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Review Article

A theoretical approach to be applied in heat exchangers by using the thermal efficiency concept and the second law of thermodynamic

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ABSTRACT: A review of the concepts of thermal efficiency and thermal and hydraulic irreversibilities is presented, applying the second law of thermodynamics and the thermodynamic Bejan number. An example problem, typical of thermal heat exchange between two fluids, is given, with a dimensionless solution for parallel and counterflow flows. The quantities of interest presented through the example are thermal efficiency, thermal effectiveness, thermal irreversibility, the relationship between outlet and inlet temperatures versus the number of thermal units, and outlet temperature for hot fluid. The theory presented in this review has been applied to numerous problems related to heat exchangers over the last three years, as per references.

KEYWORDS: thermal efficiency concept; second law of thermodynamics; Bejan number; heat exchangers

1. Introduction

A review of the second law of thermodynamics on heat and mass transfer was published during the 1980s, where the fundamental mechanisms responsible for entropy generation were analyzed[1]. In addition, it was demonstrated how to balance the irreversibility of heat transfer versus viscous dissipation and how reducing irreversibility at the component level affects the entire system[1,2].

Herwig[3] discusses and presents research suggestions related to the second law of thermodynamics and suggests applying a procedure for the analysis of thermal systems. He called this procedure the Second Law Analysis–SLA.

Heat exchanger efficiency, defined based on the second law, provides a new way to design and analyze heat exchangers.

Efficiency is defined for heat exchangers based on the second law of thermodynamics: thermal efficiency represents the true potential concerning a heat exchanger that enables the maximum heat exchange potential for the configuration under analysis. Potential does not mean the heat transfer rate between fluids, represented by thermal effectiveness, the relationship between the actual heat transfer rate, and the maximum theoretically possible heat transfer rate. When efficiency is high, the temperature difference between the fluids is high. When efficiency is minimal, the temperature difference between the fluids reaches a minimum level for the heat exchanger configuration under analysis. In this last condition, the effectiveness got its maximum possible value.

The definition discussed here was established by Fakheri[1]. It is shown that there is an ideal balanced counterflow heat exchanger that has the same product UoA, the same arithmetic mean temperature difference, and the same temperature ratio between cold and hot fluid that corresponds to the current heat exchanger. Also, the heat capacities of the ideal heat exchanger are equal to the minimum heat capacity of the actual heat exchanger. Therefore, the ideal heat exchanger generates the least entropy, is the most efficient, the least irreversible, and allows maximum heat transfer between fluids.

As with the definition of efficiency for fins, efficiency for heat exchangers has significant practical consequences since the heat exchange potential between fluids can be measured and obtained. When the thermal efficiency has reached its minimum value for a given configuration, the heat exchange potential is exhausted and has maximum thermal irreversibility. Therefore, it allows savings in heat exchanger design since it makes no sense to work beyond the minimum thermal efficiency.

Another important concept used in this chapter is the ratio between the thermal entropy generation rates and the total entropy generation rate of the heat exchanger, called the thermodynamic Bejan number[2]. When the viscous entropy generation rate is very high concerning the thermal entropy generation rate, the cost-benefit caused by viscous dissipation in the flow is very high.

From this point on, it will cover, in mathematical terms, a consolidated part of the long trajectory covered by some of the researchers who established the concepts mentioned above. They indeed overcame the obstacles and paved the way. The credits due are explicit in the references cited.

The concepts presented and discussed in this work were applied to heat exchanger problems[3–14] to understand how they can help classify, sizing, and optimize heat exchangers. Each application has its peculiarities in construction details, physical dimensions, the use of special devices, and the area of application in the industry. The initial difficulty in applying the model lies in obtaining the peculiarities. The literature available makes it possible to get such information if it is not received from the sector interested in the analysis.

2. Methodology

Bejan[2] uses the first and second laws of thermodynamics to determine the expression that makes it possible to calculate the rate of entropy generation in a system:

$$\dot{S}_{gen} = \frac{\dot{Q}\Delta T}{T^2(1+\frac{\Delta T}{T})} + \frac{\dot{m}}{\rho D_h T}(-\frac{dp}{dx}) = \dot{S}_{genT} + \dot{S}_{genf} \tag{1}$$

$\dot{S}_{genT}$ and $\dot{S}_{genf}$ correspond to the entropy generation rate associated with the temperature field and the entropy generation rate related to viscous dissipation.

