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INVESTIGATION ON THE RELATION BETWEEN EXERGY LOSS AND NTU

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Abstract: In this study, the relation between exergy losses and heat flux and their ratio are expressed as a dimensionless number for counter and parallel flow heat exchangers. The relation between the proposed dimensionless number and the number of transfer units (NTU) is formulated as a function of heat capacity and inlet temperatures. The findings based on this research are scrutinized via diagrams both as parametric case and experimental studies. The proportion of exergy losses to heat flux increases with a decreasing heat capacity rate. It has been found that the counter-flow heat exchanger has a lower exergy loss than the parallel one for the same heat exchanger. It was seen that when nanoparticles were added in the fluid used in the exchanger, the effectiveness slightly increased and therefore the necessary heat transfer area decreased. It was also observed that under condensing or boiling states the effectiveness reached its top level. Moreover, it was found that for increasing values of the temperature rates of hot and cold fluids the exergy losses also increased. The model and findings are supported both by a parametric analysis and experimental data revealed in a detailed table in the paper.
Keywords: Heat transfer, Heat exchanger, NTU, Effectiveness, Exergy loss.

EKSERJİ KAYBI VE NTU ARASINDAKİ İLİŞKİNİN İNCELENMESİ

Özet: Bu çalışmada karşıt ve paralel akışlı ısı değiştiricilerinde ekserji kaybı ve ısı akısı arasındaki ilişki ve bunların oranları bir boyutsuz sayı ile ifade edilmektedir. Oran şeklindeki bu boyutsuz sayı ile NTU arasındaki ilişki; ısıl kapasiteleri ve giriş sıcaklıklarının fonksiyonu olarak formüle edildi. Parametrik durum ve deneysel çalışmalar ile elde edilen bulgular diyagramlar aracılığıyla irdelenmiştir. Ekserji kayıplarının ısı akısına oranı, azalan ısı kapasitesi oranı ile artmaktadır. Karşıt akışlı ısı değiştiricisinin, aynı ısı değiştiricisi için paralel akışlı ısı değiştiricisinden daha düşük bir ekserji kaybına sahip olduğu tespit edilmiştir. Isı değiştiricisinde akışkana nanopartikül ilave edildiğinde, etkenlik hafifçe artar ve bu nedenle gerekli ısı transfer alanı azalır. Ayrıca yoğuşma veya kaynama durumlarında etkenliğin en üst seviyeye ulaştığı gözlemlenmiştir. Bunun yanında, sıcak ve soğuk akışkanların sıcaklıkları oranının artmasıyla ekserji kayıplarının da arttığı tespit edilmiştir. Bu çalışmada model ve bulgular hem parametrik bir analiz hem de detaylı bir tablo ile ortaya koyulan deneysel veriler ile desteklenmektedir.
Anahtar Kelimeler: Isı Transferi, Isı Değiştirici, NTU, Etkinlik, Ekserji Kaybı

Symbols

A	heat transfer area of a heat exchanger $[m^2]$
c_p	specific heat capacity at constant pressure $[J/kgK]$
C	heat capacity $[W/K]$
C_r	heat capacity ratio
e	dimeonsionless exergy loss rate, $[=\dot{E}/T_o C_{min}]$
$\dot{E}$	exergy loss $[W]$
h	specific enthalpy $[J/kg]$
$\dot{m}$	mass flow rate $[kg/s]$
NTU	number of transfer units
q	dimensionless heat flux rate, $[=\dot{Q}/C_{min}T_{c,i}]$
$\dot{Q}$	The amount of heat transfer $[W]$
s	specific entropy $[J/kgK]$
S	entropy $[J/K]$
T	absolute temperature $[K]$
T_o	ambient temperature $[K]$
U	overall heat transfer coefficient $[W/m^2K]$
$\dot{W}$	work [W]
ε	effectiveness

Subscripts

c	cold fluid
h	hot fluid
i	inlet

o	outlet
r	ratio
min	minimum
max	maximum

INTRODUCTION

Handling the design of heat exchangers solely in terms of investment costs is not sufficient. From a general point of view, one of the most important parameters affecting heat exchanger design is energy. For investment cost to be lowered, temperature difference should be greater. Moreover, the investment cost also varies depending on the type of heat exchangers.

Due to the importance of heat exchangers, remarkable studies can be found in the literature and this area has interested many researchers over the years.

Heat flows from hot to cold fluid in heat exchangers. If the temperature difference between the hot and cold fluid increases, then the heat transfer and exergy loss also increase, whereas the first investment cost of the heat exchanger decreases. Providing that fluid inlet temperatures and NTU value are constant, the heat flow and exergy losses vary according to the type of heat exchanger. For a heat exchanger to be evaluated profitable, it should transfer the maximum amount of heat and lose the minimum amount of exergy per its surface area or initial investment cost. In the literature, the exergy loss of some heat exchangers has been investigated for different purposes by Paykoc and Yuncu (1986). Moreover, the relation between dimensionless exergy loss and the heat flow and the ratio of them were found by Teke et al. (1993) and Kincay and Temir (1993). In this study, the relation between this ratio and NTU for counter-flow and parallel-flow heat exchangers was investigated, and then the most economical heat exchanger type was determined according to the exergy loss.

Sozen (2001) investigated the effects of refrigerant heat exchangers and mixture heat exchangers on the system performance in an aqua-ammonia absorption refrigeration system for three diffent cases: both of the heat exchangers are included, only the refrigerant heat exchanger is included, and only the mixture heat exchanger is included in the system. Taken into consideration were the coefficient of performance, exergetic coefficient of performance, circulation ratio (f), and non-dimensional exergy loss of each component of the system. In this study, Sozen (2001) exhibited the exergy losses for both heat exchangers for three cases and each component of the system obtained from the first and second law analyses.

Yilmaz et al. (2001) studied the the second-law-based performance evaluation criteria to evaluate the performance of heat exchangers. In this study,the classifications of second-law-based performance evaluation criteria for heat exchangers were grouped into two classes: those that use entropy as evaluation criteria and those that use exergy as evaluation criteria. Both classes were collectively presented, reviewed, and discussed. The importance of second-law-based thermoeconomic analysis of heat exchangers was emphasized and discussed briefly.

Alquaity et al. (2013) studied the closed form relations for parallel-flow heat exchangers, when the heat leak is either on the cold or hot side of the heat exchanger in the presence of kinetic energy variations. The results of this study were presented in a graphical form demonstrating the variation of effectiveness of the heat exchanger with the relevant parameters. It was also demonstrated that in limited case, the solution reduces to classical effectiveness and NTU relations for parallel-flow heat exchangers.

