Elective Design of Shell & Tube Heat Exchanger

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Abstract

This paper has been devoted to an elective design of shell and tube heat exchanger for pasteurization of cow milk by steam in a dairy plant. Milk is to flow through a bank of 1.2 cm internal diameter tubes while steam condenses outside the tubes at 1 atm. Milk is to enter the tubes at 4°C, and it is to be heated to 72°C at a rate of 15 L/sec. All calculation processes to specify the tube length and the number of tubes for the heat exchanger.

Keywords: *Elective design, Pasteurization of milk, Total surface area, Parallel flow, Counter flow, Materials involved.*

1. Introduction

According to Y. A. Cengel in his publication, Heat and Mass Transfer: a practical approach is as follows; "Heat exchangers are devices that facilitate the exchange of heat between two fluids that are at different temperature while keeping them from mixing with each other. Heat exchangers are commonly used in practice in a wide range of applications, from heating and air-conditioning systems in a household, to chemical processing and power production in large plants. Heat exchangers differ from mixing chambers in that they do not allow the two fluids involved to mix." [2].

Fig. 1. Pasteurizing milk heat exchanger

Although there are many applications for heat exchangers in manufacturing field. Especially in food industries, heat xchanger is being used to transfer heat from hot to cold or vice versa. Pasteurizing milk is one type of heat exchanger using in food applications. Design calculations are used to evaluate the efficiency of heat exchanger or to design new one for new purpose. By using this method, engineers can determine the rate of heat of both exit fluid cold and hot. Also, it can be used to estimate the power consumptions (figure 1). [1]. Elective design as a part of heat exchanger is complicated process that is composed of many aspects. This paper will discuss calculations that should be involved in thermal side.

2. Characteristic Design

In heat exchanger design, the overall surface area of heat exchanger plays role in both inlet and outlet temperatures of product fluid and heating fluid. The flow rate of the fluid is also related to the surface area. Moreover, surface area plays a role in choosing the flow direction and pressure drop. Determining the total surface area, which is composed of tubes diameter, shell diameter and other geometrics, is important in heat exchanger designing.

Heat exchangers are named according to the direction of cold fluid and hot fluid to two kinds. Parallel flow is occurred in devices that the flow of both hot and could flow is moving in the same direction. In this case, usually the outlet temperature of heating fluid is more than the outlet temperature of cold fluid. Counter flow is existed when the cold and hot fluids are moving in the opposite direction of each other. In this case, the outlet temperature of cold fluid is sometimes more than the outlet temperature of heating fluid.

3. Classification based on construction

A fixed-tube sheet heat exchanger (Fig. 2) has straight tubes that are secured at both ends to tube sheets welded to the shell. The construction may have removable channel covers, bonnet-type channel covers, or integral tube sheets. The basic advantage of the fixed tube sheet construction is its low cost because of its simple construction. In fact, the fixed tube sheet is the least expensive construction type, as long as no expansion joint is required. [5].

