Design and Performance Analysis of Shell and Tube Intercooler used in Double Acting Two Stage Reciprocating Compressor

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Abstract: ***A Compressed air system is the source of process and service air to be used for various applications such as Pneumatic instruments, cooling of shut-off rods, masking of air, drying of fuel and other in-pile assemblies, etc. An Intercooler acts as a Heat Exchanger between the multiple stages of the working compressor and helps transfer thermal energy between fluids at different temperatures. This article is concerned with the design and study of calculations involved in the performance of an Intercooler. Various parameters are responsible for the performance review under different operating conditions. Performance analysis parameters such as overall heat transfer coefficient, fluid velocity, pressure drop, and others are calculated using standard operating values. Further, considering the given operating conditions, the effectiveness of the Intercooler has been evaluated.***

Keywords: Compressed Air System, Shell and tube heat exchanger, Overall heat transfer coefficient, Performance analysis

1. Introduction

In an industry as well as a nuclear research station, reciprocating compressors are the most widely used type of air compression because of the high-pressure head delivering capability of these compressors as compared to centrifugal compressors.

Compression is done in more than one stage for improving the efficiency of the process and between each stage intercooler is provided. Intercooler improves the quality of air and reduces inlet air temperature. This improvement is achieved as by decreasing the temperature of hot discharged air from the previous stage, the volume of the air decreases and consequently the work input required for subsequent compression stage. On doing this large quantity of condensate (water) are formed. Proper separator arrangement should be made without considerable pressure drop since distraining of liquids can be a problem in intercooler systems of compressor plants.

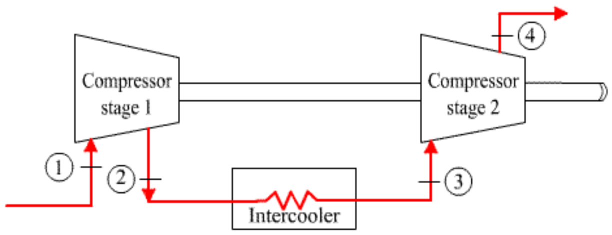

Figure 1: Two stage compression process with an intercooler

1.1 Intercooler

Intercoolers are provided between successive stages of a multi-stage compressor to remove the heat of compression hence reduces the work of compression (power requirements). The work of compression is reduced by reducing the specific volume through cooling the air. Thus intercooling affects the overall efficiency of the machine. Ideally, the temperature of the inlet air at each stage of a multi-stage machine should be the same as it was at the first stage. This is referred to as "perfect cooling" or isothermal compression. But in actual practice, the inlet air temperatures at subsequent stages are higher than the normal levels resulting in higher power consumption, as a larger volume is handled for the same duty.

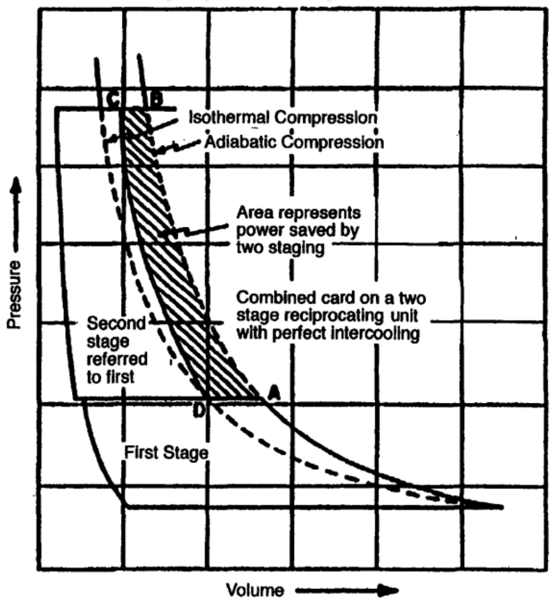

Figure 2: P-V Diagram for two stage compressor with perfect intercooling

1.2 Functions of an Intercooler

Intercooler used in air compressor performs following functions:

- Atmospheric air contains moisture, and furthermore, the air may pick up oil vapor as it passes through some compressors. Cooling the air down to or below its initial temperature will remove moisture down to the dew point, improving the quality of the air.

