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Shell and tube heat exchanger design for waste water cooling

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Abstract

This paper presents the design of a hull and tube heat exchanger, including thermal calculation, until the definition of the geometry of its mechanical components. The service to which this exchanger will be designed, and to cool the disposal of the water used in the cooling process of a steel rolling mill. Then the mechanical design is presented, where the materials are specified and the dimensions are calculated for the main components of the exchanger. The TEMA (Tubular Exchanger Manufacturers Association) and ASME (American Society of Mechanical Engineers) standards were the main international standards on which the mechanical design was based.

Keywords: Heat exchanger; Hull and tube; Cooling; Design; CAD-3D

1. Introduction

The process of heat exchange between two fluids that are at different temperatures and separated by a solid wall occurs in many engineering applications.

The equipment used to implement this exchange are called heat exchangers, and specific applications can be found in space heating and conditioning, heat recovery, chemical processes, etc. As the most common applications of this type of equipment we have: Heaters, coolers, condensers, evaporators, cooling towers, boilers, etc.

In the petroleum, chemical and petrochemical industries, the use of heat exchangers is of great importance.

2. Motivation

In order to seek alternatives to rationalize and reuse water to cool the cooling system of a steel rolling machine.

It consists of a cistern attached to the external wall of the bathroom and attached to the flush. The design will be done in such a way that the cistern causes the least visual impact to the property, but at the same time functional.

2.1 Theoretical Reference

According to Araújo (2002), Shell and Tube heat exchangers are equipment used in the most diverse fluid heating and cooling applications. They operate with one of the fluids flowing through the hull and the other through the tubes, with no contact between them. They can be used in different sectors of the industry, such as Chemical, Petrochemical, Food, Pharmaceutical, Steel, Mining, Paper and Cellulose, among others.

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These equipments can also be used in critical applications with lethal fluids, such as H2 and H2S, applications for low or high temperatures and high pressures.

They are exchangers widely used in industrial processes (heating, cooling, evaporation or vaporization and condensation of all kinds of fluids), when large areas are required for heat transmission.

- It consists of parallel tubes, through which one of the fluids circulates, mounted in a cylindrical hull, in which the other fluid circulates.
- Has manufacturing, cost, and thermal performance advantages. They can be built with large heat exchange surfaces in a relatively small volume, offering great flexibility in design and operating conditions (wide temperature and pressure range).
- Design of heat exchangers (characteristics, manufacturing, materials) **-** TEMA Standards:
- Class R: severe oil processing conditions
- Class C: moderate operating conditions, commercial application: refrigeration
- Class B: chemical processing services

Basic components of shell and tube heat exchanger:

- Hull
- Tube bundle
- Chicanas
- Heads
- Top

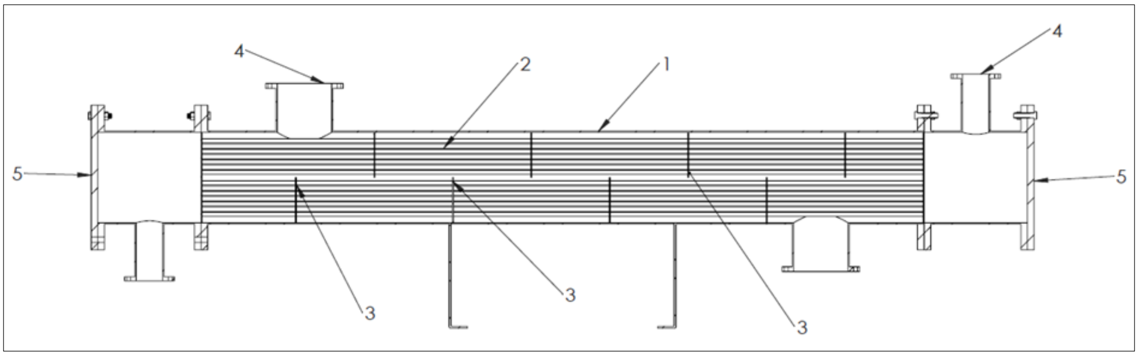

Figure 1 Heat Exchanger Main Part; (Source: Author)

Objective

For study purposes, we will develop this work where we will aim to promote good techniques for reusing water from the heat exchanger.

