Thermal Behavior of Flows in a Tube and Grille Exchanger by solar Energy

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Abstract : The objective of this work is to study the shell and tube heat exchanger for the cooling of oil by sea water. We examine the energy balance for two flow modes (parallel current and counter current) using two calculation methods (LMTD (Logarithmic Mean Temperature Difference) and NTU (Number of Transfer Unit). The first method “LMTD” is used to size the shell and tube heat exchanger. The second method “NTU method” is used to calculate the efficiency of heat exchanger. We present, also, the sizing of the shell and tube heat exchanger including thermal resistances of fluids, metal, conductivities due to the boundary layers and fouling.

Key words: Tube shell heat exchanger, cooling, LMTD, NTU, heat transfer, sea water, oil.

I. INTRODUCTION

The best way to transmit heat between two or more fluids without risk of degradation of their properties by mixing is the use of exchange surfaces or heat exchangers. Heat exchangers have been the subject for several years of research work whose main purpose is related to the improvement of their performance this work is devoted to the heat calculation of the tube shell heat exchanger, which allowed us to obtain results that have been graphically represented with analyzes on both parallel and counter flow types and both LMTD and NTU analytical methods.

In this part we will focus on the numerical study of the thermal computation of two flows of oil and sea water in stationary forced convection in the two parallel and counter flow configurations for the tubular heat exchanger.

This work is devoted to the thermal calculation of the shell and tube heat exchanger, which allowed us to obtain results with analyzes on the two types parallel current flow and against current with the two analytical methods LMTD (Logarithmic Mean Temperature Difference) and NTU (Number of Transfer Units).

II. THEORETICAL PARTY

II.1. LMTD Method "Logarithmic Mean Temperature Difference"

a. Parralel flow

Fluid Inputs and Outputs Parameters (Oil and Sea water)

II.1.1. The oil

- Inlet temperature: T_{e1} = 65 °C
- Output temperature: T_{s1} = 56 °C
- Mass flow: m_1 = 20 Kg/s
- Pressure: P_1 = 140Pa
- Convective fluid exchange coefficient: h_1 = 0.61W/m²°C
- Exchange area: S_1 = 0.5m²
- Specific heat capacity: Cp_1 = 2.3KJ/(kg°C)
- Thermal capacity flow rate of a fluid: C_1 = 46 KW/°C
- Thermal capacity flow ratio: R_1 = 0.0108

II.1.2. Sea water

- Inlet temperature: T_{e2} = 20 °C
- Output temperature: T_{s2} = 32 °C
- Mass flow: m_2 = 1020 Kg/s
- Pressure: P_2 = 40 Pa
- Convective fluid exchange coefficient: h_2 = 0.13W/m²°C
- Exchange area: S_2 = 1.5m² Specific thermal capacity: Cp_2 = 4.184 KJ/(kg°C)
- Thermal capacity flow rate of a fluid: C_2 = 4267.68 KW/°C
- Thermal capacity flow ratio: R_2 = 92.77

II.2. Type de configuration

II.2.1. Parallel flow

For an exchange surface AL is given by the heat flow Φ.

Heat output received by sea water :

$\Phi = m_2 Cp_2 (T_{s2} - T_{e2})$

Heat output lost by the oil

$\Phi = m_1 Cp_1 (T_{e1}-T_{s1})$

Heat power exchanged between the two fluids:

$$\phi = \int_0^{AL} d\phi = \int_0^{AL} K.(T_c - T_F).dAL \qquad (1)$$

$\Phi = m_2 Cp_2 (T_{s2} - T_{e2}) = m_1 Cp_1 (T_{e1} - T_{s1})$

Φ= K. S. LMTD

$$LMTD = \frac{(T_{1C} - T_{1F}) - (T_{2C} - T_{2F})}{\ln\left(\dfrac{T_{1C} - T_{1F}}{T_{2C} - T_{2F}}\right)}$$

