Iranian Journal of Chemistry and Chemical Engineering (IJCCE)

Multi-objective Optimization of Double pipe Heat Exchangers from the Point of View of Efficiency and Economics

Sayed Ehsan Alavi*, Meisam Moory Shirbani

Faculty of engineering, Shohadaye Hoveizeh Campus of Technology, Shahid Chamran University of Ahvaz, Dashte Azadegan, Iran.

Abstract

The purpose of this study is to investigate the cost and efficiency of a double pipe heat exchanger. Furthermore, by utilizing NSGA2, the operational expenses were minimized, and the efficiency was maximized. The cold fluid mass flow rate, outer tube diameter, inner tube diameter, and cold fluid outlet temperature are selected as design variables. Additionally, in this research, the Witte-Shamsundar efficiency and performance index were studied for the double pipe heat exchanger, and the impact of various parameters was examined. Optimizing results showed that the efficiency and heat transfer rate increased 21% and 43%, respectively. In addition, the operational cost experienced a reduction of approximately 58%. The findings indicate that, as the average velocity of the hot fluid increases, the number of transfer units increases, resulting in an increase in the efficiency. Also, for a given inner tube diameter, the highest Witte-Shamsundar efficiency is achieved when the hot side mass flow rate is equal to the cold side mass flow rate. The Witte-Shamsundar efficiency decreases with increasing the hot fluid inlet temperature. As the cold side mass flow rate increases, the performance index of a double pipe heat exchanger decreases. It is found that if a double pipe heat exchanger is required to increase the mass flow rate of hot fluid, provided that the mass flow rate of hot fluid is greater than that of cold fluid, the cost increase will be minimal. Moreover, it has been observed that the present investigation has yielded significantly more optimal outcomes.

KEYWORDS: *double pipe heat exchanger, economics, efficiency, performance index, multi-objective optimization.*

* Corresponding Author: +989163122381, e.alavi@scu.ac.ir.

1. Introduction

Heat transfer between cold and hot fluids is necessary for some industrial processes. However, heat exchangers are one of the most commonly devices utilized in refrigeration and heating systems in the industry. In cases where a lower heat transfer area is possible, double pipe heat exchangers are preferred. The majority of studies conducted so far in the field of enhancing the effectiveness of heat exchangers have focused on two basic strategies. The first strategy focuses on improving the factors that have a positive influence on the effectiveness of the heat exchanger. In terms of these factors, it is possible to mention the use of wide surfaces, nanofluid, and similar kinds of techniques. The second strategy aims to reduce the factors that have a negative influence on the effectiveness of the heat exchanger. The exergy loss, entropy generation, pressure drop are some of the factors that can be mentioned [1-6]. In their research, Sinaga et al [7] analyzed the second law of thermodynamics in a double pipe heat exchanger. In this experimental work, air and hot water were first mixed outside the exchanger and then entered the inner tube of a double pipe heat exchanger. Hot water and air with five different mass flow rates were investigated. The increase of the heat transfer coefficient and the number of transfer units by 33% and 38%, respectively, was one of the important results of this research. Sheikholeslami et al. [8] experimentally improved the heat transfer in a double pipe heat exchanger by installing perforated agitators in the annular space.In their research, different values of Reynolds number, open area ratio and pitch ratio were studied and the relationship between thermal efficiency, friction factor, and Nusselt number for a double pipe heat exchanger was extracted. Bahmani et al. [9] investigated heat transfer in turbulent flow in a double pipe heat exchanger. In this research, they also studied the nanofluid flow in these converters. The research results showed that increasing the volume fraction of nanofluid will increase the displacement heat transfer coefficient and Nusselt number. Among the other results of the research, the suggestion of using a double pipe heat exchanger with opposite flow in high Reynolds numbers could be mentioned. In a numerical and three-dimensional study, Huu-Quan et al. [10] investigated the turbulent forced convection heat transfer in a double pipe heat exchanger with a smooth inner tube. They observed that the effect of the geometry of the inner tube on the flow and heat transfer depends to a large extent on the Reynolds number of the inner tube. Also, it was observed that the use of a smooth inner tube will increase the efficiency of the double pipe heat exchanger by 16.8%Sivalakshmi et al. [11] in their research analyzed the effect of spiral fins on the efficiency of double pipe heat exchangers. They found that the heat transfer rate and efficiency of the double pipe heat exchanger with spiral fins are 38.46% and 35%, respectively, higher than the double pipe heat exchanger with a smooth inner tube. Maakoul et al. [12] numerically investigated the increase of heat transfer and the efficiency of the double pipe heat exchanger with spiral walls in the annular space. Also, in this research, the effect of the distance between the walls and the mass flow was also studied. It was observed that the highest thermo-hydraulic efficiency is achieved when spiral walls are used in slow flow. Sharifi et al [13] studied the effect of adding spiral wire with various arrangements on the heat transfer rate and pressure drop in a double pipe heat exchanger. They extracted friction factor, efficiency and Nusselt number of double pipe heat exchanger using CFD technique. The results showed that the addition of spiral wire will make the Nusselt number equal to 1.77 compared to the smooth double pipe heat exchanger. Mehrabian et al. [14] using experimental data investigated the overall heat transfer coefficient for a double pipe heat exchanger. Their results showed that the heat transfer coefficient of the flow in the inner tube and annular space is higher compared to the results obtained from the standard relations. It was also found that the heat transfer coefficient of the inner pipe flow in the double pipe heat exchanger is 3.4 times the heat transfer coefficient of the external pipe flow. Alkam et al. [15] improved the efficiency of a double pipe heat exchanger using porous substrates. Also, the effect of adding porous beds on the pumping power of the heat exchanger was studied in this research. The results showed that the use of porous beds increases the displacement heat transfer coefficient for both types of double pipe heat exchangers with the same and opposite flow. It was observed that the use of porous beds will increase the pressure drop in the heat exchanger in the heat exchanger. Sadighi Dizaji et al. [16] investigated heat transfer and pressure drop in a double pipe heat exchanger in an experimental study. In this study, both internal and external tubes were ribbed using a special machine. The shape of the treads was applied to the tubes in two convex and concave states. The results showed that the highest efficiency of the double pipe heat exchanger achieved when the inner and outer tubes have convex and concave treads, respectively. Taghilou et al. [17] optimized a fin-needle double pipe heat exchanger using entropy generation minimization. Their results were extracted in the state of constant heat flux and it was observed that by minimizing the entropy generation, the pumping power decreases. Also, it is found that at a constant Reynolds number and in short lengths of the double pipe heat exchanger, the irreversibility distribution ratio decreases with the increase in the length of the heat exchanger. Arjamandi et al. [18] focused on the geometrical optimization of the double pipe heat exchanger in which a combination of vortex generator and twisted tape turbulator is used. In this research, nanofluid $Al_2o_3 - \mathrm{H}_2\mathrm{o}$ was used as the base fluid in the inner tube. Also, the response surface methodology was used to obtain the optimal geometry of the desired heat exchanger. The aim of the research was to achieve the maximum Nusselt number and the minimum friction factor, and 20 experiments were conducted at different Reynolds numbers, angles and pitch ratios. The results showed that the pitch ratio has the greatest effect on the friction factor and Nusselt number, which has increased the efficiency of the heat exchanger about 5 times. It was observed that Nusselt number and friction factor decrease with the increase of vortex generator angle. Kim [19] investigated the effect of different fin shapes on the thermal performance of finned double pipe heat exchangers. Kim observed that the linear increase of the fin thickness shows almost the same performance of the finned tube heat exchanger with variable fin thickness. Also, obtained results showed that the dimensionless optimal thermal resistance is a function of the radius ratio and the dimensionless pumping power. Omidi et al. [20] reviewed the researches that have been done on double pipe heat exchangers. They stated that the main goal in many previous studies is to increase the efficiency of double pipe heat exchangers. In some studies, the heat transfer coefficient is increased by disrupting the flow or fining the inner tube. They observed that fining the inner tube was used more compared to disrupting the flow. It is important to consider that the fining of the inner tube in a double pipe heat exchanger can greatly improve heat transfer and efficiency. Iqbal et al. [21] investigated the effect of fin shape on Nusselt number in finned double pipe heat exchangers. They estimated the blade shape using interpolation polynomials. With the assumption of isothermal blade, they showed that the Nusselt number in the case where the blade with variable thickness is optimized is 138% more than the trapezoidal blade. Tian et al. [22] investigated the turbulent flow in spiral double pipe heat exchangers in their research. In this research, using a numerical method, the effect of temperature and inlet fluid velocity, volume fraction of nanofluid and type of nanofluid on heat transfer coefficient, friction factor and pressure drop were studied. The results showed that the effect of the type of nanofluid in Reynolds numbers less than 16000 can be neglected. Also, the results of optimal conditions were extracted using the genetic algorithm. Saeedan et al. [23] studied the effect of various types of nanofluids on the Nusselt number and pressure drop of double pipe heat exchangers with spiral fins using CFD method and neural network. It is found that the Nusselt number increases with the increase in volume fraction of nanofluid. Also, in this modeling, Nusselt number and pressure drop in the heat exchanger were estimated. Kalteh et al. [24] numerically investigated the laminar fluid flow in a double pipe heat exchanger. The research results showed that reducing the diameter of nanofluid and increasing its concentration is a very important factor in increasing the heat transfer rate. It was also observed that the friction factor increases with the increase in the diameter of the nanofluid. Jalili et al [25] studied the curved rectangular fin in double pipe heat exchangers. In their study, nanofluids with different concentrations were used as cold fluid in the inner tube. They compared the results of this type of heat exchanger geometry with the results of a double pipe heat exchanger with a simple rectangular fin and found that the double pipe heat exchanger with a curved rectangular fin has a higher efficiency and a lower pressure drop. Colaco et al. [26] optimized the thermal efficiency index and friction factor in a double pipe heat exchanger with internal perforated walls. Their goal in this research was to minimize the pressure drop and achieve more heat transfer. Genetic algorithm was used to extract optimization results. The results showed that the Nusselt number of this type of exchangers is about 7.93 to 8.25 times that of double pipe heat exchangers without walls. It was also found that the friction factor of the double pipe heat exchanger with internal perforated walls is about 6.5 to 9.75 times that of the double pipe heat exchanger without walls. Other related studies can be reviewed in references [27-34].

