Naval U-tube Heat Exchanger, design and CFX analysis

C L Dumitrache and D Deleanu
Maritime University of Constanta, Faculty of Electromechanics, Department of General Engineering Sciences, Mircea cel Batran street, No. 104, Constanta, 900663, Romania
E-mail: ldumitr@yahoo.com

Abstract. Heat exchangers are common devices on board’ ships, but also in the oil-chemical industry. In this paper is presented an U-tube heat-exchanger that was designed with NX Siemens. The component parts of this heat exchanger, the CAD models and their assembly mode in the final part are presented. It is known that the design of a heat exchanger is a real challenge, and the dimensions of the component parts are established according to some standards, engineering design guidelines and to the industrial domain where they are used. The dimensions of our device were established by the author by consulting specialized works. The details of how the component parts were designed, the CAD instructions used and the constraints applied to the component parts in the final assembly are presented in details. Subsequently, the final assembly is taken over in ANSYS where a CFX analysis of fluid flow is performed and a study of the heat transfer between the two fluids.

1. Introduction
The heat exchanger is the device that performs the heat transfer between different environments with different temperatures [1]. They can be classified after operation and construction as follows:

- surface exchange heat exchangers, at which the heat transfer from the primary to the secondary thermal agent is made by means of a partition wall with high thermal conductivity; taking into account that the process of recovering heat transferred by the primary heating agent is continuous, these heat exchangers are also called recuperators.
- regenerative heat exchangers, in which thermal agents pass successively through the device; in the first period, when the primary heating agent passes through the exchanger, a heating is performed of its filling (metallic or ceramic filling); in the second period, when the secondary heating agent flows through the heat exchanger, the quantity is taken over heat accumulated by the filling material in the first period of the cycle.
- heat exchangers with mixing, at which the heat transfer is carried out through mixing the two agents.

Another classification is according to the fluid-flow path through the heat exchanger:

- parallel flow, wherein both heat and cold agents flow into same direction;
- counterflow, wherein the thermal agents have the same direction but opposite directions of flow;
- crossflow and mixed wherein one of the thermal agents repeatedly changes its direction and direction of flow towards the other agent.

In our CAD NX Siemens work, a heat exchanger with pipes is designed (figure 1) mounted on tube sheet, without baffles and partition walls, these pipes are U-shaped with the role of increase flow rate
and turbulence and ensure participation the entire tubular surface upon heat transfer. The fluid flow path through the our heat exchanger corresponds to the crossflow and mixed situation and fluids flow analysis is performed with ANSYS CFX which is a high-performance computational fluid dynamics (CFD) software tool that delivers accurate solutions, quickly and robustly across a wide range of CFD and multiphysics applications.

2. The CAD models of parts

Figure 1 shows the component parts of the U-tube heat exchanger that was designed and analyzed in the present work.

This model was taken from engineering design guidelines of an international company that produces this type of heat exchanger [2]. I mention that all the dimensions of the component parts are not presented by the production company but they are original and were chosen by the author of this work. The CAD models of the component parts were made with NX Siemens. All these models share common design operations such as:

- 1. drawing sketches with geometric and dimensional constraints;

![Figure 1. Component parts of U-tube heat exchanger [2].](image-url)

![Figure 2. Mirror operations for drawing tubes sheet.](image-url)
2. extrusion operations of the sketches previously performed, thus resulting in 3D bodies of each component part;
3. operations of rounding the edges where necessary.

Of course, each component part has some design features that will be presented below. After the 3D drawing parts, we proceeded to the assembly operations of the components, taking into account the position constraints of each component in the final assembly.

In the design of the tubes sheet it can be seen from figure 1 that the U-shaped pipes are attached to the disc having holes of the same diameters. Initially in the design I made the disc separately with the holes and the U-pipes that are in pairs. For reasons that I will present in the next paragraph and which relate to the finite element discretization of these pipes and the disc with holes, I had to redesign the piece, so that the disc with holes and U-pipes would make a common body.

For the design of the tubes sheet a sketch was made on a quarter great circle (figure 2) where some small circles of the same diameters were drawn which will constitute the holes in a 3D body. On these holes, we built portions of U-pipes that gave birth to a “quarter” of the final piece (yellow color). This “quarter of the piece” was subsequently mirrored successively in relation to some reference planes, finally resulting in the 3D body of tubes sheet (figure 2).

