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Analysis and Performance Evaluation of Counter Flow Hairpin Heat Exchangers

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Abstract

Among a variety of heat exchanger types and configurations, hairpin heat exchangers are widely used in engineering processes, especially in chemical and petrochemical industries. They have several operating advantages, such as flexibility and ease of maintenance. The aim of this work is to develop a computer program that is able to evaluate and predict the performance of counter-flow hairpin heat exchangers under different flow conditions. The mathematical framework for thermal and hydraulic calculations is introduced. The developed MATLAB code has been tested for reliability and accuracy against some of the available and approved designs of single-finned tube and bare multi-tube hairpin heat exchangers. Then, it was successfully applied to analyze existing hairpin heat exchangers operating in Alsarir oil field and the Tubrok oil refinery of Arabian Gulf Oil Company (AGOCO) in Libya as case studies. The results show that by changing the operating conditions such as mass flow rates or inlet temperatures of working fluids, the thermal performance of hairpin heat exchangers can be enormously improved without exceeding the allowable pressure drop.

Keywords: Hairpin heat exchangers; counter-flow; finned tube; multi tube; heat exchanger effectiveness.

1. Introduction

Heat exchangers are thermal devices designed for the efficient heat transfer between two fluids, whether the fluids are in direct contact, where fluids are mixed, or separated by a thin solid wall, where fluids are kept unmixed. They are designed in a variety of sizes, shapes, and construction types depending on the industrial application. Hairpin heat exchangers are widely used in cooling fluid processes, especially in chemical and petrochemical industries. They have a simple structural design and an abundance of operating advantages, such as high thermal efficiency, flexibility, ease of maintenance, and low installation cost.

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The performance of heat exchangers can be improved by proper design and optimally setting operational conditions. Therefore, the continued improvement of different design aspects and the performance characteristics of heat exchangers is the main target of both researchers and manufacturers who are working in this field.

To date, numerous theoretical and numerical research efforts have been made to increase the performance of hairpin heat exchangers. For instance, an extrapolative method and data on heat transfer and pressure drop of liquids in double-pipe finned tube heat exchangers were published in [1]. This method was made popular by being incorporated into many articles and handbooks, such as [2]. A review manuscript on the thermal and hydraulic design of hairpin and finned bundle heat exchangers was published in [3]. Analytical solutions for a double-pipe heat exchanger were presented in [4]. The steady-state temperatures of the working fluids were obtained in terms of the location along the length of the heat exchanger, considering non-adiabatic circumstances at the outside surface of the outer pipe. Both counter and parallel flow arrangements were analyzed. General solutions for concentric tube heat exchangers were presented in [5], where the governing equations in non-dimensional differential form were solved analytically. Heat exchanger effectiveness was achieved as a function of the dimensionless exit temperatures for both counter and parallel flow situations. An analysis was made of countercurrent flow double pipe heat exchangers using a mixed lumped-differential formulation in [6]. Numerical results for heat transfer quantities along the thermal entry region were demonstrated as a function of the dimensionless governing parameters. A method for the design and analysis of double pipe heat exchangers was developed in [7]. This method includes modified equations that consider calculations for the pipe equipped with fins of different geometries and shapes. An engineering technique to improve the thermal performance of a concentric tube heat exchanger was introduced in [8]. The procedure is based on inserting porous substrates on both sides of the inner tube wall. The numerical results demonstrated that inserting the porous substrate can enhance the heat exchanger effectiveness considerably for both counter and parallel flow cases. A numerical investigation of heat transfer characteristics in a double-pipe helical heat exchanger under different flow rates and tube sizes was conducted in [9]. The study considered both parallel and counter-flow arrangements. The quality of simulations was approved by the comparison of the obtained Nusselt numbers in the inner tube with the available data in the literature. A numerical investigation of steady laminar mixed convection in a vertical double pipe heat exchanger was carried out for upward parallel flow in [10]. The Nusselt numbers and friction factors in the inner tube and the annular region were included in the results. The results also demonstrated the temperature and velocity profiles at different cross-sections as well as the axial evolution of bulk and wall temperatures. Thermal analysis and calculation of the overall heat transfer coefficient and pressure drop for double-tube heat exchangers were presented in [11]. In the latter reference, design procedures for a double-tube heat exchanger were also presented. Simulation and analysis of heat transfer and flow characteristics in a double pipe heat exchanger were carried out using ANSYS FLUENT software in [12]. The investigation included a study of the performance of parallel and counter flows in concentric tube heat exchangers under different flow conditions. The Discontinuous Galerkin finite element method (DG-FEM) was employed in the numerical analysis of the fully developed laminar convective heat transfer in the modern design of a finned double-pipe heat exchanger in [13]. The inner tube was equipped with longitudinal fins of variable thickness at the tip, which were subjected to the constant heat transfer rate boundary conditions. The tip thickness was controlled by the ratio of tip to base angles as a parameter whose values varied from 0 to 1, corresponding to the fin shapes varying from triangular to rectangular in cross-section. The results revealed that the tip to base angle ratio had a significant effect on improving the performance of the double-pipe heat exchanger in terms of reducing the weight, cost, and frictional loss and increasing the heat exchanger efficiency. An analytical and numerical study was performed to estimate the temperature distribution in steady-state laminar flow through a double-pipe heat exchanger in [14]. The counter and parallel flow arrangements were considered in the study, and the results of the analytical and numerical solutions were compared with each other. A comprehensive review of the main types of double-pipe heat exchangers and key factors that affect heat transfer rate, pressure drop, and various techniques used to achieve optimum effectiveness was reported in [15].