The study of Hajabdollahi et al. (2012) contributed to the literature with a multi-objective, exergy-based optimization through a genetic algorithm method to improve the performance of shell-and-tube type heat recovery heat exchangers by considering two key parameters: exergy efficiency and cost. The design parameters were tube arrangement, baffle cut ratio, tube pitch ratio, tube length, tube number and baffle spacing ratio, as well as 20 standard tubes with definite inner and outer dimaeters. It was concluded that when the exergy efficiency of a shell and tube heat exchanger increased, the total cost of heat exchanger increased as well.

In Ağra’s study (2011), a new model was formed to determine the size and the best type of heat exchanger for waste-heat-recovery systems whose technical specifications were known and available in the market at the desired saving-investments ratio. The TYO was defined as the ratio of savings obtained from the heat exchangers and the investment cost. TYO was calculated depending on investment-saving potantial ratio (E), NTU, and effectiveness. E was calculated based on technical and economical parameters of the system. As a result, for a given saving-investment ratio, effectiveness was determined one of the most important criteria in choosing the best type of heat exchanger.

Dealing with heat exchangers that have uniform wall temperatures, especially condensers and evaporators, Hermes (2012) combined the ε-NTU and EGM (entropy generation minimization) approaches used to design heat exchangers. As a result, a formulation that shows the dimensionless rate of entropy generation as a function of operating conditions, and the geometry of the heat exchanger, was produced. From the study, it was deduced that there had been a specific number of transfer units that diminished the dimensionless rate of entropy generation. It was also concluded that aiming for high effectiveness in heat exchangers did not necessarily result in the best thermal and hydraulic performance.

Pandey and Nema (2011) proposed two methods to avoid entropy generation in heat exchangers and conducted a study in an experimental setup that was made up of a three channel 1-1 pass plate heat exchanger. As result of the study, it was found that exergy loss in the sinusoidal plate heat exchanger was lower than the rectangular wavy plate heat exchanger under studied flow conditions and might be attributed to less turbulence and beter solid-fluid contact.

Teke et. al. (2010) suggested a new model to determine the area and type of the waste heat recovery heat exchanger to acquire maximum gain. To achive this, a dimensionless E number was defined based on various known technical and economic parameters. The dimensionless E numbers were illustrated in graphics as a function of NTU and ratio of heat capacities and hence corresponding heat exchanger area giving maximum net gain werereached through the graphics. Moreover, the most appropriate exchanger type and its area were chosen by comparing the gains or effectiveness of the heat exchangers at NTU max.

Wang et. al. (2010) developed the configuration of a shell and tube heat exchanger by installing sealers into the shell. The results of the experiments have shown that the heat transfer coefficient in the shell side increases 18.2%-25.5%, overall heat transfer coefficient increases 15.6%-19.7%, and exergy efficiency increases 12.9%-14.1%. Consequently, the performance of the heat exchanger has risen and it can be stated that the improvement carries significant benefits in terms of energy conservation.

Naphon (2011) carried out both experimental and theoretical investigations on the entropy generation and exergy loss of a horizontal concentric micro-fin tube heat exchanger. The experiments within the study were conducted for hot and cold water mass flow rates within the range of 0.02-0.10 kg/s, while the inlet hot water and inlet cold water were between 40°C-50°C and between 15°C-20°C, respectively, and the influence of these parameters over entropy generation and exergy loss was examined.

Johannesson et. al. (2002) presented a proof that entropy production owing to heat transfer in heat exchangers is minimum if entropy production is constant in all parts of the system.

Mathew and Hegab (2010) investigated the thermal performance of paralel flow microchannel heat exchangers exposed to constant external heat transfer. Equations were derived to determine the axial temperatures and the effectiveness of the fluids of the microchannel heat exchanger. It was found that regardless of the heat capacity ratio, for a specific NTU, external heating decreased and increased the effectiveness of the hot and cold fluids, respectively. In addition, it was seen that at a specific NTU, a decrease in the heat capacity ratio advanced the effectiveness of the fluids.

Guo et. al. (2009) produced a new optimization approach for shell and tube heat exchangers in which the dimensionless entropy generation was obtained by increasing the entropy generation on the ratio of the heat transfer rate to the inlet temperature of cold fluid that was utilized as the objective function. As a result of the paper, it is observed that the optimization design has increased the heat exchanger's effectiveness remarkably and decreased pumping capacity.

Although there are remarkable amount of studies about second law analysis of the heat exchangers in the literature, only a few of them are related to exergy loss, heat transfer, and the selection of the heat exchangers at the same time. Therefore, the aim of this study is to present analytically the relation between exergy loss and heat and their ratio as a dimensionless number for counter and parallel flow heat exchangers. The findings based on this study were evaluated as a case study. It was seen that the exergy loss of heat transfer per unit increased with an increasing heat capacity rate. It was also found that the counter-flow heat exchanger has a lower exergy loss than the parallel one for the same heat exchanger unit. Moreover, the influence of nanoparticles (Al2O3 and TiO2) in parallel and counter flow heat exchangers over exergy losses and heat transfer area was investigated.

THEORETICAL MODEL

The conventional designing method for heat exchangers is based on results of the first law of thermodynamics and the equation of mass conservation. Nevertheless, designers of heat exchangers should consider exergy equivalences as well.

Part of thermodynamics points out that all processes result in entropy generation that indicates irreversibilities that deteriorate thermal efficiency. Thus, irreversibilities are the most effective reasons of heat exchangers' decline of performance. Three remarkable sources of entropy generation (irreversibility) arises from: a) finite temperature difference, b) fluid mixing, and c) fluid friction. Exergy analysis helps us to appraise the quality of heat transfer that occurs in heat exchanger.

