of the heat exchanger is complicated since it encompasses three major sources of irreversibility, namely: the energy transfer between fluids, the energy loss due to friction, and the energy dissipation to the environment [14].

Gothe et al., [15] employed extended exergy analysis to evaluate the entropy production due to frictional source in a double-pipe heat exchanger. They pointed out the necessity of minimizing the frictional irreversibility of the fluids. However, Fard et al [14] reported that the main source of the exergy destruction is the exchanger-environment energy interaction. Wang et al. [16] quantified such destruction and found that ~ 24% of the thermal energy was destructed as a result of irreversibility due to interaction with the environment. Zühlsdorfa et al. [17] work concluded that the reduced temperature difference in the heat exchangers decrease the exergy destruction and yield higher cycle performance.

Influence of cross-section shapes of heat exchangers on entropy generation and rate of heat transfer have been numerically studied by Kurnia et al. [18]. They reported that the square shape produced larger entropy than circular or elliptical cross-section. Begag et al. [19] studied numerically the impact of corrugated surface on the irreversibility and exergy losses for a concentric heat exchanger. The results showed that a concave corrugated arrangement enhances the thermal exergy but increases the irreversibility due to friction.

Usefian and Bayareh [20] conducted a numerical simulation on a heat exchanger to investigate the effect of the flowrate on the energy and exergy deterioration using single and double segmental baffles. The revealed outcome demonstrated that entropy generation is augmented by increasing the mass flow rate of cold and hot fluids. They also found that inserting another segmental baffle would lead to a decrement in the Nusselt number as well as an increment in the exergy loss.

By utilizing a numerical approach, Ashkan [21] proposed a new dimensionless expression of exergy efficiency in correlation to hot/cold flowrates, geometrical parameters, and non-dimensional inlet temperatures. Wang et al [16] introduced an optimization correlation for operational parameters that are responsible for energy destruction. Exergy performance and efficiency was correlated by Al-Salem et al. [22] during their numerical study on double-pipe heat exchanger. The two correlations are functions of the thermal, operational, and geomatical variables. Another experimental study performed by Rahimi et al. [23] deduced the optimal operational parameters that maximize the exergy efficiency.

Xu et al. [24] attempted to optimize the Reynolds number for minimum exergy destruction using different tube sizes, flowrates, and inlet temperatures. The outcome of their study suggested that optimal Reynolds number is strongly dependent on the flowrates and tube diameter rather than fluid inlet temperatures.

Rostami et al. [25] experimentally examined the effect of Reynolds number on the irreversibility production in horizontal coiled tube heat exchanger for two-phase stream. They revealed that energy and exergy efficiencies are more sensitive to the Reynolds number of both phases.

The exergy analysis of double-pipe heat exchanger is examined by Ahmed et al. [26]. A comparison of two cases of heat capacity ratio were presented for parallel flow arrangement. The exergy loss was found to be higher when the heat capacity rate of hot fluid is lower than that of cold fluid.

Studies have frequently ignored the heat loss to the environment when exergy analysis of heat exchanger is addressed. Instead, they concentrated on the irreversibility generated by friction and fluid-fluid sources. However, it was reported that neglecting exergy losses due to heat interaction with the surroundings might not be correct if the exchanger operates at temperatures very different to the ambient [27].

Due to the scarcity of literatures that include the heat leak when they address and quantify the deterioration in exchanger performance, the outcome of the current work might instigate researchers to pursue assessing the influence of the energy leak on the operational performance of the heat exchanger. Furthermore, to the best of our knowledge, no known work has been conducted on the exergy degradation of heat exchanger. Hence, the representation of the results pertained to the energy and exergy degradation factors can serve as useful tools to the engineers and concerned sectors for better understanding, design, and optimization of any type of heat exchanger.

2. Theory

The energy and exergy characteristics and performance are estimated with the aid of experimental data from the hot and cold fluids.