In addition to the previously mentioned efforts, several experimental research studies on double-pipe heat exchangers were carried out. For example, an experimental investigation of the effect of augmentation heat transfer in double pipe heat exchangers equipped with an external helical fin on the inner tube outer surface was conducted in [16]. Different geometrical parameters were considered, and the heat transfer performance was compared with that of smooth tubes. The influence of surface finish on the deposition of suspended particles in double pipe heat exchangers was studied in [17]. The effect of augmented surfaces was also examined. Different surfaces were studied over a range of Reynolds numbers and particle concentrations. The results were compared with the results for a bare tube and other results. An experimental investigation of the friction factor, heat transfer, and exergy loss in an air/water double-pipe heat exchanger was conducted in [18]. The outer surface of the inner pipe was equipped with spring-shaped helical wires. The experiments were carried out for both counter and parallel flows within a specific range of Reynolds numbers, and the results were compared with those for a plain pipe. Heat transfer and pressure drop in turbulent flows through a concentric tube heat exchanger were studied experimentally in [19]. The turbulent flow was generated by inserting a louvered strip into the inner tube and the Reynolds number was controlled within certain values of 6,000 to 42,000. The results proved that the use of louvered strips increases heat transfer rate and friction loss in comparison with the plain tube. Flow patterns on the shell side of the double-pipe heat exchanger were analyzed experimentally in [20]. Laser Doppler Anemometer (LDA) technology was utilized to measure velocity components for the annular pipe with helical fins and pin fins. Experiments were conducted to study turbulent flow and heat transfer in a double-pipe heat exchanger using air flowing on the shell side and water through the inner pipe as working fluids [21]. Several parameters were studied, and the results showed that placing an agitator enhances the rate of heat transfer. A review of the experimental analysis of counter and parallel flow heat exchangers was published in [22]. The study summarized that heat exchanger performance differs from fluid to fluid and temperature to temperature. Also, it was concluded that heat transfer could be enhanced by changing the material of tubes, changing mass flow rates, and using liquid as a heat-absorbing medium. An experimental study was conducted to investigate heat transfer characteristics using different inner tube geometries in a double-pipe heat exchanger [23]. Different flow arrangements were considered to study the thermal performance of the heat exchanger, and the results were compared based on reference results for the flow through the circular smooth tube. The heat transfer characteristics of air in a double-pipe helical heat exchanger were investigated experimentally in [24]. The external surface of the inner tube was equipped with a copper wire fin to augment the rate of heat transfer on the shell side. The experimental results were obtained by varying the temperature and mass flow rate of both hot and cold fluids.

Recently, an experimental study using different fin geometry in turbulent flow through a double pipe heat exchanger was published in [25]. A specified range of Reynolds numbers for hot and cold fluids was considered, and the results were obtained for the pressure drop and heat transfer coefficient. The performance of a double pipe heat exchanger using helical fins was investigated experimentally in [26]. The experimental results were compared with reference data for the plain double pipe heat exchanger.

In summary, several scientific papers, chapters, and books on the design, optimization, performance characteristics, and operation of double-pipe and multi-pipe heat exchangers have been published in the literature. Some of them were based on theoretical studies, and other contributions dealt with numerical simulations. Another set of scientific works included experimental investigations into various aspects of heat exchanger performance. This variety of studies contributed enormously to heat transfer enhancement within such devices and paved the way for more investigation to improve the performance characteristics of concentric pipe and multi-pipe heat exchangers.

In the present work, the influence of several operating parameters on the performance of concentric finned tube and bare multi-tube hairpin heat exchangers is investigated. A computer program has been written and developed for thermal and hydraulic calculations using the prominent computational tool MATLAB. Initially, the developed code has been verified using available and approved hairpin designs, and then it has been used to analyze and predict the performance of the practical bare multi-tube hairpin heat exchanger.

The rest of the manuscript is organized as follows: the methodology is presented in section 2. Then the case studies and verification of the code are introduced in section 3. The results of the performance evaluation are demonstrated in section 4. Finally, the conclusions are made in section 5.

2. Methodology

As stated above, a computer program has been developed to analyze and predict the performance of practical hairpin heat exchangers. The thermal and hydraulic analysis is based on various parameters and correlations that are presented in [11]. The main calculations include heat transfer rate, Nusselt number, an overall heat transfer coefficient, heat exchanger effectiveness, terminal temperature difference, mass velocity rates for hot and cold fluids, Reynolds number, and pressure drop. The well-known Effectiveness–NTU method is utilized to predict the outlet temperatures of the hot and cold fluid streams. General and common assumptions have been considered in the thermal and hydraulic analysis, which can be summarized as follows:

- Heat exchangers are modeled as steady-flow devices.
- The infinitesimal kinetic and potential energy changes are neglected.
- Axial heat conduction along the tube is extremely small and can be neglected.
- The outer surface of the heat exchangers is assumed to be perfectly insulated.
- Finally, within a specific range of operating temperatures, the physical properties are treated as constants at

average values.

2.1. Calculation of Heat Transfer Rate

$$Q = \dot{m}_h c_{ph}\left(T_{h,in} - T_{h,out}\right) \quad (1)$$

$$Q = \dot{m}_c c_{pc}\left(T_{c,out} - T_{c,in}\right) \quad (2)$$

where, Q is the heat transfer rate and the subscripts h and c stand for hot and cold fluids, respectively. $\dot{m}_h$ and $\dot{m}_c$ are mass flow rates; c_{ph} and c_{pc} are specific heats. $T_{h,in}$ and $T_{h,out}$ are inlet and outlet temperatures for hot fluid, $T_{c,in}$ and $T_{c,out}$ are inlet and outlet temperatures of cold fluid, respectively.

2.2. Calculations of Reynolds Number

- For inner tube

$$Re = \frac{\rho V D_i}{\mu} = \frac{G D_i}{\mu} \quad (3)$$

- For annulus

$$Re = \frac{\rho V D_h}{\mu} = \frac{G D_h}{\mu} \quad (4)$$

where Re is the Reynolds number, G is the mass velocity, ρ is the fluid density, V is the fluid velocity, μ is the viscosity of the fluid, D_i is the inner tube diameter and D_h is the hydraulic diameter.

2.3. Calculation of Internal Convection Heat Transfer Coefficient

The correlations for calculating the inner and outer convection heat transfer coefficients, which are used in this work, are based on the correlations introduced in [11]. The inner convective heat transfer coefficient h_i is calculated using the Nusselt number Nu, fluid thermal conductivity k, and inner tube diameter D_i as follows:

$$h_i = Nu \times \frac{k}{D_i} \quad (5)$$

The Nusselt number Nu can be computed according to the following correlations:

- For laminar flow regime (Re < 2300),

$$Nu = 1.86\left[Re\,Pr\left(\frac{D_i}{L}\right)\right]^{0.33}(\mu/\mu_w)^{0.14} \quad (6)$$

where, Pr is the Prandtl number and $(\mu/\mu_w)^{0.14}$ is the viscosity correction factor, which represents the ratio between viscosity at the mean fluid temperature and viscosity at the mean tube wall temperature. L is the length of the heat exchanger.

