

DESIGN AND ANALYSIS OF HIGH PRESSURE FEEDWATER HEATER

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Abstract— Most of the electricity generates throughout the world is from steam power plants. The plant efficiency can be improved by regeneration which is done by using feedwater heater. It is a shell and tube heat exchanger used for waste heat recovery from steam power plant. This paper represents mathematical model of high pressure feedwater heater. The heater is divided into three zones: desuperheating, condensing and, subcooling for the modeling purpose. The same input parameters are given to the Aspen FRAN software and results are obtained. It is found that the analytical results are close compared to results given by software.

Keywords— Feedwater heaters, Aspen FRAN, heat transfer zones.

1. INTRODUCTION

Today most of the electricity generates through the world is from steam power plants and the ideal cycle for vapor power plants is the Rankine cycle. In simple Rankine cycle, steam is used as working fluid generated from saturated liquid water. This saturated steam flows through the turbine, where its internal energy is converted into mechanical work to run an electricity generating system. The Rankine cycle efficiency can be improved by regeneration, which uses sensible heat of small amount of steam taken from high pressure turbine. Basically this enables internal transfer of high temperature steam to the low temperature condensate (feed water). Regeneration is accomplished in all large scale modern power plants through the use of feedwater heaters. Figure (1) shows the Schematic of feedwater heater

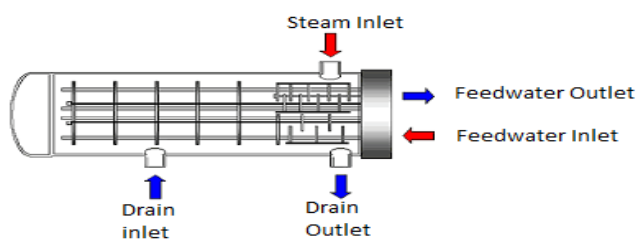


Fig.1.Schematic of feedwater heater

A feedwater heater is a heat exchanger designed to preheat boiler feedwater by means of condensing steam bled from a steam turbine. An open feedwater heater is basically a mixing chamber, where the steam extracted or bled from the turbine mixes with the feedwater exiting the pump. Another type is closed feedwater heater, in which heat is transferred from the bled steam to the feedwater without any mixing taking place. The heating process by means of extraction steam from turbine is referred to as

being regenerative. The feedwater heaters are integral portions of the powerplant thermodynamic cycle. Normally, there are multiple stages of feedwater heating. Feedwater heaters allow the working fluid to be brought up to the required temperature gradually. This minimizes the inevitable irreversibility associated with heat transfer and hence improves the thermodynamic efficiency of a turbine cycle. This decreases operating costs and also helps to avoid thermal shock to the material of the boilers. The greater the number of extraction stages, the lower the amount of thermal energy required to heat up the feedwater. A beneficial by-product of the energy extracted by the feedwater heaters is the reduced rate of rejection of energy to the environment.

The feedwater heaters mainly reduce the required heat supplied to the plant by utilizing the waste heat leaving the plant. It also increases the overall plant cycle efficiency and lowers the size of boilers. Most of heaters are of a standard shell and tube configuration, it consists of a bundle of parallel tubes inside a shell. One fluid flows through a number of tubes, while the other fluid with different temperature flows over the shell but outside the tubes to transfer heat through the tube walls. The two fluids are separated by the walls of the tube so that they never mix. Due to this design, a large number of tubes are used to enlarge the heat transfer area and transfer heat efficiently. In a word, it is an efficient way to use waste heat in order to conserve energy.

So the main objective of this work is to set a proven design methodology to design high pressure feedwater heater. The design of shell and tube configured feedwater heater requires the knowledge of a number of factors on both shell side and tube side. An exhaustive list of these items and existing standards to select these are well documented.