The exergy loss for an open system with multiple inlets, multiple exits and under steady state conditions is given by Incropera, D.P. De Witt (1985):

$$\dot{E} = \sum \dot{m}_i \left(h_i - T_o s_i\right) - \sum \dot{m}_o \left(h_o - T_o s_o\right) + \sum \dot{Q}\left(1 - \frac{T_O}{T}\right) - \dot{W} \quad (1)$$

The exergy loss for an isolated heat exchanger can be expressed as:

$$\dot{E} = \dot{m}_c \left(h_{c,I} - T_o s_{c,I}\right) + \dot{m}_h \left(h_{h,I} - T_o s_{h,I}\right) - \dot{m}_c \left(h_{c,o} - T_o s_{c,o}\right) - \dot{m}_h \left(h_{h,o} - T_o s_{h,o}\right)$$

(2)

For collecting the entropy terms together, we can also rearrange Eq. (2) as:

$$\dot{E} = \dot{m}_c\left(h_{c,i} - h_{c,o}\right) + \dot{m}_h\left(h_{h,i} - h_{h,o}\right) + T_o\left[\dot{m}_c\left(s_{c,o} - s_{c,i}\right) + \dot{m}_h\left(s_{h,o} - s_{h,i}\right)\right] \quad (3)$$

The energy transfer from hot fluid to the cold one is equal, as shown in Eq. (4).

$$\dot{Q} = \dot{m}_h\left(h_{h,i} - h_{h,o}\right) = -\dot{m}_c\left(h_{c,i} - h_{c,o}\right) \quad (4)$$

Therefore, exergy loss equations are obtained as:

$$\dot{E} = T_o\left[\dot{m}_c\left(s_{c,o} - s_{c,i}\right) + \dot{m}_h\left(s_{h,o} - s_{h,i}\right)\right] \quad (5)$$

The entropy differences for hot and cold fluids are derived by means of using specific heats are as shown in Eqs. (5) and (8).

$$s_{c,o} - s_{c,i} = c_{p_c} \ln\left(T_{c,o} / T_{c,i}\right) \quad (6)$$

$$s_{h,o} - s_{h,i} = c_{p_h} \ln\left(T_{h,o} / T_{h,i}\right) \quad (7)$$

$$\dot{E} = T_o\left[\dot{m}_h c_{p_h} \ln\left(T_{h,o} / T_{h,i}\right) + \dot{m}_c c_{p_c} \ln\left(T_{c,o} / T_{c,i}\right)\right] \quad (8)$$

The heat capacity of hot and cold fluids is defined as:

$$C_h = \dot{m}_h c_{p_h} \text{ and } C_c = \dot{m}_c c_{p_c} \quad (9)$$

The fluid, having minimum heat capacity (Cmin), is assumed as the hot one, and then the exergy loss can be written as

$$\dot{E} = T_o\left[C_{min} \ln\left(T_{h,o} / T_{h,i}\right) + C_{max} \ln\left(T_{c,o} / T_{c,i}\right)\right] \quad (10)$$

The energy rates for hot and cold fluids are shown in Eq. (11) as:

$$\dot{Q} = C_{min}\left(T_{h,i} - T_{h,o}\right) = C_{max}\left(T_{c,o} - T_{c,i}\right) \quad (11)$$

Due to the assumption of the hot fluid's minimum situation, the ideal heat transfer is the required one for the duration of hot-fluid inlet temperature's cooling to the cold-fluid inlet temperature value (The duration of the ideal heat transfer is the required duration for cooling of hot-fluid inlet temprerature to cold-fluid inlet). Thus, the heat exchanger effectiveness, ε, is defined in Eq. (12) as:

$$\varepsilon = C_{min} \ln\left(T_{h,i} / T_{h,o}\right) / C_{min}\left(T_{h,i} / T_{c,i}\right) \quad (12)$$

The temperature rates of hot and cold fluids are derived from Eqs. (11) to (13) as shown in Eqs. (14) and (15).

$$T_r = T_{h,i} / T_{c,i},\ C_r = C_{min} / C_{max},\ \left(0 \le C_r \le 1\right) \quad (13)$$

$$T_{h,o} / T_{h,i} = 1 - \varepsilon\left(1 - 1/T_r\right) \quad (14)$$

$$T_{c,o} / T_{c,i} = 1 + \varepsilon C_r\left(T_r - 1\right) \quad (15)$$

The expression of exergy loss per unit of time can be obtained by means of the replacement of temperature rates in Eqs. (14) and(15) in Eq. (10) as:

$$\dot{E} = T_o\left[C_{min} \ln\left(1 - \varepsilon\left(1 - 1/T_r\right)\right) + C_{max} \ln\left(1 + \varepsilon C_r\left(T_r - 1\right)\right)\right] \quad (16)$$

If the equation (16) is divided by $\left(T_o C_{min}\right)$, the dimensionless exergy loss expression can be calculated as a function of ε, C_r and T_r in Eq. (17).

$$e = \frac{\dot{E}}{T_o C_{min}} = \ln\left[1 - \varepsilon\left(1 - 1/T_r\right)\right] + \frac{1}{C_r}\left[1 + \varepsilon C_r\left(T_r - 1\right)\right] \quad (17)$$

The amount of heat transfer can be expressed in terms of effectiveness as follows:

$$\dot{Q} = \varepsilon C_{min}\left(T_{h,i} - T_{c,i}\right) = -\dot{m}_c\left(h_{c,i} - h_{c,o}\right) \quad (18)$$

If the two sides of Eq. (18) are divided by $\left(C_{min} T_{c,i}\right)$, the dimensionless heat flux is obtained as:

$$q = \frac{\dot{Q}}{C_{min} T_{c,i}} = \varepsilon\left(T_r - 1\right) \quad (19)$$

If the dimensionless exergy loss and dimensionless heat flux are shown as the proportion to each other, a dimensionless number is obtained as:

$$\frac{e}{q} = \frac{\ln\left[1 - \varepsilon\left(1 - 1/T_r\right)\right] + \left[1/C_r\right]\left[1 + \varepsilon C_r\left(T_r - 1\right)\right]}{\varepsilon\left(T_r - 1\right)} \quad (20)$$

The number of transfer units (NTU) is defined as a ratio of overall conductance to the tube-side fluid heat capacity rate given by

$$NTU = UA / C_{min} \quad (21)$$

The uncertainty of Eq. (20) can be eliminated with the limit expression

$\lim_{C_r \to 0}$ as

$$\frac{e}{q} = \frac{\ln\left[1 - \varepsilon\left(1 - 1/T_r\right)\right] + 1}{\varepsilon\left(T_r - 1\right)} \quad (22)$$

The effectiveness equations are calculated for counter and parallel flow heat exchangers as shown in Eqs. (23) and (24), respectively.

$$\varepsilon_{counter} = \frac{1 - \exp\left[-NTU(1 - C)\right]}{1 - C \cdot \exp\left[-NTU(1 - C)\right]} \quad (23)$$

$$\varepsilon_{parallel} = \frac{1 - \exp\left[-NTU(1 + C)\right]}{1 + C} \quad (24)$$