- For transition flow regime (2300 < Re < 4000)

$$Nu \times \frac{k}{c_p \rho v D_i} = 0.116 \left(\frac{Re^{0.66} - 125}{Re}\right)\left[1 + \left(\frac{D_i}{L}\right)^{0.66}\right] Pr^{-0.66} \left(\frac{\mu}{\mu_w}\right)^{0.14} \quad (7)$$

- For turbulent flow regime (Re > 4000),

$$Nu = 0.023\, Re^{0.8} Pr^{n} \, (\mu/\mu_w)^{0.14} \quad (8)$$

where the properties in this equation are evaluated at the average fluid temperature and the exponent n has the value of 0.4 for heating of the fluid and the value of 0.3 for cooling of the fluid.

2.4. Calculation of the Annulus Convection Heat Transfer Coefficient

The same correlations used for the internal coefficient in tube side fluid are still applicable for the annulus convection heat transfer coefficient h_o, but the internal diameter must be substituted by the annulus hydraulic diameter D_h, which is defined as

$$D_h = 4 \times \frac{flow\ area}{wetted\ perimeter}$$

In the case of the bare tube, for heat transfer calculations, the wetted perimeter corresponds to the external diameter of the internal tube. This is πD_o [2,11]. Then:

$$D_h = 4 \times \frac{\pi(D_s^2 - D_o^2)}{4\pi Do} = \frac{D_s^2 - D_o^2}{Do} \qquad (9)$$

where D_s is the internal diameter of the external tube (the shell). D_o is the external diameter of the inner tube [2] and [11].

For tubes with external longitudinal fins, the equivalent hydraulic diameter for heat transfer calculations can be calculated by:

$$D_h = 4 \times \frac{\left(\frac{\pi D_s^2}{4} - \frac{\pi D_o^2}{4} - N_f f_t f_h\right)}{\pi Do - N_f f_t + 2N_f f_h} \qquad (10)$$

where f_t, f_h and N_f are the fin thickness, fin height and number of fins per tube, respectively.

2.5. Calculation of the Overall Heat Transfer Coefficient

The overall heat-transfer coefficient typically computed based on the outside surface area of the tube as follows:

$$\frac{1}{U_o A_o} = \frac{1}{h_i A_i} + \frac{R_{fi}}{A_i} + \frac{\ln(r_o/r_i)}{2\pi kl} + \frac{R_{fo}}{A_o} + \frac{1}{h_o A_o} \qquad (11)$$

where U_o is the overall heat transfer coefficient, A_o and A_i are the outer and inner surface area of tubes, respectively; R_{fo} and R_{fi} are the fouling factors on shell side and tube side, respectively.

2.6. Terminal Temperature Difference (TTD)

This parameter provides feedback on heat exchanger's performance relative to the heat transfer and is defined as:

$$TTD = T_{h,in} - T_{c,out} \qquad (12)$$

where the increase in TTD indicates insufficient heat transfer while its decrease indicates heat transfer improvement.

2.7. The Effectiveness-NTU Method

The Effectiveness-NTU method presents numerous advantages for the analysis of specified types and sizes of heat exchangers and the computation of the heat transfer rate and the outlet temperatures of the hot and cold fluids. This method is based on the heat transfer effectiveness ε, which is a dimensionless parameter, defined as:

$$\varepsilon = \frac{Q}{Q_{max}} \qquad (13)$$

where Q is the actual heat transfer rate and Q_{max} is the maximum possible heat transfer, which is expressed by:

$$Q_{max} = C_{min}\left(T_{h,in} - T_{c,in}\right) \qquad (14)$$

where C_{min} is the smaller heat capacity rate of $C_C = \dot{m}_c c_{pc}$ and $C_h = \dot{m}_h c_{ph}$.

The effectiveness of a heat exchanger depends on the geometry of the heat exchanger and the flow arrangement. Therefore, the effectiveness of a counterflow double-pipe heat exchanger can be expressed as follows:

$$\varepsilon = \frac{1 - exp[(-UA/C_{min})(1 - C)]}{1 - C \times exp[(-UA/C_{min})(1 - C)]} \qquad (15)$$

where the dimensionless group *(UA/C_{min})* is known as the number of transfer units *NTU* and is expressed as

$$NTU = \frac{UA}{C_{min}} \qquad (16)$$

2.8. Calculations of Pressure Drop for the Internal-Tube Fluid

The common expression to calculate pressure drop through a pipe is

$$\Delta P_f = 4f\frac{L}{D_i}\rho\frac{v^2}{2}\left(\frac{\mu}{\mu_w}\right)^{\alpha} \qquad (17)$$

where α = – 0.14 for turbulent flow and α = – 0.25 for laminar flow [11].

The following expressions can be used for computing the friction factor [11]:

- For the laminar flow, the friction factor can be calculated with the following expression:

$$f = \frac{16}{Re} \qquad (18)$$

- In the turbulent region, the friction factor depends on the surface roughness of the tube. However, some simplified correlations that are valid for particular situations have been suggested.

For 3/4 or 1 in smooth tubes, a recommended expression is

$$f = 0.0014 + \frac{0.125}{Re^{0.32}} \tag{19}$$

For commercial steel heat exchanger tubes,

$$f = 0.0035 + \frac{0.264}{Re^{0.42}} \tag{20}$$

The pressure loss due to one return bend is half velocity heads based on the tube velocity;

$$\Delta P_{rb} = \frac{G^2}{4\rho} \tag{21}$$

The total pressure drop in this case is

$$\Delta P_T = \Delta P_f + \Delta P_{rb} \tag{22}$$

For heat exchangers with more than one tube, an additional pressure drop is considered due to a construction loss at entry, expansion loss at the exit, plus the return bend loss. This pressure drop can be calculated as

$$\Delta P_a = k_l \frac{G^2}{2\rho} N_p \tag{23}$$

where N_p is the number of tube passes, k_l = 0.9 for one tube pass and k_l = 1.6 for two or more tube passes.