II.2.2. Counter flow

In the case of heating against the current we arrive at the same expression:

$$\Delta T_{LMTD} = \frac{(T_{e1}-T_{s2})-(T_{s1}-T_{e2})}{\ln(\frac{T_{e1}-T_{s2}}{T_{s1}-T_{e2}})} \quad (2)$$

$$LMTD = \frac{(\Delta T_{max})-(\Delta T_{min})}{\ln\left(\frac{\Delta T_{max}}{\Delta T_{min}}\right)} \quad (3)$$

II.3. NTU Method "Number of Transfer Units"
The LMTD method requires that the fluid temperatures at the ends of the exchanger are known. In practice the inlet temperatures of the fluids are defined and the average exchange coefficient "K" estimated; it is therefore impossible to know the ΔTb. To size a separate fluid exchanger in this case, we will use the NTU method which only integrates the fluid inlet temperatures.

II.3.1. Thermal capacity of a fluid
The thermal capacity C of a fluid is the exchangeable power for a degree of deviation and evaluated for each fluid.
In an exchanger, the determination of the thermal capacity of each fluid will make it possible to write:

1. The oil: "C_{min}".
2. Seawater: "C_{max}".
   With: C = m Cp

II.3.2. Efficiency of an exchanger
It is the ratio between the power actually exchanged "Φ_{real}" and the power that it is theoretically possible to exchange "Φ_{max}" if the exchanger was perfect.

$\varepsilon = (\Phi_{max}/\Phi_{real})$ (4)
$\Phi_{max} = (m\ Cp)_{min}\ (T_{e1} - T_{e2}) = C_{min}\ (T_{e1} - T_{e2})$ (5)

The efficiency ε of an exchanger can be expressed in the form:

$$\varepsilon = \frac{m_1.Cp_1\ (Te1 - Ts1)}{C_{min}\ (Te1 - Te2)} \text{ ou } \varepsilon = \frac{m_2.Cp_2\ (Ts2 - Te2)}{C_{min}\ (Te1 - Te2)} \quad (6)$$

- By posing : : $C_1 = m_1\ Cp_1$ (7)

If $C_1 = C_{min}$ Then the efficiency becomes:

$\varepsilon = (T_{e1}-T_{s1})/(T_{e1}-T_{e2})$

- By posing: $C_2 = m_2\ cp_2$

If $C_2 = C_{min}$ Then the efficiency becomes:

$\varepsilon = (T_{s2} - T_{e2}) / (T_{e1}-T_{e2})$

Parallel flow
The efficiency of the oil side heat exchanger:

$$\varepsilon_1 = \frac{(1-exp\,[-NTU_1(1+R_1)])}{1+R_1} \quad (10)$$

The efficiency of the seawater side heat exchanger:

$$\varepsilon_2 = \frac{(1-exp\,[-NTU_2(1+R_2)])}{1+R_2} \quad (11)$$

The efficiency of the shell and tube heat exchanger:

$$\varepsilon = \frac{(1-exp\,[-NTU(1+R)])}{1+R} \quad (12)$$

The efficiency of the oil side heat exchanger:

$$NTU_1 = \frac{KA_l}{\dot{C}_1} \quad (13)$$

The efficiency of the seawater side heat exchanger:

$$NTU_2 = \frac{KA_l}{\dot{C}_2} \quad (14)$$

The efficiency of the shell and tube heat exchanger:

$$NTU = \frac{KA_l}{\dot{C}_{min}} \quad (15)$$

Or: $NTU = \frac{1}{1+R} \ln\left[\frac{1}{1-\varepsilon(1+R)}\right]$ (16)

The oil-side thermal ratio: $R_1 = (C_1/C_2)$

The thermal report on the seawater side: $R_2 = (C_2/C_1)$
$R = \min(R_1, R_2)$
Fluid Inputs and Outputs Parameters (Oil and Seawater).

III. RESULTS AND DISCUSSION

The numerical study of the thermal computation of two oil and seawater flows in forced convection stationary in both parallel and counter flow configurations for the tubular heat exchanger.

