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Selection of TEMA Type and Thermal Design Optimization of Shell and Tube Heat Exchanger

Satish Patil[1] and N.P. Gulhane[2]

[1]*M.Tech Scholar, VJTI, Matunga, Mumbai, India*
[2]*VJTI, Matunga, Mumbai, India*
E-mail: [1]patilsatish1870@gmail.com, [2]npgulhane@vjti.org.in

Abstract—*The shell and tube heat exchanger is by far the most widely used type of heat exchanger in the power and process industries. Selection of heat exchanger for particular fluid services requires several engineering judgments. Safety, reliability and cost effectiveness are the major consideration for the selection. The present study provides development of guidelines for selection of TEMA type shell and tube heat exchanger and thermal design considerations to optimize design with minimum iterations. The thermal design of shell and tube heat exchangers is done by the computer based software viz. HTRI, ASPEN.*

The actual design of the exchanger begins with selection of TEMA type and the geometrical parameters. The thermal design comprises of estimation of optimum surface area to meet thermo-hydraulic performance by deciding Number of tubes, tube length, selection of tube outer diameter, tube pitch, baffle type, baffle cut, baffle spacing, nozzle sizing, number of passes, flow directions, multiple shells, and orientation. The paper explains the guidelines for selections and optimizing of shell and tube heat exchanger. In this paper a typical shell & tube exchanger for petrochemical applications is designed to explain the effects of above parameters on to the optimize design.

Keywords*: Shell and Tube Heat Exchanger, TEMA, Guidelines, Thermal Design, Design Optimization.*

1. INTRODUCTION

The heat exchangers are always been at the heart of industrial heat recovery systems but the latest advances in their design makes them even more central to manufacturing and industrial processes. Raw and unprocessed crude oil supplied in refinery is dirty, corrosive, and hazardous. This corrodes and fouls the heat transfer equipment. Thus selection of better heat exchanger for particular fluid service is prime important before actual design starts. Safety, reliability, performance and cost are the major consideration for the selection of heat exchanger. Thermal design of heat exchanger starts with the selection of exchanger TEMA type. Thermal design has goal at fixing the maximum number of tubes in given exchanger design. Thermal design typically consists of determination of number of tubes, tube pitch, baffle spacing, baffle cut, and stream analysis, and their effect on heat transfer and pressure drop.

2. TEMA DESIGNATION OF HEAT EXCHANGERS

Shell and tube heat exchangers are selected based on TEMA classification. TEMA is set of standards commonly used for the designing and manufacturing of shell and tube heat exchangers. This provides a three letter nomenclature. The first letter identifies the front head, the second letter identifies the shell type and the third letter identifies the rear head type [1] as shown in Fig. 1.

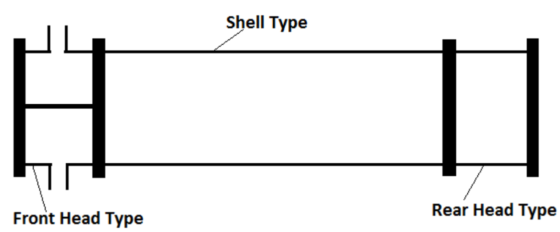

Fig. 1: Heat exchanger nomenclature.

3. SELECTION OF TEMA TYPE

3.1 Front Head Types

There are five front head types namely A, B, C, N and D type (see Fig. 2).

3.1.1 A type. It is the most common type of header, has two flange joints, easy to repair and replace. Cleaning of inside of the tubes is easy while two seals increase the risk of leakage.

Recommended for: Dirty tubeside fluid, petroleum refineries.

3.1.2 B type. One end of B type head is flanged while other end is permanently welded in semi-elliptical head.

Recommended for: Clean tubeside fluid, high pressure duties compare to A-type.

3.1.3 C type. One end of C type head is flanged while other end is welded to the tubesheet and extended to form a flange. C type is difficult to repair and replace because the tube bundle is an integral part of header [2].

Recommended for: Removable bundles, service requiring frequent shellside cleaning, high pressure applications (>100 bar) and hazardous tubeside fluid.