Based on previous research, it appears that the optimization of the double pipe heat exchanger using NSGA2 with objective functions of efficiency and operational cost has not been done. Additionally, the effect of process and geometric parameters on thermodynamic parameters and the operational cost of the double pipe heat exchanger have not been investigated. Furthermore, the mass flow rate ratio of two fluids has not been included in previous analyses of the double pipe heat exchanger. Hence, this parameter is also taken into account in this research. The study examines a parameter known as Witte-Shamsundar efficiency, which is utilized to examine the quality and quantity of energy. The performance index is a parameter that considers the heat transfer rate and pumping power. Therefore, this parameter is also studied in this investigation.

2. Thermodynamic modeling

The heat exchangers used in this research are for double pipe heat exchangers in series, as shown in Figure 1.

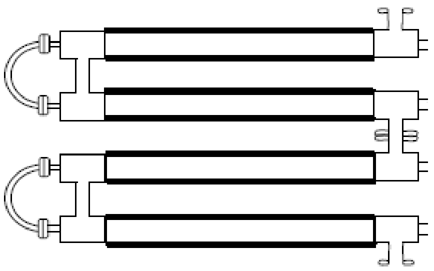


Fig. 1.Double pipe heat exchangers in series.

The assumptions used in the research are [35]:

- The hypothesis is that the condition is in a steady state.

- Every direction has a uniform flow of cold and hot fluids.
- There is no noticeable heat loss.
- The cold and hot fluids' temperatures are supposed to be uniform.
- The velocity is assumed to be uniform in the region of the entry and exit.
-The fouling resistance is taken as a fixed amount.

The hydraulic diameter and the equivalent diameter in the double pipe heat exchanger are calculated from the following correlations, respectively [36].

$$D_{\mathrm{h}} = D_i - d_o \tag{1}$$

$$\mathrm{D_e} = \frac{\mathrm{D_i^2 - d_o^2}}{\mathrm{d_o}} \tag{2}$$

Where d and D are the inner tube diameter and the outer tube diameter. D_{h} and D_e represent the hydraulic diameter and the equivalent diameter.

The fluid pressure drop on the pipe side is obtained from the following equation [36].

$$\Delta p_{\mathrm{h}} = 4f \frac{2L}{d_i} N_{hp} \rho \frac{u_m^2}{2} \tag{3}$$

Where L and N_{hp} are the length of the heat exchanger and the number of hairpins, respectively. f is the coefficient of friction in a circular duct with smooth surfaces, which is obtained from the following relationship [36].

$$\mathrm{f_h} = \left(1.58\mathrm{LnRe_h} - 3.28\right)^{-2} \tag{4}$$

Also, u_m is the average velocity of the fluid, which is obtained from the following relationship [36].

$$\mathrm{u_m} = \frac{\dot{m}}{\rho \mathrm{A_c}} \tag{5}$$

Where A_c and ρ are the net flow cross-sectional area and the fluid density, respectively. The fluid pressure drop on the side of the annular space is obtained from the following equation. [36].

$$\Delta p_{\mathrm{c}} = 4f \frac{2L}{D_{\mathrm{h}}} N_{hp} \rho \frac{u_m^2}{2} \tag{6}$$

Friction factor of turbulent flow in annular space with smooth surfaces is obtained from the equation (7) [36].

$$f_c = \left(3.64 Log_{10} Re_c - 3.28\right)^{-2} \tag{7}$$

The Nusselt number for turbulent flow in a circular duct is obtained from the equation (8) [35].

$$\mathrm{Nu_h} = \frac{\left(\frac{f_h}{2}\right)(\mathrm{Re_h})\mathrm{Pr_h}}{1 + 8.7\left(\frac{f_h}{2}\right)^{0.5}(\mathrm{Pr_h} - 1)} \tag{8}$$

The heat transfer coefficient in a circular duct is obtained from the following equation [35].

$$h_h = \frac{Nu_h\, k}{d_i} \tag{9}$$

The Nusselt number for turbulent flow in annular space is obtained from the following equation [35].

$$Nu_c = \frac{\left(\frac{f_c}{2}\right)(Re_c)Pr_c}{1 + 8.7\left(\frac{f_c}{2}\right)^{0.5}(Pr_c - 1)} \tag{10}$$

The heat transfer coefficient in the annular space is as follows [37]:

$$h_c = \frac{Nu_c\, k}{D_h} \tag{11}$$

The efficiency of a double pipe heat exchanger is determined using the following correlation [35]:

$$\varepsilon = \frac{1 - e^{-NTU(1-C^*)}}{1 - C^* e^{-NTU(1-C^*)}} \tag{12}$$

In relation (1), the number of transfer units and the heat capacity ratio are [39]:

$$NTU_{max} = \frac{UA_{tot}}{C_{min}} \tag{13}$$

$$C^* = \frac{C_{min}}{C_{max}} \tag{14}$$

The overall heat transfer coefficient based on the outer surface of the inner tube is obtained from the following equation [35]:

$$\frac{1}{U} = \frac{d_o}{d_i h_i} + \frac{d_o R_{fi}}{d_i} + \frac{d_o Ln\left(d_o/d_i\right)}{2k} + R_{fo} + \frac{1}{h_o} \tag{15}$$

The heat transfer level for each two-branch heat exchanger is obtained from the following equation [35]:

$$A_{hp} = 2\pi d_o L \tag{16}$$

The pumping power is obtained from the following relationship [38]:

$$P = \frac{\Delta P.\, \dot{m}}{\eta_P \rho} \tag{17}$$

Where η_P is the pump efficiency.