The total pressure drop in this case is

$$\Delta P_T = \Delta P_f + \Delta P_a \tag{24}$$

2.9. Calculations of Pressure Drop for the Annulus Fluid

The same expressions for the internal tube fluid are still valid. However, the equivalent hydraulic diameter must be used instead of the internal diameter.

- The equivalent hydraulic diameter for a bare tube is given by:

$$D_h = 4 \times \frac{\pi(D_s^2 - D_o^2)/4}{\pi(D_s + D_o)} \tag{25}$$

- The equivalent hydraulic diameter for a longitudinal finned tube is given by:

$$D_h = 4 \times \frac{(\frac{\pi D_s^2}{4} - \frac{\pi D_o^2}{4} - N_f f_t f_h)}{\pi Ds + \pi Do - N_f f_t + 2N_f f_h} \quad (26)$$

The pressure loss due to one return bend housing is half velocity heads based on the annulus velocity;

$$\Delta P_{rb} = \frac{G^2}{4\rho} \quad (27)$$

Nozzle losses are calculated using the following expression:

$$\Delta P_n = k_n \frac{G^2}{2\rho} \quad (28)$$

where k_n = 1.0 for inlet and k_n = 0.5 for outlet.

The total pressure drop in the shell side (ΔP_{Ts}) is equal to:

$$\Delta P_{Ts} = \Delta P_f + \Delta P_{rb} + \Delta P_n \quad (29)$$

2.10. Computer Program Description and Limitations

The MATLAB code was written and developed based on the Effectiveness-NTU method, which is a widely used technique to analyze the performance of heat exchangers. The program consists of three main parts, which are:

- The input program, where the primary information is entered and then the data is supplied to the main program.
- The main program, which is the core of the performance calculations, a closed-loop is used to obtain the thermal and hydraulic parameters in two major steps. In the first step, an estimation of the parameters is made depending on the physical properties at the inlet temperatures of the fluids. Then the final results are achieved at an average temperature of each fluid within the second step.
- The physical properties sub-program, where the essential physical properties for water and kerosene are provided by this sub-program to perform the necessary calculations. The stored data bank for water covers a range of temperatures between 0 ^{o}C and 316 ^{o}C (from 32^{o}F to 600 ^{o}F) and for kerosene between 21^{o}C and 60 ^{o}C (from 70 ^{o}F to 140 ^{o}F).

In general, the developed program can perform performance calculations for both concentric pipe and multi-pipe hairpin heat exchangers for a wide range of Reynolds numbers, including laminar and turbulent flow regimes. The program is capable of being run for gas to liquid and liquid to liquid heat exchangers. However, in the current study, the investigations are limited to turbulent flows through practical hairpin heat exchangers with water to water and kerosene to water.

3. Case Studies and Verification of the Program

Data for two actual hairpin heat exchangers has been considered to verify the developed program. The first heat exchanger is a finned tube double-pipe heat exchanger located in the Alsarir oil field of the Arabian Gulf Oil Company (AGOCO) in Libya. The second exchanger is a bare multi-tube heat exchanger located in the Tubrok refinery of the same company in Libya. Then the program was utilized to evaluate and predict the performance of the second heat exchanger under different operating parameters.

3.1. Finned Tube Double Pipe Heat Exchanger

Table (1) presents a summary of the design data for the finned tube double pipe heat exchanger as extracted from the worksheet supplied by the vendor. The tabulated comparison between the present code results and the available specifications of the heat exchanger is shown in Table (2).

Table 1: Design Data for the counter flow finned tube double pipe heat exchanger located in the Alsarir oil field in Libya as extracted from the worksheet supplied by the vendor.

Unit data	Shell side	Tube side
Working fluids	Kerosene	Water
Flow rates (lb/hr)	3773	2849
Inlet temperatures, (^oF)	140	93
Outlet temperatures, (^oF)	100	120
Tube inside diameter,(ft)	-	0.0652
Tube outside diameter,(ft)	-	0.083
Number of tubes	1	1
Number of fins per tube	20	
Fin height (in)	7/16	
Fin thickness (in)	0.035	
Shell inside diameter, (ft)	0.17225	-
Total length of exchanger, (ft)	203	
Number of units	4	
Unit arrangement	Series	Series
Overall coefficient, (Btu/hr.ft^2. ^oF)	17.8	
Heat flow, (Btu/hr)	77000	
Fouling factor	0.001	0.002
Allowable pressure drop (psi)	10	10
Actual pressure drop in (psi)	3.3	8.7

An excellent agreement can be noted between the thermal parameters calculated via the present code and the actual design data of the heat exchanger. The percentage of error did not exceed 3% in all parameters except that for the pressure drop on the shell side, where it was about 6%. The small differences in the values of some parameters are related to the assumptions and simplifications that have been made and because of the approximation of the physical properties of the working fluids. The current effectiveness of the heat exchanger under the actual operating conditions is found to be approximately 85%. This relatively high percentage indicates that the heat exchanger is working very well.

Table 2: Comparison between the present results and design data extracted from the worksheet of the vendor.

Comparison parameters	The present work	Reported
Shell side Fluid (Cold fluid)	Kerosene	Kerosene
Tube side Fluid (Hot fluid)	water	Water
Tube inside diameter,(ft)	0.0652	0.0652
Tube outside diameter,(ft)	0.083	0.083
Number of tubes	1	1
Shell inside diameter, (ft)	0.1723	0.1723
Total length of exchanger, (ft)	203	203
Number of units	4	4
Number of fins per tube	20	20
Units arrangement	series	series
Inlet temperature of tube side fluid,(^{o}F)	93	93
Outlet temperature of tube side fluid,(^{o}F)	119.9	120
Specific heat of tube side fluid, (Btu/lb. ^{o}F)	0.749	-
Viscosity of tube side fluid, (centipoises)	1.54	-
Thermal conductivity of tube side fluid, (Btu/hr.ft.^{o}F)	0.367	-
Inlet temperature of shell side fluid,(^{o}F)	140	140
Outlet temperature of shell side fluid,(^{o}F)	100.3	100
Specific heat of shell side fluid, (Btu/lb.^{o}F)	0.675	-
Viscosity of shell side fluid, (centipoises)	0.797	0.69 - 0.9
Thermal conductivity of shell side fluid, (Btu/hr.ft. ^{o}F)	0.02	-
Tube side heat transfer coefficient, (Btu/hr.ft^{2}. ^{o}F)	883.30	-
Shell side heat transfer coefficient, (Btu/hr.ft^{2}. ^{o}F)	18.84	-
Overall coefficient, (Btu/hr.ft^{2}. ^{o}F)	17.4	17.8
Heat flow, (Btu/hr)	76383	77000
Shell side mass flow rate, (lb/hr)	2849	2849
Tube side mass flow rate, (lb/hr)	3773	3773
Shell side flow velocity, (ft/s)	1.41	1.4
Tube side flow velocity, (ft/s)	3.83	3.8
Tube side Reynolds number	3374.4	-
Heat exchanger effectiveness, (%)	84.5	85.2
Tube side pressure drop in (psi)	8.67	8.7
Shell side pressure drop in (psi)	3.49	3.3