3.1.4 N type. N type head is similar to C-type head but integral tubesheet is not welded to form flange instead welded to the shell. This has same disadvantage as that of C-type head.

Recommended for: High pressure application (>150 bar), hazardous fluid on tubeside.

3.1.5 **D type:** D type is specially designed, non-bolted, closure. Difficult to repair and replace as tube bundle is an integral part of the header. This is most expensive type of header.

Recommended for: High pressures (>150 bar) applications [2].

3.2 Shell Type

There are seven shell types in TEMA specification namely E, F, G, H, J, K and X (see Fig. 2).

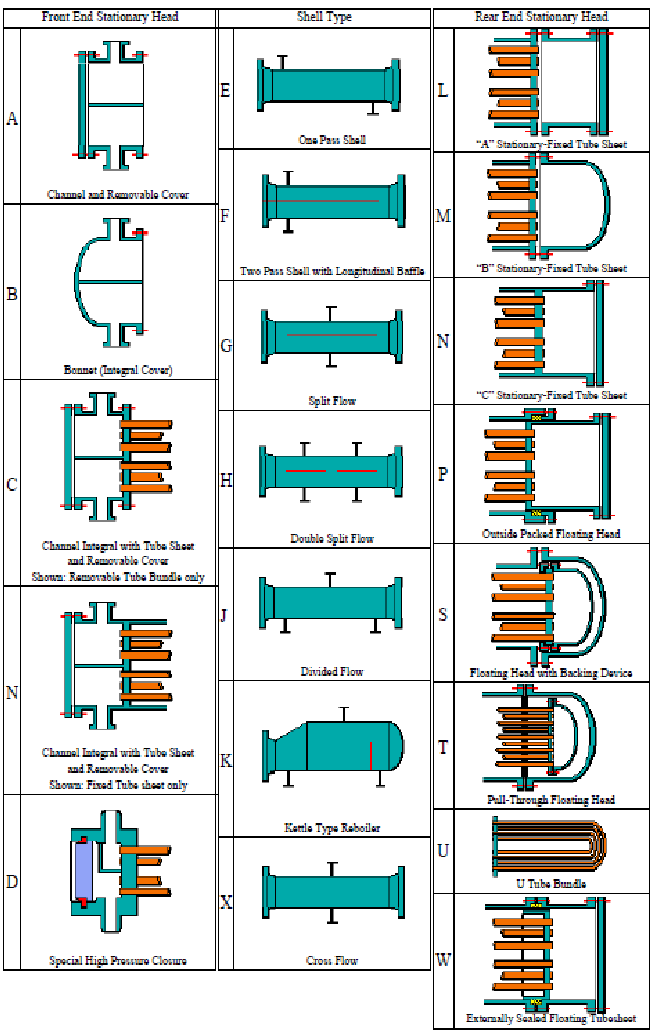

Fig. 2: TEMA designation system. [1]

3.2.1 E type: E is one pass shell, the most common shell type, used almost for all duties and applications.

3.2.2 F type: F is two-pass shell with longitudinal baffle. Shellside temperature range is limited to 175 degree C due to thermal and hydraulic leakages across the longitudinal baffle [3].

Recommended for: Pure counter-current flow, U-tube type removable bundles.

3.2.3 G and H type: G is split flow while H is double split flow, both the shell types are with central support plates. H-type is greater in length than G-type, used where the larger units are required. Shellside pressure drop is much lower as flow is divided.

Recommended for: Horizontal thermosyphon reboilers [4].

3.2.4 J type: J is divided flow on the shellside reduces the flow velocities over the tubes and hence reduces the pressure drop and tube vibration.

Recommended for: when shellside pressure drop exceeds in an E-type shell, condensing and boiling services.

3.2.5 K type: K is the kettle type reboiler in which shell diameter is larger than tube bundle. It provides large disengagement space in order to minimize shellside liquid carry over. Alternatively K-type shell may be used as chiller.

Recommended for: Reboilers, condensing or boiling services.