2.1 Energy analysis

The Reynolds number of both streams is determined from the hydraulic diameter of the carrier and the volume flowrate, density, and viscosity of each fluid as follows:

$$Re = \frac{4\rho\dot{V}}{\pi\mu D_h} \tag{1}$$

where $D_h = d_i$ for hot fluid and
$D_h = d_o - d_i$ for cold fluid
The heat absorbed by the cold fluid and the heat discharged from the hot fluid are calculated using the corresponding measured inlet and exit temperatures:

$$\dot{Q}_c = \rho_c \dot{V}_c C_{p(c)} (T_{co} - T_{ci}) \tag{2a}$$

$$\dot{Q}_h = \rho_h \dot{V}_h C_{p(h)} (T_{hi} - T_{ho}) \tag{2b}$$

The heat lost to the surrounding atmosphere, which is normally ignored, is considered in the current study and determined from the following expression:

$$\dot{Q}_l = \dot{Q}_h - \dot{Q}_c \tag{3}$$

The effectiveness of the concentric heat exchanger is obtained from the experimental observations data using the following equation [28]:

$$\varepsilon = \frac{(\dot{Q}_h + \dot{Q}_c)/2}{(\rho\dot{V}C_p)_{min}(T_{hi} - T_{ci})} \tag{4}$$

Then, The Energy Degradation Factor (τ) is defined as [29]:

$$\tau = \frac{\varepsilon_{NHL} - \varepsilon}{\varepsilon_{NHL}} \tag{5}$$

ε_{NHL} is the effectiveness of the double pipe heat exchanger when there is no heat loss to the environment, calculated theoretically from the following equation:

Parallel Flow: $$\varepsilon = \frac{1 - e^{[-NTU(1+c)]}}{1+c} \tag{6a}$$

Counter Flow: $$\varepsilon = \frac{1 - e^{[-NTU(1-c)]}}{1 - c \times e^{[-NTU(1-c)]}} \tag{6b}$$

where c stands for the heat capacity rate ratio which is defined as the ratio between the minimum to the maximum heat capacity rate (C_{min}/C_{max}). NTU, on the other hand, is the Number of Transfer Unit defined as the dimensionless group UA_s/C_{min}.

2.2 Exergy analysis

A typical heat exchanger has three main sources of exergy losses associated with its operational condition. In the present analysis, irreversibility due to temperature difference and energy interaction with the atmosphere due to heat loss are accounted. Heshmatian et al [30] have reported that irreversibility resulting from friction is insignificant for tubes of sizes larger than 10 mm. Hence, the thermal source dominates the total entropy generation. Assuming the condition is steady state with uniform thermophysical properties, the total irreversibility is obtained from the following expression, [31]:

$$\dot{I} = \dot{I}_{\Delta T,HE} + \dot{I}_{\Delta T,heat\ loss} \tag{7a}$$

Each source of irreversibility can be calculated from the following equations [31]:

$$\dot{I}_{\Delta T,HE} = T_o \left[\rho_c \dot{V}_c C_{p(c)} \ln\left(\frac{T_{co}}{T_{ci}}\right) - \rho_h \dot{V}_h C_{p(h)} \ln\left(\frac{T_{hi}}{T_{ho}}\right)\right] \tag{7b}$$

$\dot{I}_{\Delta T,HE} = T_o \left[\rho_c \dot{V}_c C_{p(c)} \left(\ln\frac{T_{b,c}}{T_o} - \frac{T_{b,c} - T_o}{T_o}\right) + \right.$
$\rho hVhCp(h)$ln$Tb,hTo-Tb,h-ToTo$ (7c)

The change in exergy of the cold and hot fluids are respectively obtained from the following:

$$\Delta\dot{E}_c = \dot{Q}_c - \rho_c \dot{V}_c T_o (s_{co} - s_{ci}) \tag{8a}$$

$$\Delta\dot{E}_h = \dot{Q}_h - \rho_h \dot{V}_h T_o (s_{hi} - s_{ho}) \tag{8b}$$

where:

$s_{co} - s_{ci} = C_{p(c)} \ln\left(\frac{T_{co}}{T_{ci}}\right)$ and

$$s_{hi} - s_{ho} = C_{p(h)} \ln\left(\frac{T_{hi}}{T_{ho}}\right) \tag{8c}$$

The exergetic efficiency (ψ) is defined as:

$$\psi = \frac{\Delta\dot{E}_c}{\Delta\dot{E}_h} \tag{9}$$

Finally, the exergy degradation factor is defined as:

$$\epsilon = \frac{\psi_{I=0} - \psi_I}{\psi_{I=0}} \tag{10}$$

where:

$$\psi_{I=0} = \frac{(\Delta \dot{E}_c)_{I=0}}{(\Delta \dot{E}_h)_{I=0}} \tag{11}$$

3. Experimental Set-up

The experimental equipment used in the current research is manufactured by the Edibon International Company based in Spain. The setup consists of two units, namely: the base unit and the concentric heat exchanger unit. The heat exchanger unit (Figure 1) consists of a meter-long tube coaxial double-pipe. The cold fluid tank is placed separately, and with the help of another impeller pump, is pumped into the heat exchanger unit.