3.2. Bare Multi-Tube Heat Exchanger

In the second test, the present code was verified by comparison with the actual multi-pipe heat exchanger located in the Tubrok refinery of the Arabian Gulf Oil Company in Libya. Table (3) provides a summary of design specifications as extracted from the worksheet supplied by the vendor.

A comparison between the present code results and the existing data for the heat exchanger is shown in table (4). A very good agreement can be seen between the thermal parameters calculated via the developed code and the actual design data of the heat exchanger, where the percentage of error is limited within a small range of 1% to 6% for all parameters. The slight differences in the values of some parameters are owing to the approximation of the physical properties of the oily water and to the assumptions and simplifications that have been made.

As a final point, it is found that the effectiveness of the heat exchanger is relatively low at only 66%, which is a sign of poor performance of the heat exchanger under the current operating conditions. This particular point encourages us to investigate the performance of this heat exchanger under a new set of operating parameters.

Table 3: Design Data for the counter flow bare multi- tube heat exchanger located in the Tubrok refinery in Libya as extracted from the worksheet supplied by the vendor.

Unit data	Shell side	Tube side
Working fluids	Fresh Water	Oily Water
Flow rates (lb/hr)	14600	14600
Inlet temperatures, (^oF)	80	247
Outlet temperatures, (^oF)	190	138
Tube inside diameter,(ft)	-	0.0589
Tube outside diameter,(ft)	-	0.0729
Number of tubes	1	7
Shell inside diameter, (ft)	0.3354	-
Total length of exchanger, (ft)	200	
Number of units	4	
Unit arrangement	Series	Series
Overall coefficient, (Btu/hr.ft^2. ^oF)	86.2	
Heat flow, (Btu/hr)	1606000	
Fouling factor	0.003	0.003
Allowable pressure drop (psi)	20	20
Actual pressure drop in (psi)	-	3.2

Table 4: Comparison between the current results and design data extracted from the worksheet of the vendor.

Comparison parameters	The present work	Reported
Shell side Fluid (Cold fluid)	Fresh water	Fresh water
Tube side Fluid (Hot fluid)	Oily water	Oily water
Tube inside diameter,(ft)	0.0589	0.0589
Tube outside diameter,(ft)	0.0729	0.0729
Number of tubes	7	7
Shell inside diameter, (ft)	0.3354	0.3354
Total length of exchanger, (ft)	200	200
Number of units	4	4
Number of fins per tube	20	20
Units arrangement	series	series
Inlet temperature of tube side fluid,(^oF)	247	247
Outlet temperature of tube side fluid,(^oF)	140	138
Specific heat of tube side fluid, (Btu/lb. ^oF)	1.012 – 1.004	-
Viscosity of tube side fluid, (centipoises)	0.569 – 0.755	-
Thermal conductivity of tube side fluid, (Btu/hr.ft.^oF)	0.396 – 0.391	-
Inlet temperature of shell side fluid,(^oF)	80	80
Outlet temperature of shell side fluid,(^oF)	188	190
Specific heat of shell side fluid, (Btu/lb.^oF)	0.998	-
Viscosity of shell side fluid, (centipoises)	2.08 – 1.235	-
Thermal conductivity of shell side fluid, (Btu/hr.ft. ^oF)	0.355 – 0.375	-
Tube side heat transfer coefficient, (Btu/hr.ft^2.^oF)	334	-
Shell side heat transfer coefficient, (Btu/hr.ft^2.^oF)	339	-
Overall coefficient, (Btu/hr.ft^2.^oF)	81	86.2
Heat flow, (Btu/hr)	1,573,646	1,606,000
Heat exchanger effectiveness, (%)	64.7	66
Shell side mass flow rate, (lb/hr)	14600	14600
Tube side mass flow rate, (lb/hr)	14600	14600
Tube side pressure drop in (psi)	6.71	7
Shell side pressure drop in (psi)	11.05	-

4. Results and Discussion

In this section, the actual hairpin counter flow heat exchanger in the second test mentioned above is considered for thermal and hydraulic analysis. This particular heat exchanger has been chosen for the investigation because of its low effectiveness, which has a value of 66%. Heat exchanger specifications and data collected for the current operating conditions summarized in tables (3) and (4) are used as a reference. The computations are performed based on increasing and decreasing the mass flow rate of each fluid and the inlet temperature of the cold fluid.

4.1. Effect of Changing the Mass Flow Rate of the Hot Fluid

In the first performance test, the mass flow rate of the oily water on the tube side was gradually decreased and increased by 40% with a constant step of 10% in each computation, whereas the other operating parameters were kept constant. Table (5) shows the results of the program computations.