- Research methods and processes
- Service Definition
- Study of thermal load
- Calculations and dimensioning;
- Design in CAD -3D

3. Results and conclusion

3.1 Service Definition

The service that will be considered is a cooling of a waste water stream resulting from a cooling process in the rolling process.

In this project, the shell and tube heat exchanger will be dimensioned to cool waste water. It was previously determined that water flows through the interior of the tubes due to the increased tendency of water to corrode and encrust. Waste water is the hot process fluid that must be cooled from 120 °C to 40 °C. The waste water flow data are well known, as it is precisely this that will define the characteristics of the equipment to be designed, in order to meet the thermal objectives of the process. Below are tables with properties and characteristics of hot and cold fluids, taken from the service data sheet, provided as input for this project.

Table 1 Thermophysical Properties of Water

Cold Fluid			
Properties	**Symbol**	**Value**	**Value**
Inlet temperature	tl	31	^{0}C
Outlet temperature	t2	47	^{0}C
Mass flow	mwater	33,28	Kg/s
Specific mass at the entrance	ρ1 Water	995	Kg/m3
Specific mass at the output	ρ2 Water	986,07	Kg/m4
Dynamic Viscosity at the inlet	µ1 water	0,00078	Pa.s
Dynamic Viscosity at the output	µ2water	0,00051	Pa.s
Specific heat at the inlet	cl	4178,43	J/(Kg^{0}C)
Specific heat at the outlet	c2	4181,72	J/(Kg^{0}C)
Thermal Conductivity at the Inlet	k1 water	0,6164	J/(s m^{0}C)
Thermal conductivity at the output	k2 water	0,65	J/(s m^{0}C)
Manometric pressure	P water	676,66	J/(s m^{0}C)
Permissible load loss	ΔP water	68,65	k Pa
Deposit Ratio	Rf Λ	0,000688	(s m2^{0}C)/J

Source: Adapted by the author

3.2 Laminating Machine

Lamination can be defined as a process of forming metals where they pass between two rotating rollers that compress them, and have their thickness reduced and their length increased. It is a process that allows to obtain high productivity and good dimensional accuracy, in addition to a certain variety of shapes.

In this process, the material is subjected to high compressive stresses, resulting from the direct action of two rolls, and to surface shear stresses, resulting from friction between the rolls and the material. These frictional stresses are also responsible for pulling the material out of the space between the rollers.

The hot rolling process is one of the options for working steel. Considered as the basis of any production process, this type of lamination manufactures both final products and raw materials for other uses, both ends being used in different segments.

Aiming at this type of lamination, the lamination machine by inductive heating, needs cooling of the laminator head, thus the need for a heat exchanger.

Figure 2 Laminating Machine; Fonte: EUROTHERMO Aquecimento

4. Sizing

4.1 Process features

4.1.1 Cold fluid: liquid water. Hot fluid: distilled water in a liquid state.

$$T_{c1} = 20°C \quad T_{h1} = 120°C$$

$$P_{c1} = 30 \text{ kPa} \quad P_{h1} = 300 \text{ kPa}$$

$$m_c = 30 \text{ kg/s} \quad m_h = 10 \text{ kg/s}$$

$$T_{c2} = 40\ ^0C$$

Below is a step-by-step sizing:

- First Step: define the type of heat exchanger, in this case we define the shell and tube heat exchanger. We will fix the passage of the cold fluid through the hull and the fluid through the tubes. A heat exchanger with such a configuration can be connected for pure countercurrent or pure concurrent flow. The largest representative temperature difference between the hot fluid and the cold fluid is obtained with pure countercurrent flow.

We will fix the heat exchanger connection with countercurrent flow.

In the equation: $q = A.U.\Delta.Tr$

$\Delta.Tr$ = Representative temperature difference between the hot fluid and the cold fluid inside the heat exchanger. Each type of flow presents a different equation deduced for its calculation. The equation deduced on a case-by-case basis for calculating the representative temperature difference can be rewritten in the form:

$$\Delta.Tr=F.\ \Delta Tml$$

where F is a factor correction, which ranges from 0 to 1.0 given by an equation and which can be read in figures.

$$q = A.U.F\Delta.Tml$$

ΔTml is the logarithmic mean temperature difference between the hot fluid and the cold fluid, given by the expression:

$$\Delta Tml=\Delta Ta-\Delta Tb/In\Delta Ta/\Delta TB$$

and the temperature differences are those between the hot and cold fluid at the ends of the heat exchanger:

For countercurrent flow, the representative temperature difference is equal to the logarithmic mean temperature difference and F=1.0

- Second step: Through the energy balance, determine the missing variable to fully characterize the process conditions; in this case the hot fluid design outlet temperature (Th2). Also determine the heat load of the heat exchanger.

