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Research Article

Thermal Design of Shell & Tube Heat Exchanger for Concentrating Solar Power Application

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Abstract

In the first part of this paper, a simplified approach to Theoretical Thermal Design of Shell & Tube Heat Exchanger for Concentrating Solar Power Application. The thermal design of STHE involves evaluation of required effective surface area (i.e. number of tubes) and finding out log mean temperature difference [LMTD]. The design was carried out by referring ASME/TEMA standards, available at the company. In the second part of this paper, theoretically (analytical) thermal design of STHE compared with the Software (i.e. HTFS) basis design. Theoretical thermal design validate on the basis of HTFS.

Keywords*: Heat Exchanger; LMTD, TEMA, HTFS.*

1. Introduction

Solar thermal power plants produce electricity in much the same way as conventional power stations. The difference is that they obtain their energy input by concentrating solar radiation and converting it to high temperature steam or gas to drive a turbine or engine. Four main elements are required: a concentrator, a receiver, some form of heat transport media or storage, heat exchanger and power conversion. Many different types of systems are possible, including combinations with other renewable and non-renewable technologies.

Heat Exchangers are devices used to enhance or facilitate the flow of heat. Every living thing is equipped in some way or another with heat exchangers. They are widely used in space heating, refrigeration, air conditioning, power plants, chemical plants, petrochemical plants, petroleum refineries, natural gas processing, and sewage treatment. The design of STHE including thermodynamic and fluid dynamic design, cost estimation and optimiza- tion, represents a complex process containing an inte- grated whole of design rules and empirical knowledge of various fields.

Shell and tube heat exchangers are designed using commercially available software such as those developed by co-operative research organizations such as Heat Transfer and Fluid Flow Service (HTFS) and Heat Transfer Research Inc. (HTRI) and by computer service companies such as B-JAC International. These programs offer design and cost analysis for all primary heat exchanger types and incorporate multiple design codes and standards from the American Society of Mechanical Engineers (ASME),Tubular Exchangers Manufacturers Association (TEMA) and the International Standards Organisation (ISO).

2. Problem Definition

The most common problems in heat exchanger design are rating and sizing. The rating problem is evaluating the thermo-hydraulic performance of a fully specified exchanger. The sizing problem, however, is concerned with the determination of the dimensions of the heat exchanger.

Table I gives process design data and Table II gives geometric design data required of shell and tube heat exchanger. With considration of above process and geometric prameters, the heat exchanger is designed which gives required tube side fluid outlet temperature.

Table 1 Process Design Data

Parameter	Unit	Tube Side	Shell Side
Medium		Steam	Dowthe-rm A
Mass Flow	kg/s	36.07	313.8
Temperature inlet	^oC	315.1	393.3
Temperature outlet	^oC	386.1	378.7
Oprating pressure	bar	17.2	105.7
Velocity	m/s	4.506	1.1
Fouling resistance	$m^2.K/W$	0.000088	0.000176
Allowable pressure drop	bar	1	1

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Table 2 Geometric Design Data

Parameter	Unit	Tube Side	Shell Side
Inside diameter(d_i)	mm	19.5	2000
Outside diameter(d_o)	mm	25.4	2010
Length of tube(L)	mm	7300	
Number of passes(N_p)		2	
Baffle cut		20%	
Tube layout		Squre(90°)	
Tube Pitch	mm	50	

3. Theoretical Thermal Design Method

The thermal design of heat exchangers is directed to calculate an adequate surface area to handle the thermal duty for the given specifications whereas the hydraulic analysis determines the pressure drop of the fluids flowing in the system, and consequently the pumping power or fan work input necessary to maintain the flow.

Petukhov-Popov and Bell-Delaware methods will be presented by performing Thermal Analysis and Hydraulic Analysis separately for the tube-side and for the shell-side.

3.1 Petukhov-Popov Method

Tube-Side Nusselt Number

Nusselt number is a function of Reynolds number (Re) and Prandtl number (Pr). However, there are equations developed according to the type of flow. For turbulent flow, the following equation developed by Petukhov-Popov (1963) can be used.