Table 5: Observation table for the effect of changing the mass flow rate of the hot fluid

% of change in hot fluid flow rate	Hot fluid mass flow rate (lb/hr)	Tube side			Shell side					
		$T_{h,in}$ (^oF)	$T_{h,out}$ (^oF)	ΔP_T (*psi*)	$T_{c,in}$ (^oF)	$T_{c,out}$ (^oF)	ΔP_S (*psi*)	*TTD* (^oF)	*Q* (*Btu/hr*)	ε (*%*)
-40%	8760	247	107.9	2.71	80	163.9	11.09	83.1	1.2204×10^6	83.3
-30%	10220	247	116.9	3.55	80	171.5	11.08	75.5	1.3326×10^6	77.9
-20%	11680	247	125.4	4.51	80	177.8	11.07	69.2	1.4241×10^6	72.8
-10%	13140	247	133.3	5.56	80	182.9	11.06	64.1	1.4991×10^6	68.1
Reference data	**14600**	**247**	**140**	**6.71**	**80**	**188**	**11.05**	**59**	$\mathbf{1.5736\times10^6}$	**64.7**
+10%	16060	247	147	7.97	80	190.7	11.05	56.3	1.6129×10^6	66.3
+20%	17520	247	152.9	9.32	80	193.7	11.04	53.3	1.6565×10^6	68.1
+30%	18980	247	158.2	10.77	80	196.2	11.03	50.8	1.6937×10^6	69.6
+40%	20440	247	163	12.32	80	198.4	11.03	48.6	1.7257×10^6	70.9

From the observation table, it can be seen that the decrease in the hot fluid mass flow rate by 10% to 40% causes a decrease in the outlet temperature of both cold and hot fluids and the heat transfer rate as well. Although the effectiveness of the heat exchanger has been increased from 64.7% at a flow rate of 14600 (lb/hr) to 83% at a flow rate of 8760 (lb/hr), the terminal temperature difference has also been increased from 59 (^{o}F) to 83.1 (^{o}F) which causes a large undesirable change in the performance of the heat exchanger. The pressure drop on the tube side decreases as the mass flow rate decreases from 10% to 40%.

On the other hand, it can be observed that the increase in the hot fluid mass flow rate from 14600 (lb/hr) to 20440 (lb/hr) causes an increase in the outlet temperature of both cold and hot fluids. This is associated with an increase in both the heat transfer rate and the effectiveness of the heat exchanger. The terminal temperature difference has decreased from 59 (^{o}F) to 48.6 (^{o}F) which is a sign of improvement in the heat exchanger performance. The pressure drop on the tube side gradually increases with the increase in the mass flow rate of oily water. However, it did not exceed the allowable pressure drop.

Figure (1) shows the variation of the overall heat transfer coefficient versus the hot fluid mass flow rate. It is clear that the overall heat transfer coefficient increases as the mass flow rate of the oily water increases and vice versa.

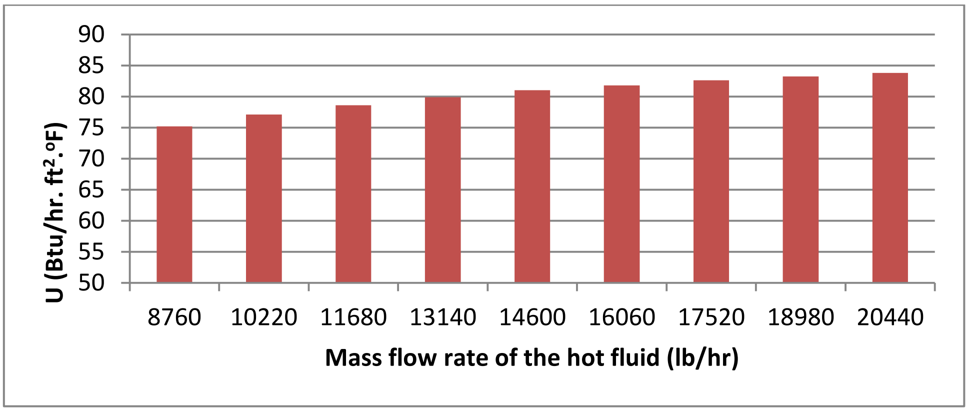

Figure 1: The overall heat transfer coefficient as a function in hot fluid mass flow rate.

4.2. Effect of Changing the Mass Flow Rate of the Cold Fluid

In the second performance test, the mass flow rate of freshwater was decreased and increased with a 10% step in each computation between 8760 and 20440 (lb/hr). This represents up to ±40% in comparison with the reference value of 14600 (lb/hr). The obtained results are demonstrated in table (6) and figure (2). It can be seen that the decrease in the cold fluid mass flow rate causes an increase in the outlet temperature of both cold and hot fluids and the heat exchanger effectiveness as well. In this case, the pressure drop on the shell side decreases as the mass flow rate reduces from 14600 to 8760 (lb/hr). Even though the heat transfer rate has decreased with the decrease in the mass flow rate of the cold fluid, the terminal temperature difference has also decreased, which has a positive effect on the heat exchanger performance. On the other hand, it can be observed that the increase in the cold fluid mass flow causes a decrease in the outlet temperature of both cold and hot fluids. This is associated with an increase in terminal temperature difference, the heat transfer rate, and the effectiveness of the heat exchanger. The pressure drop on the shell side rapidly increases until it exceeds the allowable pressure drop when the percentage increase in mass flow rate of the cold fluid reaches 40%.

Table 6: Observation table for the effect of changing the mass flow rate of the cold fluid.

% of change in cold fluid flow rate	Cold fluid mass flow rate (lb/hr)	Tube side			Shell side			*TTD* (^oF)	*Q* (*Btu/hr*)	*ε* (%)
		$T_{h,in}$ (^oF)	$T_{h,out}$ (^oF)	ΔP_T (*psi*)	$T_{c,in}$ (^oF)	$T_{c,out}$ (^oF)	ΔP_S (*psi*)			
-40%	8760	247	168.3	6.63	80	211.9	4.13	35.1	1.1542×10^6	79
-30%	10220	247	159.9	6.66	80	205.3	5.6	41.7	1.2777×10^6	75
-20%	11680	247	152.5	6.68	80	198.8	7.2	48.2	1.3853×10^6	71.2
-10%	13140	247	146.1	6.7	80	192.8	9	54.2	1.4792×10^6	67.5
Reference data	**14600**	**247**	**140**	**6.71**	**80**	**188**	**11.05**	**59**	$\mathbf{1.5736\times10^6}$	**64.7**
+10%	16060	247	135.6	6.73	80	181.9	13.3	65.1	1.6327×10^6	66.7
+20%	17520	247	131.2	6.74	80	177	15.7	70	1.6955×10^6	69.3
+30%	18980	247	127.4	6.75	80	172.5	18.3	74.5	1.7508×10^6	71.6
+40%	20440	247	124.1	6.76	80	168.3	21.2	78.7	1.7998×10^6	73.6

Figure 2 shows the variation of the overall heat transfer coefficient as a function of the mass flow rate of the cold fluid. The overall heat transfer coefficient is directly proportional to the cold mass flow rate. Increasing the mass flow rate from 14600 to 20440 (lb/hr) increases the overall heat transfer coefficient by 16%, and decreasing it from 14,600 to 8760 (lb/hr) results in a decrease in the overall heat transfer coefficient by 22.8%.

