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Parametric Study of Various Coolants by Using Shell and Tube Heat Exchanger

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ABSTRACT

In power plants, the heat generated at the Turbine/Alternator bearings is to be dissipated continuously in order to maintain the lube oil properties constant. This is critical in nature, as any variation in the designed lube oil temperature will affect the oil film thickness between turbine shaft and bearings resulting in mal-operations of turbine rotor.

The thermal analysis of shell and tube Heat Exchanger is desired with the bell manual method and for the same the numerical Analysis have been carried out based on the prescribed pressure drop criteria. The thermal analysis of shell and tube Heat Exchanger and performance of evaluation is presently established technique used in power plant industry. In this project the thermal analysis on various tube fluids flow pressure drop variations for shell and tube heat Exchanger are obtained. The pressure drop values for shell and tube heat Exchanger are obtained by using "MAT LAB" and compared with Bell Manual Method values.

INTRODUCTIONTOHEAT EXCHANGER:

Heat exchanger is a device used for affecting the process of heat exchange between two fluids that are at different

Temperatures. Heat exchangers are use full in many engineering processes like those in refrigerating and air conditioning systems, power plants, food processing industries,
chemical reactors and space and aeronautical applications

A Heat Exchanger in which two fluids exchange heat by coming in direct contact is called a direct heat exchanger. Examples of this type are open feed water heaters and jet condensers. Recuperators (closed type exchangers) are heat exchangers in which fluids are separated by a wall. The wall me be a simple plane wall or a tube or a complex configuration involving fins, baffles and multi-pass of tubes.

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CLASSIFICATIONOFHEAT EXCHANGERS:

Three basic types of heat exchangers are:

1.Closed type heatexchangers(or) Recuperates

2. Regenerators

3. Open type exchanger (or) mixed type

1. Closed type heat exchangers:

These exchangers are those in which heat transfer occurs between two fluids that do not mix and there is no physical contact between the two fluids. A pipe or tube wall or any other surface, which may be involved in heat transfer path, separates the fluid streams. Heat transfer thus occurs by convection from the hotter fluid to the solid surface and then to the cooler fluid. Most of the heat exchangers come under this category

2. Regenerators:

These exchangers are those in which the hot and cold fluids flow through the same space alternatively with a little mixing occurring between the two streams. In such heat exchangers the surface, which receives thermal energy and then releases thermal energy is important. Surface materials properties as well as fluid flow properties of the fluid stream along with the geometry are the main parameters, which must be known. The analysis needs knowledge of unsteady state of conduction and convection. In steam power plants the air preheaters are usually rotor regenerator type.

3. Open type exchangers:

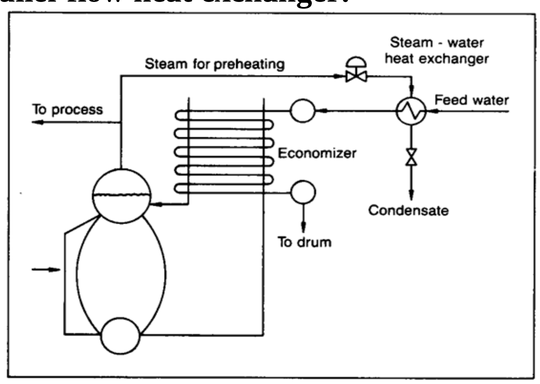
These exchangers are devices wherein the entering fluid stream flows into the open chambers and complete mixing of the two streams occurs. Hot and cold fluid streams entering such an exchanger leave as a single stream. Analysis of open type involves the law of conservation of mass and law of thermodynamics. Jet condensers used for cooling the water circulated through the condensers in power plants come under this category.

CLASSIFICATION BASED ON FLUID FLOW ARRANGEMENT:

On the basis of fluid paths, heat exchangers are classified as follows

1. Parallel flow heat exchangers
2. Counter flow heat exchangers
3. Cross-flow heat exchangers

1. Parallel-flow heat exchanger:



In parallel flow heat exchanger the two fluids enter the exchanger at the same end, and travel in parallel to one another to the other side.

2. Counter-flow heat exchanger:

In counter flow heat exchanger the fluids enter the exchanger from opposite ends. The counter current design can transfer most heat from the heat (transfer) medium because the counter current design is most efficient.

3. Cross flow heat exchanger:

In a cross-flow heat exchanger, the fluids travel roughly perpendicular to one another through the exchanger.