The formula is written:

$$\Delta T_{LMTD} = \frac{\Delta T_a - \Delta T_b}{\ln(\frac{\Delta T_a}{\Delta T_b})} = \text{LMTD} \quad (17)$$

Avec : $\Delta T_a = T_{e1}-T_{e2}$ et $\Delta T_b = T_{s1}-T_{s2}$ (18)

$$LMTD = \frac{(65-20)-(56-32)}{\ln\left(\frac{65-20}{56-32}\right)} = 33{,}87°C$$

$\Delta T_a = T_{e1}-T_{e2} = 65\text{-}20 = 45°C = \Delta T_0 = \Delta T_{max}$
$\Delta T_b = T_{s1}-T_{s2} = 56\text{-}32 = 24°C = \Delta T_L = \Delta T_{min}$
Φ = K S LMTD=150 2 33.87=10,161Kw

The necessary condition for co-current flow is to have a heat outlet temperature greater than or equal to that of cold $T_{s1} \geq T_{s2}$.
For our case: the exit temperature of the oil is greater than or equal to the outlet temperature of the sea water.

In an ideal co-current exchanger, one could obtain at best "T_{s2}" approaching "T_{s1}" without ever reaching it.

Note that the heat flow is important at the inlet of the exchanger (see Figure 1) and, at the outlet of the exchanger it becomes low, because of the cooling of the oil by the sea water up to their outlet temperatures are close to the limit temperature of 47 °C.

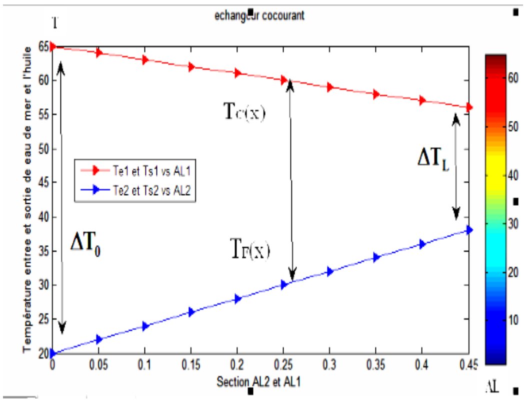

Figure (1) : Temperature profile in the parallel flow tubular heat exchanger

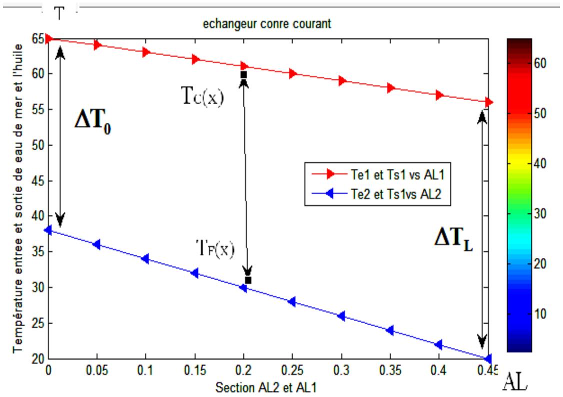

Figure (2): Temperature profile in the counter flow tubular heat exchanger

With: $\Delta T_{max} = \Delta T_a = T_{e1} - T_{s2}$
and $\Delta T_{min} = \Delta T_b = T_{s1} - T_{e2}$
$\Delta T_a = T_{e1} - T_{s2} = \Delta T_{max} =$ 65-32 = 33°C
$\Delta T_b = T_{s1} - T_{e2} = \Delta T_{min} =$ 56-20 = 36°C

$$\Delta T_{LMTD} = \frac{(65-32)-(56-20)}{\ln\left(\frac{65-32}{56-20}\right)} = 37.5°C$$

Knowing that:
$m_2\ Cp_2\ (T_{s2} - T_{e2}) = m_1\ Cp_1\ (T_{e1} - T_{s1})$
We then distinguish two cases :

1. For the fluid cooling:
   If $m_2\ Cp_2 < m_1\ Cp_1$ then:
   Then $(T_{s2} - T_{e2}) > (T_{e1} - T_{s1})$
   Let: $T_{s1} - T_{e2} > T_{e1} - T_{s2}$

2. For the fluid heating:
   If $m_2\ Cp_2 > m_1\ Cp_1$ then:
   Then: $(T_{s2} - T_{e2}) < (T_{e1} - T_{s1})$
   Let: $T_{s1} - T_{e2} < T_{e1} - T_{s2}$

In our case, to cool the oil by sea water, one must have the condition of:

- If $m_2\ Cp_2 < m_1\ Cp_1$. This is the cold fluid that "controls the transfer" since it has the lowest thermal unit rate.