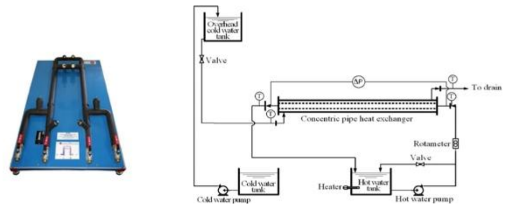


Fig. 1. Schematic diagram of the heat exchanger unit

Hot water is circulated through the internal tube while cold water flows through the annulus area lying between the rubber-insulated external stainless tube and the stainless internal tube. Flow direction may be configured for co-current (parallel) or counter-current (opposite direction) of the hot/cold streams. The transition between the two configurations may be implemented without shutting down the pumps, but it is recommended to reverse the cold flow for safety reasons.

Temperatures at the inlet and outlet of both fluids are measured using J-type thermocouples. Six thermocouples are distributed strategically to measure the temperature of the hot and cold fluids. One of six thermocouples is placed into the hot tank to measure the temperature of the water tank, which can be regulated through PID built-in software with a limit up to 60°C. The complete setup is monitored and controlled by the PLC-SCADA system. Parallel and counter flow patterns are investigated in the current study by adjusting a ball valve mounted on the base unit. The volume flowrates of both fluids are electronically measured using flow sensors placed along each flow line. The various components and specifications of the heat exchanger unit are listed in Table 1.

Table 1
Components and specifications of the base and the heat exchanger units

Specification/Component	Inner Tube	Outer Tube
Material	Stainless steel	Stainless Steel
External diameter (mm)	18	28
Internal diameter (mm)	16	26
Length (mm)	1000	1000
Heat Transfer External area (m^2)	0.0565	0.088
Heat Transfer Internal area (m^2)	0.0503	0.0817
Thickness (mm)	1	
Base unit net weight (kg)	30 kg	
Heat exchanger net weight (kg)	20 kg	
Thermocouples	Type: J Quantity: 6	
Electric Supply	220 V 50 HZ or (110 V 60 Hz)	

4. Uncertainty Analysis

Uncertainty of the measured parameters commonly exists during experimental study. The source of the uncertainty often emerges from the inaccuracy of the various tools, sensors, and devices that are employed during the measurement process. Hence, the recommendation to conduct an uncertainty analysis is essentially justified. The method proposed by Kline and McClintock [32] is used in this research to estimate the experimental uncertainty for Reynolds number (Re) and Nusselt number (Nu).

The uncertainty of the abovementioned parameters is evaluated using the following expressions:

$$\frac{\delta Re}{Re} = \left[\left(\frac{\delta \dot{m}}{\dot{m}}\right)^2 + \left(\frac{\delta D_h}{D_h}\right)^2\right]^{1/2}$$

$$\frac{\delta Nu}{Nu} = \left[\left(\frac{\delta h}{h}\right)^2 + \left(\frac{\delta D_h}{D_h}\right)^2 + \left(\frac{\delta k}{k}\right)^2\right]^{1/2}$$

4.1 Uncertainty of Re

The flow meter used in this experiment has a precision of 0.25% and the heat exchanger dimensions have a precision up to 0.6%. Employing Eq. 15, the uncertainty in Reynolds number would be:

$$\frac{\delta Re}{Re} = [(0.0025)^2 + (0.006)^2]^{1/2} = 0.65$$

4.2 Uncertainty of Nu

First, we have to evaluate the uncertainty of the convection heat transfer coefficient (h):

$$\frac{\delta h}{h} = \left[\left(\frac{\delta Q_{avg}}{Q_{avg}}\right)^2 + \left(\frac{\delta D_h}{D_h}\right)^2 + \left(\frac{\delta L}{L}\right)^2 + \left(\frac{\delta(\Delta T_{lm})}{\Delta T_{lm}}\right)^2\right]^{1/2}$$

where:

$$\frac{\delta Q_{avg}}{Q_{avg}} = \left[\left(\frac{\delta m_h}{m_h}\right)^2 + \left(\frac{\delta(\Delta T_h)}{\Delta T_h}\right)^2 + \left(\frac{\delta m_c}{m_c}\right)^2 + \left(\frac{\delta(\Delta T_c)}{\Delta T_c}\right)^2\right]^{1/2}$$