![Figure 3. Sections viewing of hot inlet-outlet flange and the shell side.](image)

The hot inlet-outlet flange and the shell side have been designed using simple extrusion operations to which "chamfer" and "edge blend" edges have been added (figure 3).

![Figure 4. General and section viewing of U-tube heat exchanger assembly.](image)
The final assembly (figure 4) was made by attaching the component parts using localization constraints which involved the concentric and align constraints and touching of the tubes sheet-shell and hot inlet-outlet flange-shell.

3. Basic equations in design of heat exchangers
The thermal calculation of heat exchangers is based on two equations:
- the equation of heat balance on the appliance;
- the equation of heat transfer in the appliance.

These equations correlate eight main parameters:
- the thermal load of the appliance (Q);
- the heat exchange surface area (S);
- the inlet and outlet temperatures of the hot fluid (t1’, t1’’);
- the inlet and outlet temperatures of the cold fluid (t2’, t2’’);
- the flow rates of thermal fluids (m1, m2).

Because the thermal load does not represents an independent variable by the thermal calculation two unknown quantities can be determined depending on the remaining five parameters, whose values must be known (given, chosen or calculated).

3.1. The heat balance equation
The heat balance equation on a heat exchanger has the form [1, 3]:

\[
Q_i = Q_2 + Q_p = \frac{Q_i}{\eta_r}
\]

in which:
- \(Q_i\), \(Q_2\) - the thermal flow yielded by the primary thermal agent (hot fluid), respectively received by the secondary thermal agent (cold fluid), [W];
- \(Q_p\) - heat flux lost in the environment, [W];
- \(\eta_r\) - coefficient of heat retention in the appliance, \(\eta_r = 0.980-0.995\);
- \(m_1, m_2\) - the mass flow rate of the hot and cold fluid, [kg/s];
- \(c_{p1}, c_{p2}\) - the specific heat at constant pressure of the hot and cold fluid [J/kgK]. These values \(c_{p1}\), \(c_{p2}\) is considered constant throughout the fluid flow through the device, in calculations using their average values;
- \(C_1, C_2\) - the thermal capacity of the hot and cold fluid, respectively, [W/K];
- \(\Delta t_1, \Delta t_2\) - temperature variation of the hot and cold fluid, respectively, [°C] or [K];
- \(i_1', i_1''\) - enthalpy of the hot fluid at the entrance, respectively the exit from the device, [J/kg];
- \(i_2', i_2''\) - enthalpy of the cold fluid at the entrance, respectively the exit from the device, [J/kg].

3.2. The heat transfer equation
This equation expresses the thermal load of the device (Q) in the form:
where $k_s$ - represents the global coefficient of surface heat exchange, considered constant over the entire heat exchange surface of the device [W/m²K]; $S$ - the area of the heat exchange surface of the device [m²]; $\Delta t_{med}$ - average temperature difference of thermal agents [°C].

In the case of the heat exchanger designed in this work, it is provided with tubular surfaces and the heat transfer equation can be written:

$$Q = k_l \Delta t_{med} l \quad [W]$$

where $k_l$ - is the global coefficient of linear heat exchange, considered constant in the device [W/m²K], and $l$ - represents the summed length of the heat exchanger pipes, [m].

Practical situations can be encountered when certain thermophysical properties of fluids change greatly with temperature and the global coefficient of heat exchange can no longer be considered constant over the whole surface (e.g. changing the viscosity can cause the flow regime change). In this case the heat transfer equation can be expressed:

$$Q = \sum_{i=1}^{n} \Delta Q_i = \left( \sum_{i=1}^{n} k_{s,i} \Delta S_i \right) \Delta t_{med} \quad [W]$$

in which the heat exchange surface is divided into "n" parts.

4. The average temperature difference

In general, $\Delta t_{med}$, the average temperature difference between the two heat agents varies along the heat exchange surface. It is determined differently for devices with parallel and counterflow arrangements and for devices with non-parallel flow pattern as shown in the device in this paper (multipass crossflow and mixed).

4.1. The average temperature difference for the devices with parallel flow and counterflow arrangements

The variations of the temperatures of the thermal agents along the heat exchange surface in the case of the parallel flow (wherein both heat and cold agents flow into same direction, figure 6) and counterflow (wherein the thermal agents have the same direction but opposite directions of flow, figure 7) devices, where $L$ is the length of the heat exchange surface are presented.