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For efficiency, heat exchangers are designed to maximize the surface area of the wall between the two fluids, while minimizing resistance to fluid flow through the exchanger. The exchanger's performance can also be affected by using fins or corrugations in one or both directions, which increase surface area and may channel fluid flow or induce turbulence. The driving temperature across the heat transfer surface varies with position, but an appropriate mean temperature can be defined. In most simple systems this is the "log mean temperature difference"

(LMTD). In some conditions the direct knowledge of the LMTD is not available, so at that time the NTU method is used.

GENERAL FEATURES OF HEAT EXCHANGERS APPLICABLE CODES AND STANDARDS:

TEMA:

TEMA stands for Tubular Heat Exchanger Manufacturers Association. This is a design standard which gives the basic details of heat exchangers used in different power, petrochemical and refinery operations. This is classified in to TEMA 'R', 'C', AND 'B' section based on utility and functional aspects. Each section deals with baffles, tubes, and tube holes in baffle or tube sheets, tie rods and other constructions features of tubular exchangers.

1.9.2. ASME SECTION VIII DIV 1

ASME stands for American Society of Mechanical Engineers. This deals with the mechanical design, fabrication, inspection, testing of pressure vessels and heat exchangers. Design information includes calculation of shell, channel thickness for the internal and external pressure, reinforcement requirements, post weld heat transfer requirement etc.

Design fouling resistances:

The purchaser should attempt to select an optimal fouling resistance that will result in a minimum sum of fixed, shutdown and cleaning costs. The following tabulated values of the fouling resistances allow for over sizing the heat exchanger so that it will meet performance requirements with reasonable intervals between shutdowns cleaning.

These values do not recognize the time related behavior of fouling with regard to specific design and operational characteristic of particular heat exchangers.

Usually the scale is of an unknown or complicated composition and the fouling factor must be known as a result of analyzing the performance of fouled exchangers. If the performance tests are made on a clean exchanger and repeated later after the unit has been in service for some time, the thermal resistance of the deposit can determined from the relation.

The thermal resistance of the deposition $R_f = 1/U_1 - 1/U_2$

Where

R_f = Thermal resistance or Fouling resistance

U_1 = Unit conductance of the clean surface

U_2 = Unit conductance after fouling of surface

Thermal analysis calculations:

The starting point of any heat transfer calculation is the overall energy balance and the rate equation. Assuming only sensible heat is transferred, we can write the heat transfer Q

- Mass of steam flowing through the heat exchanger in kg/sec

= 19.74 Tons/hr.

= $(19.74 \times 1000)/3600$

= 5.483 kg/sec

- Specific heat of the steam in J/kg

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K = 2125.7 J/kg K

- Inlet temperature of the steam= 485 °C = 758 K

– Outlet temperature of the steam

- Mass of water flowing through the heat exchanger in kg/sec

= 17.635 Tons/hr.

= (17.635 x1000)/3600

= 4.8986 kg/sec

Specific heat of the water in J/kg

K = 4297.75 J/kg K

Inlet temperature of the water= 120 °C =393 K Outlet temperature of the water = 145 °C = 418 K

Heat transfer rate:

Determining the outlet temperature of steam from Energy balance equation

Q = 5.483 x 2125.7 x (758 –)

= 4.8986 × 4297.75 × (418 – 393)

5.483 × 2125.7 × (758 –)

= 526323.95 W

= 712.842 K 713 K = 440°C

Q = UAF

Where ,

U – Overall heat transfer coefficient in W/m² K

A – Heat transfer surface area in m²

The new symbol, F stands for a correction factor that must be used with the log mean temperature difference, for a countercurrent heat exchanger to accommodate the fact that the flow of the two streams here is more complicated than simple countercurrent or concurrent flow.

Logarithmic mean temperature difference:

The convention in shell-and-tube heat exchangers is as follows:

T_1= Inlet temperature of the shell-side (or hot) fluid i.e., steam =485°C

T_2= exit temperature of the shell-side (or hot) fluid =440°C

= Inlet temperature of the tube-side (or cold) fluid i.e., water =120°C

= exit temperature of the tube-side (or cold) fluid =145°C

Tube side heat transfer coefficient:

The inside heat transfer coefficient, can be evaluated using the standard approach for predicting heat transfer in flow through tubes. Typically, turbulent flow can be expected, and a good design would aim to arrange for turbulent flow, because of the substantial enhancement in heat transfer provided by eddy transport.