3.3 Rear Head Type

Based on construction, it is divided into three types: fixed tubesheet (L, M and N), U tube and floating head type (P, S, T, U and W) (see Fig. 2).

3.3.1 Fixed Tubesheet Type. This consists of L, M and N type rear heads corresponding to A, B, N-type front head channels. The tubesheet is welded to the shell therefore shellside cannot be accessed. The inside of the tubes can be access without removing any pipework. Bellows or an expansion roll is required to allow for thermal expansions which limit the permitted operating temperature and pressure.

Recommended for: Clean shellside fluid, when tubeside fluid cleaned mechanically for L type while chemically for M and N type.

3.3.2 U-Tube Type. The U tube is cheapest of all removable bundle designs. It permits thermal expansion of tube, tubesheet and shell. It has tightest bundle to shell clearances. It cannot accommodate pure counter-flow unless F-type shell is used.

Recommended for: Thermal expansion [5], services with steam or other clean fluids in the tubes, high pressure applications.

3.3.3 Floating Head Type. All the floating heads permits the thermal expansion of tube bundle, tubesheet and shell. P type is an outside packed floating rear header with a low cost floating head design which allows access to the tubes and the bundle to be removed for cleaning. It is limited to low pressure, non-hazardous fluids. S type is the most expensive of all the floating head types, it has smaller bundle to shell clearances than the other floating head types. T type has largest bundle to shell clearance of all the floating head types. It is more expensive than fixed head and U-tube types. W type is limited to low pressure and non-hazardous fluids.

Recommended for: Thermal expansion, dirty services on both shellside and tubeside, S type for refinery services [3], for maintenance T is preferred

4. GUIDELINES FOR PRIORITY IN SELECTION OF TEMA TYPE

The order of priority in selection of front head, shell type and rear head type are as follows:

4.1 Front head

The order of priority in selection based on safety, reliability, company practice, ease of maintenance and cost are as follows:

1. If tubeside fluid is hazardous consideration should be given to use C or N-type or B-type head welded to the tubesheet.
2. If tubeside pressure is very high, consideration should be given to use D-type or B-type welded to the tubesheet.
3. For the ease of maintenance A-type head is standard for most oil companies.
4. B-type head is the cheapest.

4.2 Shell type

1. E-type shells are standard
2. G and H are normally specified only for horizontal thermosyphon reboilers.
3. J and X-types if the allowable pressure drop cannot be accommodated in E-type design.
4. F-type is used for services requiring multiple shells with removable bundles.

4.3 Rear head

P, T, W type would be considered only for special cases.

1. For normal service fixed tubesheet (L, M, N-types) can be used provided that there is no overstressing due to differential thermal expansion and the shellside will not require mechanical cleaning.
2. For thermal expansion in fixed tubesheet header, use bellows provided that shellside fluid is not hazardous, shellside pressure does not exceed 35 bar, shellside will not require mechanical cleaning and tube to shell metal temperature difference greater than 50 degree C [2].
3. Use of U-tube rear header provided that the shellside will not require mechanical cleaning and counter current flow is not required (can achieve only F-type shell).
4. S-type floating head is used when thermal expansion is to be overcome.

4.4 TEMA configuration

On considering all the TEMA guidelines for front head, shell type and rear head type, it is possible to select number of TEMA configurations. Fig. 3 shows TEMA AES Configuration.

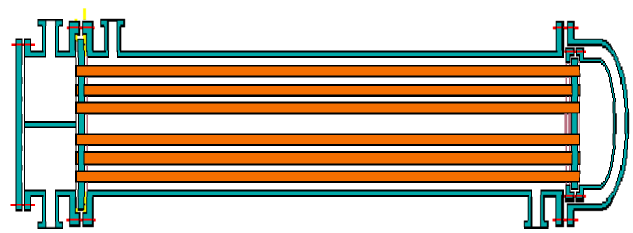

Fig. 3: TEMA type AES.

If both fluids are clean:

a. Consider BEM for low pressure and temperature differences.
b. Consider BEU for high pressure or high metal temperature differences.