Entropy generation due to fluid friction is [40]:

$$\dot{S}_{gT} = \left(\frac{\dot{m}\Delta P}{\rho}\right)_h \frac{\ln\left(T_{h,o}/T_{h,i}\right)}{T_{h,o} - T_{h,i}} + \left(\frac{\dot{m}\Delta P}{\rho}\right)_c \frac{\ln\left(T_{c,o}/T_{c,i}\right)}{T_{c,o} - T_{c,i}} \tag{18}$$

The entropy generation due to the limited temperature difference is as follows [40]:

$$\dot{S}_{gP} = \dot{m}_h C_{p,h} \ln(T_{h,o}/T_{h,i}) + \dot{m}_c C_{p,c} \ln(T_{c,o}/T_{c,i}) \tag{19}$$

Total entropy generation is the sum of entropy generation caused by limited temperature difference and fluid friction:

$$\dot{S}_g = \dot{S}_{gT} + \dot{S}_{gP} \tag{20}$$

The Witte-Shamsundar efficiency is defined as [40]:

$$\text{WSE} = 1 - \frac{\dot{S}_g T_\infty}{q} \tag{21}$$

The performance index can be obtained from the following equation [41]:

$$\text{Performance Index} = \frac{q}{P} \tag{22}$$

Operation cost of double pipe heat exchanger is [40]:

$$\text{Cost} = C_1 \frac{Q'}{Q} + C_2 \frac{P_t}{P'} + C_3 M_r \tag{23}$$

Where C_1, C_2, C_3 are the constant coefficients that are selected based on industrial considerations. P',Q' and M_r represent the lowest value of pumping power, standard heat transfer rate and ratio of hot mass flow rate to cold mass flow rate .

3. Results

The processing properties of each of the hot and cold fluids are summarized in Table 1. In order to confirm the findings for the double pipe heat exchanger, the thermal and hydraulic results were compared with the reference findings shown in Table 2. It is evident that the conclusive outcomes of this study are in excellent conformity with the mentioned reference.

Table 1. Characteristics of hot and cold fluids [36].

Process information	Hot fluid	Cold fluid
fluid density(kg/m^3)	870	880
specific heat ($j/kg.K$)	1817	1778
fluid viscosity($pa.s$)	0.00041	0.0005
fluid thermal conductivity($W/m.K$)	0.1471	0.1574
Prandtl number	5.064	5.648
Fouling Factor($m^2.W/K$)	0	0

Table 2. Validation of results.

parameter	Current Work	Ref [42]
$\Delta P_{\mathrm{h}}(Pa)$	460.1	464.5402
$\Delta P_c(Pa)$	2876.4	2981.1
$h_{\mathrm{h}}(W/m^2K)$	4911	4912.3
$h_c(W/m^2K)$	1345	1344.6
$P_{\mathrm{h}}(W)$	0.84	0.8469
$P_{\mathrm{c}}(W)$	5	5.1946
Nu_{h}	375.393	375.3
Nu_{c}	89.06	89

This section presents the optimization of the double pipe heat exchanger with objective functions of operating cost and efficiency. The range of design variables are shown in Table 3. The Pareto front diagram of the result of optimization using the NSGA2 method is shown in Figure 2. Since efficiency and cost are opposite goals, point A on the pareto front is introduced as the optimal point. The optimization results are presented in Table 4. By optimizing the double pipe heat exchanger, the efficiency of the heat exchanger and its heat transfer rate have gone up by 21% and 43%, respectively, and the operating cost has gone down by 58%. Furthermore, the findings indicate that the pressure drop of both hot and cold fluids has decreased by 70% and 97%, respectively. Also, the optimization results demonstrate that the pumping power of hot and cold fluid has been significantly reduced. Table 5 compares the optimization outcomes of the present work with those of previous work conducted by other methods. It is evident that the present research has yielded significantly more optimal outcomes.

Table 3.Range of optimization variables.

variable	Lower limit	Upper limit
$d_i(mm)$	20	60
$D_i(mm)$	70	110
$\dot{m}_{\mathrm{c}}(kg/s)$	1	6
$T_{\mathrm{c,o}}(K)$	320	340

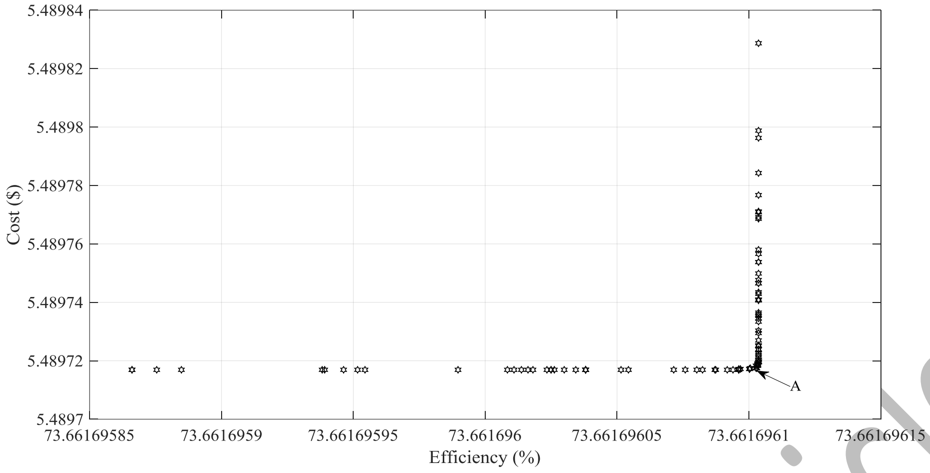


Fig.2. Pareto front.

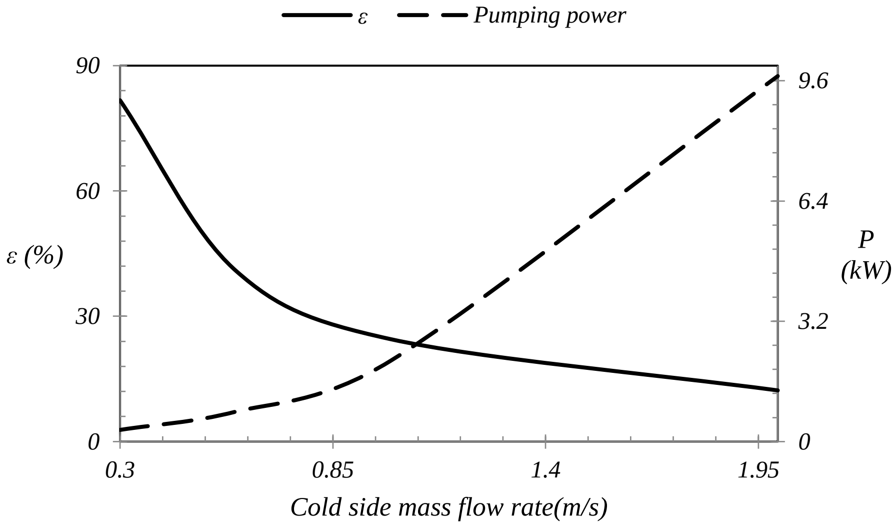
Table 4. Comparison of the results before and after optimization.

Parameter	D_i(m)	$d_i(m)$	$T_{c,o}$ (K)	$\dot{m}_c(kg/s)$	$P_c(W)$	$P_h(W)$	q(W)	ε(%)	Cost ($)
Initial value	0.06	0.042	323	1.512	331.45	55.33	61832	52.27	13.06
Optimal value	0.11	0.06	339.98	1.0039	11.32	8.36	107490	73.66	5.48

Table 5.The optimization results compared to other references.

Parameter	q(W)	$\Delta P_h(Pa)$	$\Delta P_c(Pa)$	Cost ($)
Optimal value (Current Work)	107490	5795.2	5271.5	5.48
Optimal value (Ref [36])	62868	27051	62314	10.0755
Optimal value (Ref [43])	105650	9273	51819	10.1457

In Figure 3, the effect of cold fluid mass flow rate on the efficiency and pumping power of the double pipe heat exchanger has been investigated. It can be seen that, increasing the cold side mass flow rate, decreases the efficiency of the double pipe heat exchanger. The reason is that according to the range of changes in mass flow rate of cold fluid, the heat capacity of cold fluid will always be lower than the heat capacity of hot fluid. In other words, the heat capacity of the cold fluid in such conditions will always be the minimum heat capacity. Therefore, increasing the cold side mass flow rate decreases the number of transfer units. Since the number of transfer units is directly related to the efficiency, the efficiency of the double pipe heat exchanger will also decrease. It can also be found from this graph that increasing the cold side mass flow rate, increases the pumping power. According to equation (17), pumping power has a direct relationship with fluid mass flow rate. Therefore, increasing the mass flow rate of the cold fluid, the pumping power also increases.



Figure

3.Performance and pumping power in terms of cold fluid mass flow rate.