Hence:

$$\frac{\delta Q_{avg}}{Q_{avg}} = \Big[(0.0025)^2 + \left(\frac{0.1}{3.5}\right)^2 + (0.0025)^2 +$$
0.13.521/2=4.06%

and:

$$\frac{\delta h}{h} = \Big[(0.0406)^2 + (0.006)^2 + (0.006)^2 +$$
0.13.521/2=5.03%

Finally, the uncertainty in Nu is:

$$\frac{\delta Nu}{Nu} = [(0.0503)^2 + (0.006)^2]^{1/2} = 5.07\%$$

The instruments' precisions together with the calculated uncertainties are summarized in Table 2.

Table 2
Accuracy of data processing.

Parameter	Precision and Uncertainty
Flow rate, *m*	±0.25%
Diameter D_h	±0.6%
Length, *L*	±0.6%
Temperature, T	±0.1K%[+]
Convection heat transfer coefficient, *h*	±5.03%[*]
Re	±0.65 %[*]
Nu	±5.07%[*]

5. Validation of Experimental Results

Reliability of the data obtained from experimental equipment are prominent for accuracy and confidence purposes. The overall heat transfer coefficient based on the outer surface area (U_o) and the Number of Transfer Units (NTU) are the target parameters of the validation analysis. Variation of U_o evaluated using temperatures of hot and cold fluids obtained from experimental measurements at different Reynolds numbers is compared with their corresponding values calculated from empirical correlations as follows:

5.1 Actual value of U_o

The overall heat transfer coefficient based on the outside tube area U_o is evaluated by:

$$U_o = \frac{\dot{Q}_{ave}}{A_o \Delta T_{lm}} \tag{12}$$

where:

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \tag{12a}$$

ΔT_1 and ΔT_2 are the temperature difference dependent on the flow configuration.

5.2 Theoretical value of U_o

Gnielinski and Dittus-Boelter correlations are employed to evaluate Nusselt numbers of the hot and cold streams:
Gnielinski correlation:

$$Nu_D = \frac{(f/8)(Re_D - 1000)Pr}{1 + 12.7\left((f/8)^{0.5}(Pr^{2/3} - 1000)\right)} \tag{13}$$

for $0.5 \le Pr \le 2000$ and $3000 \le Re_D \le 5 \times 10^6$
where Darcy friction factor is determined from Petukhov correlation:

$$f = (0.79 \ln(Re_D) - 1.64)^{-2} \tag{13a}$$

Dittus-Boelter correlation:

$$Nu_D = 0.023 Re_D^{0.8} Pr^n \tag{14}$$

for $0.6 \le Pr \le 160$ and $Re_D > 10000$

The convection heat transfer coefficient of both fluids is then determined from:

$$h = \frac{Nu_D \times k}{D_h} \tag{15}$$

Finally, the theoretical overall heat transfer coefficient is calculated using the following expression:

$$\frac{1}{U_o} = \frac{1}{h_o} + \frac{d_o ln(d_o/d_i)}{2k_{wall}} + \frac{d_o}{d_i} \times \frac{1}{h_i} \qquad (16)$$

The number of transfer units (NTU) is selected as a parameter of interest for accuracy investigation of the experimental facilities and is illustrated in Figure (2). The results obtained from the experimental measurement and those predicted by the empirical correlations are in good agreement, giving that the error is in the range of 4% for the Gnielinski correlation and 7% for the Dittus-Boelter correlation. The outcome of the validation revealed the credibility of the experiment facility as well as the measurement techniques.

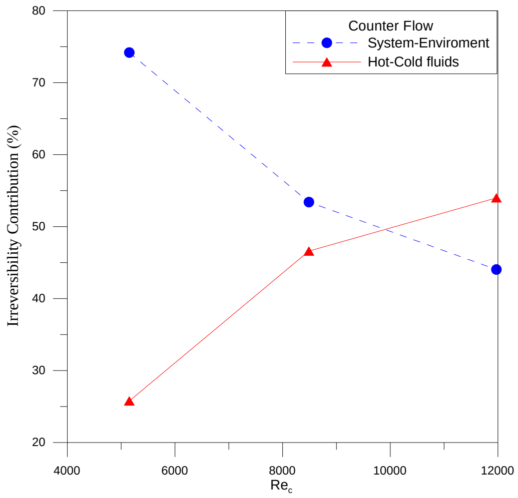


Fig. 2. Comparison of NTU value variation with Reynolds number.