The ε and NTU values are equal to each other at the same limit expression for parallel and counter-flow heat exchangers $\left(C_r = 0\right)$ as given in Eqs. (25) and (26). Additionally,the effectiveness is described by Eqs. (27) and (28) for the heat capacity ratio as $C_r = 1$.

$$\varepsilon_{parallel} = \varepsilon_{counter} = 1 - \exp\left(-NTU\right) \quad (25)$$

$$\mathrm{NTU}_{parallel} = \mathrm{NTU}_{counter} = -\ln - (1-\varepsilon) \quad (26)$$

$$\varepsilon_{counter} = \frac{\mathrm{NTU}}{1+\mathrm{NTU}} \quad (27)$$

$$\varepsilon_{parallel} = \frac{1}{2}\left(1-\exp\left(-2\cdot \mathrm{NTU}\right)\right) \quad (28)$$

The e/q expression in Eq. (22) takes the following form by eliminating the expression of uncertainty with the limit term, $\lim\limits_{\varepsilon \to 0}$

$$\frac{e}{q} = 1 - \frac{1}{T_r} \quad (29)$$

The relation between e/q and NTU dimensionless numbers, which are in the form of rates, is analyzed as ratio of heat capacity that varies from 0.0 to 1.0 and the ratio of inlet temperatures varies from 1.2 to 2.0 for parallel and counter flow heat exchangers.

For Cr =0 and 1, the relation between e/q and NTU for both types of heat exchangers is presented in the figures.

EXPERIMENTAL SETUP

The illustration of the experimental facility is given in Fig. 1. The system contains a temperature-controlled inlet tank, a temperature controlled outlet tank, and a test section which is a double pipe heat exchanger. The nanofluid's temperature is attained before arriving to the test section with cooling coils and a heater. Then, a pump is used for transportation of the nanofluid to the test section. The temperature of nanofluid is measured at four points in the inlet of the test section and at four points in the outlet of it, using T-type thermocouples with an accuracy of 0.1 °C. After heating, the nanofluid enters the temperature-controlled outlet tank to circulate. The detailed information on the experimental setup, preparation of nanofluids and determination of their physical properties can be seen from authors' previous publications. (Nitiapiruk et al. (2013), Duangthongsuk and Wongwises (2008), Duangthongsuk and Wongwises (2009), Duangthongsuk and Wongwises (2009), Duangthongsuk and Wongwises (2010), Duangthongsuk and Wongwises (2010)).

A CASE STUDY: EVALUATION FOR COUNTER AND PARALLEL FLOW HEAT EXCHANGERS

In a heat exchanger, the inlet temperatures of hot and cold fluids are assumed as 84.6 °C and 15°C, respectively, and taken as the flow rates of 6000 kg/h and 8000 kg/h, respectively. If the specific heat is at a constant pressure, the overall heat transfer coefficient and the heat exchanger area are assumed to be constant, and an examination finds the minimum exergy loss in the heat exchanger. The calculated results for NTU analysis and details of calculations can be seen step by step as shown below.

NTU can be found from Eq. (21) as:

$$\mathrm{NTU} = \mathrm{UA}/\mathrm{C}_{min} = \frac{\left(1200\ \mathrm{W/m^2K}\right)\cdot\left(23\ \mathrm{m^2}\right)}{6976\ \mathrm{W/K}} = 3.956$$

Inlet temperature ratios of hot and cold fluids can be calculated using Eq. (13) as:

$$\mathrm{T}_r = \mathrm{T}_{h,i}/\mathrm{T}_{c,i} \quad \text{and} \quad \mathrm{T}_r = \left(\frac{273+84.6}{273+15}\right) = 1.242$$

The heat capacity rate can be calculated from Eq. (13) as:

$$\mathrm{C}_r = \mathrm{C}_{min}/\mathrm{C}_{max} \ \text{ and } \ \mathrm{C}_r = \left(\frac{6976\ \mathrm{W/K}}{9302\ \mathrm{W/K}}\right) = 0.75$$

The heat exchanger effectiveness [3] can be calculated by calculations as:

$$\varepsilon_{parallel} = 0.571 \ \text{ and } \ \varepsilon_{counter} = 0.871$$

The values of the dimensionless number e/q are calculated according to the results above.

RESULTS AND DISCUSSION

In this study, the relation between dimensionless exergy losses and heat flow and the rate of them that can be applied to all the heat exchangers was found [2]. The relation between this dimensionless rate and NTU was investigated for parallel and counter-flow heat exchangers. The minimum exergy loss for a heat exchanger type is the counter flow one used in an application shown in Fig. 2 and Fig. 3.

Figs. 2a and 2b show that when Al2O3 and TiO2 have been added in the fluid, Cr markedly decreases and thus efficiency (ε) slightly increases. Also, as expected, (ε) is higher under counter-flow conditions compared to parallel-flow conditions. In addition, from Fig 2b, it is understood that ε varies less at parallel-flow preference compared to counter-flow conditions at rising NTU.

Fig 3a illustrates that when condensing or boiling occurs, (Cr= 0) efficiency (ε) has reached its highest level and it also shows that under counter flow conditions (Cr= 1) efficiency has been found to be higher than parallel flow. From Fig. 3b, it is deduced that Cr and consequently the heat transfer area of the heat exchanger diminish when nanoparticles (Al2O3 and TiO2) have been included in the water at a ratio of 4%.

In Figs. 4a and 4b for (Cr= 0) and (Cr= 1), respectively, the exergy losses per unit heat flux are illustrated. The figures demonstrate that for increasing NTU values and under the (Cr= 1) condition, exergy losses per unit heat flux (e/q) draw more rapid decreases compared to the condition (Cr= 0). Additionally, it has been observed that for increasing Tr values exergy losses increase.

In Figs. 5a and 5b under counter- and parallel-flow conditions, respectively, and for (Cr= 0.75 and 1) and (Tr= 1.2-2), exergy losses per unit heat flux (e/q) are presented. According to these figures, it is observed that when (Cr= 1) exergy losses show more rapid falls and become less. It is also seen that for increasing Tr values exergy losses are higher.

Figs. 6a and 6b illustrate the exergy losses per unit heat flux (e/q) in a heat exchanger that performs a counter flow that contains Al2O3 and TiO2, respectively, in the ratio of 4%. From the figures, it can be seen that for (Cr= 1) and for decreasing values of Tr, exergy losses per unit heat flux (e/q) show more rapid falls compared to the condition (Cr= 0.66).