- Another purpose of the intercooler is to improve the efficiency of compression. This is done by reducing the work of compression (power requirements).
- As the air comes out from the compressor is at a higher pressure as well as at higher temperature. This higher temperature may create a problem for pneumatic tools, so intercoolers are used to reduce the outlet temperature of the compressed air.
- Every 4 ℃ rise in inlet air temperature results in a higher energy consumption by 1 percent to achieve equivalent output. Hence the intake of cool air improves the energy efficiency of a compressor.

1.3 Shell and tube type intercooler

A heat exchanger such as an intercooler is a mechanical device which is used for the purpose of exchange of heats between two fluids at different temperatures. There are various types of intercoolers available in the industry, however, the Shell and Tube Type intercooler is probably the most used and widespread type of the intercooler's classification. It is used most widely in various fields such as oil refineries, thermal power plants, nuclear stations, chemical industries and much more. This high degree of acceptance is due to the comparatively large ratio of heat transfer area to volume and weight, easy cleaning methods, easily replaceable parts etc. Shell and tube type intercooler consists of a number of tubes through which one fluid flows. Another fluid flows through the shell which encloses the tubes and other supporting items like baffles, tube header sheets, gaskets etc. The heat exchange between the two fluids takes through the wall of the tubes.

1.4 Principle components of Shell and tube type intercooler

Some of the very basic components of a shell and tube type intercooler are as given below:

1.4.1 Tubes

The tubes are the basic components of a shell and tube type intercooler. The outer surfaces of the tubes are the boundary along which heat transfer takes place. It is therefore recommended that the tubes materials should be highly thermal conductive otherwise proper heat transfer will not occur. The tubes of Copper, Aluminium and other thermally conductive materials are commonly used in practice.

1.4.2 Tube sheets

The tubes are held in place by being inserted into holes in the tube sheet and they are either expanded into grooves cut into the holes or welded to the tube sheet where the tube protrudes from the surface. The tube sheet is usually a single round plate of metal that has been suitably drilled and grooved to take the tubes (in the desired pattern), the gaskets, the spacer rods and the bolt circle where it is fastened to the shell. However, where mixing between the two fluids (in the event of leaks where the tube is sealed into the tube sheet) must be avoided, a double tube sheet may be provided.

1.4.3 Shell: The shell is simply the container for the shell side fluid, and the nozzles are the inlet and exit ports. The shell normally has a circular cross section and is commonly made by rolling a metal plate of the appropriate dimensions into a cylinder and welding the longitudinal joint ("rolled shells").

1.4.4 Impingement plates

When the fluid under high pressure enters the shell there are high chances that if the fluid will directly impinge over the tubes then their breakage or deformation may occur. To avoid the same the impingement plates are installed to reduce the kinetic energy of fluid to some extent so that the fluid may impact the tubes with lower velocity.

1.4.5 Channel covers

The channel covers are round plates that bolt to the channel flanges and can be removed for the tube inspection without disturbing the tube side piping. In smaller intercoolers, bonnets with flanged nozzles or threaded connections for the tube side piping are often used instead of the channel and channel covers.

1.4.6 Baffles

Baffles serve two functions; Most importantly, they support the tubes in the proper position during assembly and operation and prevent vibration of the tubes caused by flow-induced eddies, and secondly, they guide the shell side flow back and forth across the tube field, increasing the velocity, turbulence and heat transfer coefficient.