In Figure 4, the effect of the diameter of the inner tube on the overall coefficient of conduction heat transfer and the number of transfer units is investigated. It can be seen that with the increase in the diameter of the inner tube, the overall coefficient of conduction heat transfer increases. Based on equation (15), the general coefficient of conduction heat transfer has a direct relationship with the inner tube diameter. Therefore, by increasing the diameter of the inner tube, the overall coefficient of conduction heat transfer also increases. It can also be seen that with the increase in the diameter of the inner tube, the number of transfer units also increases. The number of transfer units has a direct relationship with the overall conduction heat transfer coefficient. Therefore, any parameter that increases the overall heat transfer coefficient will also increase the number of transfer units.

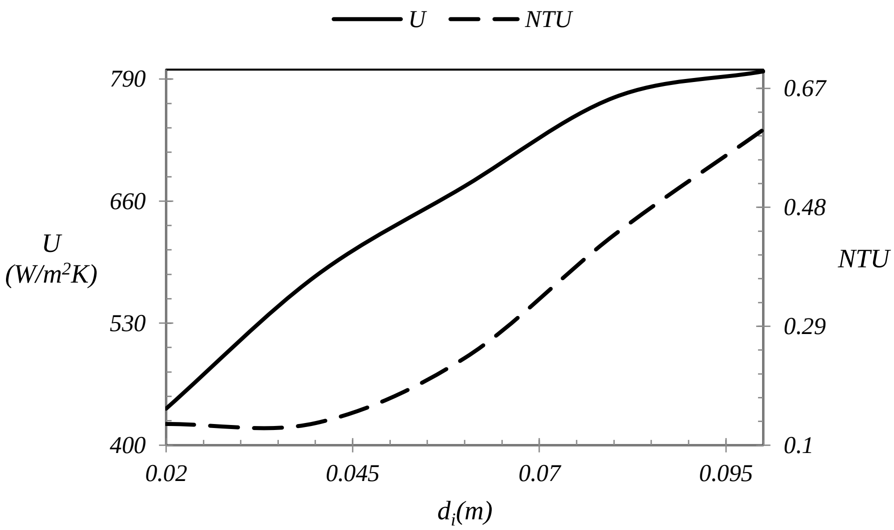


Fig. 4.The overall heat transfer coefficient and the number of transfer units in terms of the diameter of the inner tube.

Figure 5 shows the effect of increasing the diameter of the inner tube on the Nusselt number of hot and cold fluids. It can be seen that with the increase in the diameter of the inner tube, the Nusselt number of the hot fluid increases and the Nusselt number of the cold fluid decreases. Because, increasing the diameter of the inner tube, the Reynolds number of the hot fluid increases. Since the Reynolds number has a direct relationship with the Nusselt number, the Nusselt number of the hot fluid increases with the increase of the inner tube diameter. Increasing the diameter of the inner tube, the hydraulic diameter in the double pipe heat exchanger decreases and therefore the Reynolds number of the cold fluid also decreases. As the Reynolds number of the cold fluid decreases, the Nusselt number of the cold fluid also decreases.

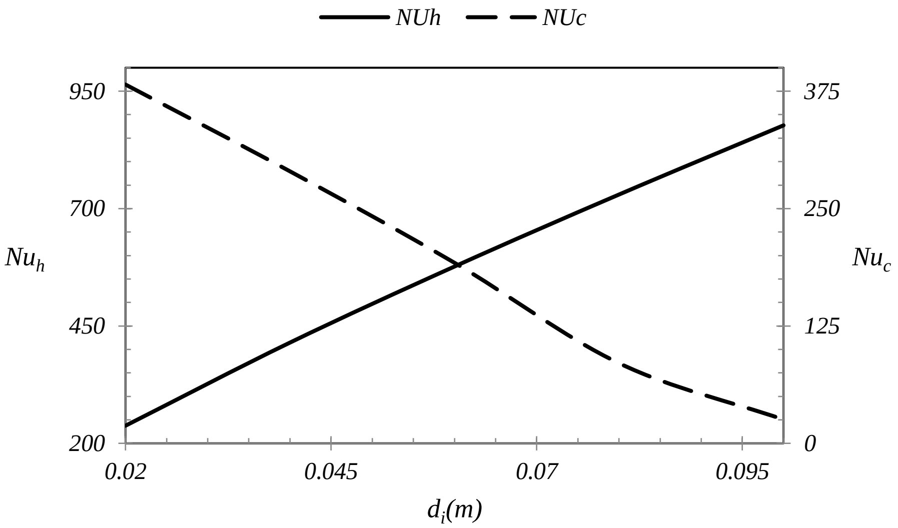


Fig.5. Nusselt number of hot and cold fluids in terms of the diameter of the inner tube.

Figure 6 shows the effect of the inner diameter of the outer tube on the Reynolds number and the friction factor of the cold fluid. As can be seen, with the increase in the diameter of the outer tube, the Reynolds number increases and the fluid friction factor decreases. Increasing the diameter of the outer tube increases the hydraulic diameter. With the increase of the hydraulic diameter, the Reynolds number of the fluid increases. Also, equation (7) shows that the Reynolds number and the fluid friction factor have an inverse relationship, so it can be concluded that the cold fluid friction factor has an inverse relationship with the outer tube diameter.

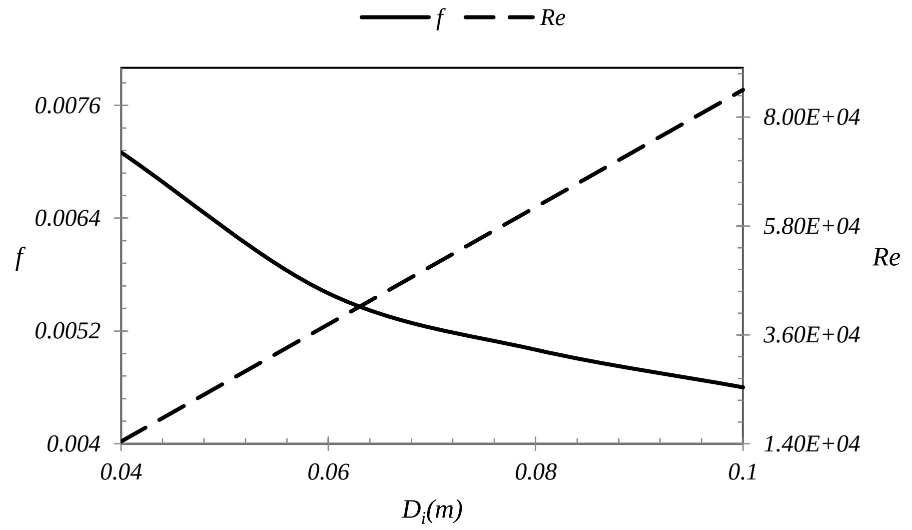


Fig.6.Friction factor and Reynolds number in terms of the inner diameter of the outer tube.

In Figure 7, the effect of increasing the average velocity of the cold fluid on the friction factor and the Nusselt number of the cold fluid is investigated. It can be seen that with the increase in the average velocity of the cold fluid, the friction factor decreases and the Nusselt number of the cold fluid increases. In fact, increasing the average velocity of the cold fluid increases the Reynolds number, and increasing the Reynolds number decreases the friction factor of the cold fluid. Also, according to equation (10), an increase in the Reynolds number causes an increase in the Nusselt number. Therefore, it can be concluded that the average velocity of the cold fluid has an inverse relationship with the friction factor and a direct relationship with the Nusselt number.