Equation (1) can be better interpreted in the context of our work if we associate common concepts in heat transfer and fluid mechanics, that is:

$$f = \frac{\rho D_h}{2G^2}(-\frac{dp}{dx}) \tag{2}$$

f is the friction factor. D_h is the hydraulic diameter related to the heat exchanger $\frac{dp}{dx}$ corresponds to the pressure drop.

$$G = \frac{\dot{m}}{A} \tag{3}$$

$\dot{m}$ is the mass flow rate and A the cross-sectional area through which the fluid flows.

$$Re = \frac{GD_h}{\mu} \tag{4}$$

Re is the Reynolds number associated with the flow.

$$D_h = \frac{4A}{p} \tag{5}$$

p is the perimeter associated with the cross-sectional area.

$$St = \frac{\frac{\dot{Q}}{(p^2 \Delta T)}}{C_p G} \tag{6}$$

St is the Stanton number and $\frac{\dot{Q}}{(p^2 \Delta T)}$ corresponds to the average heat transfer coefficient.

Finally, we can rewrite the equation for the rate of entropy generation:

$$\dot{S}_{gen} = \frac{\dot{Q}^2}{4T^2 \dot{m} C_p St} + \frac{2\dot{m}^3 f}{\rho^2 T A^2} \tag{7}$$

The Stanton number is the prevalent factor for thermal irreversibility, and the friction factor is dominant for viscous irreversibility. It is a fact that the Stanton number and friction factor grow simultaneously, and what reduces thermal irreversibility increases viscous irreversibility.

Returning to the crucial parameter for our analyses, the definition of thermal efficiency for heat exchangers, we have:

$$\eta_T = \frac{\dot{Q}}{\dot{Q}^{\eta}_{max}} \tag{8}$$

where,

$$\dot{Q}^{\eta}_{max} = U_O A(\bar{T}_h - \bar{T}_c) \tag{9}$$

$\dot{Q}^{\eta}_{max}$ is the maximum heat transfer rate and $(\bar{T}_h - \bar{T}_c)$ is the arithmetic mean temperature difference between the fluids.

Fakheri[1] clarifies that Equations (8) and (9) establish that any actual heat exchanger with the same product $U_o A$ and the same arithmetic mean temperature difference has a thermal efficiency lower than 1. The maximum heat transfer rate occurs in a balanced counterflow heat exchanger.

Through Equations (8) and (9), it is evident that if the thermal efficiency is determined, the heat transfer rate can be obtained through the maximum heat transfer rate.

Thermal efficiency can be determined for usual heat exchangers through the following expression:

$$\eta_T = \frac{tanh(Fa)}{Fa} \tag{10}$$

where, Fa is the fin analogy number.

Equation (10) means that usual heat exchangers have the same functional expression for the thermal efficiency as the expression used to determine the efficiency of an insulated constant-area fin. If the expression is applicable, there is now an opening for thermal efficiency calculation for heat exchangers that makes it possible to determine the heat transfer rate, which is closely linked to thermal irreversibility. Fakheri[1] presents expressions for calculating the thermal efficiency for usual heat exchangers, which depend on two parameters, NUT and C*, similarly to what happens with the thermal effectiveness in the ε-NTU procedure.

One of the significant obstacles to developing theoretical models in the area is that the NTU parameter is strongly dependent on the Nusselt number and the latter on the Reynolds and Prandtl numbers. The Nusselt number, or the correlated quantities, suffers variations with the heat exchanger type, physical and geometric constitution, and special devices that improve performance. The expression for the Nusselt number for a particular configuration is usually obtained from experimental data, which generates empirical correlations. Fortunately, there is a vast literature on the various types of heat exchangers, and the difficulty can be overcome through a thorough search.