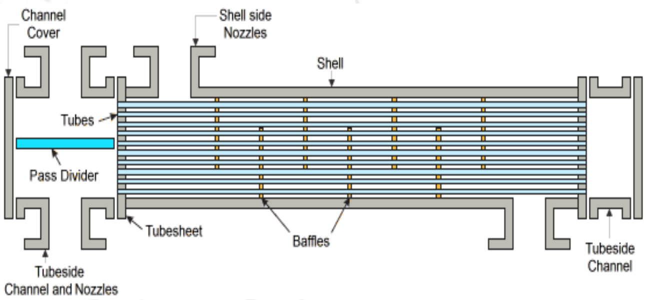

Figure 3: Schematic diagram of principle components of shell and tube type intercooler

2. Literature Review

Ebieto, C.E and Eke G.B. [1] in their experimental paper carried out the performance analysis of shell and tube heat exchanger and used an analytical method to develop a correlation for the performance analysis.

The thermal analysis of a shell and tube heat exchanger or an intercooler involves the determination of the overall heat-transfer coefficient from the individual film coefficients, and (Kern, 1965). The shell-side coefficient presents the greatest difficulty due to the very complex nature of the flow in the shell. In addition, if the exchanger employs multiple tube passes, then the LMTD correction factor must be used in calculating the mean temperature difference in the exchanger. For the turbulent flow regime ($Re \geq 104$), the following correlation is widely used (serth, 2007).

$$Nu=Re^{0.8}\,Pr^{1/3}(\mu/\mu_w)^{0.14} \qquad (1)$$

Where,

Nu = Nusselt Number = hd/k
Re = Reynold's Number = DVρ/μ
Pr = Prandtl Number = $c_p\mu/k$
D = Inside diameter of the pipe
V = average fluid velocity.
C_p,μ,ρ,k = Fluid properties at average bulk temperature.
μw = Fluid viscosity evaluated at average wall temperature. Eq. (1) holds good for 0.5 ≤ Pr ≥ 17,000 & for pipes L/D ≥10

However, for short pipes 10 ≤ L/D ≥ 60, the right-hand side of the equation is often multiplied by the factor $[1+(D/L)^{2/3}]$ to correct for the entrance and exit effects. (Serth 2007).

For laminar flow in circular pipes (Re < 2100), the seider-Tate correlation takes the form:

Nu = $1.86[RePrD/L]^{1/3}$ $(\mu/\mu w)^{0.14}$. This equation is valid for 0.5< Pr <17000 and
$[RePrD/L]^{1/3}$ $(\mu/\mu_w)^{0.14}$ (2)

For low in he transition region (2100 < Re < 104), the Hausen correlation is:

$$Nu = 0.116(Re^{2/3} - 125)\, Pr^{1/3}(\mu/\mu_w)^{0.14}\, (1+(D/L)^{2/3}) \tag{3}$$

In computing the tube-side coefficient h_i it is assumed that all tubes in the exchanger are exposed to the same thermal and hydraulic conditions. The value of h_i is then the same for all tubes, and the calculation can be made for a single tube. Equations (1), (2), or (3) were used, depending on the flow regime. The tube fluid heat transfer coefficient, hi, can be calculated using;

$$h_i = (NuK/Di) \tag{4}$$

The Delaware method (Serth, 2007) was used to compute the shell-side heat transfer coefficient, h_o. In the equation for the overall heat transfer coefficient, the temperature difference, Δtm, is the mean temperature difference between the two fluid streams. Since is independent of position along the exchanger, Δtm is the logarithmic mean temperature difference (Serth, 2007);

$$\Delta tm = LMTD = \frac{(T1-t2)-(T2-t1)}{\ln(\frac{(T1-t2)}{(T2-t1)})} \tag{5}$$

Equation (5) is valid regardless of whether counter flow or parallel flow is employed. In multi-pass shell and tube exchangers, the flow pattern is a mixture of co -current and countercurrent flow. For this reason, the mean temperature difference is derived by introducing a correction factor, F, which is termed the LMTD correction factor;

$$\Delta tm = F * \Delta tm \tag{6}$$

The correction factor is a function of the shell and tube fluid temperatures, and the number of tube and shell passes. This is corrected using two dimensionless temperature ratios (Serth, 2007); let, N = No. of shell side passes then

$$S = \frac{t2-t1}{T1-t1} \tag{7A}$$

$$R = \frac{T1-T2}{T2-t1} \tag{7B}$$

Then F is calculated as:

$$F = \frac{(R^2+1)^{0.5}\ln\left(\frac{1-S}{1-RS}\right)}{(R-1)\ln\left[\frac{\left[(2-S(R+1-(R^2+1)^{0.5})\right]}{\left[(2-S(R+1-\sqrt{R^2+1}))\right]}\right]} \tag{8}$$