![Figure 6. The temperature variations in the parallel flow heat exchanger and both single-phase fluids [3].](image6.png)

![Figure 7. The temperature variations in the counterflow heat exchanger and both single-phase fluids [3].](image7.png)
When establishing the expression of the average temperature difference, the following hypotheses are made:

- heat losses in the environment are negligible which means that \( Q_p = 0 \) and \( \eta_r = 1 \) in the relation (1);
- the global heat exchange coefficient \( (k_s) \), the fluid flows \( (m_1, m_2) \) and their specific heat \( (c_{p1}, c_{p2}) \) are constant along the heat exchange surface;
- the axial conduction along the surface is negligible;
- the area of the heat exchange surface varies linearly with its length.

Next is written the thermal balance equations for two differential elements of the two fluids (the hot and the cold), each element being of length \( dx \) and area of the heat exchange surface \( dS \):

\[
\frac{dQ}{dt} = -\dot{m}_1 c_{p1}\frac{dt_1}{dt} = \pm \dot{m}_2 c_{p2}\frac{dt_2}{dt} \\
\frac{dQ}{dS} = k_s (t_1 - t_2)
\]

It should be mentioned that in relation (6) the minus sign indicates a reduction of the temperature along the surface, the plus sign indicating an increase of it. The temperature of the primary fluid \( t_1 \) will always decrease along the surface, while the temperature of the secondary fluid \( t_2 \) increases for parallel flow and decreases in the case of counterflow.

It follows from equation (6) that:

\[
dt_1 = -\frac{dQ}{\dot{m}_1 c_{p1}}; \quad dt_2 = \pm \frac{dQ}{\dot{m}_2 c_{p2}}; \quad d(t_1 - t_2) = -\frac{dQ}{\dot{m}_1 c_{p1} \pm \frac{1}{\dot{m}_2 c_{p2}}}
\]

Substituting \( dQ \) from relation (7) into relation (8) and separating the variables is obtained:

\[
\frac{d(t_1 - t_2)}{t_1 - t_2} = -k_s\left(\frac{1}{\dot{m}_1 c_{p1}} \pm \frac{1}{\dot{m}_2 c_{p2}}\right) dS
\]

By integrating equation (9) for the parallel flow situation:

\[
\int_{t_1 - t_2}^{t_1' - t_2'} \frac{d(t_1 - t_2)}{t_1 - t_2} = \int_0^{S_0} -k_s\left(\frac{1}{\dot{m}_1 c_{p1}} \pm \frac{1}{\dot{m}_2 c_{p2}}\right) dS \Rightarrow \ln\frac{t_1' - t_2'}{t_1 - t_2} = k_s\left(\frac{1}{\dot{m}_1 c_{p1}} + \frac{1}{\dot{m}_2 c_{p2}}\right) S_0
\]

From the heat balance equation:

\[
\dot{m}_1 c_{p1} = -\frac{Q}{t_1' - t_1}; \quad \dot{m}_2 c_{p2} = \frac{Q}{t_2'' - t_2'} \quad \Rightarrow \ln\frac{t_1' - t_2'}{t_1 - t_2} = \frac{k_s}{Q}\left(t_1' - t_1'' + t_2'' - t_2\right) S_0 \Rightarrow
\]

\[
Q = k_s S_0 \left(\frac{t_1' - t_2'}{t_1 - t_2'} - \frac{t_1'' - t_2''}{t_1'' - t_2''}\right) \ln\frac{t_1' - t_1''}{t_1'' - t_2''}
\]

If we compare equation (11) with the heat transfer equation (3), it results that the average temperature difference between the two thermal agents for the parallel flow situation is:
\[ \Delta t_{med,cc} = \frac{(t'_1 - t'_2) - (t''_1 - t''_2)}{\ln \frac{t'_1 - t'_2}{t''_1 - t''_2}} \Rightarrow \Delta t_{med,cc} = \frac{\Delta t_f - \Delta t_H}{\ln \frac{\Delta t_f}{\Delta t_H}} \; ; \; \Delta t_f = t'_1 - t'_2 ; \; \Delta t_H = t''_1 - t''_2 \] (12)