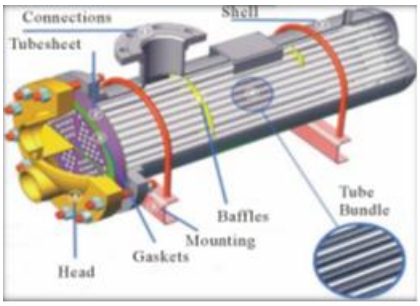

Fig.1 Shell and tube heat exchanger BEU

$$Nu_t = \frac{(f/2)Re_t Pr_t}{C + 12.7\left(\frac{f}{2}\right)^{\frac{1}{2}}\left(Pr_t^{\frac{2}{3}} - 1\right)}$$

Where C and ƒ is the friction factor turbulent flow in smooth duct

Petukhov-Popov Correlation predicts the results in the range

$4000 \le Re_t \le 5 \times 10^6, 0.5 \le Pr_t \le 10^6$ with accuracy $\pm 5\%$.

Tube-Side Heat Transfer Co-efficient

$$h_t = Nu_t \frac{k_t}{d_i}$$

Hydraulic Analysis for Tube-Side

The pressure drop encountered by the fluid making passes through the heat exchanger is a multiple of the kinetic energy of the flow. Therefore, the tube-side pressure drop is calculated by

$$\Delta p_t = \left(4f\frac{LN_p}{d_i} + 4N_p\right)\frac{\rho_t u_t^2}{2}$$

3.2 Bell-Delaware Method

Bell Delaware method is a rating analysis. The Bell-Delaware method offers the most widely accepted method.

Simplified Mechanisms of Shell-Side Flow

As can be seen from Fig., five different streams are identified on the shell-side. Stream B is the main cross flow stream flowing through one window across the cross flow section and out through the opposite window.

However, there are four other streams because of the mechanical clearances required in a shell-and-tube heat exchanger. One of them is the A stream that leaks through the clearance between the tubes and the baffle. There is also the C stream which is the bundle bypass stream. The E stream flows through the clearance between the baffles and the inside diameter of the shell. Finally, the F stream flows through any channels within the tube bundle caused by the provision of pass dividers in the exchanger header.

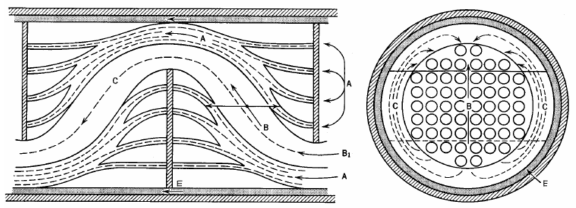

Fig.2 Diagram indicating leaking paths for shell side flow

In Bell Delaware method correction factors were introduced for the following elements:

(a) Leakage through the gaps between the tubes and the baffles and the baffles and the shell, respectively.
(b) Effect of the baffle configuration.
(c) Bypassing of the flow around the gap between the tube bundle and the shell.
(d) Effect of adverse temperature gradient on heat transfer in laminar flow.

Shell-Side Heat Transfer Coefficient

In the Bell–Delaware method, the shell side heat transfer coefficient h_s is determined using Eq. () by correcting the ideal heat transfer coefficient h_{id} for various leakage and bypass flow streams in a segmental baffled shell-and-tube exchanger.

$$h_{id} = 0.27 \frac{k_s}{d_o} Re^{0.63} Pr^{0.36} \left(\frac{Pr_\infty}{Pr_W}\right)^{0.25}$$

Shell side mass velocity as,

$$G_s = \frac{\dot{m}_s}{A_{o,cr}}$$

Shell side Reynolds number as,

$$Re_s = \frac{G_s d_o}{\mu_s}$$

It is then corrected by five correction factors as follows:

$$h_s = h_{id} J_c J_l J_b J_s J_r$$

Hydraulic Analysis for Shell-Side

Similar to shell-side heat transfer, the shell-side pressure drop is also affected by various leakage and bypass streams in a segmentally baffled exchanger.

$$\Delta p_s = \Delta p_{cr} + \Delta p_w + \Delta p_{i-o}$$

$$\Delta p_s = \left[(N_b - 1)\Delta p_{b,id}\zeta_b + N_b \Delta p_{w,id}\right]\zeta_l + 2\Delta p_{b,id}\left(1 + \frac{N_{r,cw}}{N_{r,cc}}\right)\zeta_b \zeta_s$$

Overall heat transfer coefficients

$$\frac{1}{U_o} = \frac{1}{h_s} + R_{of} + \frac{d_o}{k} \ln \frac{d_o}{d_i} + \frac{d_o}{d_i} R_{fi} + \frac{d_o}{d_i} \frac{1}{h_t}$$

Heat transfer area

$$A_o = \frac{Q}{U_o LMTD}$$

The inputs are given to HTFS software and results are obtained. For the same process conditions shown in Table I and Table II heat exchanger is designed and developed by using software. From the measured Parameters heat duty is calculated. Result of therotical and software designs are compared.