The required overall heat transfer coefficient is given by;

$$U_{req} = \frac{Q}{AF\Delta Tm} \tag{9}$$

The clean overall heat transfer coefficient is given by;

$$U_c = \left[\frac{Do}{hiDi} + \frac{Do\ln(\frac{Do}{Di})}{2k} + \frac{1}{ho}\right]^{-1} \tag{10}$$

And the design overall heat transfer coefficient is given by;

$$U_D = (\frac{1}{U_c + R_D})^{-1} \tag{11}$$

The effect of fouling is allowed for in the design by including the inside and outside fouling coefficients. Kern (1965) presented typical values for the fouling factors for common process service fluids used in plane tubes (not finned tubes). The fouling factor for the exchanger is given as (Serth, 2007);

$$R_D = R_{Di}\left(\frac{Do}{Di}\right) + R_{Do} \tag{12}$$

Design problems frequently include specifications of the maximum allowable pressure drops in the two streams. In that case, pressure drops for both streams would have to be calculated in order to determine the hydraulic suitability of the heat exchanger. The pressure drop due to fluid friction in the tubes is given by Equation (15) with the length of the flow path set to the tube length times the number of tube passes (Serth, 2007).

$$\Delta_{pf} = \frac{fn_pLG^2}{2\rho D_i s\varnothing} \tag{13}$$

Δ_{pf} = Pressure drop (Pa)
f = Darcy friction factor (dimensionless)
n_p = No. of tube passes (dimensionless)
L = Tube length (m)
G = Mass flux (kg/s m2)
Di= Tube inside diameter (m)
ρ = Density of water (kg/m3)
s = Fluid specific gravity (dimensionless)

$\varnothing$ = Viscosity correction factor (dimensionless)

$\varnothing = (\mu/\mu_w)^{0.14}$ for turbulent and transition flow.

$\varnothing = (\mu/\mu_w)^{0.28}$ For laminar flow

For laminar flow friction factor is given by;

$$f = 64/Re \tag{14}$$

for turbulent flow (RE > 3000) following equation can be used

$$f = 0.4137\, R_e^{-0.2385} \tag{15}$$

The minor losses on the tube side are estimated using the following equation:

$$\Delta_{pf} = 0.5\text{x}10^{-4} a_r G^2/s \tag{16}$$

where a_r is the number of velocity heads allocated for minor losses. Serth (2007) proposed the following expression for computing the shell-side pressure drop:

$$\Delta_{pf} = \frac{f d_{op} G^2 (n_b + 1)}{2\rho D_p s \varnothing} \tag{17}$$

The shell side friction factor formula is given by;

$$f' = 144\left\{ f_1 - 1.25\left(1 - B/Do\right)\left(f_1 - f_2\right)\right\} \tag{18}$$

B.Jayachandriah & K. Rajasekhar [2] An attempt is made in this paper is for the Design of shell and tube heat exchangers by modeling in CATIA V5 by taking the Inner Diameter of shell is 400 mm, length of the shell is 700 mm and Outer diameter of the tube is 12.5mm, the length of Tube is 800mm and Shell material as Steel 1008, Tube Material as Copper and Brass.
By using modeling procedure Assembly Shell and Tube with water as a medium is done. By using ANSYS software, the thermal analysis of Shell and Tube heat exchangers is carried out by varying the Tube materials. The comparison is made between the Experimental results, ANSYS. With the help of the available numerical results, the design of Shell and Tube Heat Exchangers can be altered for better efficiency.
Analysis has been done by varying the tube materials and it is found that copper material gives the better heat transfer rates than the brass material.