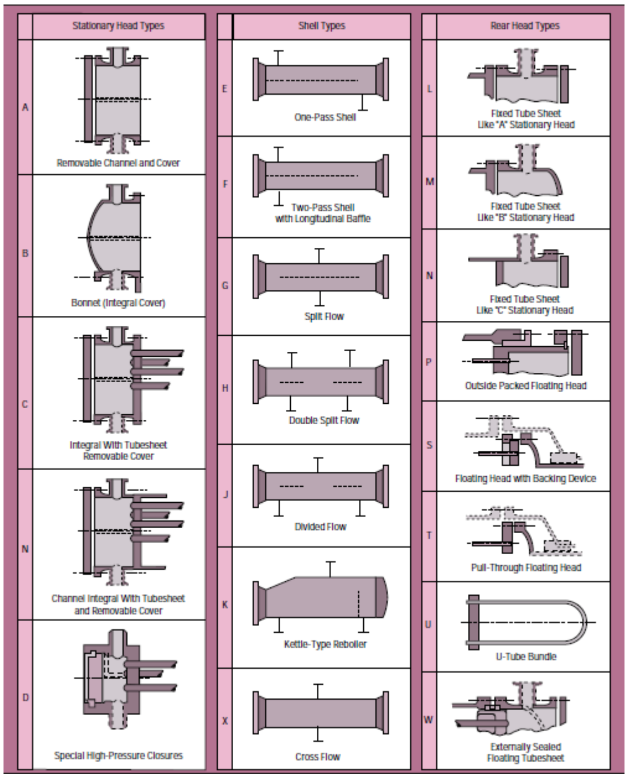


Fig. 2. Designation of Shell- and- Tube Heat Exchanger

Other advantages are that the tubes can be cleaned mechanically after removal of the channel cover or bonnet, and that leakage of the shell-side fluid is minimized since there are no flanged joints. A disadvantage of this design is that since the bundle is fixed to the shell and cannot be removed, the outsides of the tubes cannot be cleaned mechanically. Thus, its application is limited to clean services on the shell-side. However, if a satisfactory chemical cleaning program can be employed, fixed-tube sheet construction may be selected for fouling services on the shell-side. In the event of a large differential temperature between the tubes and the shell, the tube sheets will be unable to absorb the differential stress, thereby making it necessary to incorporate an expansion joint. This takes away the advantage of low cost to a significant extent.

4. Calculation Processes

$T_{in,Milk} = 4\ ℃$

$T_{out,Milk} = 72\ ℃$

$\dot{m}_{total} = 15\ L/sec$

$d_i = 1.2\ cm$

$T_f = \frac{72+4}{2} \gg T_f = 38℃$

It is assumed that:

1. *The number of tubes is 30 tubes.*
2. *The properties of milk equal to the water properties.*

$C_p = 4178.5 \frac{J}{Kg.K}$

$Pr = 4.572$

$\rho = 993.7 \frac{kg}{m^3}$

$K = 0.627 \frac{W}{m.K}$

$A_c = \frac{\pi}{4} D_i^2$

$Ac = 1.131x10^{-4} m^2$

$\mu = 0.759\ x\ 10^{-3} \frac{kg}{m.s}$

$v = \frac{\mu}{\rho}$ *"Kinematic viscosity"*

$v = 7.638\ x\ 10^{-7} \frac{m^2}{s}$

$\dot{Q} = \dot{m}\, C_p \left(T_{c,out} - T_{c,in}\right)$

Thus;

1 kg/sec = 1.0753 litter/sec

$\dot{m}_{30\ tubes} = \frac{15}{1.0753} = 13.95\ kg/sec$

$\dot{m}_{one\ tube} = \frac{13.95}{30} = 0.465\ kg/sec$

$\dot{Q} = 0.465 \frac{Kg}{s}\ x\ 4178.5 \frac{J}{Kg.K} (72\ ℃ - 4\ ℃)$

$\dot{Q} = 132.124\ Kw$

$\dot{m} = \rho\, V_{avg}\, A_c$

$V_{avg} = \frac{\dot{m}}{\rho . A_c} >> V_{avg} = \frac{0.465}{993.7\ x\ 1.131\ x\ 10^{-4}}$

$V_{avg} = 4.138 \frac{m}{s}$

$Re = \frac{V_{avg} . D_i}{v} >> Re = \frac{4.138\ x\ 1.2\ x\ 10^{-2}}{7.638\ x\ 10^{-7}}$

$Re = 65011.78 >> \quad Re > 10{,}000$

Since Reynolds number is more than 10,000, the flow is TURBULENT.

Nusslet Number is:

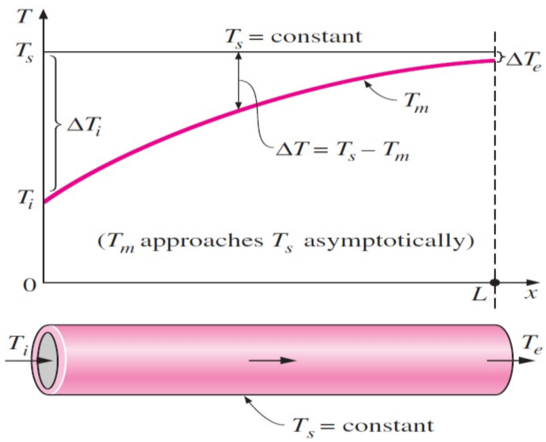
$Nu = 0.023 Re^{0.8} Pr^{n} >>$

$Nu = 0.023 (65011.78)^{0.8} (4.572)^{0.4} >>$ *for heating, n is 0.4*

$Nu = 299.34$

From the equation $Nu = \frac{h_i . D_i}{K}$ *th*

$$h_i = 15640.515\ \frac{W}{m^2.K}$$



$$h_i = \frac{Nu.K}{D_i} >> h_i = \frac{299.34\ x\ 0.627}{1.2\ x\ 10^{-2}}$$

From the equation, $\dot{Q} = hA_s\Delta T_{lm} >> A_{s\ one\ tube} = \frac{Q}{h\Delta T_{lm}}$