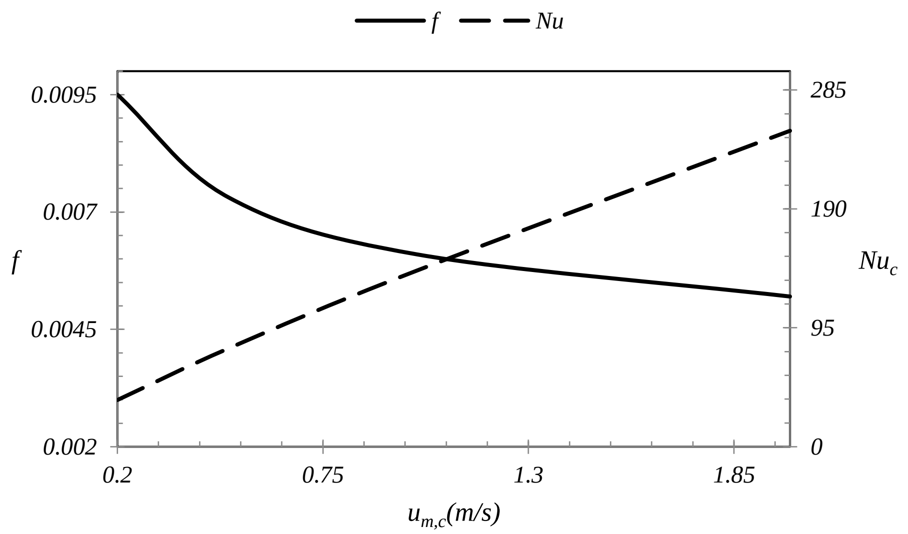


Fig.7.Friction factor and Nusselt number in terms of cold fluid velocity.

In Figure 8, the effect of increasing the average velocity of the cold fluid on the number of transfer units and the efficiency of the double pipe heat exchanger has been investigated. It can be found that increasing the average velocity of the cold side decrease the number of transfer units and the efficiency of the heat exchanger. Equation

(13) shows that the number of transfer units is related to the general heat transfer coefficient of conduction, heat transfer cross-section and minimum heat capacity of two hot and cold fluids. As the average velocity of the cold fluid increases, the heat transfer coefficient of the cold fluid increases. Therefore, based on the equation (15), the overall conduction heat transfer coefficient also increases. On the other hand, in a constant heat flux, with the increase of the overall heat transfer coefficient, the heat transfer cross-sectional area decreases. The interaction between these two parameters ultimately has no effect on the number of transfer units. Increasing the average velocity of the cold fluid up to a velocity of 1.2 m/s, the heat capacity of the cold fluid is always the minimum heat capacity. Therefore, with an increase in the average velocity of the cold fluid, the minimum heat capacity also increases, and since the minimum heat capacity has an inverse relationship with the number of transfer units, hence, with an increase in the average velocity of the cold fluid, the number of transfer units decreases. Also, since the number of transfer units has a direct relationship with the efficiency of the heat exchanger, so, with the increase in the average velocity of the cold fluid, the efficiency of the double pipe heat exchanger also decreases. Increasing the average velocity of the cold fluid to more than 1.2 m/s, the heat capacity of the hot fluid will have a minimum value, and therefore, as can be seen from the diagram, in such conditions, the increase in the average velocity of the cold fluid will not affect the number of transfer units and the efficiency of the double pipe heat exchanger.

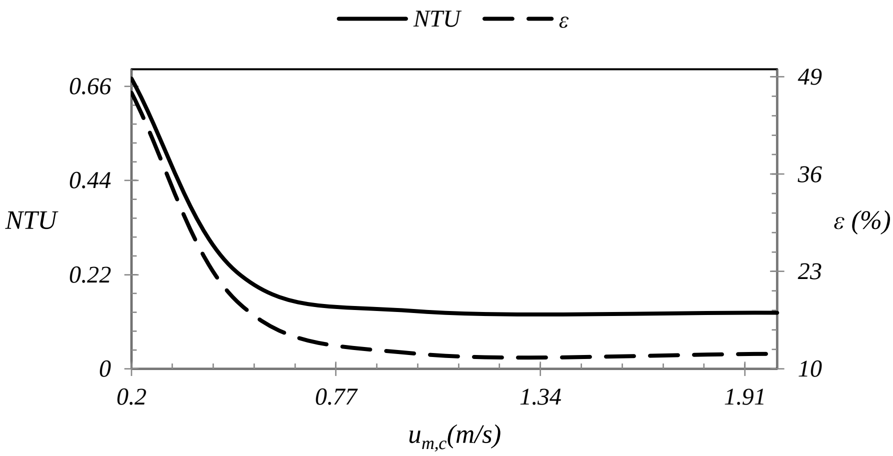


Fig.8.The number of transfer units and efficiency in terms of the velocity of the cold fluid.

In Figure 9, the effect of increasing the average velocity of the hot fluid on the number of transfer units and the efficiency of the double pipe heat exchanger has been investigated. Increasing the hot fluid average velocity increase the number of transfer units and the efficiency of the double pipe heat exchanger. By increasing the average velocity of the hot fluid, the Reynolds number of the hot fluid increases and therefore increases the heat transfer coefficient of the hot fluid. As the heat transfer increases, the term UA also increases. Hence, the number of transfer units and the efficiency of the double pipe heat exchanger also increase. In other words, the average velocity of the hot fluid has a direct relationship with the number of transfer units and the efficiency of the heat exchanger.

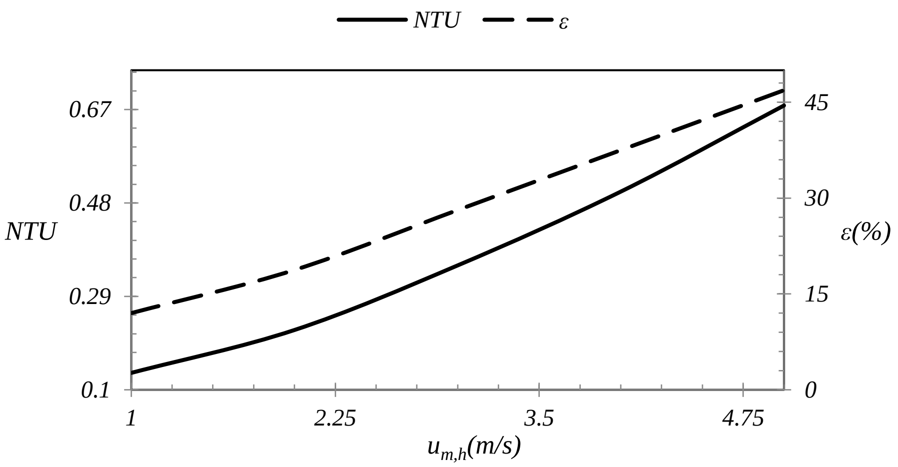


Fig.9. The number of transfer units and efficiency in terms of the velocity of the hot fluid.

In Figure 10, an examination of the impact of the cold fluid inlet temperature on the Witte-Shamsundar efficiency in a constant heat flux is conducted for varying values of the mass flow rate ratio. It is observed that, the Witte-Shamsundar efficiency increases with the increase in the cold fluid inlet temperature for all values of M_r. As the inlet temperature of the cold fluid increases in a constant heat flux, the entropy generation decreases, resulting in an increase in Witte-Shamsundar efficiency. Also, the results show that at a given cold fluid inlet temperature, the highest Witte-Shamsundar efficiency is achieved when the mass flow rate of both hot and cold fluids is equal.

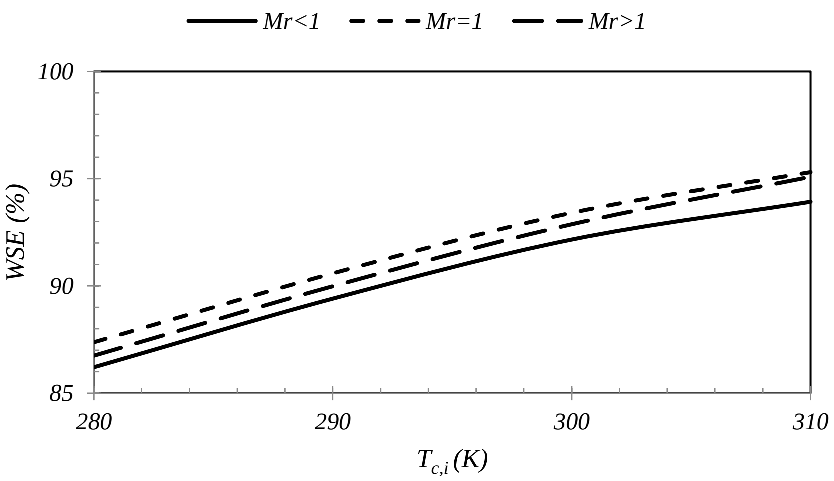


Fig.10. The Witte-Shamsundar efficiency in terms of the cold fluid inlet temperature.