The usual expressions for determining are[2]:

$$Fa = \frac{NTU(1 - C^*)}{2} \ counter\ flow \tag{11}$$

$$Fa = \frac{NTU(1 + C^*)}{2} \ parallel\ flow \tag{12}$$

$$Fa = \frac{NTU}{2} \ single\ stream \tag{13}$$

$$Fa = \frac{NTU\sqrt{1 + C^{*2}}}{2} \ single\ shell \tag{14}$$

The relevant fact in our analysis is that the concepts of thermal effectiveness and thermal efficiency can be related through Equations (13) and (14) with Equations (8) and (9), which integrate two distinct procedures in favour of a better understanding of the phenomena related to heat exchangers. So, we have:

$$\eta_T = \frac{\varepsilon \dot{Q}^{\varepsilon}_{max}}{\dot{Q}^{\eta}_{max}} \tag{15}$$

soon,

$$\eta_T = \frac{\varepsilon_T C_{min}(Th_i - Tc_i)}{U_o A(\bar{T}_h - \bar{T}_c)} \tag{16}$$

It is possible to obtain an expression for the effectiveness as a function of thermal efficiency. There is an expression for the arithmetic mean temperature difference as a function of the difference in outlet temperatures.

The arithmetic mean temperature difference between the fluids can be obtained by the following expression.

$$(\bar{T}_h - \bar{T}_c) = \frac{Th_i + Th_o}{2} + \frac{Tc_i + Tc_o}{2} \tag{17}$$

From Equation (17) and Equations (19) and (20), an expression is determined for the arithmetic mean temperature difference as a function of the inlet temperature difference between the fluids and the thermal effectiveness, that is:

$$(\bar{T}_h - \bar{T}_c) = \frac{(Th_i - Tc_i)[2 - \varepsilon_T(1 + C^*)]}{2} \tag{18}$$

It can be shown, therefore, that thermal effectiveness can be obtained from thermal efficiency:

$$\varepsilon_T = \frac{1}{\dfrac{1}{\eta_T NTU} + \dfrac{(1 + C^*)}{2}} \tag{19}$$

soon,

$$\dot{Q} = \frac{C_{min}(Th_i - Tc_i)}{\dfrac{1}{\eta_T NTU} + \dfrac{(1 + C^*)}{2}} \tag{20}$$

One of the advantages of using the concept of thermal efficiency for heat exchangers is that the use of complex diagrams and equations can be avoided. It is an approach to the analysis of heat exchangers that allows a more comprehensive, systemic view when thermal and viscous irreversibilities are included.

Assuming that there is no heat exchange with the medium and that the specific heat of both fluids is constant, the entropy generation rate can be obtained as a function of thermal irreversibilities by the following equation:

$$\dot{S}_{gen} = C_{\min}\sigma_T + C_{\min}\sigma_f = \dot{S}_{genT} + \dot{S}_{genf} \tag{21}$$

where,

$$\sigma_T = \frac{C_h}{C_{min}} \ln\left(\frac{Th_o}{Th_i}\right) + \frac{C_c}{C_{min}} \ln\left(\frac{Tc_o}{Tc_i}\right) \tag{22}$$

$$\sigma_f = -\frac{C_h}{C_{min}} \mathrm{Rln}\left(\frac{P_{ho}}{P_{hi}}\right) - \frac{C_c}{C_{min}} R \ln\left(\frac{P_{co}}{P_{ci}}\right) \tag{23}$$

σ_T and σ_f are the thermal and viscous irreversibilities, respectively. P_{ho} and P_{hi} are the outlet and inlet pressures of the hot fluid.

$$R = \frac{(Th_i - Th_o)}{(Tc_0 - Tc_i)} \tag{24}$$

Finally, we have:

$$Be = \frac{\dot{S}_{genT}}{\dot{S}_{genT} + \dot{S}_{genf}} \tag{25}$$

Be is the thermodynamic Bejan number, which compares the thermal entropy generation with the total entropy generation of the system.