Heat to be received:

q_receivd =mc.cpc.(tc2-Tc1)

Data: Table B.2 Kakaç cpc at 30 °C = 4.179 kJ/kg.K

q_received=304190(40-20)=2507400W

If there are no heat losses in the system,

q_received= q_ceded

q_received=mh.cph(Th1-Th2)= 2507400=10cph(120-Th2)

We do not know Th2 and we do not know the arithmetic mean temperature of the hot fluid inside the heat exchanger. The heat capacity of liquids tends to follow a linear model (cp = a + bT). In this case the average cp in a temperature range becomes equal to the cp in the average temperature.

We have above an equation that can be solved by trials. Estimates the average temperature; get the cp at this temperature; calculates the outlet temperature, etc...

Data: Table B.2 Kakaç cp at 90ºC = 4.209 kJ/kg.K

And so, Th2 = 60.4 °C

Third step: Pre-estimation of the global heat exchange coefficient (U).

The U can be estimated by the convection heat transfer coefficients, which depend on a series of information related to the heat exchanger in question, such as the flow velocity. However, such variables are only defined when the exchanger is dimensioned. That is, we need U to dimension the heat exchanger, but we can only estimate the value of U when the exchanger is already dimensioned. This difficulty is overcome by an iterative process.

The starting point is then to pre-estimate the U with the help of previous experience conveyed in technical tables.

Table 8.4 pg 301 Kakaç. Sensible heat transfer with water: 5,000 W/m2K < h < 7,500 W/m2K

We will assume that an arithmetic mean value within the predicted range is acceptable for our case. *h = 6250 W/m2K*. For this case, initially ignoring the phenomenon of fouling on the exchange surfaces, the expression:

Table 8.5 pg 302 Kakaç. Overall water-to-water heat transfer coefficient without phase change and clean wall.

1,300 W/m2K < U < 2,500 W/m2K

We can use the tabulated values of U or h. We chose here to use the values of h.

For each heat exchanger configuration, for example, finned or not, the deduction of a specific equation for the calculation of U is necessary.

For this case, initially ignoring the phenomenon of fouling on the exchange surfaces, the expression:

1/C=do/(di.hi)+(do.In(do/di))/(2.k)+ 1/ho

Where Uc is valid for absolutely clean wall and for calculation of the exchange area referring to the external surface of the non-finned tubes.

The choice of the outer diameter of the tubes takes into account costs (which favors the use of small diameters) and ease of cleaning (which favors larger diameters). We will opt for a usual diameter,

do = 3/4" = 0.01905 m.

The choice of wall thickness and therefore of the internal diameter, takes into account the internal and external pressures, other data of structural strength and durability against corrosion. We will arbitrate a standard BWG 10 pipe.

di = 0.01224 m.

The choice of material of construction of the tubes allows the determination of its thermal conductivity. It depends on suitability for the corrosive characteristics of the fluids and on other information such as the working temperature range.

According to *Kaká (2002),* Table A1, page 469 makes it possible to know the conductivities. We will arbitrate a material of construction of the tubes with k = 76 W/m.K

So the calculated U for the clean wall is: Uc = 2153 W/m2.K (Within the expected range.)

However, when the heat exchanger comes from the factory, it is already slightly oxidized and/or dirty with oil. Even if clean, process fluids will oxidize over time and various impurities will be deposited on exchange surfaces, reducing transfer and equipment efficiency over time. For proper sizing; for the equipment to operate satisfactorily, it must be oversized so that desirable process conditions are achieved between maintenance stops. This can be accomplished using tabulated 'fouling resistances' (Rf), which can be employed with the equation:

1/Uf=1/Uc+Rfi+Rfo

From table 5.11 pg 177 Kakaç Rfi = Rfo = 0.000088 m2.K/w

The U for dirty wall is the one that must be considered for calculation purposes. Otherwise, the heat exchanger may not operate properly, even immediately after startup.