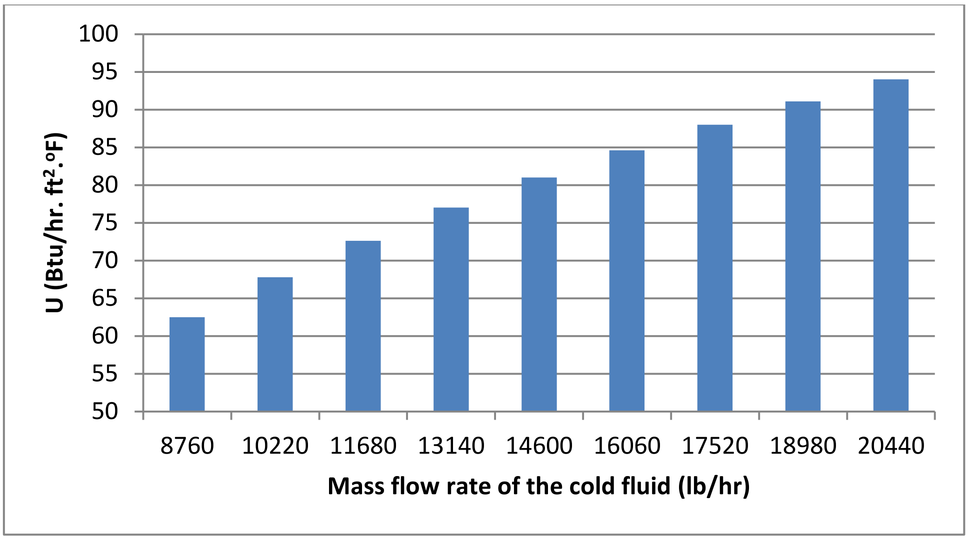

Figure 2: The overall heat transfer coefficient as a function in cold fluid mass flow rate.

4.3. Effect of Changing the Inlet Temperature of the Cold Fluid

In the third performance test, the inlet temperature of the cold fluid was changed within the range of 70 to 90 ^oF by a step of 5 ^{o}, while the other operating parameters were kept constant.

Table 7: Observation table for the effect of changing the inlet temperature of the cold fluid.

% of change in the inlet temperature of cold fluid	shell side $T_{c,in}$ (^oF)	$T_{c,out}$ (^oF)	ΔP_S (*psi*)	Tube side $T_{h,in}$ (^oF)	$T_{h,out}$ (^oF)	ΔP_T (*psi*)	*TTD* (^oF)	*U* ($Btu/hr.ft^2.^oF$)	*Q* (*Btu/hr*)	ε (%)
+12.5%	90	180.5	11.07	247	147.2	6.69	66.5	80.4	1.3187×10^6	57.6
+6.25%	85	183.8	11.06	247	143.8	6.7	63.2	80.7	1.4396×10^6	61
Reference data	**80**	**188**	**11.05**	**247**	**140**	**6.71**	**59**	**81**	$\mathbf{1.5736\times10^6}$	**64.7**
- 6.25%	75	190.5	11.04	247	137.1	6.72	56.5	81.3	1.6829×10^6	67.2
-12.5%	70	193.8	11.03	247	133.8	6.73	53.2	81.5	1.8039×10^6	69.9

Table (7) illustrates the results for the outlet temperature of both fluids, terminal temperature difference, heat transfer rate, effectiveness, and pressure drop on both sides of the heat exchanger as a function of the percent change in the inlet temperature of the freshwater.

It can be noticed that the decrease in the inlet temperature of the water from 80 to 70 ^{o}F causes a considerable drop in the outlet temperature of the hot fluid and the terminal temperature difference by 4.42% and 9.8%, respectively. This is associated with the increase in the heat transfer rate. Consequently, the effectiveness of the heat exchanger was increased by 8%. Whereas the pressure drop on both sides is almost unchanged. This action causes an improvement in the heat exchanger's overall performance. Conversely, when the inlet temperature of the cold fluid rises from 80 to 90 ^{o}F, the terminal temperature difference increases by 12.7%, and other performance parameters decrease. These give a warning about a decrease in the overall performance of the heat exchanger.

5. Conclusions

In this work, the prediction and analysis of the performance of hairpin heat exchangers were conducted effectively. A computer program was developed based on the famous Effectiveness-NTU method. The program was tested and then applied to study the performance of a practical multi-tube heat exchanger. In general, it is confirmed that the effectiveness of the heat exchangers highly depends on the mass flow rate of the working fluids as well as their inlet temperatures. The following conclusions are obtained:

- The heat transfer rate is directly proportional to the mass flow rate of the hot fluid, whereas the terminal temperature difference is inversely proportional to the mass flow rate of the hot fluid.
- Both the heat transfer rate and the terminal temperature difference are directly proportional to the mass flow rate of the cold fluid.
- The heat transfer rate and the effectiveness of the heat exchangers are inversely proportional to the inlet temperature of the cold fluid. Whereas, the terminal temperature difference is directly proportional to the inlet temperature of the cold fluid.
- For the particular case study considered in this paper, it is found that the heat exchanger does not work at its optimum operating conditions. However, the performance of the heat exchanger can be improved by either increasing the mass flow rate of the hot fluid, decreasing the mass flow rate of the cold fluid, decreasing the inlet temperature of the cold fluid, or by a combination of these parameters as well.

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References

[1] B. Delorenzo, and E. D. Anderson. "Heat Transfer and Pressure Drop of Liquids in Double Pipe Fin-Tube Exchangers", *ASME Trans*, vol 67, pp. 697-702, 1945.