6. Results and discussion

It is well-known from the theory of the heat exchanger that fluid of lower heat capacity rate undergoes greater temperature difference than the fluid with higher heat capacity rate. Since one of the research objectives is to study the behavior of the exchanger near $C_c/C_h = 1$, the volume flow rate of the cold fluid is adjusted to ascertain the magnitude of C_c/C_h ratio is sequentially less than, near, and greater than unity. Hence, increasing the flowrate of the cold stream (increasing Re) doesn't necessarily result in a decrement in the minimum heat capacity ratio (c_{min}), but more specifically, when the increase in the cold fluid flowrate causes c_{min} to change from C_c/C_h to C_h/C_c. Such exchange impacts the heat exchanger performance dramatically. It is worth mentioning that the physical and thermal properties, including specific heat, of both fluids are assumed to be constant during the analysis.

6.1 Exchanger Energy Performance (Effectiveness)

As it is mentioned earlier, the effectiveness of the heat exchanger is directly related to the heat capacity rate of both fluids (C_c and C_h). Figure (3) presents the influence of the heat capacity ratio on the exchanger effectiveness for parallel flow. The figure indicates that the effectiveness drops to its minimum limit as the capacity ratio approaches unity for both theoretical and experimental cases. This trend can be attributed to the fact of the inverse relation between the minimum heat capacity ratio and the number of transfer units, which is one of the prominent factors that affect the effectiveness. When the heat capacity rates of both fluids are close to each other, the potential of transferring energy between the fluids dwindles [26].

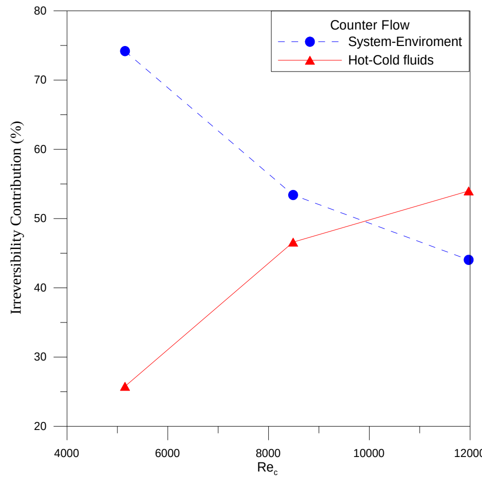


Fig. 3. Comparison of exchanger actual effectiveness with the theoretical corresponding value for parallel flow profile.

6.2 Production of Irreversibilty

The entropy production due to the thermal sources at different flowrates of cold fluid for the two configrations is depicted in Fig. (4).

Evidently, the total irrversibitly augments with the increase in Reynolds number for both profiles. However, the counter flow arrngment produces less irreversibilty compared to the parallel case at the prescribed Reynolds number. Within the range of the cold fluid's Re, the increment in the thermal entropy generation (Eqs. 7a, 7b, 7c) for parallel and counter arrangements approaches 70% and 62%, respectively. Bahiraei et al. [33] reported compatible conculsions in their study on the influence of flowrate on the exergy loss in counter flow heat exchanger. The outcome supported the fact that the overall exergy destruction in the heat exchanger increases by the increase of the flow rate. Begag et al. [19] performed a numerical study on concentric heat exchanger (parallel flow arrangement) to address the effect of hot and cold fluids' mass flow rate on the exergy efficiency and found that the entropy generation continues to augment with increasing mass flow ratio for all cases under study. A numerical study on shell and tube heat exchanger, conducted by Usefian and Bayareh [20], revealed that for a given mass flow rate of inner fluid, the exergy loss enhances as the Reynolds number of the annulus fluid increases. In their study on different types of heat exchangers, Thakur et al. [34] reported that losses due to irreversibility increase as Re increases.