This means that the temperature difference between the fluids is minimal on the inlet side of the hot fluid, and that this gap is even smaller than the exchange surface AL is greater (see Figure 2). For an infinitely long heat exchanger, the cold fluid outlet temperature is equal to the inlet temperature of the hot fluid. [6, 7].
S = 1.7m^2 we will have $T_{e1} = T_{s2}$ = 65 °C, we deduce that by increasing the exchange section, the exit temperature of the seawater increases rapidly until it is equal to the temperature oil inlet; the heat exchange is canceled :
This is the limit value of the exchanger if $T_{e1} > T_{s2}$. It is impossible to get physically but or may have $T_{s2} > T_{s1}$ because it is the principle of heat transfer in the counter flow. The heat transfer in the counter flow.

In the tubular exchanger, the transferred heat flux is always high against the current because the Logarithmic Mean Temperature Difference is high.
In an ideal parallel flow exchanger, one could obtain at best "T_{s2}" approaching "T_{s1}" without ever reaching it.
On the other hand, for the configuration of pure counter current, "T_{s2}" is now more than "T_{s1}" (which shows that this exchange is more efficient than the previous one. In our case:
$R = R_1 = (C_1/C_2) = (46303)/(4267.68\ 10^3)$
R= 0.010.
The grouping ($K\ S/C_{min}$) is dimensionless and is noted NTU or Number of Unit Transfer: it is representative of the exchange power of the device.

$$\varepsilon = \frac{(1-exp\,[-NTU\,(1+R)])}{1+R}$$

It emerges that the efficiency of an exchanger depends on the thermal ratio $R = (C_{min}/C_{max})$ on the one hand, and NUT on the other hand $NTU=(K\ S/C_{min})$. This latter ratio can be understood as the ratio between the surfaces actually offered AL and surface reference (K/C_{min}). [15]
The efficiency is great when NUT is large and tends towards the maximum, the thermal ratio being small.
In both cases limit:

- R=0, in this case $R=(C_{min}/C_{max})=0$ however $C_{min} = 0$, corresponds to a zero mass flow rate on one of the fluids in our case it is the oil, and in transfer a zero power ; the expression of efficiency [15] is then reduced to : ε=1-exp(-NTU) so the efficiency is maximum.
- R=1, in this case $R=(C_{min}/C_{max})=1$ however $C_{min}=C_{max}$, corresponds to two equal mass flows; in our case the mass flow rate of the oil is equal to the mass flow of the sea water, the expression of the efficiency is reduced to : ε=(½)-exp(-NTU) so efficiency and at a minimum.

The following figures (3), (4) and (5) show that the lower the thermal ratio, the greater the efficiency is important because we are in where the thermal ratio is low, the mass flow rate of the oil is weak in front of the sea water in the tube; in our case we will have a good thermal efficiency for the cooling of the oil.

If the thermal ratio is important, the mass flow of oil and sea water are almost equal, which gives us a low thermal efficiency for cooling the oil.

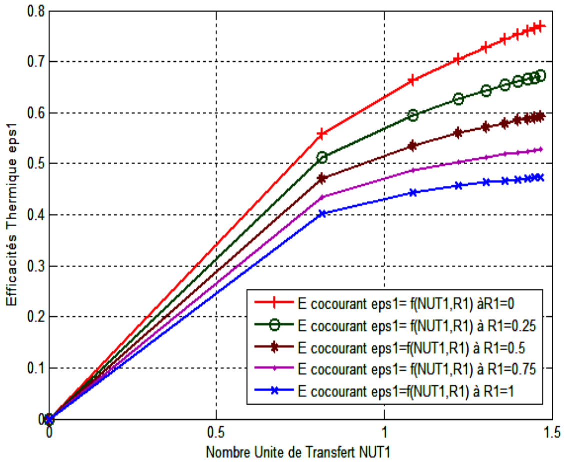

Figure (3): Change in oil efficiency as a function of NTU_1 and R_1 parallel flow.