Before ending the chapter, let's learn a little more with Bejan[2] applying the concepts of entropy generation using usual dimensionless parameters that represent the momentum diffusivity and the thermal diffusivity, the Reynolds Re, and Nusselt number Nu . The development below applies to turbulent flow in ducts with a circular cross-section, and in this case, we have:

$$D_h = D, \quad A = \pi\frac{D^2}{4}, \quad P = \pi D \tag{26}$$

$$Re = \frac{4\dot{m}}{\pi D\mu} \quad and \quad Nu = StRePr \tag{27}$$

Pr is the Prandtl number.

For the configuration under analysis, Equation (7) becomes:

$$\dot{S}_{gen} = \frac{\dot{Q}^2}{\pi T^2 DkNu} + \frac{32\dot{m}^3 f}{\pi^2\rho^2 TD^5} \tag{28}$$

For a fully developed turbulent regime, we have the following usual expressions for Nusselt number and friction factor:

$$Nu = 0.023\, Re^{0.8}\, Pr^{0.4} \quad and \quad f = 0.046\, Re^{-0.2} \tag{29}$$

When Equation (29) is introduced in Equation (28), we have an expression for entropy generation that depends only on the Reynolds number for a fixed Prandtl number. We can then obtain the Reynolds number that minimizes the entropy generation, that is:

$$Re_{opt} = 2.023\, Pr^{-0.071}\, {B_o}^{0.356} \tag{30}$$

where,

$$B_o = \dot{m}\dot{Q} - \frac{\rho}{\mu^{5/2}(kT)^{1/2}} \tag{31}$$

The minimum value for entropy generation rate can be obtained in a completely turbulent flow regime in a duct with a circular cross-section.

$$\dot{s}_{gen,min} = \dot{s}_{gen}(Re_{opt}) \tag{32}$$

It can be shown that the effect of the Reynolds number on entropy generation is obtained through the following relationship:

$$\frac{\dot{s}_{gen}}{\dot{s}_{gen,min}} = 0.056\left(\frac{Re}{Re_{opt}}\right)^{-0.8} + 0.144\left(\frac{Re}{Re_{opt}}\right)^{4.8} \tag{33}$$

3. Basic example of an application for thermal analysis of counterflow and parallel flow heat exchangers

3.1. Formulation

$$C_{max}\ defined\ hot\ fluid \tag{34}$$

C_{max} is the thermal capacity of the hot fluid.

$$Th_i = 2.0 \quad defined \tag{35}$$

Th_i represents the inlet temperature of the hot fluid numerically.

$$Tc_i = 1.0 \quad defined \tag{36}$$

Tc_i represents the inlet temperature of cold fluid.

$$C_{min} = C^* C_{max} \tag{37}$$

C_{min} is the thermal capacity of cold fluid, and C^* is the ratio of thermal capabilities.

$$Fa = \frac{NTU(1 - C^*)}{2}\ counter\ flow \tag{38}$$

$$Fa = \frac{NTU(1 + C^*)}{2}\ parallel\ flow \tag{39}$$

Fa is the fin analogy number.

$$\eta_T = \frac{tanh(Fa)}{Fa} \tag{40}$$

η_T is the thermal efficiency.

$$\varepsilon_T = \frac{1}{\frac{1}{\eta_T NTU} + \frac{(1 + C^*)}{2}} \tag{41}$$

ε_T is the thermal effectiveness.

$$\dot{Q} = \varepsilon C_{min}(Th_i - Tc_i) \tag{42}$$

$\dot{Q}$ is the heat transfer rate.

$$Th_O = Th_i - \frac{\dot{Q}}{C_{max}} \tag{43}$$

Th_O is the outlet temperature of the hot fluid.

$$Tc_O = Tc_i + \frac{\dot{Q}}{C_{min}} \tag{44}$$

Tc_O is the outlet temperature of the cold fluid.

$$\sigma_T = \frac{C_h}{C_{min}} \ln\left(\frac{Th_o}{Th_i}\right) + \frac{C_c}{C_{min}} \ln\left(\frac{Tc_o}{Tc_i}\right) \tag{45}$$

σ_T is thermal irreversibility.

3.2. Results and discussions

The presented results deal with a dimensionless simulation with reference data defined a priori. The main parameters for data analysis are the Number of Thermal Units (NTU) and the ratio between the fluids' minimum and maximum thermal capacities (C^*). The thermal efficiency of the heat exchanger, the central magnitude of the recommended analytical method, depends only on these two parameters. All other quantities are defined and determined based on thermal efficiency.