The impact of the hot fluid inlet temperature on the Witte-Shamsundar efficiency is investigated in Figure 11 for a constant heat flux and for varying values of the mass flow rate ratio. It is found that, for all values of M_r, the Witte-Shamsundar efficiency decreases with the increase in the hot fluid inlet temperature. By increasing the hot fluid inlet temperature in a constant heat flux, the ratio of the hot fluid outlet temperature to its inlet temperature also increases. The ratio of the hot fluid outlet temperature to its inlet temperature has an inverse correlation with the Witte-Shamsundar efficiency. As a result, the Witte-Shamsundar efficiency decreases as the hot fluid inlet temperature increases. Also, the findings show that for a given hot fluid inlet temperature, the Witte-Shamsundar efficiency is at its lowest point when the hot side mass flow rate is lower than the cold side mass flow rate.

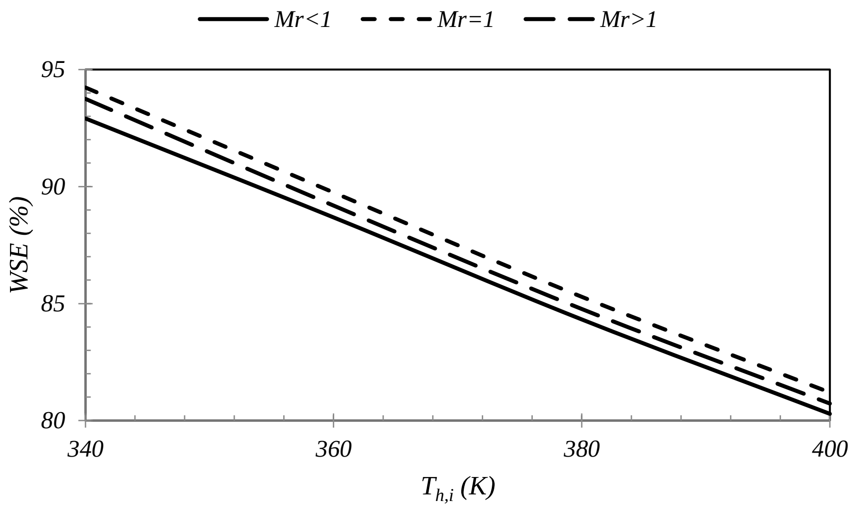


Fig.11. The Witte-Shamsundar efficiency in terms of the hot fluid inlet temperature.

For different values of mass flow rate ratio, Figure 12 illustrates the effect of hot fluid mass flow rate on Witte-Shamsundar efficiency in a constant heat flux. It has been discovered that, as the hot side mass flow rate increases, the Witte-Shamsundar efficiency decreases for all values of M_r. By increasing the hot side mass flow rate in a constant heat flux, the hot fluid outlet temperature also increases. Since the Witte-Shamsundar efficiency exhibits an inverse correlation with the outlet temperature and mass flow rate of hot fluid, it decreases with the increase of these parameters. Results also reveal that for a given hot side mass flow rate, the highest Witte-Shamsundar efficiency appears when the hot side mass flow rate is greater than the cold side mass flow rate.

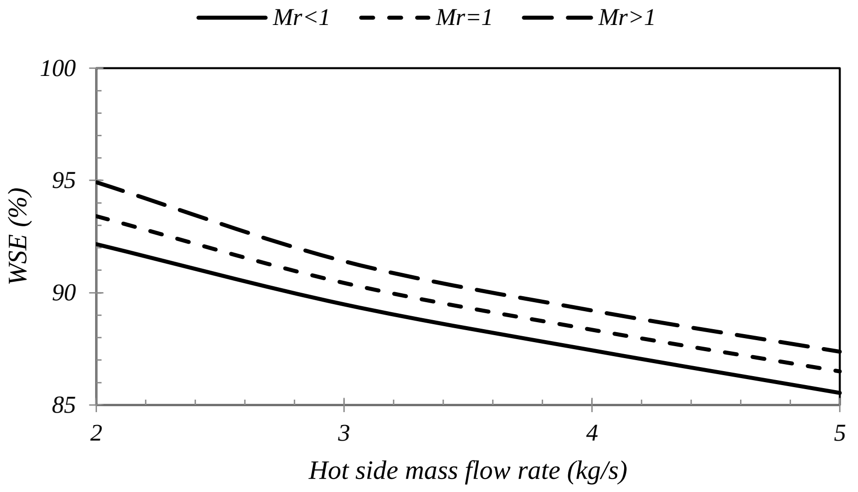


Fig.12. The Witte-Shamsundar efficiency in terms of the hot side mass flow rate.

In Figure 13, the effect of inner tube diameter on Witte-Shamsundar efficiency in a constant heat flux is investigated for various values of mass flow rate ratio. The findings indicate that, in the event that the hydraulic diameter remains constant with the increment of the inner diameter of the tube, the Witte-Shamsundar efficiency increases for all values of M_r. By increasing the inner tube diameter, the pressure drop of the hot fluid decreases. Since pressure drop has an inverse correlation with Witte-Shamsundar efficiency, the Witte-Shamsundar efficiency increases with the increase in inner tube diameter. Furthermore, the results show that for a given inner tube diameter, the highest Witte-Shamsundar efficiency is observed in the case where the hot side mass flow rate is equal to the cold side mass flow rate.

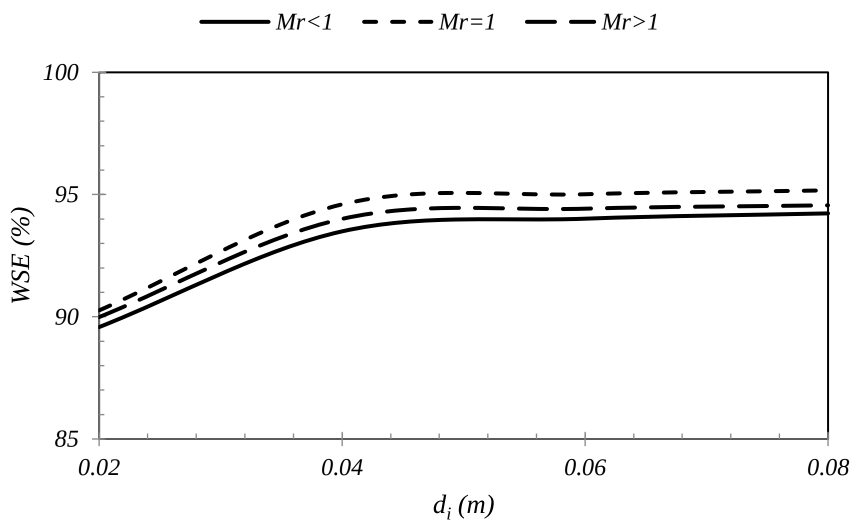


Fig.13. The Witte-Shamsundar efficiency in terms of the inner tube diameter.

Figure 14 depicts the impact of hot fluid inlet temperature on performance index for various values of mass flow rate ratio. It can be seen that the performance index of the double pipe heat exchanger increases with the increase in the hot fluid inlet temperature for all values of M_r. As the hot fluid inlet temperature increases, the heat transfer rate increases. As a result, the performance index also increases. Moreover, the outcomes show that for a given hot fluid inlet temperature, the highest performance index is observed when the hot side mass flow rate is greater than the cold side mass flow rate.

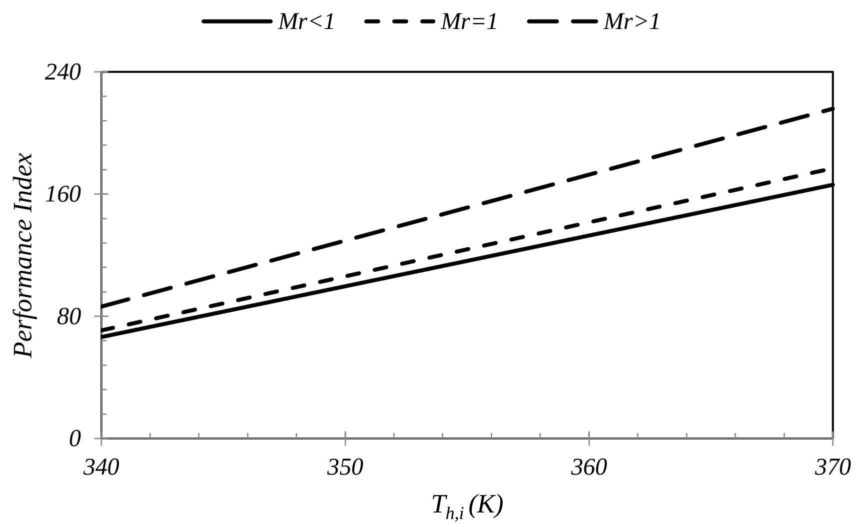


Fig.14. The performance index in terms of the hot fluid inlet temperature.