1) The rate of heat transfer can be improved by varying the tube diameter, length and no of tubes.
2) By changing the pitch lay out rate of heat transfer can be improved.
3) By changing the temperature of tubes and the medium, the rate of heat transfer can be improved.
4) By changing the materials of tubes heat transfer rate can be improved.

Rajeev Mukherjee [3] explains the basics of exchanger thermal design, covering such topics as STHE components; classification of STHEs according to construction and according to service; data needed for thermal design; tube side design; shell side design, including tube layout, baffling, and shell side pressure drop; and mean temperature difference. The basic equations for tube side and shell side heat transfer and pressure drop. Correlations for the optimal condition are also focused and explained with some tabulated data. This paper gives an overall idea to design optimal shell and tube heat exchanger. The optimized thermal design can be done by sophisticated computer software, however, a good understanding of the underlying principles of exchanger designs needed to use this software effectively.

Jiangfeng Guo et.al[4] took some geometrical parameters of the shell-and-tube heat exchanger as the design variables and the genetic algorithm is applied to solve the associated optimization problem. It is shown that for the case that the heat duty is given, not only can the optimization design increase the heat exchanger effectiveness significantly, but also decrease the pumping power dramatically.

3. Design Method

3.1 Approach

Shell and tube heat exchanger or in this case intercooler is designed normally by using either Kern's method or Bell-Delaware method. Kern's method is mostly used for the preliminary design and provides conservative results whereas; the Bell- Delaware method is a more accurate method and can provide detailed results. It can predict and estimate pressure drop and heat transfer coefficient with better accuracy. Here, Kern's method of designing has been described in detail. The steps of designing are described as follows by considering an actual industrial operation:

3.2 Input Data

Hot fluid inlet temperature (T1) = 160°C
Hot fluid outlet temperature (T2) = 40°C
Cold fluid inlet temperature (t1) = 32°C
Cold fluid outlet temperature (t2) = 38°C
Fouling factor of hot fluid (R_{da}) = 0.00035 m^2K/W (for air)
Fouling factor of cold fluid (R_{dw}) = 0.00057 m^2K/W (for water)
P_{inlet} (for hot fluid) = 2.67 kg/cm^2
P_{inlet} (for cold fluid) = 5 kg/cm^2
Δp_{max} (for hot fluid) = 0.35 kg/cm^2
Δp_{max} (for cold fluid) = 0.5 kg/cm^2
Flow rate of hot fluid (m_a) = 18.1 m^3/min
Flow rate of cold fluid (m_w) = 8.5 m^3/hr
(Subscripts a for air and w for water).

Fluid properties at temperatures of 100°C for air and 35°C for water:

Viscosity:
μw =0.71932 cp
μa =0.02195 cp

Density:
ρw =994 kg/m^3
ρa =2.446 kg/m^3

Thermal conductivity:
kw =0.62350 W/m*K
ka =0.03145 W/m*K

Specific heat capacity:
Cw = 0.99863 cal/g*k
Ca = 0.24200 cal/g*k

Specific gravity:
Sw = 0.994
Sa = 2.029

3.3 Energy Balance

Assuming no heat loss to surroundings, we have

$$Q_a = Q_w = m_a C_a (T1 - T2) = m_w C_w (t1 - t2) \quad (19)$$

So, $Q_w = 5.06 * 10^4$ Kcal/hr

3.4 Calculation of Log Mean Temperature Difference (LMTD)

For 1 shell pass and 1 tube pass shell and tube intercooler with following dimensions and considerations.