$\Delta T_e = T_s - T_{milk\ out} >> \Delta T_e = 100\ °C - 72\ °C = 28$

$\Delta T_i = T_s - T_{milk\ in} >> \Delta T_i = 100\ °C - 4.0\ °C = 96$

$\Delta T_{lm} = \frac{\Delta T_e - \Delta T_i}{\ln(\Delta T_e/\Delta T_i)} >> \Delta T_{lm} = \frac{28-96}{\ln(28/96)}$

$\Delta T_{lm} = 55.188$

$A_{s\ one\ tube} = \frac{Q}{h\Delta T_{lm}} >> A_s = \frac{132124.17}{15640.515 * 55.188}$

$\boldsymbol{A_s = 0.153\ m^2}$

From the equation, $A_s = n\pi D_i L$ *where n=1*

$L_{one\ tube} = \frac{A_s}{n\pi D_i} >> L = \frac{0.153}{\pi\ x\ 1.2\ x\ 10^{-2}}$

$\boldsymbol{L = 4.058\ m}$

The frication factor is:

$f = 0.184\ Re^{-0.2} >> f = 0.184\ (65011.78)^{-0.2} >>$

$\boldsymbol{f = 0.02}$

$\dot{W} = \dot{V}\Delta P$

$\Delta P = f\frac{L}{D}.\frac{\rho v_m^2}{2} >> \Delta P = 0.02\frac{4.058}{1.2\ x\ 10^{-2}}.\frac{993.7\ (4.138)^2}{2}$

$\boldsymbol{\Delta P = 57539.63}$

$\dot{V} = V_m.A_c n >> \dot{V} = 4.138\ x\ 1.131\ x\ 10^{-4}\ x\ 30$

$\boldsymbol{\dot{V} = 0.014\frac{m^3}{s}}$

$\dot{W} = \dot{V}\Delta P >> \dot{W} = 0.014\ x\ 57539.63$

$\boldsymbol{\dot{W} = 805.554\ W}$

The power of pump is W/745.7 (Hp)

See (Fig. 3).

5. Conclusion

The calculation of thermal design had assisted to the design and operation efficiency of heat exchangers [3,4]. Is usually used routinely LMTD method to estimate the changes in temperature in the fluid and the size of heat exchangers. The method is based on the assumption that the properties of liquids are constant along the heat exchanger (but we did not use LMTD due to lack of knowledge steam temperature in / out of our system). To overcome these assumptions, the design calculations must be performed using computational techniques. Estimated coefficient of heat transfer surface could be the most difficult design because of its dependence on the flow of fluid, thermal properties, flow conditions, heat exchanger engineering, and surface roughness. Modern tools such as computational fluid dynamics are able to help determining the values of surface heat transfer coefficient.

Nomenclature

A	AREA OF HEAT EXCHANGE SURFACE (M^2)
C_p	specific heat of fluid (J/kg K)
F	correction factor (dimensionless)
H	surface heat transfer coefficient (W/m^2 K)
K	thermal conductivity (W/m K)
L	length of a cylindrical tube (m)
LMTD	log mean temperature difference (ºC)
$\dot{m}$	mass flow rate (kg/sec)
Q	heat flow rate (W)
Re	Reynolds number (dimensionless)
Pr	Prandtl number (dimensionless)
Nu	Nusslet Number (dimensionless)
Rt	total thermal resistance of heat exchange surface (K/W)
Rf	fouling factor (m^2 K/W)
T	temperature (Celsius or Kelvin)
ΔT_{lm}	log mean temperature difference (ºC)
U	overall heat transfer coefficient (W/m^2 K)

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