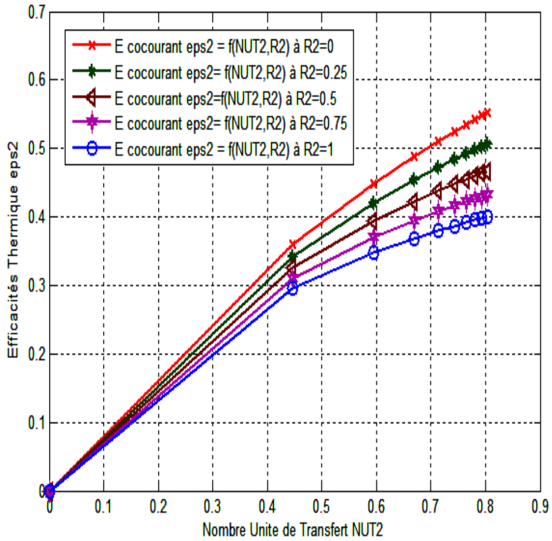

Figure (4): Change in seawater efficiency as a function of NTU_2 and R_2 in parallel flow

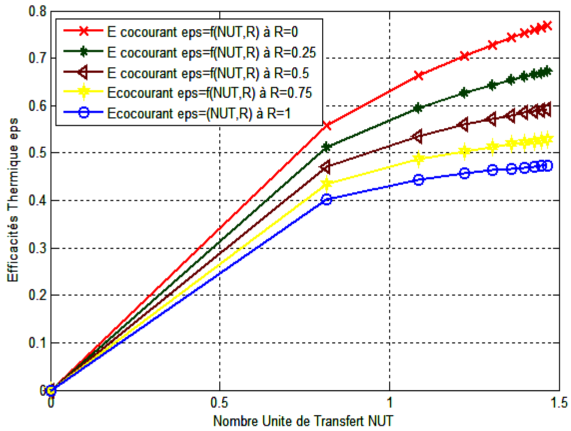

Figure (5): Variation of the efficiency of the exchanger according to NTU and R in parallel flow

III.1. Comparison between parallel and counter flow

Note on the graphs (6), (7) and (8) that the flow of the countercurrent fluid is more efficient than in parallel flow because the maximum heat transferable flow by an exchanger is reached by a heat exchanger against current infinitely long. In such a configuration, one of the fluids experiences the highest temperature change. If one is in the case where $C_c < C_f$ this temperature difference is reached by the hot fluid which sees its temperature evolve from T_{ce} towards T_{fe}, the maximum flux being given by:

$\Phi_{max} = m_c\, Cp_c\, (T_{ce}\text{-}T_{fe}) = Cc\, (T_{ce}\text{-}T_{fe})$

If, on the other hand, we find ourselves in the case where: $C_f < C_c$, the cold fluid undergoes the greatest temperature difference from T_{fe} ver T_{ce}, leading to the maximum heat flow.

$\Phi_{max} = m_f\, Cp_f\, (T_{ce}\text{-}T_{fe}) = C_f\, (T_{ce}\text{-}T_{fe})$.

Thus the flow [16]:

$\Phi_{max} = m_{min}\, Cp_{min}\, (T_{ce}\text{-}T_{fe}) = C_{min}\, (T_{ce}\text{-}T_{fe})$.

The notion of efficiency of flow of this last value characterizes the relation between the flow actually transmitted and the maximum transferable flow and is written:

$\varepsilon = \Phi/\Phi max = m_c\, Cp_c(T_{ce}\text{-}T_{cs})/(m_{min}\, Cp_{min}(T_{ce}\text{-}T_{fe}))$

$m_f\, Cp_f\, (T_{fs}\text{-}T_{fe})\, /\, m_{min}\, Cp_{min}\, (T_{ce}\text{-}T_{fe})$