NOMENCLATURE:

A	area, m ²
C _p	specific heat, kJ/kg.K
d	tube diameter, m
D _s	shell inner diameter, m
g	gravitational force, m/s ²
h	heat transfer coefficient, W/m ² K
h _{fg}	latent heat of vaporization, kJ/kg
k	thermal conductivity, W/m.K
L	length of the tube, m
m	mass flow rate, kg/sec
Q	heat transfer rate, W
R _f	resistance, m ² . K/W
t _h	temperature of hot fluid, °C
t _c	temperature of cold fluid, °C
t _{sat}	saturation temperature, °C
t _{wall}	wall temperature, °C
t _{film}	film temperature, °C
u	velocity, m/sec
μ	dynamic viscosity, kg/m.sec
ρ	density, kg/m ³

Dimensionless:

Hg	Hagen number
Lq	Leveque number
N _u	Nusselt number
N _t	number of tubes
Pr	Prandtl number
Re	Reynolds number
X _d *	ratio of the diagonal pitch to the tube outside diameter
X _l *	ratio of the longitudinal pitch to the tube outside diameter
X _t *	ratio of the transverse pitch to the tube outside diameter

Subscripts:

c	cold fluid (water)
h	hot fluid (steam)

i	inlet
o	outlet
s	shell side
t	tube side

2. PROBLEM DEFINITION

Table I gives process design data and Table II gives geometric design data required of feedwater heater. With consideration of these process and geometric parameters, the feedwater heater is designed.

TABLE I. PROCESS DESIGN DATA

Sr. No.	Parameters	Unit	Value
1	Tube side fluid flow rate	kg/sec	26.66
4	Shell side fluid flow rate	kg/sec	3.12
3	Tube side fluid inlet temperature	°C	115
4	Tube side fluid outlet temperature	°C	187
5	Shell side fluid inlet temperature	°C	343
6	Shell side fluid outlet temperature	°C	125
7	Tube side fluid inlet pressure	bar	16.7
8	Shell side fluid inlet pressure	bar	117.6
9	Tube side fluid inlet velocity	m/sec	1.6

TABLE II. GEOMETRIC DESIGN DATA

Sr. No.	Parameters	Unit	Value
1	Shell material	SA 516 GR 70	
2	Tube material	SA 556 GR A2	
3	Tube outside diameter	m	0.0158
4	Tube wall thickness	m	0.0016
5	Tube inside diameter	m	0.0126
6	Tube pitch	m	0.021
7	Tube layout	Triangular (60°)	

3. DESIGN PROCEDURE

Design is an activity aimed at providing complete description of the feedwater heater as a component. This description can be accomplished using a well-defined design methodology. The thermal design is directed to calculate an adequate surface area to handle the thermal duty for the given specification while the hydraulic analysis determines the pressure drop of the fluids flowing in the system.

A. Mathematical model

Mathematical model helps to determine unmeasured parameters of the heater. The first attempts to provide methods for calculating pressure drop and heat transfer coefficient is the well known Kern method [5].

Now considering the desuperheating zone, condensing zone, and the subcooling zone of feedwater heater separately,

1) *Desuperheating zone (DS)*: In the desuperheating section, the superheated steam is cooled in a contact with dry tubes of the U-tube bundle to the saturation temperature. The heat transfer is written as,

$$Q_I = m_s C_{ps} (t_{hi} - t_{ho}) = m_t C_{pt} (t_{ci} - t_{co}) \quad (1)$$

The properties of tube side and shell side fluids are calculated at the arithmetic mean of fluid temperatures at the inlet and outlet of zone.

Logarithmic mean temperature difference,

$$LMTD = \frac{(t_{hi}-t_{co})-(t_{ho}-t_{ci})}{\ln \frac{(t_{hi}-t_{co})}{(t_{ho}-t_{ci})}} \quad (2)$$

a) Thermal analysis for tube-side:

Number of tubes:

The flow rate inside the tube (m_t) is a function of the density of the fluid (ρ_t), the velocity of the fluid (u), cross-sectional flow area of the tube (A_c), and the number of tubes (N_t).

$$m_t = \rho_t \cdot u_t \cdot A_c \cdot N_t \quad (3)$$

Tube side Reynolds number:

$$Re = \frac{\rho_t u_t d_i}{\mu_t} \quad (4)$$

Tube side Nusselt number:

The Nusselt number is a function of Reynolds number (Re) and Prandtl number (Pr).