Calculating: Uf = 1,561 W/m2.K

Step: Basic sizing.

The logarithmic mean temperature difference is calculated as ΔTml = 58ºC and F = 1.0.

The estimated exchange area of the outer surface of the required heat exchanger tubes is:

$$S = 27.7 \text{ m}^2$$

This area is obtained by a certain number of tubes of a certain length.

S=π.do.L.Nt

So we have an equation and two unknowns. We need to fix one to calculate the other.

Kakaç usually exemplifies heat exchangers with a length-to-diameter ratio between 5 and 15.

Let's arbitrate a useful length of the exchange sector (without taking into account the heads) that culminates in the relationship indicated above (trial and error).

Setting L = 3.0 m

We calculated Nt ± 154 tubes.

Such a number of tubes will be arranged to be distributed within the hull. The arrangement or arrangement may be triangular or square. The use of triangular arrangement is more frequent and will be used here. Pitch is the spacing between the pipe axes. We will employ a Pt = 1 in. We have already seen that the outside diameter of the tubes is 3/4 in and that we have a pass through the tubes and the shell.

Table 8.3 pg 293 Kakaç allows you to read the inner diameter of the standard hull.

Di = 17 1/4 "

The exchanger in question has a useful length of 3.0 m and a diameter of 0.45 m, which results in a ratio of 7, consistent with the practice of the author of the textbook.

The spacing of the transverse baffles is of the same order of magnitude as the diameter of the exchanger. Setting a number of baffles of Nc = 7, the spacing (B) between the baffles is given by:

1/(Nc+1)=B and B=0.375 m

Fourth step: Calculation of the U estimate of the transfer coefficient for the fluid circulating inside the tubes. (In our case the hot fluid.)

The tables in chapter 3 of Kakaç have numerous equations appropriate for each specific condition.

Calculating the Reynolds number:

$$Re=4m/((Nt/Np).\mu\pi.di) = 2.23x10\text{-}4$$

Knowing that μ= 3.03x10-4 Pas

The Prandt is : Pr=cpu/k = 1.88,

Remember that k is the conductivity of the fluid.

The friction factor is given by: $f=(1.58.In.Re-3.28)^{(-2)}= 〖6.36\ x10〗^{(-3)}$

A suitable Nusselt equation for this range of Re and Pr is:

$$Nu=((f/2)Re.Pr)/(1.07+12.7(f/2)^{0.5}.(〖Pr〗^{(2/3)}-1))=92.28$$

The transfer coefficient is given by: $h=(K.Nu)/di = 5.103 W/m^2K$

K=0.677W/Mk

Fifth step: Estimation of the transfer coefficient for the fluid that circulates between the tubes and the hull. (In our case the cold fluid.) There are some alternatives, from the simplest to the most sophisticated, for estimating the transfer coefficient in the hull. One of the simplest, calculates Nusselt as a function of a modified Reynolds, Prandlt and a factor that takes into account the variation of fluid properties within the boundary layer. For liquids without phase change:

$$((ho,De)/k)=0.36.((De.Gs)/\mu)^{0.55}.((cp\ \mu)/k)^{(1/3)}.((\mu)/(\mu w))^{0.14}$$

Numerical values are reality-fit values. The equation works with the variables: heat transfer coefficient (ho, what you want to determine), apparent equivalent diameter of the channels through which the fluid flows, fluid conductivity (k), apparent mass velocity of the fluid through the channels (Gs) , heat capacity of the fluid (cp), average viscosity of the circulating fluid (μ), viscosity of the fluid at the average temperature of the exchange surface (μw).