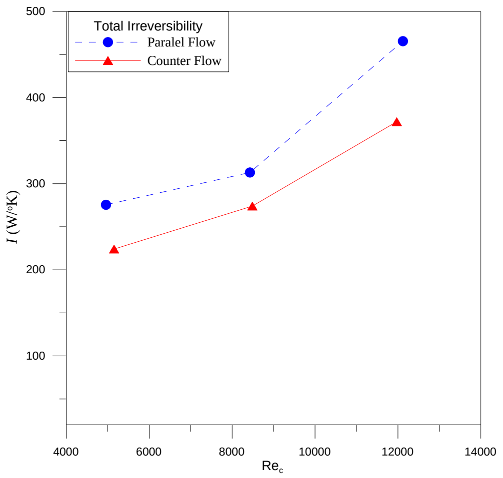


Fig. 4. Total irreversibility generation versus Reynolds number.

However, the two main sources that contribute in irreversibility production of the heat exchanger are: 1) the entropy generation due to the heat transfer from the hot fluid to the cold fluid; and 2) the entropy generation due to the energy interaction with the surrounding atmosphere. One of the objectives of the current study is to evaluate the contribution of each source to the energy destruction. Hence, both sources are calculated for counter flow at various Reynolds number of the cold fluid and presented in Fig. (5).

The figure shows that by tripling the Reynolds number, the relative irreversibility due to heat exchange is augmented by 120%.

In contrast, the contribution of irreversibility due to heat leak to the surrounding drops by 40% at the same range of Reynolds number. Bianco et al. [35] experimental study on the effect of flowrate on the exergy destruction due to heat loss to surrounding supports the outcome of the present work. They observed that greater mass flow rate dwindles the entropy generation due to heat loss to environment. This could be attributed to the reduction in the difference between wall and bulk average temperatures, which causes a decrease in the entropy generation, [35].

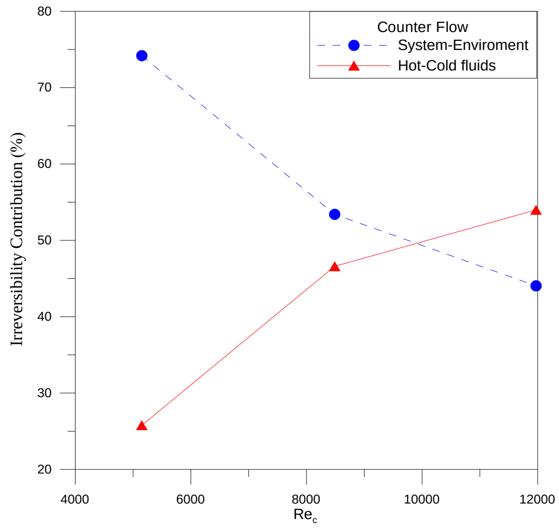


Fig. 5. Contribution of irreversibility thermal sources against Reynolds number for counter flow arrangement.

For parallel flow arrangement (Fig. 6), the increment in irreversibility contribution due to fluid-fluid energy interaction is about 55% while the decrement in the contribution of the system-environment source approaches 22%.

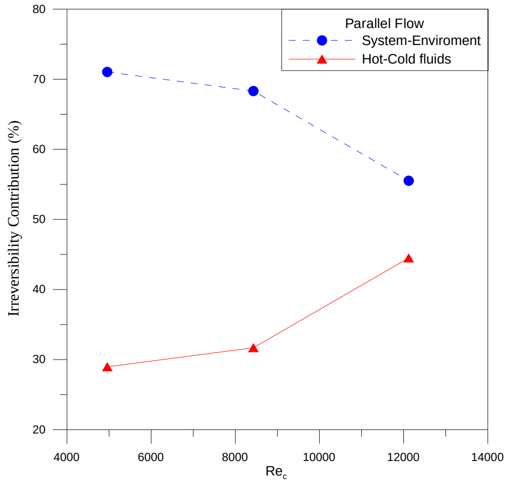


Fig. 6. Contribution of irreversibility thermal sources against Reynolds number for parallel flow arrangement.