By integrating equation (9) for the counterflow flow situation:

\[ \int_{t'_1 - t'_2}^{t''_1 - t''_2} \frac{d(t'_1 - t'_2)}{t''_1 - t''_2} = \frac{S_0}{k_s} \left( \frac{1}{\dot{m}_1 c_{p1}} + \frac{1}{\dot{m}_2 c_{p2}} \right) dS \Rightarrow \ln \frac{t'_1 - t'_2}{t''_1 - t''_2} = k_s \left( \frac{1}{\dot{m}_1 c_{p1}} - \frac{1}{\dot{m}_2 c_{p2}} \right) S_0 \] (13)

From the heat balance equation:

\[ \dot{m}_1 c_{p1} = -\frac{Q}{t'_1 - t'_2} ; \; \dot{m}_2 c_{p2} = -\frac{Q}{t''_1 - t''_2} \; eq.13 \ln \frac{t'_1 - t'_2}{t''_1 - t''_2} = k_s \frac{Q}{S_0} \left( t'_1 - t'_2 + t''_1 - t''_2 \right) \] (14)

If we compare equation (14) with the heat transfer equation (3), it results that the average temperature difference between the two thermal agents for the counterflow situation is:

\[ \Delta t_{med,cc} = \frac{(t'_1 - t'_2) - (t''_1 - t''_2)}{\ln \frac{t'_1 - t'_2}{t''_1 - t''_2}} \Rightarrow \Delta t_{med,cc} = \frac{\Delta t_f - \Delta t_H}{\ln \frac{\Delta t_f}{\Delta t_H}} \; ; \; \Delta t_f = t'_1 - t'_2 ; \; \Delta t_H = t''_1 - t''_2 \] (15)

4.2. The average temperature difference for the devices with multipass crossflow and mixed arrangements

The heat exchanger presented in this work is a device where the hot thermal agent will pass through the U-shaped pipes while the cold thermal agent will pass through the shell side, the fluid flow situation being crossflow and mixed situation (figure 8).

For this situation, the average temperature difference is determined by the relation:

\[ \Delta t_{med} = \Delta t_{med,CI(CM)} = F \Delta t_{med,CC} \] (16)

where F is a correction factor and \( \Delta t_{med,CC} \) is the average temperature difference for the counterflow situation. This factor F is calculated according to two ratios noted with P and R and by the type of flow:

\[ P = \frac{\Delta T_2}{\Delta T_{2,max}} = \frac{T'' - T'_2}{T'_1 - T'_2} \] (17)

where \( \Delta T_2 \) represents the heating degree of the secondary agent in the device (cold fluid), and \( \Delta T_{2,max} \) is the maximum difference available between the two hot and cold fluids. This P ratio has the role of thermal efficiency and is always a subunit value.

\[ R = \frac{C_2}{C_1} \] (18)

**Figure 8.** The temperature variations in the crossflow heat exchanger and both single-phase fluids [3].
where $C_2$ and $C_1$ represent the thermal capacities of the two thermal agents; $R$ can be smaller, greater than or equal to 1.

Both $P$ and $R$ ratios are calculated with the relations (17) and (18) regardless of the space in the device through which the two fluids flow.

The correction factor $F$ is subunit value increasing with the decrease of $R$ and $P$. For the four temperatures of the thermal agents, the average difference of the maximum temperature is obtained for the counterflow situation, and the minimum for the parallel flow and for the other types of flows are between these limits.

5. The flow CFX analysis

After the assembly was made (figure 4), it was imported into the CFX Fluid Flow module from Ansys.

It is also worth mentioning that in the construction of the assembly, the non-metallic gaskets that should have been used in the contact between the hot fluid inlet-outlet flange with tubes sheet and the contact between the shell’ flange with tubes sheet were not taken into account.

With the "Fill" option, the spaces occupied by the hot and the cold fluid were well defined (figure 9). Then, we went to "mesh" discretization operations of each component part of the assembly in finite elements. It should be mentioned that the disk with the U-pipes are part of a single common body after they were initially separated. Because there were some errors in establishing the boundary conditions, more precisely when establishing the contact surfaces between the U-pipes and the disk, a common body between the U-pipes and the disk was created.