$$Nu_t = \frac{(f/2) Re_t Pr_t}{1.07+12.7(f/2)^{1/2}(Pr_t^{2/3}-1)} \quad (5)$$

Where f is the friction factor which can be calculated from

$$f = (1.58 \ln Re - 3.28)^{-2} \quad (6)$$

Tube side heat transfer coefficient is given as,

$$h_t = Nu \frac{k_t}{d_i} \quad (7)$$

b) Thermal Analysis for Shell-Side:

Shell side heat transfer coefficient:

Martin correlation of Hagen number for flow normal to staggered plain tube bundles is given as [6],

$$Hg = Hg_{lam} + Hg_{turb} [1 - \exp(1 - \frac{Re_s + 200}{1000})] \quad (8)$$

Where,

$$Hg_{lam} = 140 Re_s \frac{(X_t^{*0.5} - 0.6)^2 + 0.75}{X_t^{*1.6} (\frac{4X_t^* X_t^*}{\pi} - 1)} \quad (9)$$

$$Hg_{turb} = \left\{ \left[1.25 + \frac{0.6}{(X_t^* - 0.85)^{1.08}} \right] + 0.2 \left(\frac{X_t^*}{X_t^*} - 1 \right)^3 - 0.005 \left(\frac{X_t^*}{X_t^*} - 1 \right)^3 \right\} Re_s^{1.75} \quad (10)$$

Leveque Number is expressed as,

$$L_q = 0.92 Hg \cdot Pr \left(\frac{(4X_t^* X_t^* / \pi) - 1}{X_t^* X_d^*} \right) \quad (11)$$

Nusselt Number can be calculated with help of Leveque number as follows,

$$Nu = 0.404 * L_q^{1/3} \quad (12)$$

The heat transfer coefficient for the shell-side is expressed as follows:

$$h_s = Nu \cdot \frac{k_s}{d_o} \quad (13)$$

Overall heat transfer coefficient:

The overall heat transfer coefficient is given by

$$U_o = \frac{1}{\frac{1}{h_s} + R_{f_o} + \frac{r_o}{k_t} \ln \frac{r_o}{r_i} + \frac{r_o}{r_i} R_{f_i} + \frac{r_o}{r_i} \frac{1}{h_t}} \quad (14)$$

Now total heating surface area can be find out from,

$$Q_I = U_I \cdot A_I \cdot \Delta T_{mI} \quad (15)$$

Length of tube in this zone can be calculated using,

$$L_I = \frac{A_I}{\pi d_o N_t} \quad (16)$$

Shell Diameter can be expressed as, [5]

$$D_s = 0.637 \sqrt{\frac{CL}{CTP} \left(\frac{A_o PR^2 d_o}{L_I} \right)^2} \quad (17)$$

Where,

CTP is tube count calculation constant, PR is tube pitch ratio given as $\frac{P_t}{d_o}$, CL is tube layout constant, and A_o is outside heat transfer area and it can be calculate from $\pi d_o N_t L$.

2) Condensing zone (CD): In the condensing zone, the steam condenses in a contact with tubes where it changes its phase from saturated vapor to saturated liquid and releases the latent heat of vaporization.

Heat transfer in condensing zone is latent heat exchange,

$$Q_{II} = m_s \cdot h_{fg} = m_t C_{pt} (t_{ci} - t_{co}) \quad (18)$$

The driving force for condensation is the difference between the temperature of cold wall surface and the bulk temperature of the saturated vapor.

$$\Delta T_{Driving} = T_{sat} - T_{wall} \quad (19)$$

The viscosity and other properties used in the condensing correlations are evaluated at the film temperature, a weighted mean of the cold surface (wall) temperature and the vapor saturation temperature.

$$T_{film} = T_{sat} - \frac{3}{4}(T_{sat} - T_{wall}) \quad (20)$$

Where,
$$T_{wall} = \frac{T_{sat} + \frac{t_{co} + t_{ci}}{z}}{2} \quad (21)$$

Now the heat transfer coefficient for condensation is given as, [7]

$$h_{cond} = 0.725 \left\{ \frac{k_f^3 \cdot \rho_f \cdot (\rho_f - \rho_v) \cdot g \cdot h_{fg}}{\mu_f (T_{sat} - T_{wall}) d_o} \right\} \quad (22)$$

The tube side heat transfer coefficient h_t and overall heat transfer coefficient U_{II} can be calculate similar to the desuperheating zone.