Exergy losses per unit of heat flux in a parallel flow heat exchanger that uses water that contains Al2O3 and TiO2 in a ratio of 4% are given in Figs. 7a and 7b, respectively. According to these figures, as Cr increases, exergy losses decrease, and as Tr increases, exergy losses increase. Furthermore, it is understood that in parallel flow heat exchangers, exergy losses per unit heat flux do not draw rapid changes and keep constant after a definite point, compared to the counter flow heat exchangers (Figs. 6a and 6b).

Figures of parametric case study are shown from Figs. 2 to 7. Experimental data, shown in Table 1 for tube side measurements of pure water and nanofluid flow having TiO2 particles and Table 2 for annulus side measurements for pure water flow of counter flow heat exchanger, are used for the preparation of Figs. 8 and 9. As it can be seen from Fig. 8a, addition of nanoparticles increases the total heat transfer coefficient and effectiveness values. There is a consensus in the literature on volumetric concentration of nano particles in its base fluid as 4% due to its known negative specifications as aggregation and sedimentation. It is also known that not only nano particles enhance heat transfer coefficients but also they increase pressure drop, and cause some increase in pumping power, in other words, energy consumption. From Figs. 9a and 9b, alteration of dimensionless exergy loss rate and heat flux rate with Reynolds number are shown according to pure and nanofluids having 0.2% and 1% nano particles. The charactestics of trend lines supports the parametric findings partly. It should be noted that the trend lines are not linear due to the effects of transition regime in tube flow and similar trend lines for friction factors especially regarding this flow regime exist in the literature.

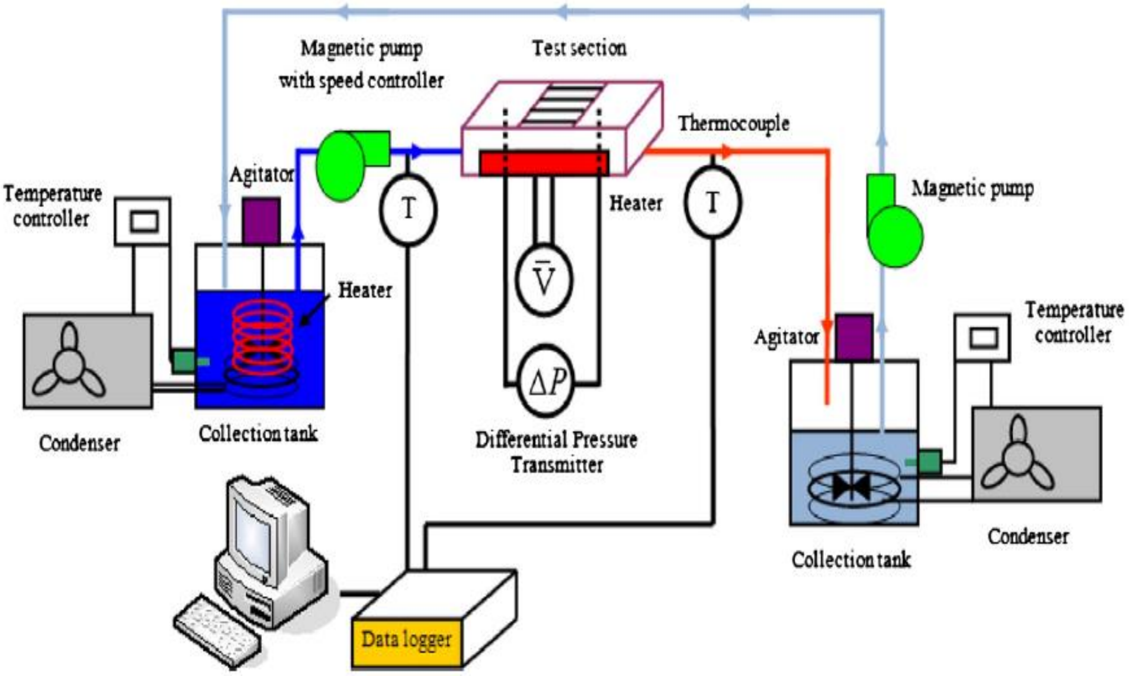

Figure 1. Schematic diagram of experimental apparatus [From Nitiapiruk et al. (2013), with permission from Elsevier.]

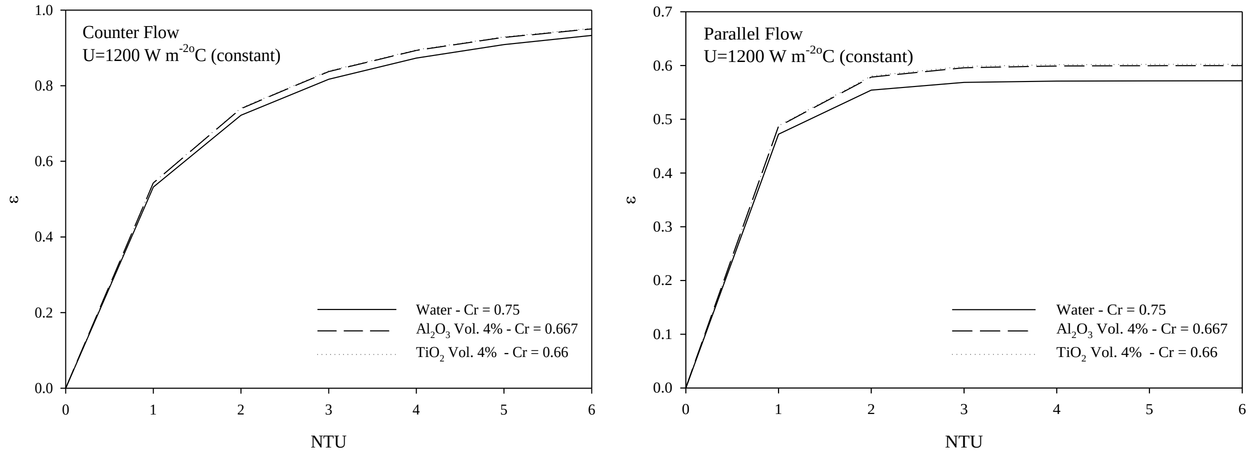

Figure 2. Efficiency vs NTU for counter (a) and parallel flows (b)