After the "mesh" operation, the cold fluid will have 351742 nodes and 1797784 finite elements and the hot fluid will have 264366 nodes and 1226098 finite elements (figure 9).

![Figure 9. Fluids meshed domains: a-hot fluid; b-cold fluid.](image)

![Figure 10. The vertical middle section plane (temperature) through U-tube heat exchanger.](image)
At the inlet of the hot fluid with the temperature of 70°C and the pressure of 3 bar, as well as at the entrance of the cold fluid with the temperature of 20°C and the pressure of 2 bar, boundary condition of “inlet” type was used and at the exit of the two fluids they used “opening” type boundary condition with an opening temperature of 15°C and an air opening pressure of 1 bar (figures 10 and 11).

Figure 11. Another vertical section plane (temperature) through U-tube heat exchanger.

The convection surfaces of the assembly, namely the shell surface, the surface of the disc with pipes and the surface of the flange at the inlet and outlet of the hot fluid, were taken into account, using the "wall" boundary condition. For the shell and disk surfaces were considered a heat transfer coefficient of 10 [W/m²K], and for the flange at the inlet and outlet of the hot fluid was used a heat transfer coefficient of 8 [W/m²K] (figures 12 and 13). For all these calculations, outside temperature of 15°C was considered.

Figure 12. Temperature horizontal section plane through hot fluid inlet zone.
Shell-U tubes heat exchangers are the most widespread type in the industry also including the naval field due to its constructive simplicity, high reliability and relatively low cost.

This type of heat exchangers have some advantages [4]:
- provides maximum heat transfer surface area per given shell and tube size;
- capable of withstanding thermal shock;
- allows for differential thermal expansion between shell and tubes as well as between individual tubes;
- the welded carbon steel shell side construction provides maximum durability;

but have also limitations:
- individual tube replacement not always possible;
- tube side can be cleaned by chemical means only.

![Figure 13. Temperature horizontal section plane through hot fluid outlet zone.](image)

The U-pipes for heat exchangers are laminated and specially designed for their construction [5]. The most used materials are: steels for medium or low temperatures; copper; copper-nickel alloys in different compositions (eg. 70/30%, or 90/10%); copper-aluminium alloys in different compositions (eg. 93/7%, or 91/9%); different types of zinc alloys 22 and 40%; stainless steel.

The pipes can have inner ribs or star-shaped core to ensure an increase of the heat transfer and the quality of the materials used to make them must be very good because during their operation they are subject to corrosion and stresses that can destroy them or decrease the heat transfer capacity.

The shell must be of sufficiently large diameter to allow the tube sheet to be fixed. It is made of steel, the thickness of its wall is calculated in such a way that it can withstand the pressure of the fluid flowing through the pipes but also the pressure of the fluid flowing inside the shell.

The hot inlet-outlet flange has the role of circulating the hot fluid through the inside of the U-pipes. This is executed by casting using as cast iron and their shape obtained by casting must ensure the desired flow rates.

6. Conclusions
The CAD model realized and presented in this paper is a complex one and brings us a series of information related to how the heat exchange between the two fluids with different temperatures takes...
It should be mentioned that the fluid used for static flow analysis is water with \( T = 70^\circ C \) at the hot inlet zone and \( T = 20^\circ C \) at the cold inlet zone.

The energy efficiency of the heat exchanger can be easily observed from the results of flow analysis and the results are satisfactory. These results derive primarily from the way this heat exchanger was designed but also from the boundary condition imposed on the static flow analysis.

The efficiency of the heat exchanger is expressed on the energy efficiency that is established on the basis of the relation (1) and represents the measure of the quality of the thermal insulation of the heat exchanger. In the case of the device designed in this paper the temperatures \( T_{\text{hotoutlet}} = 66.6^\circ C \) and \( T_{\text{coldoutlet}} = 25.2^\circ C \) these temperatures being measured at the diameter point of the circular surfaces hot outlet and cold outlet.

For the better efficiency of this heat exchanger, the following solutions would be required:

- the use of U-pipes with smaller diameters;
- the U-pipes can have much smaller wall thicknesses to make heat exchange much faster;
- the shell diameter may be larger to enter a larger amount of cooling fluid to create larger turbulences with considerably increased cooling effect;
- the pressure of the hot inlet fluid may be lower to allow the cooling fluid to have a longer heat transfer time.

7. References

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