Floating tube plate type:
a. 9.5 mm OD tubes with 12.7 mm triangular pitch (Pt)
b. Tube length (Lt) = 1810 mm
c. Tube ID=7.1 mm with a wall thickness of 1.2 mm
d. Flow area per tube= 39.59 mm^2

The log mean temperature correction factor (FT) for 1-1 shell and tube intercooler will be taken as 1 since both the shell and tube side fluid make a single traverse through the intercooler. For counter flow, we have:

$$\Delta tm = LMTD = \frac{(T1 - t2) - (T2 - t1)}{\ln(\frac{(T1 - t2)}{(T2 - t1)})}$$

LMTD = 41.84 °C

3.5 Determining the heat transfer area, A and Number of tubes

The value of overall heat transfer coefficient (Uo,assm) of 215 Kcal/hr/m^2/°C (≈250 W/m^2K) is assumed for the design calculation of the compressed air and water intercooler. The approximate range of overall heat transfer coefficient depending on the hot and cold fluid can be found out from available resources.

$$A = \frac{Q_w}{U_{assum} LMTD * F}$$

A=5.62 m^2
Calculating no. of tubes (n_t):

$$N_t = \frac{A}{\pi d_o L_t} \quad (20)$$

N_t= 104
This is taken corresponding to the closest standard shell ID of 148 mm for floating tube sheet, 1-shell and 1-tube pass intercooler with 9.5 mm tube OD on 12.7 mm triangular pitch. The tube counts are available in heat transfer resources.

3.6 Fluid Velocity

$$R_e = \frac{4m_a\left(n_p / n_t\right)}{\pi d_i \mu} \quad (21)$$

R_e = 57965.24 > 10^4
So now,

$$u = \frac{R_e \mu_a}{d_i \rho_a} \quad (22)$$

u = 48.16 m/s

3.7 Determination of Heat Transfer Coefficient

3.7.1 Tube side heat transfer coefficient:

$$j_H = \frac{h_i d_i}{k_a}\left(\frac{\mu_a C_a}{k_a}\right)^{-1/3}\left(\frac{\mu}{\mu_w}\right)^{-0.14} \quad (23)$$

j_H=170 for the tube side fluid at Re=57965.24 (**[5] page 834**)

Considering $\left(\frac{\mu}{\mu_w}\right)$ = 1, we have,

h_i = 671.07 W/m^2K

3.7.2 Shell side heat transfer coefficient:

Assumptions:
a) 25% cut segmental baffles
b) Baffles spacing, B= 74 mm (half of the shell ID is selected).
c) Equivalent diameter for the shell side:

For triangular pitch,

$$D_e = \frac{4\left(0.5P_T * 0.86P_T - 0.125\pi d_o^2\right)}{0.5\pi d_o} \quad (24)$$

D_e= 9.09 mm
d. Shell side cross flow area:

$$a_s = \frac{CBD_S}{P_T} \quad (25)$$

Where C is tube clearance and C=P_T-d_o
So, a_s = 0.00276 m^2
e. Mass Velocity:

$$G_s = \frac{m_w}{a_s} \quad (26)$$

G_s = 850.342 Kg/m^2s

f. $$R_e = \frac{D_e G_s}{\mu_w} \quad (27)$$

R_e = 10745.71
Now, for shell side:

$$j_H = \frac{h_o D_e}{k_w}\left(\frac{\mu_w C_w}{k_w}\right)^{-1/3}\left(\frac{\mu}{\mu_w}\right)^{-0.14}$$

j_H = 60 for the shell side fluid at Re=35668 with 25% cut segmental baffles (**[5] page 838**)

So, h_o = 6992.11 W/m^2K

3.8 Overall heat transfer co-efficient $U_{o,\,Cal}$:

$$U_{o,cal} = \left[\frac{1}{h_o} + R_{dw} + \frac{A_o}{A_i}\left(\frac{d_o - d_i}{2k_c}\right) + \frac{A_o}{A_i}\left(\frac{1}{h_i}\right) + \frac{A_o}{A_i} R_{da} \right]^{-1} \quad (28)$$

Let select, Cupro-nickel 70:30 as tube material with thermal conductivity, k_c = 30 W/mK,

$U_{o,cal}$ = 245.28 W/m^{2}K

Now, $\frac{U_{assum} - U_{cal}}{U_{assum}} * 100$ =1.88% < 30%

3.9 Pressure drop calculation:

3.9.1 Tube side pressure drop

Friction factor f = 0.00018 x 144 = 0.02592 ft2/ ft2 (for Re=57965.24 (**[5] page 836]**).