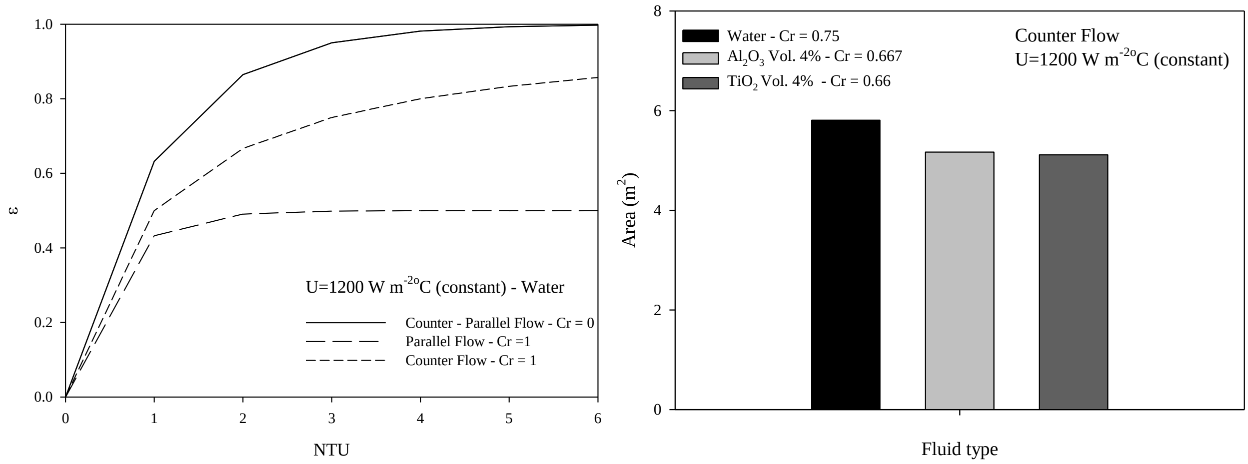

Figure 3. Efficiency vs NTU (a) and calculated surface area according to fluid types (b)

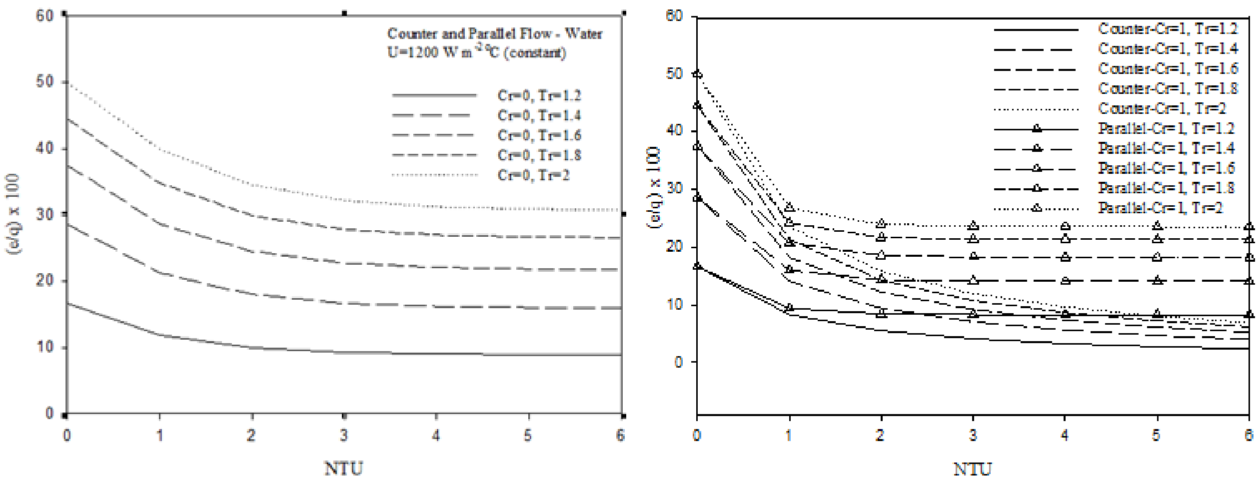

Figure 4. (e/q) values vs NTU are the same for both heat exchangers according to Cr = 0 (a) and Cr = 1 (b) for the counter flow and parallel flow of water

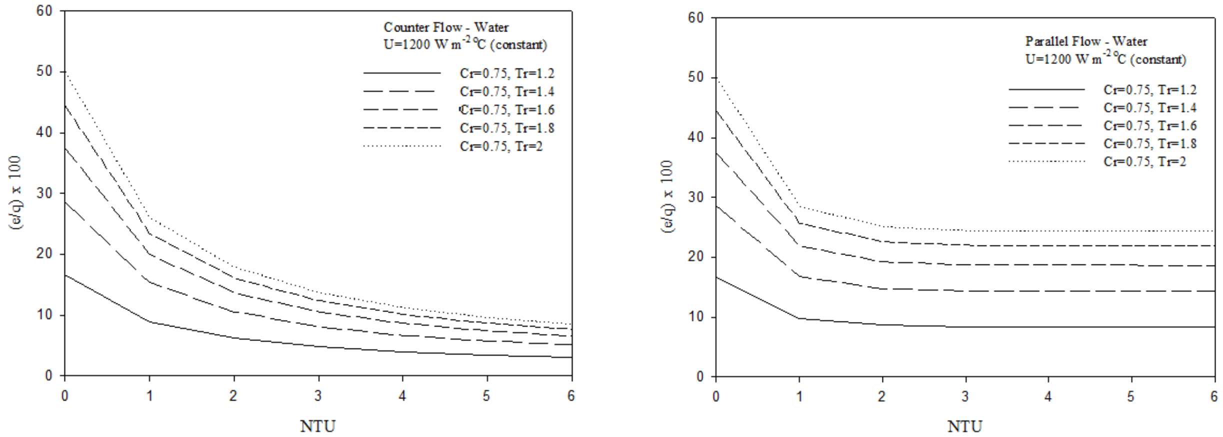

Figure 5. (e/q) values vs NTU according to water flow for counter (a) and parallel (b) flows