1. If the shell side is clean and tubeside is dirty consider AEL or AEM.
2. If the shellside is dirty and the tubeside is clean consider a BEU.
3. If both fluids are dirty consider an AES.

5. THERMAL DESIGN CONSIDERATION

The thermal design of a shell and tube exchanger is an iterative process which is normally carried out using computer programs from organizations such as the Heat Transfer Fluid Flow Service (HTFS) or Heat Transfer Research Incorporated (HTRI). In thermal design heat exchanger is sized, all the principle construction parameter such as shell type, shell diameter, number of tubes, tube OD, tube pitch, number of passes, baffle spacing and cut are determined.

5.1 Shell ID and Tube OD

Choose the shell diameter for given design that can fit maximum number of tubes to maximise turbulence. The preferred tube length to shell diameter ratio is in between 5 to 10. Standard tube lengths as per TEMA standard RCB-2.1 are 96, 120, 144, 196, and 240 inch. The standard tube outside diameters for general use are 1/2, 5/8, 3/4 and 1 inch and for process industries 3/4 inch (19.05 mm) [1].

5.2 Number of tubes and Number of passes

The number of tubes are selected such that the tube-side velocity is from 0.9-2.4 m/s and the shell-side velocity from 0.6-1.5 m/s for water and similar liquids [2]. Greater the number of passes, greater the heat transfer coefficient. Excessive tube side velocities lead to erosion of the tube material, therefore for a safe limit ρv^2 should be less than 10000 (where v is in m/s and tubeside density ρ is in kg/m3) [1].

5.3 Tube pitch and Tube layout

The tube layout is a definite arrangement of tubes with respect to the direction of shellside fluid. There are 4 tube layout specified by the TEMA as shown in fig.

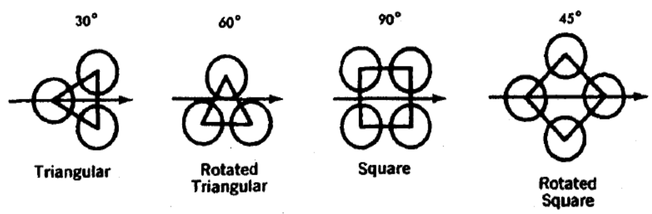

Fig. 4: Types of tube layout [1].

5.3.1 Triangular (30°) and Rotated triangular (60°). A triangular (or rotated triangular) accommodates more tubes than square (or rotated square) pattern. Triangular layout produces high turbulence. A triangular layout pattern is limited to use in clean services on the shellside.

For triangular (or rotated triangular) layout TEMA specifies, Minimum tube pitch=1.25 times the Tube OD [1].

5.3.2 Square (90°) and Rotated square (45°). It is usual practice to use square layout pattern for dirty services on shellside.

For square (or Rotated square) layout TEMA specifies, Minimum tube pitch=Larger of (1.25 times the Tube OD or tube OD + 6.35mm) [1].

5.4 Baffle type and geometry

Baffles are used to increase velocity of the fluid flowing on shellside and to support the tubes. Higher velocities have advantage of increasing heat transfer and decreasing fouling (material deposit on the tubes), but have the disadvantage of increasing pressure drop. The amount of pressure drop on the shellside is a function of baffle spacing, baffle cut, baffle type, and tube pitch.

5.4.1 Baffle spacing and baffle cut. Baffle spacing is the centreline to centreline distance between adjacent baffle. Baffle spacing is increased when it is necessary to decrease pressure drop. A limit must be imposed to prevent tube sagging or flow-induced tube vibration. The TEMA standards specify the minimum baffle spacing as 1/5 of shell inside diameter or 2 inch, whichever is greater.

Baffle cut is the height of the segment that is cut in each baffle to permit the shellside fluid to flow across the baffle. It is expressed as a percentage of shell inside diameter. Baffle cut vary in between 15-45 %, but recommended value is in between 20-25% [3].