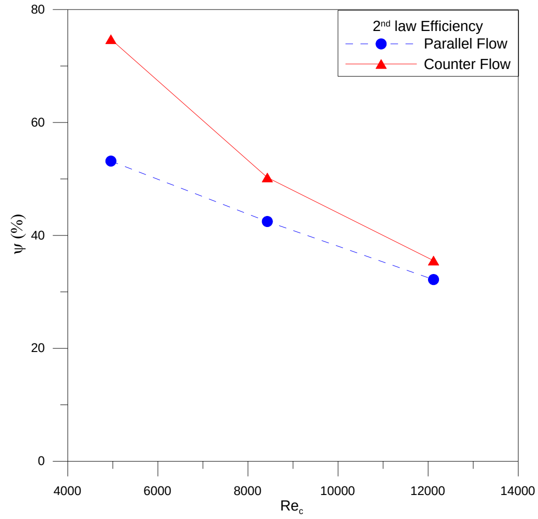


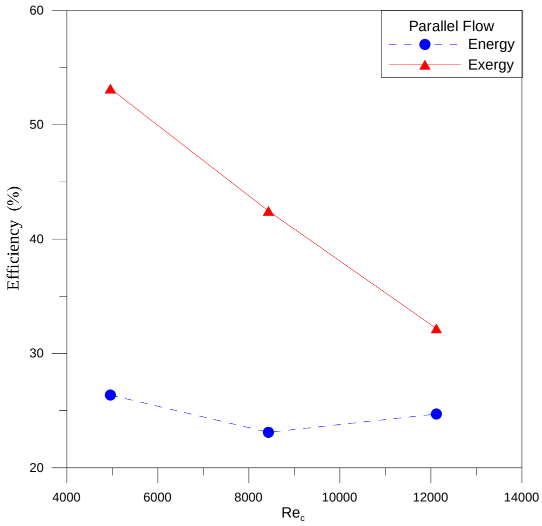
Fig. 7. Exergy efficiency variation against annulus flow Reynolds number.

6.3 Exchanger Exergy Performance

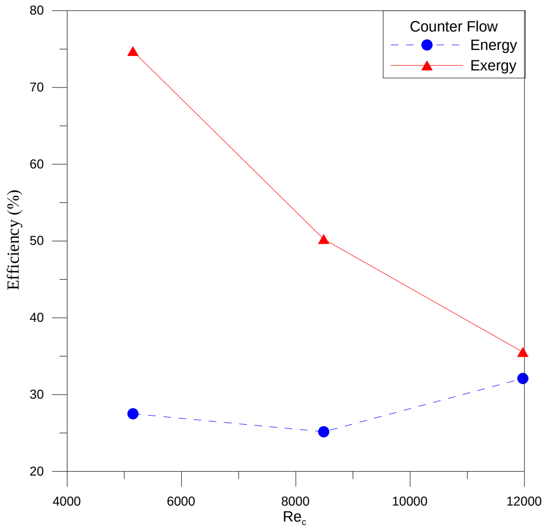
In the process of designing thermal energy systems, optimization analysis emerges as a powerful tool that effectively contributes in evaluating the system performance. Exergy is the foremost optimization criterion used to evaluate the quality of energy and to quantify the rate of deterioration due to irreversibility.

Figure (7) represents the influence of cold-fluid flowrate on the exergy efficiency for parallel and counter flow arrangements. It is obvious from the figure that when the Reynolds number of the annulus side (the cold fluid) is tripled, the system availability drops by 49% for parallel and 52% for counter arrangements. Haseli et al. [27] implemented a complete exergy analysis on the shell and tube heat exchanger and found that exergy efficiency decreases as the mass flow rate increases. The result also agrees with experimental studies on different types of heat exchangers reported by Kumar et al. [36] and Bahiraei et al. [33]. Both studies revealed that exergetic efficiency declines systemically with the increase in Reynolds number.

The first and second efficiency (energy and exergy effectiveness) are consolidated and illustrated in Fig. (8a and 8b) to present the correlation between the efficiencies and the Reynolds number. The analysis showed that although the reduction in exergy efficiency has a linear relation with the Reynolds number of the cold fluid for both arrangements, the energy efficiency has different trends resulting from the direct relation with the heat capacity ratio but not Re.



(a) Parallel flow



(b) Counter flow

Fig. 8. Consolidation of the energy and exergy performance.

6.4 Energy and Exergy degradation

In the theory section, we defined the energy degradation coefficient (τ) as the deterioration in the theoretical effectiveness due to heat interaction with the surrounding. Figure (9) presents the variation of energy degradation coefficient against Reynolds number of the cold fluid for parallel and counter profiles. The Λ-shaped trend observed in the figure substantiates the influence of the heat capacity rate on the energy loss. The peak value in the degradation curve at C_c/C_h~1 occurs as a result of increase in the imbalance in heat exchange between the two streams which causes more heat flow to the surrounding [27].