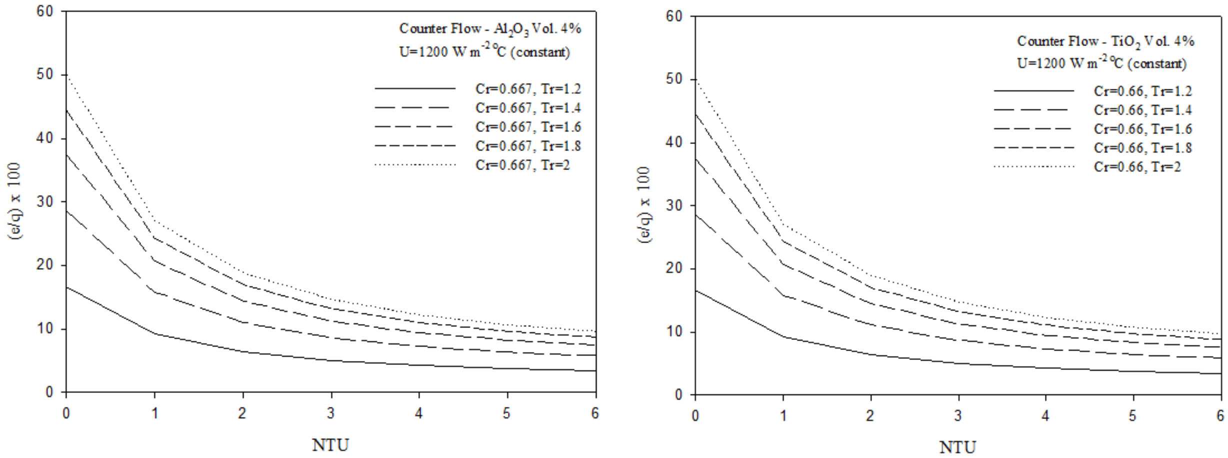

Figure 6. (e/q) values vs NTU according to nanofluid flow including Al2O3 (a) and TiO2 (b) nanoparticles for counter flow

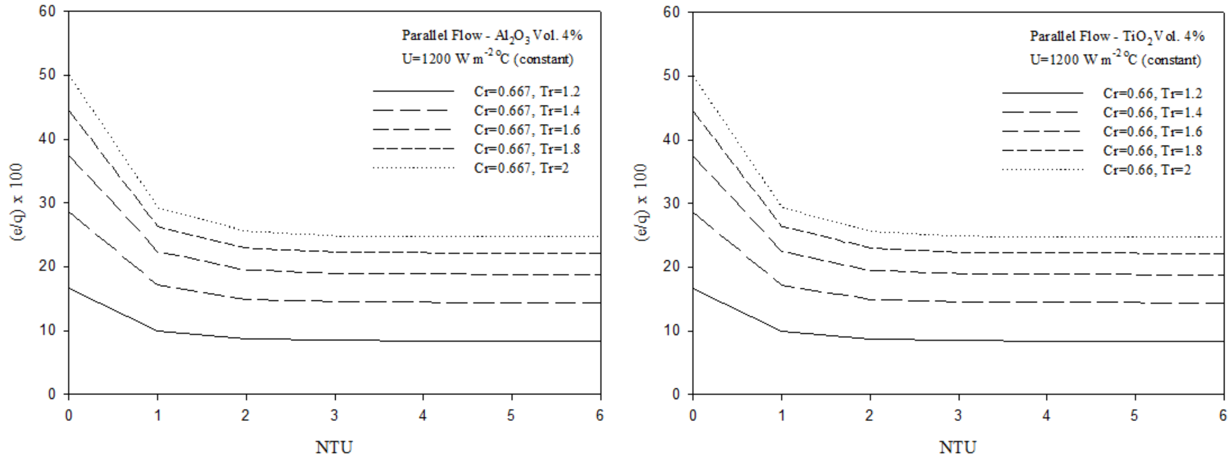

Figure 7. (e/q) values vs NTU according to nanofluid flow including Al2O3 (a) and TiO2 (b) nanoparticles for parallel flow

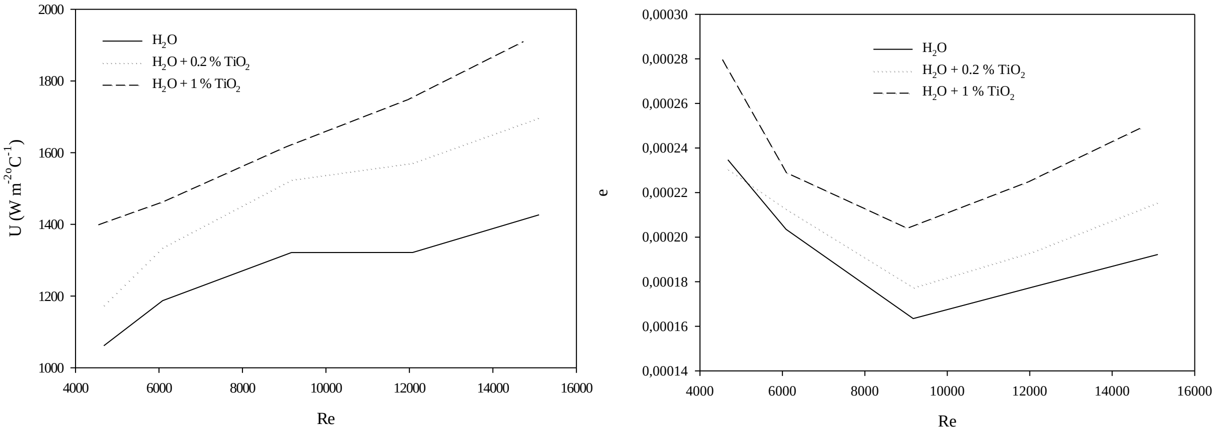

Figure 8. Heat transfer enhancement on total heat transfer coefficient (a) and effectiveness according to Reynolds number

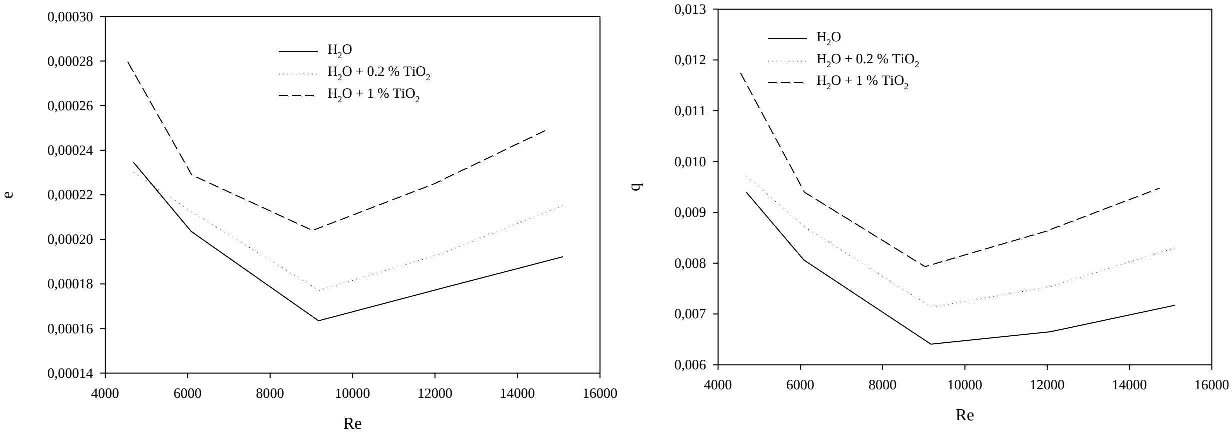

Figure 9. Alteration of dimeonsionless exergy loss rate (a) and heat flux rate (b) with Reynolds number