For various values of mass flow rate ratio, Figure 15 illustrates the impact of cold fluid inlet temperature on performance index. It is apparent that for all values of M_r, as the cold fluid inlet temperature increases, the performance index of the double pipe heat exchanger decreases. The cold fluid inlet temperature has an inverse correlation with the performance index. Therefore, the performance index decreases with its increase. The results indicate that at a given cold fluid inlet temperature, the highest performance index is achieved when the cold fluid mass flow rate is greater than the hot fluid mass flow rate.

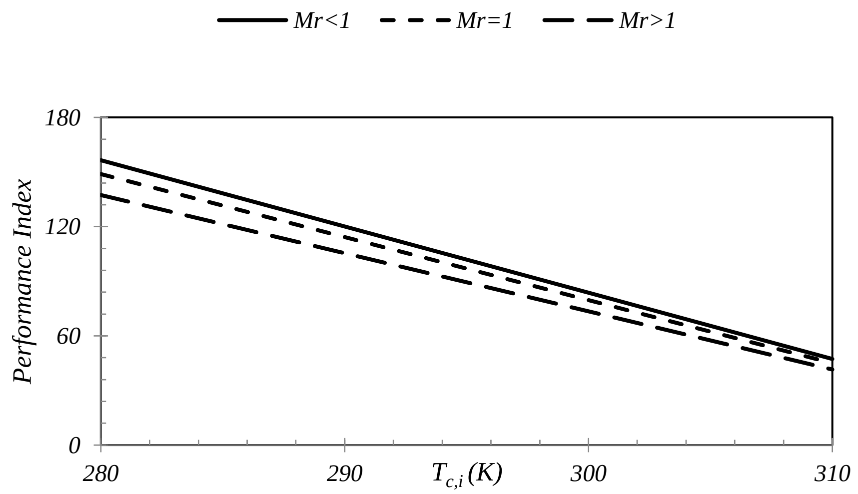


Fig.15. The performance index in terms of the cold fluid inlet temperature.

Figure 16 demonstrates the effect of the cold side mass flow rate on the performance index for various values of the mass flow rate ratio. It is evident that for all values of M_r, the increase in the cold side mass flow rate decreases the performance index of a double pipe heat exchanger. The heat transfer rate and pressure drop on the cold side increase by increasing the cold side mass flow rate. Therefore, with an increase in the cold side mass flow rate, the pumping power will increase significantly, resulting in a decrease in the performance index. The results depict that for a given cold side mass flow rate, the lowest performance index occurs when the hot side mass flow rate is greater than the cold side mass flow rate.

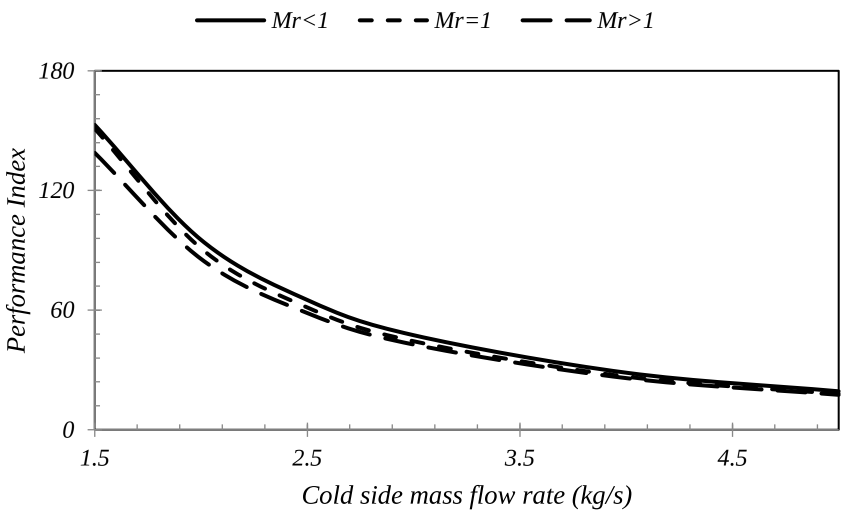


Fig.16. The performance index in terms of the cold side mass flow rate.

The effect of the inner diameter of the outer tube on the performance index is shown in Figure 17 for various values of the mass flow rate ratio. It has been observed that the performance index of the double pipe heat exchanger increases with the increment in the inner diameter of the outer tube for all values of M_r. By increasing the inner diameter of the outer tube, the heat transfer rate remains unchanged; however, a decrease in the cold side pressure drop is observed. Hence, by enhancing the inner diameter of the outer tube, the pumping power decreases, resulting in an increase in the performance index. Furthermore, the findings indicate that for a given inner diameter of the outer tube, the highest performance index is achieved when the hot side mass flow rate is greater than the cold side mass flow rate.

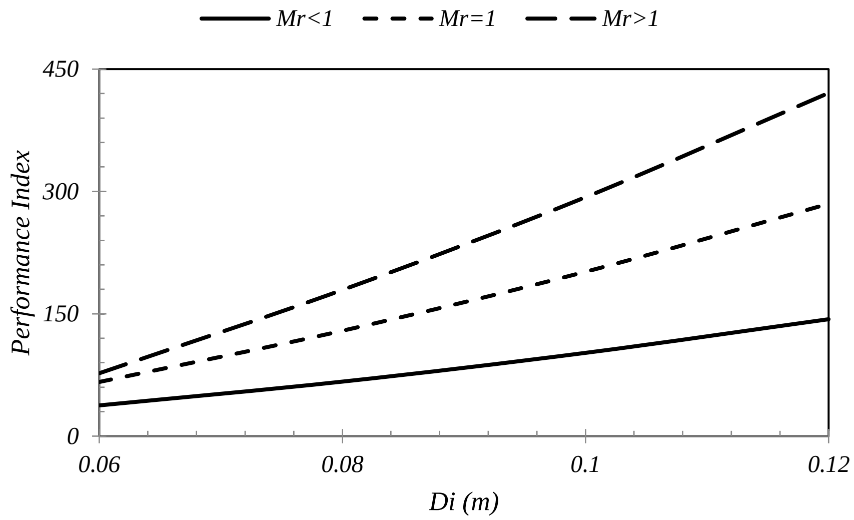


Fig.17. The performance index in terms of the inner diameter of outer tube.

In Figure 18, an investigation has been conducted to examine the impact of the hot fluid inlet temperature on the total cost of the heat exchanger for a constant heat flux and for varying values of M_r. It can be seen that the total cost decreases with the increase in hot fluid inlet temperature. The heat transfer cross-sectional area and the length of the heat exchanger decrease with the increase of the hot fluid inlet temperature, since the heat flux is constant. Therefore, the pressure drop of the hot fluid and the pumping power are both reduced. The total cost decreases when the pumping power is reduced. The results also show that the lowest heat exchanger cost occurs for values of $M_r > 1$ for a given hot fluid inlet temperature.

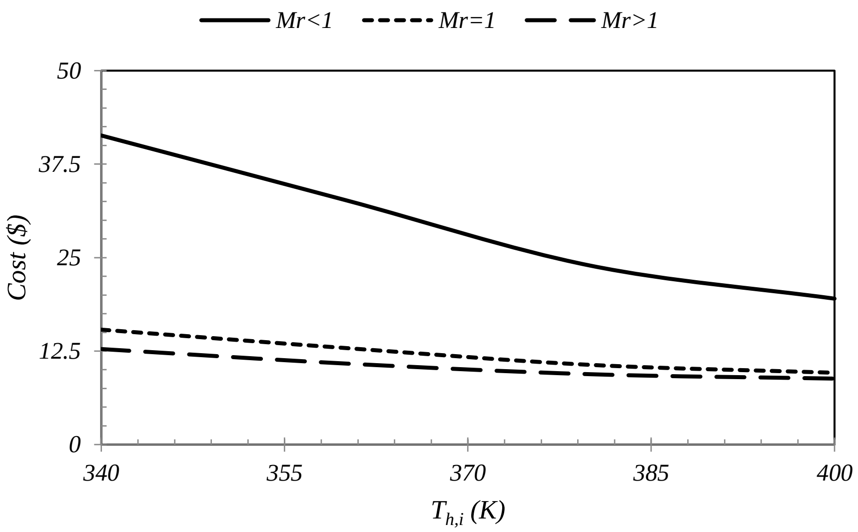


Fig.18.The operation cost in terms of the hot fluid inlet temperature.