Figure 1 shows the efficiency of the heat exchanger for configuration against the flow as a function of the number of thermal units and having as parameters the relationship between the thermal capacities. Thermal efficiency tends to its maximum value when the number of thermal units approaches zero, regardless of the thermal capacity ratio. That is, thermal efficiency is high for low NTU values. When the heat capacity of the heat exchanger tends to its maximum value, that is, equal to 1, the thermal efficiency tends to its minimum value, regardless of the number of thermal units. Thermal efficiency decreases with the decrease in the ratio between the thermal capacities and the increase in the number of thermal units. As already mentioned, thermal efficiency measures the potential for heat exchange between fluids because the greater the thermal efficiency, the greater the temperature difference between them. However, it does not provide information about the order of magnitude, for example, the rate of heat transfer.

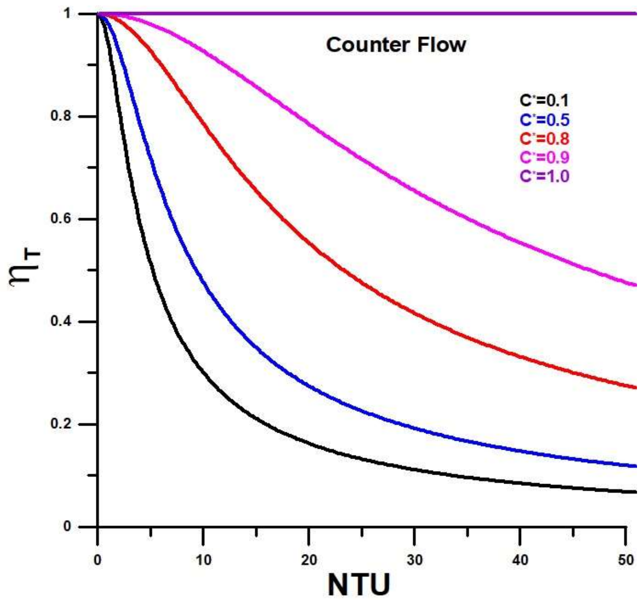

Figure 1. Thermal efficiency versus the number of thermal units for counter-flow configuration.

Figure 2 shows the thermal efficiency of the heat exchanger for a parallel flow configuration. There is a great difference between the two configurations under analysis in this case. For configuration parallel flow, the efficiency is only high for a low number of thermal units, regardless of the relationship between the thermal capacities. The result shows that the potential for heat exchange between fluids is only high for low NTU.

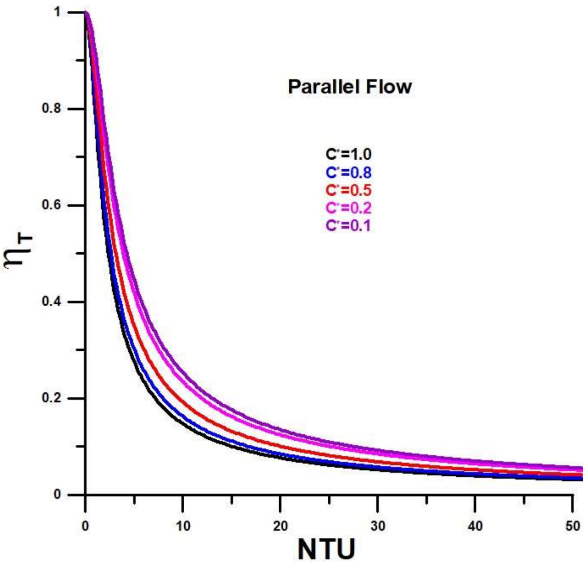

Figure 2. Thermal efficiency versus the number of thermal units for parallel flow configuration.