a_t = (no. of tubes)×(flow area per tube)/(no. of passes) =0.004117 m^2

Tube side mass velocity, G_t = (m_a/ a_t) = 179.226 Kg/m^{2}s

Frictional pressure drop:

$$\Delta P_f = \frac{f n_p L G^2}{2\rho D_i s \varnothing}$$

= 21257.46 Pa = 0.2167 Kg/cm^2

Return loss ΔP_{rt} (due to change in flow direction of the tube side fluid):

$$\Delta p_{rt} = 1.334 * 10^{-13}(2n_p - 1.5)\frac{G_t^2}{S_a} \quad (30)$$

= 0.000574 psi = 0.00004 Kg/cm^2

Total tube side drop neglecting nozzle loss = 0.21674 Kg/cm^2 < 0.35 Kg/cm^2

Therefore the tube side pressure drop is within the maximum allowable pressure drop.

3.9.1 Shell side pressure drop

No. of baffles = n_b = 18

Friction factor f = 0.0022 x 144 = 0.3168 ft2/ ft2 with 25% cut segmental baffles (**[5] page 839**).

$$\Delta P_f = \frac{f D_s G_s^{\,2}(n_b + 1)}{2\rho D_e s_w \varnothing}$$

= 35860.89 Pa = 0.365 Kg/cm^2

ΔP_{rs} =0 (in case of single shell pass flow)

Total shell side drop neglecting nozzle loss = 0.365 Kg/cm^2<0.5 Kg/cm^2

Therefore the shell side pressure drop is also within the maximum allowable pressure drop.

3.10 Over Surface and Over Design

Over surface = $\frac{U_C - U_{o,cal}}{U_C}$

The clean overall heat transfer coefficient: $U_C = \frac{h_o * h_{io}}{h_o + h_{io}}$

$h_{io} = h_i * \frac{d_i}{d_o}$ = 501.53 W/m^{2}K

U_c = 467.96 W/m^{2}K

% Over surface = 47.585 %

% Over Design = $\frac{A - A_{reqd}}{A_{reqd}} * 100$

The design area of heat transfer (where n_t=104):

$A = \pi d_o L_t n_t$ = 5.58 m^2

The required heat transfer area (where n_t =100)

$A_{reqd} = \pi d_o L_t n_t$ = 5.37 m^2

%Over Design = 3.91 %

4. Calculation of Effectiveness

The intercooler effectiveness ε is defined as

$$\varepsilon = \frac{Q}{Q_{\max}} \quad (31)$$

where Q is the actual rate of heat transfer from the hot to cold fluid and Q_{max} is the maximum possible rate of heat transfer for given inlet temperatures of the fluids.

$$Q_{\max} = C_{\min}(T_{h,i} - T_{c,i}) \quad (32)$$

Here, C_{min} is the smaller of the two heat capacity rates

For given conditions, we have:

Q =5.06 * 10^4 Kcal/hr

$T_{h,i}$ = 160°C

$T_{c,i}$ = 32°C

$C_{min} = m_a c_a =$ 642.838 Kcal/hr*k

Q_{max} = 8.22 * 10^4 Kcal/hr

Therefore, ε = 0.6155

5. Conclusion

After the above discussion, it is easy to say that the shell & tube type intercoolers to be used for multi-stage compressors have been given a great respect among all the classes due to their virtues like comparatively large ratios of heat transfer area to volume and weight and much more. Moreover well designed as well as described methods are available for its designing and analysis. The literature survey also shows the importance of this class of heat exchangers. Further, it may be seen that a lot of factors affect the performance of the intercooler and the analysis obtained by the formulas depicts the cumulative effect of all the factors over the performance of the intercooler.

6. Future Scope

Currently, a plain tube intercooler has been designed to meet the requirement. An enhanced surface such as finned tube intercooler may also be designed thereby increasing the heat transfer area as well as the overall rate at the same time reducing the total number if tubes required. This, therefore, reduces the overall size and volume of the equipment and hence the cost.

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Author Profile

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