Table 1. Tube side measurements of pure water flow and nanofluid flow having TiO2 particles

Exp. No.	Particle Concentration (% weight)	Tube Diameter	Tube length	Velocity	Kinematic viscosity	Dynamic viscosity	Density	Thermal Conductivity	Specific Heat	Fluid Inlet Temperature	Fluid Outlet Temperature	Reynolds Number
	ϕ	d (m)	L (m)	v (m/s)	u (m^2/s)	μ (Pa.s)	ρ (kg/m^3)	k (W/m K)	Cp (kJ/kg K)	T_1 (K)	T_2 (K)	Re
1	0	0.00813	1.5	0.508	8.601E-07	0.000857	996.6919	0.5965	4179.0873	297.841	300.642	4730.921
2	0	0.00813	1.5	0.677	8.63E-07	0.00086	996.7512	0.5963	4179.1528	297.841	300.242	6281.757
3	0	0.00813	1.5	1.017	8.658E-07	0.000863	996.8101	0.5961	4179.2204	297.941	299.743	9412.182
4	0	0.00813	1.5	1.357	8.701E-07	0.000867	996.8982	0.5957	4179.3256	297.841	299.243	12489.725
5	0	0.00813	1.5	1.706	8.716E-07	0.000869	996.9274	0.5956	4179.3616	297.841	299.043	15681.363
6	0.2	0.00813	1.5	0.515	8.942E-07	0.000897	1002.9046	0.6111	4172.0129	298.241	301.141	4680.051
7	0.2	0.00813	1.5	0.672	8.977E-07	0.0009	1002.9791	0.6109	4172.0892	298.141	300.742	6087.552
8	0.2	0.00813	1.5	1.018	9.02E-07	0.000905	1003.0681	0.6107	4172.1850	298.141	300.142	9173.593
9	0.2	0.00813	1.5	1.344	9.049E-07	0.000908	1003.1271	0.6106	4172.2513	298.141	299.743	12076.489
10	0.2	0.00813	1.5	1.690	9.094E-07	0.000912	1003.2152	0.6104	4172.3548	297.941	299.343	15106.808
11	1	0.00813	1.5	0.502	8.978E-07	0.000923	1028.2625	0.6212	4144.2456	298.041	301.541	4546.316
12	1	0.00813	1.5	0.676	9.01E-07	0.000926	1028.3365	0.6210	4144.3200	298.141	300.942	6102.149
13	1	0.00813	1.5	1.008	9.074E-07	0.000933	1028.4836	0.6205	4144.4783	297.941	300.142	9028.470
14	1	0.00813	1.5	1.339	9.1E-07	0.000936	1028.5420	0.6203	4144.5452	297.941	299.743	11963.647
15	1	0.00813	1.5	1.653	9.126E-07	0.000939	1028.6002	0.6201	4144.6141	297.841	299.443	14731.052

Table 2. Annulus side measurements of pure water flow

Exp. No.	Mass flow rate (kg/s)	Fluid Inlet Temperature T_1 (K)	Fluid Outlet Temperature T_2 (K)	Reynolds Number Re
1	0.049710789	307.6456	306.15	2204.863
2	0.049711648	307.6456	306.05	2204.901
3	0.049713362	307.6456	305.85	2204.977
4	0.049713362	307.7465	305.75	2204.978
5	0.049715073	307.7465	305.55	2205.053
6	0.049710789	307.7465	306.25	2204.863
7	0.049711648	307.84652	306.05	2204.901
8	0.049713362	307.84652	305.75	2204.977
9	0.049713362	307.9465	305.65	2204.978
10	0.049715073	307.84652	305.45	2205.053
11	0.049710789	308.04648	306.1	2204.863
12	0.049711648	308.04648	306	2204.901
13	0.049713362	308.14646	305.8	2204.977
14	0.049713362	308.14646	305.5	2204.978
15	0.049715073	308.14646	305.2	2205.053

6. CONCLUSION

Heat exchangers are parts in many engineering systems . Hence, calculations of heat exchangers are directly affected by each system's properties and should be made according to the system's optimization rather than the optimization of the heat exchanger. We may express the importance of the analysis of a heat exchanger as follows: For a group of presented data (e.g., flow rates and fluid temperatures at inlets and inner tubes' and shell-side's geometry and relevant information), the results are obtained through heat transfer and fluids' flow balances. Therefore, while determining the system's optimum values, features of the heat exchanger should also be thoroughly understood.

The design of heat exchangers as a component of a system is a major subject of engineering. In numerous engineering systems, heat transfer between two sources that are at unequal temperatures occur through separated walls. We use heat exchangers to perform this exchange. Generally, no external heat or work exchange take place in these devices. Typical applications can be observed in space heating/cooling and power plants that recover or reject heat systems and chemical processes. From this point of view, one can understand that the enhancement of heat exchangers is not only important with regard to heat transfer but also in terms of exergetic efficiency. In the present paper, this point of view tried to be fulfilled.

The following results have been acquired via this study:

a- The minimum exergy loss for a heat exchanger type was determined for the counter flow one as shown in this application.

b- It was found that efficiency of the heat exchanger increased slightly when nanoparticles such as Al_2O_3 and TiO_2 were added in the flowing water in the heat exchanger.

c- Including nanoparticles reduces the heat transfer area of the heat exchangers.

d- The efficiency has reached its highest point under the condition of condensing or boiling (C_r= 0).

e- At increasing values of T_r , exergy losses have also become higher.

f- It is understood that in parallel flow heat exchangers, exergy losses per unit heat flux do not draw rapid changes and keep constant after a definite point, compared to the counter flow heat exchangers.

g- Researchers will be able to benefit from the reliable data of pure and nanofluid flow belonging to a double pipe heat exchangers given in this study.

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