Apparent equivalent diameter for triangular layout is estimated by the equation:

De=(4((〖Pt〗^2.√3)/4-(π.〖do〗^2)/8-)/(π.do))/2

and the De calculated, with the help of the variables previously fixed is:

De=0.01829m

spacing between the outer walls of the tubes (C) is determined by the expression:

and its calculated value is: C = $6.35.10^{-3}$m

The apparent cross-sectional area (As) through which the fluid flows is given by the expression:

AS=(Di.C.B)/Pt

and its calculated value is: As = 0.041 m2

The apparent mass velocity of the circulating fluid is given by:

Gs=m/As

and its calculated value is: Gs=730.34 kg/m2.S

Consequently, the apparent Reynolds value is

Re=(Gs.De)/μ= 16.393

that is within the validity range of the equation used. The validity range of this McAdams method is: 400<Re s<106

The viscosity at the arithmetic mean temperature of the cold fluid in the hull is:

μ = 8.15.10-4 Pa.s

And the viscosity of the cold fluid at the arithmetic mean temperature of the wall (approximately equal to the arithmetic mean temperature of the fluid circulating inside the tubes): μw = 4.66.10-4 Pa.s.

The cp at the arithmetic mean temperature of the fluid in the hull is: cp = 4179 J/kg.k and the conductivity: k = 0.612 W/m.k.

In this way ho is calculated as ho = 4801 W/m2.k

We now have the information needed to calculate the U:

1/Uc=do/(di.hi)+(do.In(do/di))/2k+1/ho

the U ignoring fouling (not usable) is: Uc = 1759 W/m2.K

1/Uf=1/Uc+Rf+Rfo

And the U considering the performance reduction by the fouling is: Uf=1343w/m2. K

Sixth step: Basic sizing, the heat load of the heat exchanger does not vary, as the fixed operating conditions do not vary. The logarithmic mean temperature difference is also the same and we remain with the pure countercurrent flow option. (ΔTml = 58 °C and one pass through the shell and tubes; F = 1.0.)

$q = A.U.F\Delta.Tml$

The estimated exchange area of the outer surface of the required heat exchanger tubes is: S = 32.2 m2

This area is obtained by a certain number of tubes of a certain length.

$S=\pi.do.L.Nt$

We can keep the tube length constant (L) and vary the number of tubes (Nt) up to an acceptable sizing. The variation in the number of tubes can imply a change in the diameter of the hull and a significant fluctuating variation of the heat transfer coefficient in the hull. In addition, the shell is usually the most expensive part of the heat exchanger. For convergence in general, it is more suitable to keep the number of tubes constant (as well as the layout, Di, do, Pt...) and vary the length of the tubes until the area of the dimensioned heat exchanger allows heat transfer. desired to the process.

do = 0.01905 m (3/4"), 154 tubes.

Calculating the new length: L= 3.49 m

Seventh step: Determination of pressure drops in the heat exchanger. Process characteristics can limit the allowable pressure drop of the two fluids as they pass through the heat exchanger. The pressure drop of the fluids is a relevant factor in the operating costs of the system, as such fluids need to be pumped or compressed to pass through the exchanger. There are limit reference values published in the technical bibliography on the subject.

For the pressure drop in the tubes and the estimate of the head loss in the heads, there is the expression:

ΔPtubes=(4fLNp/di+4Np) (pu^2)/2

f=〖(1.58.In Re-3.28)〗^(-2)

Re=(μ.di.ρ)/μ

For the estimation of the pressure drop of the fluid that circulates between the tubes and the hull there is the expression:

ΔHulls=(f.〖Gs〗^2 (Nc+1).Di)/(2ρ.De.Ø)

f=exp(0.576-0.19In Res S)

Re s=(Gs.De)/u

The validity range of this McAdams method is: 400<Re s<106

Eighth step: Modeling and detailing of the project generated in CAD.

By putting the information obtained in the calculations, in the generation of the CAD drawings, it was observed that the best arrangement of the tubes with the diameter of 450 mm was 152 tubes instead of 154, as well as to maintain a symmetry, instead of 7 baffles. 8 baffles were used.