[2] D. Q. Kern. *Process Heat Transfer*. McGraw-Hill, 1951.

[3] G. P. Purohit. "Thermal and Hydraulic Design of Hairpin and Finned Bundle Heat Exchangers", *Chemical Engineering*, vol 90(10), pp. 62-70, May1983.

[4] R.C. Prasad. "Analytical solution for a double-pipe heat exchanger with non-adiabatic condition at the outer surface", *International Communications in Heat and Mass Transfer*, vol. 14(6), pp. 665-672, 1987.

[5] R.C. Prasad. "Generalized solution and effectiveness for concentric tube heat exchangers", *International Journal of Heat and Mass Transfer*, vol. 31(12), pp. 2571-2578, 1988.

[6] F. S. Neto and R.M. Cotta. "Counter flow double-pipe heat exchanger analysis using a mixed lumped-differential formulation", *International Journal of Heat and Mass Transfer*, vol. 35(7), pp. 1723-1731, 1992.

[7]J. Taborek. "Double Pipe Heat Exchanger," in *Heat Exchanger Design Hand Book*, vol. 3. G. F. Hewitt, Executive Editor, by Begell House, Inc 1998.

[8] M. K. Alkam, and M.A. Al-Nimr. "Improving the performance of double-pipe heat exchangers by using porous substrates", *International Journal of Heat and Mass Transfer*, vol. 42(19), pp. 3609-3618, 1999.

[9] T. Rennie, and V. Raghavan. "Numerical studies of a double-pipe helical heat exchanger", *Applied Thermal Engineering*, vol. 26, pp. 1266-1273, 2006.

[10]I. Voicu, T. Maré, N. Galanis, J. Miriel, I. Colda. "Mixed convection in a vertical double pipe heat exchanger", *International Journal of Thermal Sciences*, vol. 46(6), pp. 540-550, 2007.

[11]E. Cao. *Heat transfer in process engineering*. McGraw-Hill Companies, Inc. 2010.

[12] D. Bhanuchandrarao, M. A. Chakravarthy, Y. Krishna, V. V. Subba Rao, and T. H. Krishna. "CFD analysis and performance of parallel and counter flow in concentric tube heat exchangers", *International Journal of Engineering Research &Technology*, vol. 2(11), pp. 2782- 2792, Nov. 2013.

[13] K.S. Syed, Muhammad Ishaq, Zafar Iqbal, Ahmad Hassan. "Numerical study of an innovative design of a finned double pipe heat exchanger with variable fin-tip thickness", *Energy Conversion and Management*, vol.98, pp.69-80, 2015.

[14] Burak Kursun, Bülent Nafi Örnek, Korhan Ökten. "Analytical model for estimation of temperature distribution in parallel and counter flow double pipe heat exchangers", *Sigma Journal of Engineering and Natural Sciences*, vol 35 (3), pp. 557-569, 2017.

[15] Mustafa M. Gabir and Dhirgham Alkhafaji (2021). "Comprehensive Review on Double Pipe Heat Exchanger Techniques". *Journal of Physics: Conference Series*. [On- line]. 1973 012013. Available:https://iopscience.iop.org/article/10.1088/1742-6596/1973/1/012013 [Oct. 5, 2021].

[16] Shou-Shing Hsieh, Chihng-Tsung Liauh, and Anthony C Ku. "Heat transfer coefficients of double pipe heat exchanger with helical type roughened surface", *Heat Recovery Systems and CHP*, vol. 7(2), pp119-127, 1987.

[17]W. B Freeman, J. Middis and H.M Müller-Steinhagen. "Influence of augmented surfaces and of surface finish on particulate fouling in double pipe heat exchangers", *Chemical Engineering and Processing: Process Intensification*, vol. 27(1), pp1-11, 1990.

[18] Ebru Kavak Akpinar. "Evaluation of heat transfer and exergy loss in a concentric double pipe exchanger equipped with helical wires", *Energy Conversion and Management*, vol. 47(18–19), pp 3473-3486, 2006.

[19] Smith Eiamsa-ard, Somsak Pethkool, Chinaruk Thianpong, and Pongjet Promvonge, "Turbulent flow heat transfer and pressure loss in a double pipe heat exchanger with louvered strip inserts", *International Communications in Heat and Mass Transfer*, vol. 35(2), pp 120-129, 2008.

[20]Li Zhang, Wenjuan Du, Jianhua Wu, Yaxia Li, and Yanwei Xing. "Fluid flow characteristics for shell side of double-pipe heat exchanger with helical fins and pin fins", *Experimental Thermal and Fluid Science*, vol. 36, pp 30-43, 2012.

[21] Mohsen Sheikholeslami, M. Hatami, M. Jafaryar, F. Farkhadnia, Davood Domiri Ganji, and Mofid Gorji-Bandpy. "Thermal management of double-pipe air to water heat exchanger", *Energy and Buildings*, vol. 88, pp 361-366, 2015.

[22] A. Magadum, and T. C. Mestri. "Review of experimental analysis of parallel and counter flow heat exchanger", *International Journal of Engineering Research & Technology*, vol. 5(2), pp 395-397, Feb. 2016.

[23] E. F. Abbas, S. R. Aslan, and T. A. Ridha. "Experimental Investigation of Heat Transfer Enhancement Methods on the Thermal Performance of Double Pipe Heat Exchanger", *International Journal of Engineering Research & Technology*, vol. 6(2), pp 169-174, Feb. 2017.

[24] Davood Majidi, Hashem Alighardashi, and Fatola Farhadi. "Experimental studies of heat transfer of air in a double-pipe helical heat exchanger", *Applied Thermal Engineering*, vol. 133, pp 276-282, Mar. 2018.

[25] Osama A. Mohsen, Muhannad A. R. Muhammed, and Basim O. Hasan. " Heat Transfer Enhancement in a Double Pipe Heat Exchanger Using Different Fin Geometries in Turbulent Flow", *Iranian Journal of Science and Technology, Transactions of Mechanical Engineering*, vol. 45, pp 461–471, Jun. 2020.

[26] S. Sivalakshmi, M. Raja, G. Gowtham. "Effect of helical fins on the performance of a double pipe heat exchanger", *Materials Today: Proceedings*, vol. 43(2), pp 1128-1131, 2021.