Figure 3 shows thermal effectiveness for both the settings under analysis. There are great differences between both. Effectiveness grows with the increase in thermal units in all situations under study. The value of thermal effectiveness is generally high for counterflow configurations compared to parallel flow configurations. When the relationship between thermal capabilities is low, the thermal effectiveness achieves values close to the maximum theoretically possible for low thermal units for both settings. With the increase in the relationship between thermal capabilities, the maximum value reached for effectiveness decreases but remains relatively high for counterflow configuration. The relevant fact is that in a parallel flow configuration, the potential for heat exchange remains stagnant for a higher number of thermal units, and increasing the number of thermal units is unproductive. Concerning the configuration in parallel flow, the results demonstrate that it is advantageous to invest in diminishing the number of thermal units, if necessary.

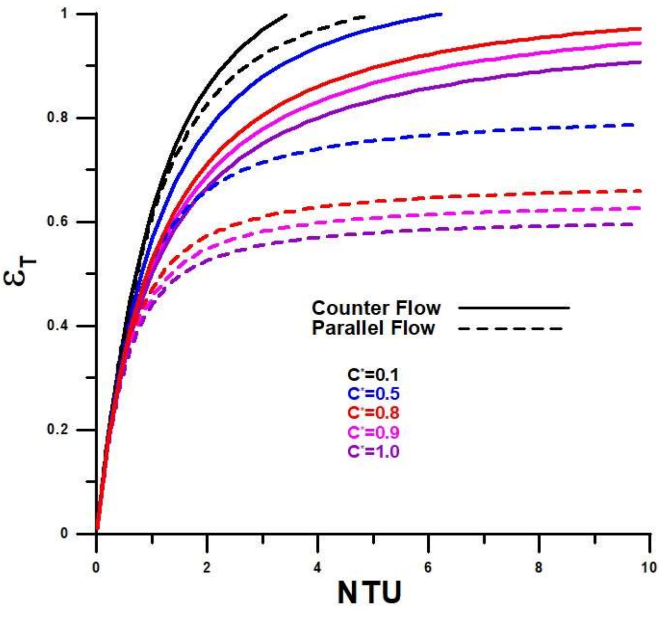

Figure 3. Thermal effectiveness versus the number of thermal units.

Figure 4 shows thermal irreversibility and a qualitative trend similar to thermal effectiveness. In a parallel flow configuration, irreversibility remains stagnant for a high number of thermal units, demonstrating that the heat exchange potential has been exhausted. For configuration in counterflow, however, irreversibility grows for low values of thermal units. It decreases slightly after undergoing a maximum value, meaning that there is heat exchange potential for high thermal unit numbers.

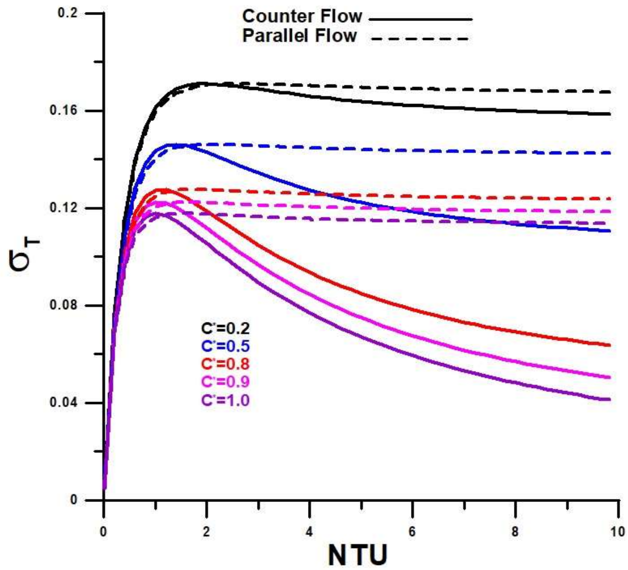

Figure 4. Thermal irreversibility versus the number of thermal units.

The situation already analyzed is reflected in the relationship between the fluid outlet temperatures (**Figure 5**). The temperature difference is maximum for the number of thermal units tending to zero. Still, it decreases rapidly for low values, demonstrating the great potential for heat exchange in this region. This decrease is more significant for counterflow configuration. The smallest differences between temperatures occur for high values for the relationship between thermal capacities.