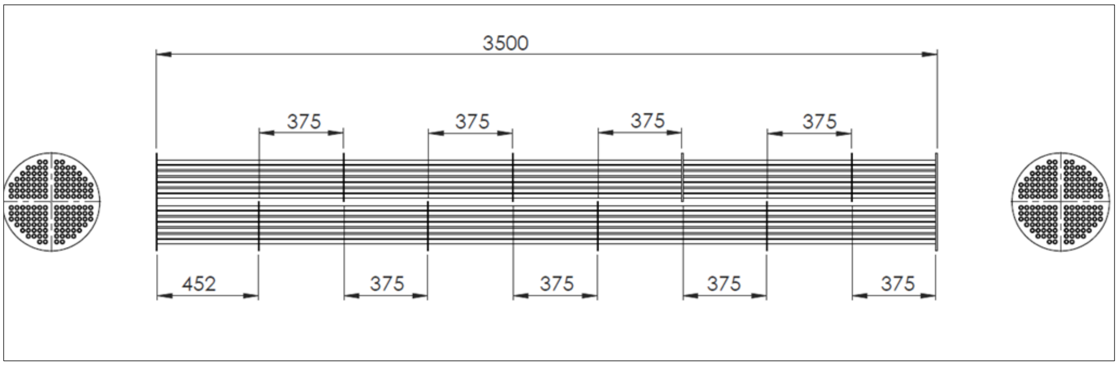

Figure 3 Heat Exchanger Main Parts; (Source: Author)

The heat exchanger was designed in the CAD program, Solidworks®, although the graphic part observed in Figure 3, where it appears in yellow, is agreed in accordance with the ABNT standard. NBR 6493/1994, that it is not painted, only insulated with rock wool or another type of thermal insulation.

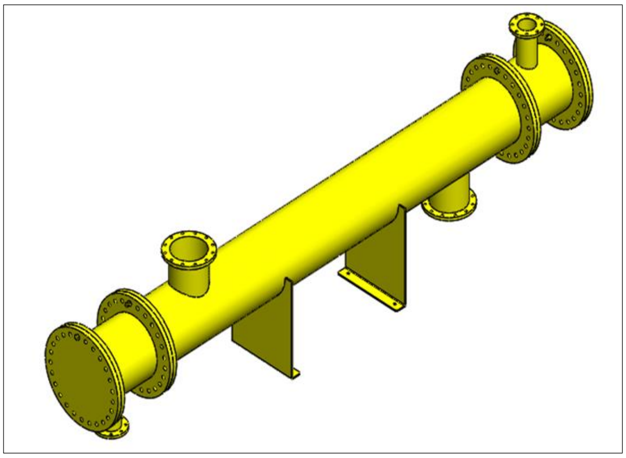

Figure 4 Heat Exchanger Main Parts Isometric view; (Source: Author)

The heat exchanger shell was designed with ASTM 36 material. According to Allgayer (2017), ASTM A36 steel is a structural steel widely used in various sectors of the Metalworking industry, as it has good weldability and medium mechanical strength. The structural profiles made of steel sheets have medium strength and are applied in various structural components, from the most common to the most elaborate, such as: metal structures in general, machinery, sawmills, walkways and agricultural implements, in addition to road and rail implements.

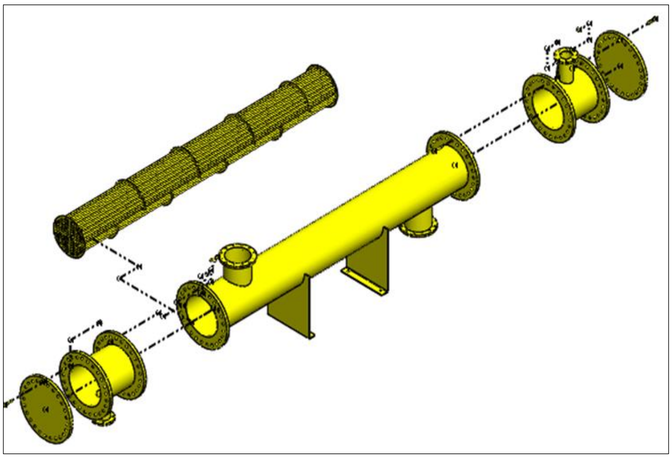

Figure 5 Heat Exchanger Main Parts Isometric view explode; (Source: Author)

The tube bundle of the exchanger consists of a sequence of tubes arranged and welded, in a perforated disc, followed by a guide for them.