4. Results and Discussion

The heat exchanger is for the 30MW solar thermal power plant. The validation of therotical thermal design is based on HTFS Software results.

The analytical and software results for heat transferred (Fig. 3), Log mean temperature difference (Fig. 4), Pressure drop (Fig. 5), Heat transfer area (Fig. 6) are compared and relative difference is checked.

The graph in fig. 3, fig. 4, fig. 5, fig. 6 shows that we have get heat transfer 10 % more than that analytical method. Also pressure drop & Heat transfer area get minumum.

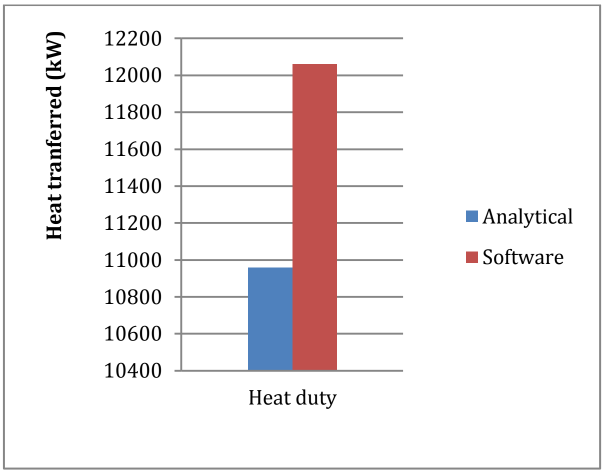

Fig.3 Heat transferred for heat exchanger

In same dimension means no. of tube, tube daimeter, shell daimeter, tube length we get optimised area & pressure drop, increase in heat duty.

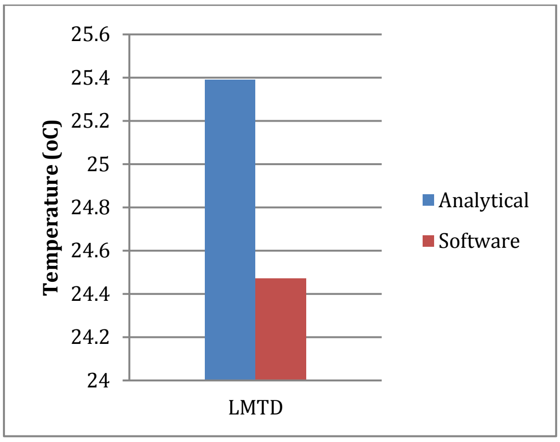

Fig.4 LMTD for heat exchanger

Mass velocity strongly influences the heat-transfer coefficient. For turbulent flow, the tubeside heat-transfer coefficient varies to the 0.8 power of tubeside mass velocity, whereas tubeside pressure drop varies to the square of mass velocity. Thus, with increasing mass velocity, pressure drop increases more rapidly than does the heat-transfer coefficient. Consequently, there will be an optimum mass velocity above which it will be wasteful to increase mass velocity further.

Furthermore, very high velocities lead to erosion. However, the pressure drop limitation usually becomes controlling long before erosive velocities are attained. The minimum recommended liquid velocity inside tubes is 1.0 m/s, while the maximum is 2.5–5.0 m/s.

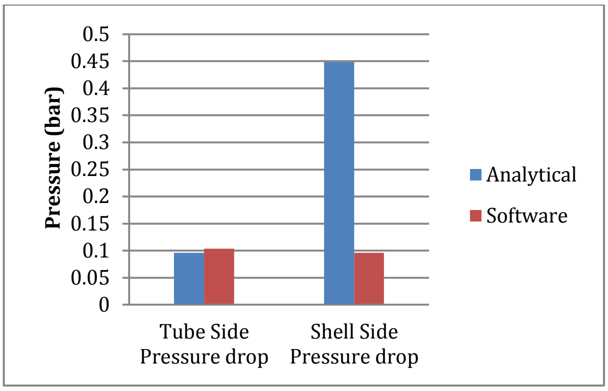

Fig.5 Pressure drop for heat exchanger

It is strongly recommended that only baffle cuts between 20% and 35% be employed. Reducing baffle cut below 20% to increase the