Properties to be evaluated at BULK MEAN temperature for the internal flow through the tubes

Property values of water at bulk mean temperature

PROPERTY	VALUE
Density, (kg/m³)	923.25
Kinematic viscosity, (m²/s)	0.22575
Thermal conductivity, k (W/m K)	0.68425
Prandtl number, Pr	1.318

Mass flow rate through the tubes

M=AV

Where

- Mass of water flowing in kg/sec=4.8986 kg/sec

-density of water at temperature, in kg/m³

A – Cross sectional area of the tubes, in m²

A = number of tubes cross sectional area of each tub

A = 61 = 0.0431 m²

V – Velocity of water, in m/s

From above eqn

4.8986 = 923.25 0.0431V

V = 0.123 m/s

Calculating Reynolds number, based on diameter

The **Dittus-Boelter Equation** is applicable to both liquids and gases with Reynolds Number Re>10000, Prandtl Number 0.7<Pr<160 and L/D>10 ie suitable for applications with shorter tube lengths.

Dittus-Boelter Equation is given by

Nu = 0.023

n = 0.4 for heating of fluids

n = 0.3 for cooling of fluids

Hence we consider the value of n = 0.4, since the water is heated in the tubes.

From eqn (3.11)

Therefore, Nu = 0.023

Nu = 0.023

Nu = 60.183

Also,

i.e., = 1373.58 W/m² K

Shell side heat transfer coefficient,

Pitch dimensions (as stated in article 5.3):

Pitch transverse the flow, = 100 mm

Pitch along the flow, = 100 mm

Predicting the shell-side heat transfer coefficient is more involved, because the flow passage is not simple, even in the absence of baffles. Heat transfer correlations for flow through tube banks assume flow normal to the long axes of a set of tubes.

Properties to be evaluated at BULK MEAN temperature for the internal flow through the tubes

Property values of steam at bulk mean temperature

PROPERTY	VALUE
Density, (kg/m³)	19.2984
coefficient of viscosity, (Ns/m²)	27.02
Thermal conductivity, k (W/m K)	0.0637
Prandtl number, Pr	0.9

Overall heat transfer coefficient,

Finally, calculating the overall heat transfer coefficient, based on the outer area of the pipe

Where is the heat transfer coefficient for the fluid flowing in the shell, is the heat transfer coefficient for the fluid flowing through the tubes, And are the inside and outside radii of the tube, respectively, is the fouling resistance for the shell side is the fouling resistance for the tube side is the thermal conductivity of the tube material

Values of = 420.732 W/ m² K

= 1373.58 W/ m² K

= 0.01905 m

= 0.01499 m

= 39.8 W/m² K

Fouling resistances (from table 5 in appendix)

For the shell side, = 0.0000877 m² K / W

For the tube side, = 0.0001754 m² K / W

On substituting the values in the above equation we obtain

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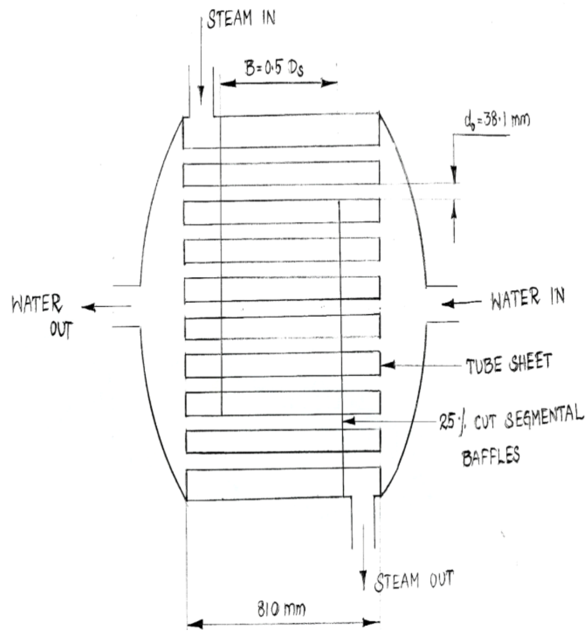

shell and tube heat exchanger cross section

Pressure drop calculations:

In designing heat exchangers, pressure drop considerations are usually quite important. There are quite a few correlations and charts are available for the calculation of pressure drops over the tube and shell sides of a heat exchanger. One method is described below:

Tube-side pressure drop:

Calculation of the tube-side pressure drop is made by first estimating the (Darcy) friction factor for flow through the tubes from the value of the Reynolds number and the relative roughness, and applying the viscosity correction. Then, this friction factor is used to evaluate the pressure drop for flow through the tubes from

$\Delta P = f\ L/d_i(1/2\rho V^2)$number of tube passes (3.21)

Where L –The length of the tubes= 0.81 m

d_i – Inner Diameter of the tubes = 0.02998 m

ρ_t – Density of the tube-side fluid = 923.25 kg/m³

V – Average flow velocity through a single tube = 0.123 m/s

f – friction factor, given by (also obtained from figure 8 in appendix)

f = 0.184 = 0.0264

Here, we are considering only one tube pass. Therefore, number of tube passes is 1.