In Figure 19, a study was conducted to examine the impact of the cold fluid inlet temperature on the total cost of the heat exchanger for a constant heat flux and for varying values of the mass flow rate ratio. It is evident that the total cost decreases with the increase in the hot fluid inlet temperature. The heat transfer cross-sectional area and heat exchanger length decrease with the rise in the hot fluid inlet temperature, as the heat flux remains constant. So, the pressure drop of the cold fluid and the pumping power are both increased. The total cost increases when the pumping power is increased. The results also indicate that the highest heat exchanger cost occurs for values of $M_r < 1$ for a given cold fluid inlet temperature.

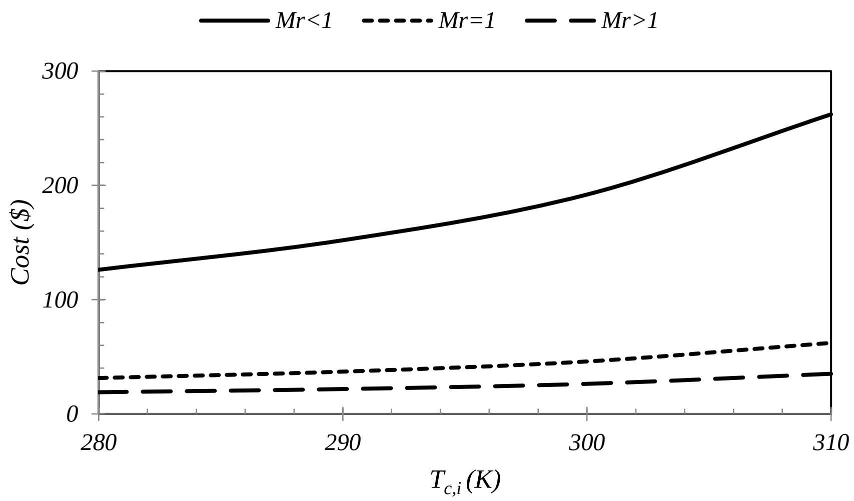


Fig.19.The operation cost in terms of the cold fluid inlet temperature.

In Figure 20, the effect of the hot fluid mass flow rate on the total cost of the heat exchanger for a constant heat flux and for varying values of the mass flow rate ratio. It can be seen that the total cost increases with the increase in the mass flow rate of hot fluid. As a result of the increase in mass flow rate, the power required to pump the fluid increases. One significant conclusion that can be discerned from this diagram is that in the event that it becomes necessary to augment the mass flow rate of the hot fluid in a double pipe heat exchanger for any reason, the increase in cost will be minimal, provided that $M_r > 1$.

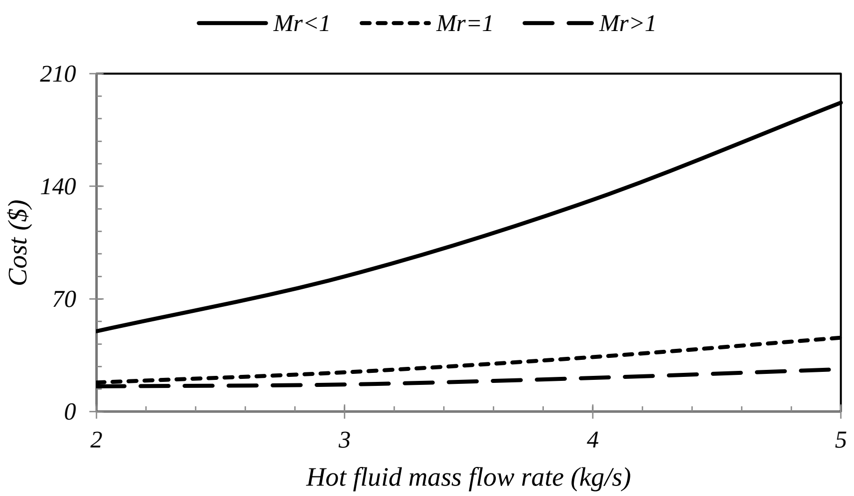


Fig.20.The operation cost in terms of the hot fluid mass flow rate.

4. **Conclusion**

In this study, the optimization of the double pipe heat exchanger was conducted utilizing the NSGA2 method, focusing on both efficiency and operational cost. Four parameters were selected as design variables in the optimization. Through the optimization of the double pipe heat exchanger, its efficiency was increased by 21% and its operational cost was reduced by 58%. Additionally, the heat transfer rate improved by 43%. Furthermore, the pressure drop of both hot and cold fluids has decreased by 70% and 97%, respectively. Moreover, the impact of process and geometric parameters on thermodynamic behavior and the operational cost of the double pipe heat exchanger thoroughly examined. Other main research results are as follows:

- Increasing the cold side mass flow rate, decreases the efficiency of the double pipe heat exchanger.
- As the inner tube's diameter increases, the Nusselt number of the hot fluid increases, while that of the cold fluid decreases.
- Increasing the diameter of the outer tube, the cold fluid friction factor decreases.
- Increase in the average velocity of the cold fluid, the cold fluid friction factor decreases and the Nusselt number increases.
- Increasing the average velocity of the cold fluid to more than 1.2 m/s will not affect the number of transfer units and the efficiency of the double pipe heat exchanger.
- Increasing the hot fluid average velocity increase the number of transfer units and the efficiency of the double pipe heat exchanger.
- The highest Witte-Shamsundar efficiency appears when the hot side mass flow rate is greater than the cold side mass flow rate for a given hot side mass flow rate.
- The lowest heat exchanger cost occurs for values of $M_r > 1$ at a given hot fluid inlet temperature.
- Increasing the hot side mass flow rate, decreases the Witte-Shamsundar efficiency for all values of M_r.
- The highest performance index is observed when the hot side mass flow rate is greater than the cold side mass flow rate for a given hot fluid inlet temperature.
- Increasing the inner diameter of the outer tube, the performance index of the double pipe heat exchanger increases for all values of M_r.
- It was also observed from the findings that in a constant heat flux, the operational cost increases with the rise in the cold fluid inlet temperature.
- If, for any reason, it becomes necessary to increase the mass flow rate of the hot fluid in a double pipe heat exchanger, the cost increase will be minimal, provided that $M_r > 1$.

Nomenclature

A_{tot}	Overall heat transfer area (m^2)		***Greek abbreviation***
A_c	Cross-sectional area (m^2)	ΔP	Pressure drop (kPa)
A_{hp}	Hairpins area(m^2)	μ	Viscosity (Pa. s)
C_P	Specific heat capacity (J/kg. K)	ρ	Density (kg/m^3)
$C*$	Heat capacity rate ratio (-)	ε	Effectiveness (-)
d	Inner tube diameter (m)		***Subscrtipts***
D	Outer tube diameter (m)	h	Hot
D_e	Equivalent diameter	c	Cold
D_h	Hydraulic diameter	i	input
f	Friction factor (-)	o	output
h	Heat transfer coefficient (W/m^2.K)		
K	Conductivity (W/m.K)		
L	Length (m)		
$\dot{m}$	Mass flow rate (kg/s)		
M_r	Mass flow rate ratio of hot fluid to cold fluid (-)		
NTU	Number of transfer units (-)		
Nu	Nusselt number (-)		
P	Pumping power (W)		
P'	Lowest value of pumping power (W)		
Pr	Prandtl number (-)		
q	Heat transfer rate (w)		
Q'	standard heat transfer rate (w)		
Re	Reynolds number (-)		
R_f	Fouling resistance(m^2kw−1)		
T	Temperature (K)		
U	Overall heat transfer coefficient(W/m^2.K)		
u_m	Average velocity of the fluid (m/s)		
WSE	Witte-Shamsundar efficiency (-)		

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