The heat transfer is written as,

$$Q_{II} = U_{II} \cdot A_{II} \cdot \Delta T_{mII} \quad (23)$$

Length of tube in condensing zone can be calculated using, [5]

$$L_{II} = \frac{A_{II}}{\pi d_o N_t} \quad (24)$$

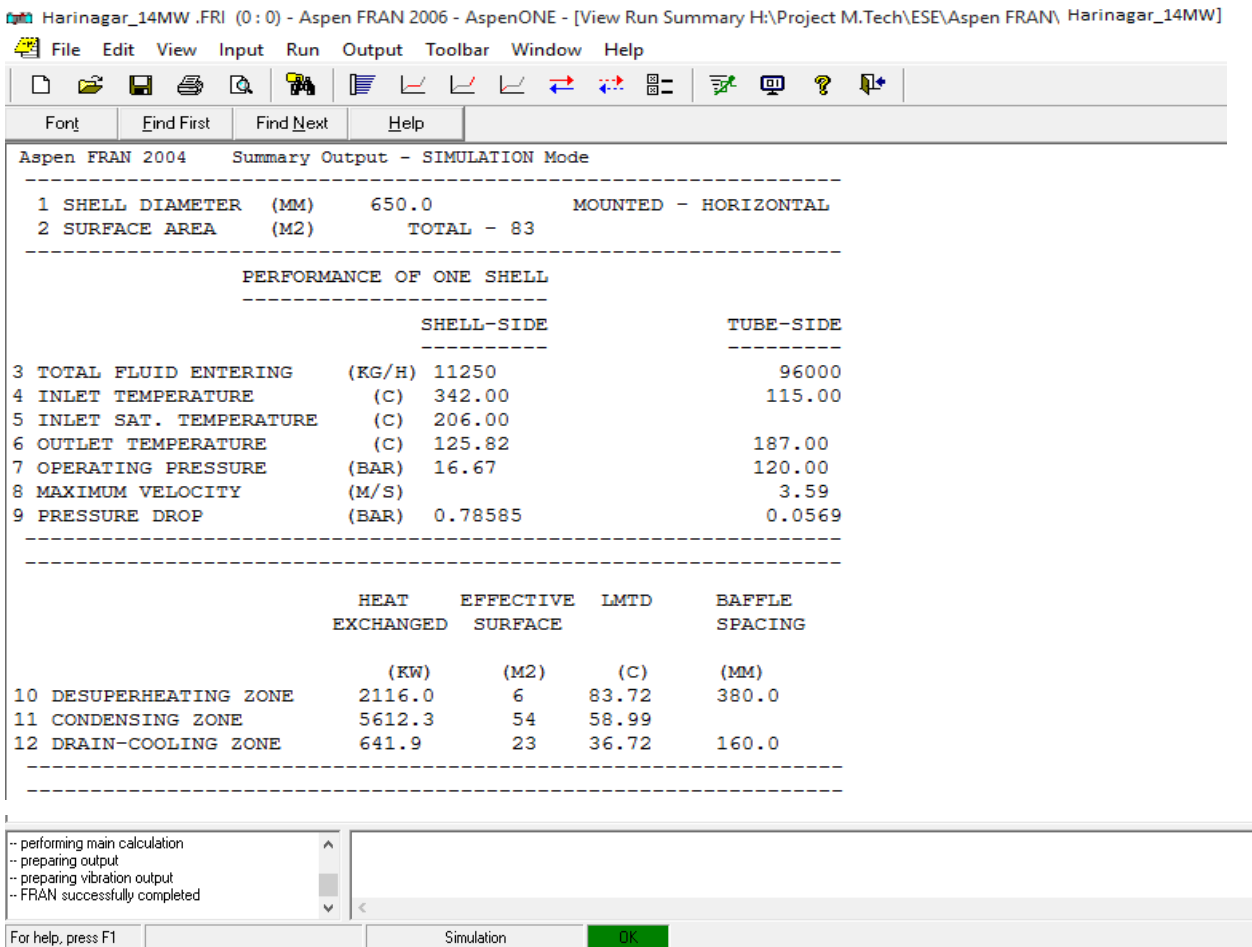


Fig.2. Results obtained by Aspen FRAN software

3) *Subcooling zone (SC)*: In the subcooling zone, the tube side and shell side fluids are liquids i.e. single phase and the heat exchange occurs in form of sensible heat transfer. The values of heat transferred, LMTD, overall

heat transfer coefficient, area of zone and length of tube can be calculate using same equations as that of the desuperheating zone.

The same input conditions shown in TABLE I and TABLE II are given to the Aspen FRAN software and the high pressure feed water heater is designed. The results obtained by analytical calculation and software are compared. The figure (2) shows results obtained by Aspen FRAN software.

4. RESULTS AND DISCUSSION

A) Temperature profile of fluid along the length of feedwater heater

Figure (3) shows the temperature variation for both shell and tube side along the length of heater. The superheated steam shows a large drop in temperature in the desuperheating zone. The horizontal line in the condensing zone indicates constant temperature and finally a gradual decrease in temperature in the subcooling zone. While the temperature of tube side fluid rises gradually along the length of the tube.

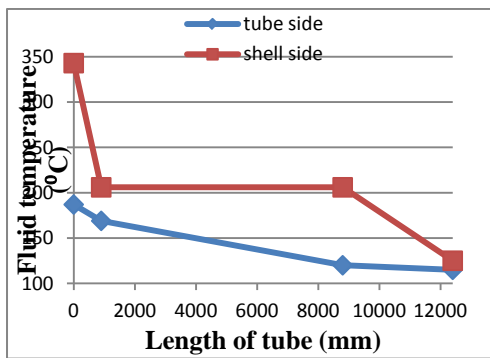


Fig.3. Fluid temperature variations along the length of heater

B) Three zones of feedwater heater:

In feedwater heater, the heat is transferred from hot fluid on shell side to cold fluid on tube side. The major heat is transferred in the condensing zone due to latent heat transfer while sensible heat transfers in desuperheating and subcooling zone.

Figure (4) gives areas of three zones of heater. It shows that condensing zone accounts for maximum area than other two zones.

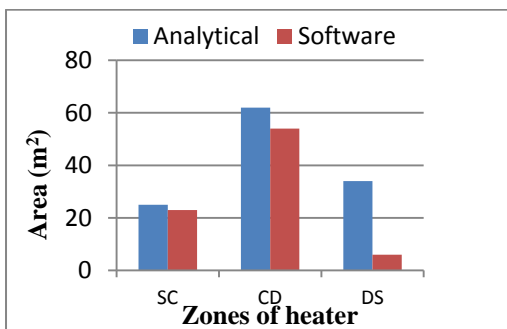


Fig.4. Area of three zones of feedwater heater

Variation in analytical and software results is seen due to change in velocity of fluid on shell side

changes along the heater because of phase change. The variation is also seen due to change in properties of fluid with change in properties of fluid with change in temperature cannot be considered in analytical method.

5. CONCLUSIONS

Sizing of high pressure feedwater heater is done analytically and by using Aspen FRAN software to perform required heat duty.

The following conclusions are drawn,

- The maximum amount of heat duty is handled in the condensing zone of feedwater heater.
- Shell side heat transfer coefficient has a large influence on overall heat transfer coefficient and hence area of feedwater heater.
- Fouling factor affects in calculation of surface area to a large extent.
- Condensing zone accounts for maximum amount of area of feedwater heater than other two zones.
- Pressure drop on shell side is maximum in subcooling zone as compared to other two zones. While pressure drop on tube side of feedwater heater is negligible.

ACKNOWLEDGMENT

The authors acknowledge the support facility provided by “Walchandnagar Industries Limited” and also thankful for providing the valuable guidance and encouragement to carry out the work.

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