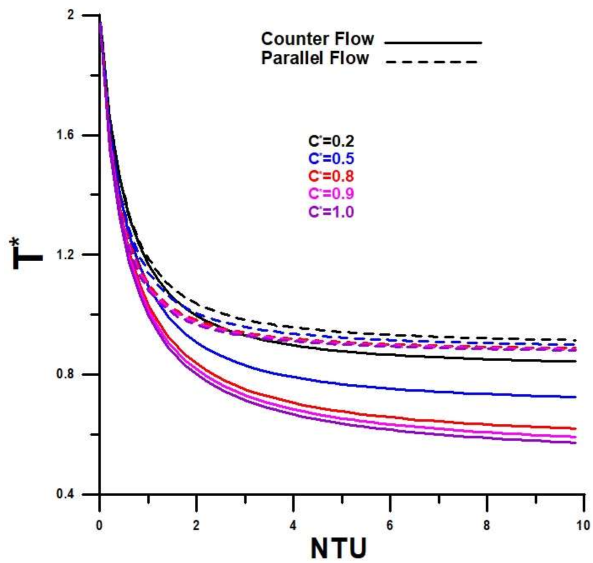

Figure 5. Relationship between outlet and inlet temperatures versus the number of thermal units.

Furthermore, the temperature difference between the fluids in the counter-flow configuration is significant and almost irrelevant for the configuration in a parallel flow configuration. The relationship between the thermal capacities has the greatest influence on the counterflow configuration. Again, it is observed that there is no justification for increasing the number of thermal units for the parallel flow configuration since the temperature differences between the fluids do not change.

To consolidate the results and complete the analysis, present results for the absolute hot fluid's output temperature in the temperature unit (**Figure 6**).

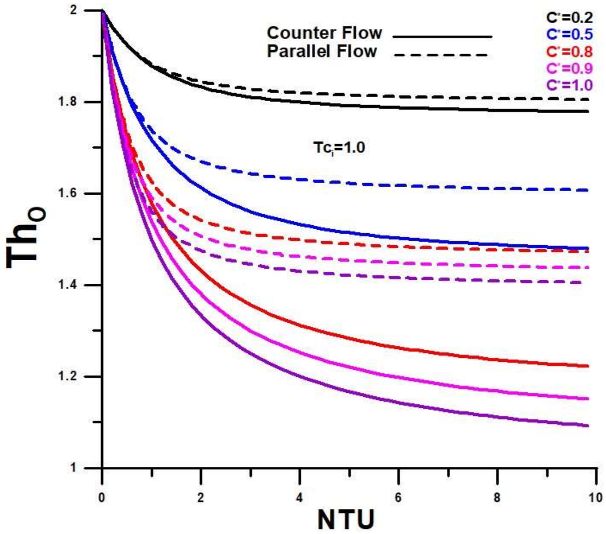

Figure 6. Outlet temperature for hot fluid.

The relevant drop in the hot fluid temperature occurs for smaller values of the number of thermal units for all the relationships between the thermal capacities. For high values of the heat capacity ratio, the hot fluid outlet temperatures tend towards the cold fluid inlet when the number of heat units is high. There is a significant difference in the hot fluid outlet temperature concerning the configurations under analysis. The important fact, in final terms, is that, regardless of the configuration, the best result obtained for the outlet temperature occurs for high values of the relationship between the thermal capacities, that is, when both values of thermal capacity approach.

4. Conclusion

A standard procedure for solving problems associated with heat exchangers was presented using classical and dimensionless analyses. When available, these procedures depend on physical and geometric characteristics and input variables, such as fluid flows, inlet temperatures, and outlet temperatures, but this last is not essential.

The example presented makes evident the importance of dimensionless analysis. Furthermore, the use of the concept of thermal efficiency of heat exchangers is a valuable instrument for a better understanding of the quantities involved in the study of heat exchangers.

It is established that there is a solid, simple, and functional dimensionless procedure for the rating, sizing, and optimization of heat exchangers.

For those initiated in the procedure described, it is suggested to revise the steps presented in this work. In addition, it can be a valuable auxiliary tool for troubleshooting.

Conflict of interest

The author declares no conflict of interest.

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