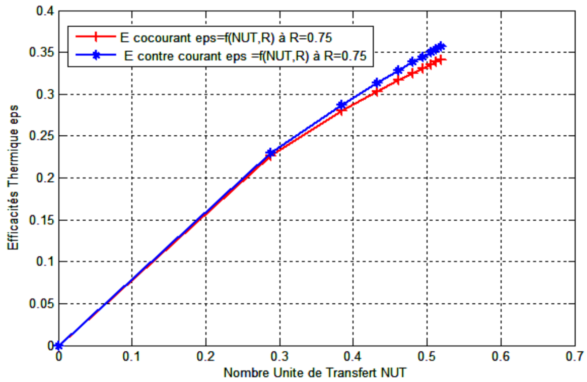

Figure (6): Comparison between two parallel and counter flow of the efficiency of the exchanger according to NTU for R = 0.75

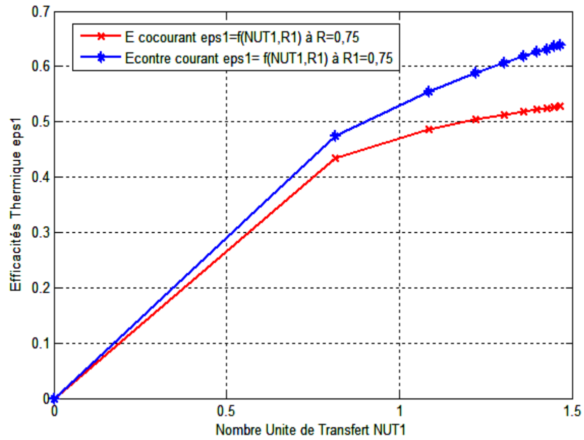

Figure (7): Comparison between parallel and counter flow of oil efficiency as a function of NTU_1 for $R_1 = 0.75$

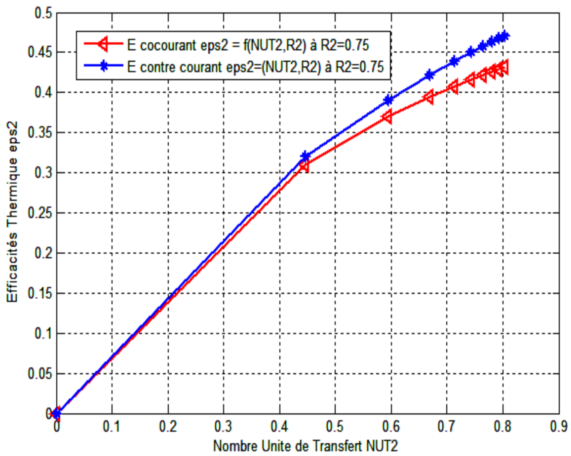

Figure (8): Comparison between parallel and counter flow of sea water efficiency versus NTU_2 for $R_2 = 0.75$

IV. CONCLUSION

In this work we made a heat balance of the shell and tube heat exchanger to cool oil by sea water with two fluids flow arrangements: parallel and counter, and with two calculation methods LMTD and NTU.

These two methods of calculation are equivalent because they make it possible to reach the same result. However the second knows a greater use in practice because it is closer to the constraints of the designer. Indeed, the lack of knowledge of the exit temperatures quickly eliminates the LMTD method in favor of the efficiency NTU which only involves the input temperatures.

And it has also been concluded that the countercurrent flow method is more efficient than in parallel flow.

When NTU <1: The thermal transfer is short, that is to say economical in investment, but incomplete from the thermal point of view

When NTU>1: The thermal transfer is said to be long, so complete from the thermal point of view.

When NTU is large one has a good thermal efficiency and a low thermal ratio.

When NTU is small it has a low thermal efficiency and a good thermal ratio.

In an ideal parallel current exchanger one could get at best "T_{s2}" approaching "T_{s1}" without ever reaching it. On the other hand, for the configuration of pure countercurrent, "T_{s2}" is now more than "T_{s1}" (which shows that this exchange is more efficient than the previous one.

For our shell and tube heat exchanger, to cool oil with seawater, the counter flow will be used to obtain the best thermal energy result and the NTU calculation method because it is easier and close to the constraints of the designer.

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