From above equation

$\Delta P = 0.02641$

$\Delta P = 4.98$ N/m²

Shell-side pressure drop:

a shell with segmental baffles, the following equation may be used

$= f_s G_s 2 D_s (N_B+1)/2g\rho_s D_e \Phi_s$

Where

P = Pressure drop, kg/m^2

f_s = friction factor, given by (also obtained from figure 9 in appendix)

For staggered arrangement,

$f_s = [0.25+(0.118/(p-d_0/d_0)]\ Re^{-0.16}$

pt- Pitch along the flow = 100 mm

d_0- outer diameter of tube = 38.1 mm

Re – Reynolds number for the flow on shell side = From equation

f_s = = 0.0477

G_s = Shell-side fluid mass velocity = 15.869kg/m² s

D_s= diameter of shell = 1 m

N_B= number of baffles = 2

g = acceleration = 9.8 m/s2

ρ_s = density of the shell fluid =19.2984 kg/m³

D_e= hydraulic diameter of the shell=0.194m

Φ_s = dimensionless viscosity ratio for the shell-side fluid

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$\Phi_s = (\mu_{b/}\ \mu_w)^{0.14}$

μ_b=coefficient of viscosity of steam at bulk mean temperature

= 27.02 N s/m^2

μ_w=coefficient of viscosity of steam at wall temperature= 12.746 N s/m^2

From equation (3.25)

Φ_s = = 1.11

= 0.442 kg/ m^2

P = 4.336 N/

3.4. PROGRAM OUTPUTS:

ENTER THE VALUE OF HEAT DUTY(Q) :: 526323.95
ENTER THE VALUE (Tho) :: 713
ENTER THE VALUE (Tci) :: 393
ENTER THE VALUE SPECIFIC HEAT OF COOLENT (Cpc) :: 4297.75
ENTER THE VALUE (Cps) :: 2125.7
ENTER THE DENSITY OF COOLENT :: 923.25
ENTER THE FLOW RATE OF COOLENT :: 0.08
ENTER THE VISCOSITY OF COOLENT :: 0.00000022575
ENTER THE VISCOSITY OF WATER INSIDE :: 0.00085
ENTER THE THERMAL CONDUCTIVITY OF COOLENT :: 0.68425
ENTER THE FOULLING FACTOR OF COOLENT :: 0.0005
ENTER THE DENSITY OF STEAM:: 19.2984
ENTER THE FLOW RATE OF STEAM:: 5.483
ENTER THE FOULLING FACTOR OF STEAM:: 0.003
ENTER THE VALUE do:: 0.0381
ENTER THE VALUE di:: 0.02998
ENTER THE VALUE OF Pr1 :: 1.318
ENTER THE VALUE OF new :: 0.00000022575
ENTER THE VALUE OF np :: 1
ENTER THE LENGTH L:: 0.81
K1 VALUE IS
K1 =0.7308
Bi VALUE IS
Bi = 1.0001
DP1 VALUE IS
DP1 = 9.5905e-012
TUBE SIDE PRESSURE DROP IS
Tpd = 4.9883
SHELL SIDE PRESSURE DROP IS
Tps =0.4420
DTm VALUE IS
DTm =320.3388
F VALUE IS
F =1.0000
Ht VALUE IS
Ht =1.3758e+003
Hs VALUE IS
Hs =708.3070
Uf VALUE IS
Uf =162.4226
Tpd VALUE IS
Tpd =4.9883
Tps VALUE IS
Tps = 0.4420

CONCLUSIONS

The thermal and pressure drop calculations for a given heat exchanger are carried out manually the calculated results are compared with water and ammonia.

Ammonia is to be preferred in tube side because calculated pressure drop is less than allowable pressure drop

If design is based on pressure drop criteria ammonia is to be used in the tube side of heat exchanger due to more heat transfer coefficient and less pressure drop compared to water.

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