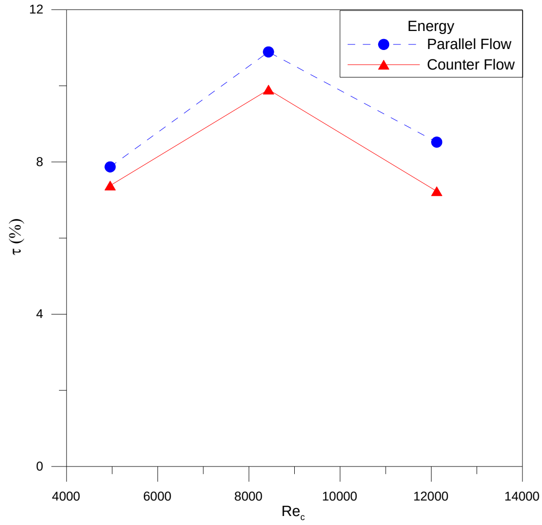


Fig. 9. Energy Degradation Factor (EDF) variation for parallel and counter flow.

Similarly, the exergy degradation coefficient (∈), defined as the deterioration in the theoretical exergy efficiency due to irreversibility production, is demonstrated against Re of the cold fluid in Fig. (10). Unlike the energy degradation coefficient, no peak is observed in the curve of exergy degradation factor, supporting the conclusion that there is no direct impaction of heat capacity ratio on the exergy deterioration reported in the literatures such as [37 and 38].

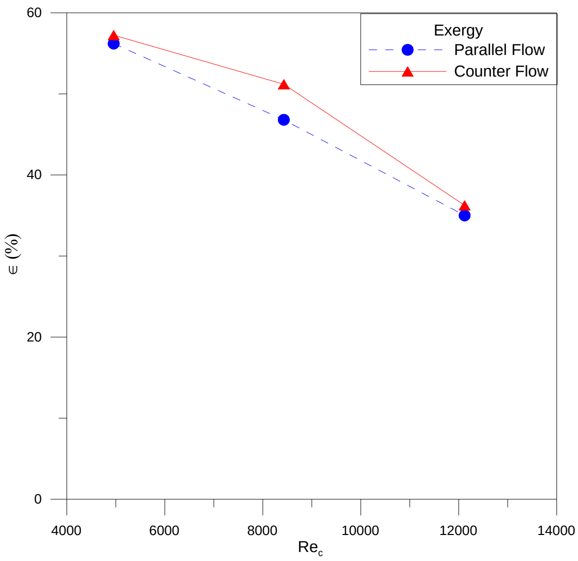


Fig. 10. Exergy Degradation Factor (ExDF) variation for parallel and counter flow.

7. Conclusions

In the present paper, an experimental analysis was carried out to study the effect of Reynolds number, heat capacity ratio, and heat leak on the energy transfer, effectiveness, exergy destruction, and exergy efficiency of a concentric double-tube heat exchanger. The study introduced two new factors, namely: Degradation of Energy and Degradation of Exergy coefficients to measure the deterioration in availability caused by two sources of irreversibility.

The results of the study are summarized as the following:

- Observation found that the performance when the exchanger operates as counter-flow is better than parallel flow case for all flowrates under consideration.
- A V-shaped variation of the thermal effectiveness of the exchanger was discerned with the increase in the flowrate.
- No direct relation is observed between minimum capacity ratio (c_{min}) and exergy.
- 1st effectiveness is directly related to the heat capacity ratio of the fluid rather than Re.
- With the increase in the Reynolds number of the annulus fluid, the exergy destruction due to heat leak to the surrounding decreases while the exergy destruction due to fluid-fluid heat transfer increases.
- The total exergy destruction augments with the increase of Reynolds number since the increment in the exergy destruction due to heat interchange between the fluids is estimated to be higher than the decrement in exergy loss resulting from the heat lost to the environment.
- It is observed that the total exergy destruction of parallel configuration expanded by 70% when the flowrate of the cold fluid triples. Counter flow analysis showed a 62% increment in the exergy destruction for the same range of flowrate.
- The exergy efficiency is declined by 49% for parallel-flow and 52% for counter-flow with the tripling of the Reynolds number.
- The energy degradation factor took a Λ-shaped variation against Reynolds number for both parallel and counter profiles.
- The exergy degradation factor continuously decreased with the increase in the Reynolds number.

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