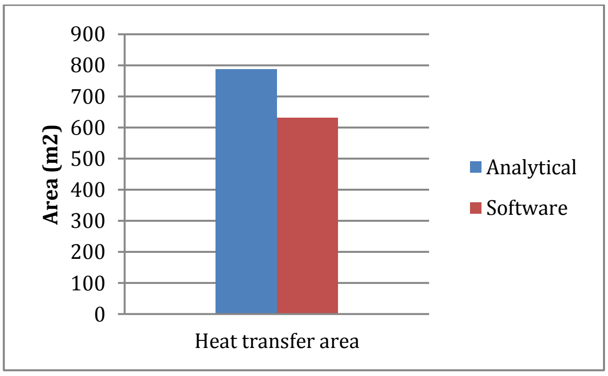

Fig.6 Heat transfer area for heat exchanger

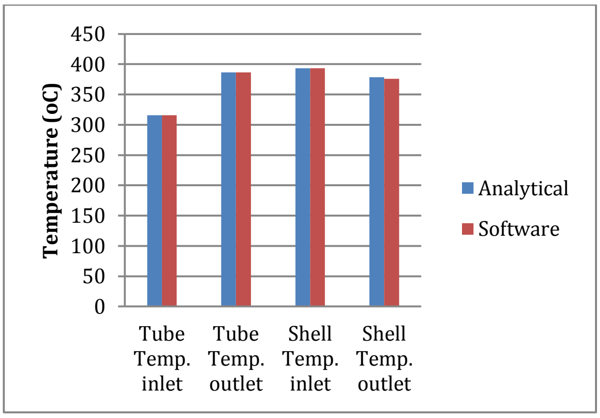

Fig.7 Temperature of heat exchanger

Shell side heat-transfer coefficient or increasing the baffle cut beyond 35% to decrease the shellside pressure drop usually lead to poor designs.

The graph in fig. 7, fig. 8 show that outlet temperature of heat exchanger achived by simulation mode is exact to the analytical method & tube side heat transfer coefficient is more, shell side heat transfer coefficient is less & overall heat transfer coefficient is also less than that of analytical method.

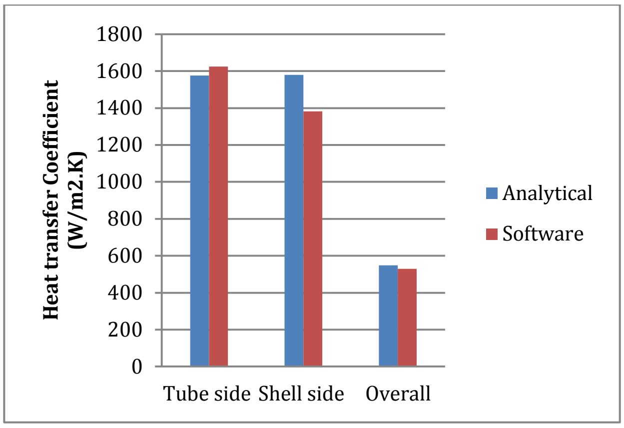

Fig.8 Heat transfer coefficient for heat exchanger

Heat transfer coefficients are depends on Reynolds number, Prandtl Number, velocity, diameter. Heat transfer coefficient varies as per these parameter varies.

Conclusions

Rating & Sizing of heat exchanger is done analytically and by using the Aspen HTFS software to perform required heat duty. Also it is validated using Aspen HTFS Software.

The following conclusion are drawn,

1) Inlet & outlet temp. of heat exchanger are achived by software is exact to calculated by anaylatical method
2) Overall heat transfer coefficient & Tube side heat tansfer coefficient is more than calculated by anaylatical method.
3) Pressure drop, LMTD & Area of heat exchanger are smaller than calculated.
4) Fouling factor has large influence in the calculation of surface area.So steps like periodic cleaning to maintain lower fouling condition may be less expensive than purchasing additional surface area.
5) The relative difference between analytical results and software results is very less (<10%).
6) In simulation mode we have get exact results as in rating mode.

Thus, proven thermal design methodology is set to Thermal Design of Shell & Tube Heat Exchanger for Concentrating Solar Power Application by studying various parameters affecting on it.

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