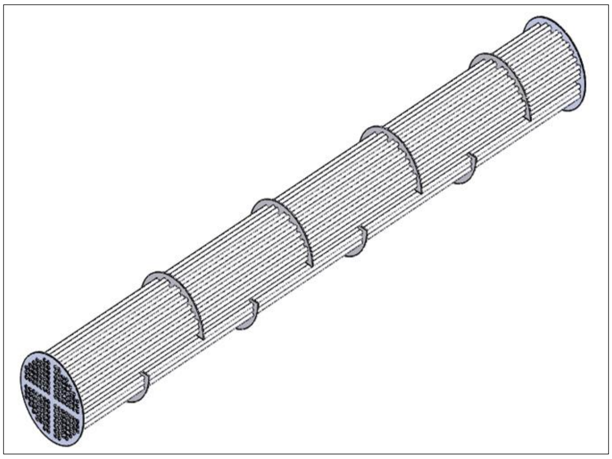

Figure 6 Heat Exchanger Main - Tube bundle; (Source: Author)

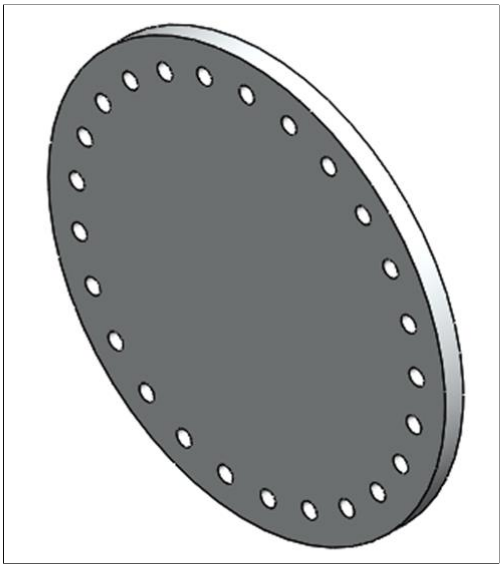

Figure 7 Heat Exchanger Main – Cover; (Source: Author)

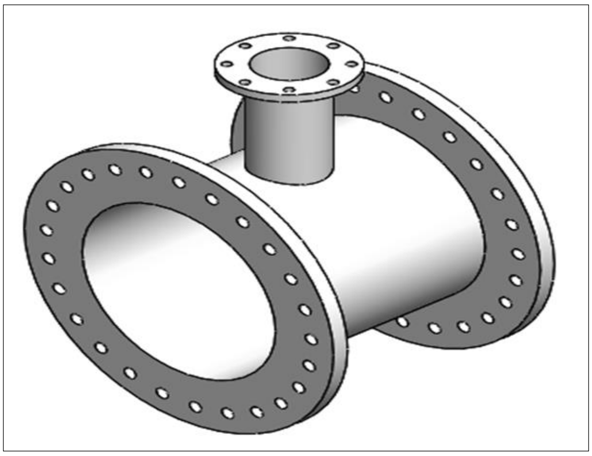

Figure 8 Heat Exchanger Main – Head Top; (Source: Author)

The heat exchanger head was also designed with ASTM 36 material, being manufactured in multiparts and then welded.

The tops were designed to also use ASTM 36 steel, using the laser cutting process.

5. Conclusion

In this article, a heat exchanger was calculated for an induction steel rolling machine to cool the waste water used in the cooling of the rolling mill head.

After applying all the theoretical methodology for calculating the heat exchanger, a configuration was reached for the exchanger, consisting of 154 tubes, 7 baffles, at a distance of 375 mm, with tubes of 19 mm of internal diameter and 12 mm of internal diameter.

When placing the information obtained in the calculations, the CAD program, it was observed that a better adaptation of the geometry to obtain symmetry, the exchanger ended up being designed with 152 tubes and 8 baffles.

Compliance with ethical standards

Acknowledgments

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Disclosure of conflict of interest

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References

[1] ALLGAYER, Alexandre André. Caracterização das propriedades mecânicas e metalúrgicas de uma junta de aço ASTM 36 soldado pelo processo de arco submerso (saw) com diferentes aportes térmicos. UNIVATES-Lajeado. 2017.

[2] ARAÚJO ECC. Trocadores de calor. 1. ed. São Carlos: Editora Edufscar. 2002.

[3] INCROPERA FP, DEWITT DP. Fundamentos de transferência de calore massa. 6. ed. Rio de Janeiro: LTC. 2008.

[4] KAKAC S. Heat Exchangers – Selection, Rating, and Thermal Design. 2. ed. IE – CRC Press. 2002.