5.5 Stream analysis

As per the model proposed by Tinker [6], there are five streams on shellside, a main cross flow stream and four leakage or bypass streams as listed below.

1) B stream – the main cross flow stream
2) A stream – baffle hole-tube leakage stream
3) C stream – bundle bypass stream
4) F stream – pass-partition lane bypass stream
5) E stream – shell-baffle leakage stream

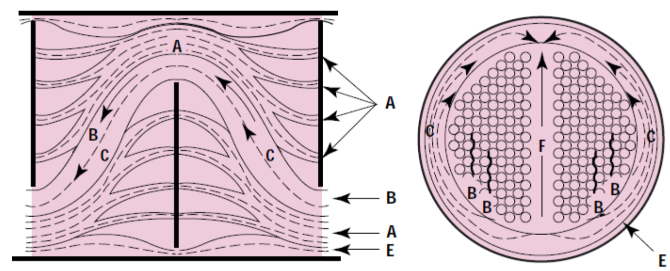

Fig. 5: Shellside flow distribution [3].

The B stream is highly effective for heat transfer, while others streams are not as effective. The A stream is fairly efficient, because the shellside fluid is in contact with the tubes. The C stream is in contact with the peripheral tubes around the bundle, and the F stream is in contact with the tubes along the pass-partition lanes. Consequently, these streams also experience heat transfer, although at a lower efficiency than the B stream.

However, since the E stream flows along the shell wall, where there are no tubes, it encounters no heat transfer at all.

These streams represent the fraction of total flow on shellside. Based upon the efficiency of each of these streams, the overall shellside stream efficiency and thus the shellside heat-transfer coefficient are estimated.

6. OPTIMIZATION OF SHELL AND TUBE HEAT EXCHANGER DESIGN

Consider a shell and tube heat exchanger has process parameter as given below in Table 1.

Table 1: Problem specification

Parameters	Shellside	Tubeside
Process stream	Hydrocarbon	Hydrocarbon
Flow rate, kg/hr	540000	180000
Temperature in/out, degree C	210/227	320/270

Heat duty, MM kcal/hr	6.3	6.3
Density in/out, kg/m3	732/717	650/705
Viscosity in/out, Ns/m2	0.0005/0.00045	0.00025/0.0004
Specific heat in/out, kJ/kg.K	2.659/2.721	3.224/3.015
Thermal conductivity in/out, w/mk	0.0686/0.0674	0.057/0.064
Allowable pressure drop, (kgf/cm2)	1	0.7
Shell ID, mm	780	
Tube OD * thickness * length, mm	25.4 * 2.1 * 7315	
No of tubes * no of tube passes	388 * 4	

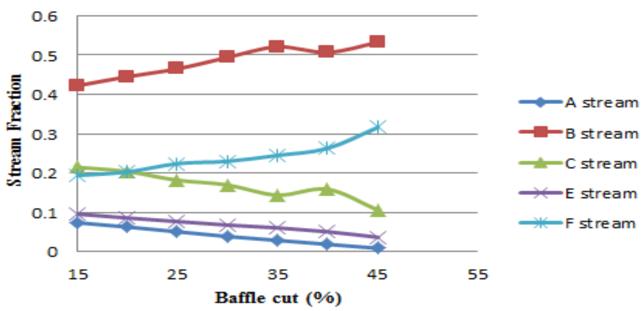

Fig. 6: Stream fraction vs % baffle cut.

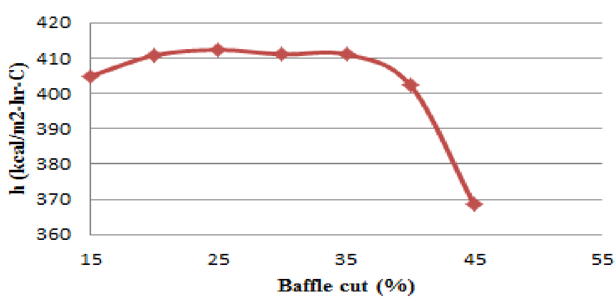

Fig. 7: h vs % baffle cut.

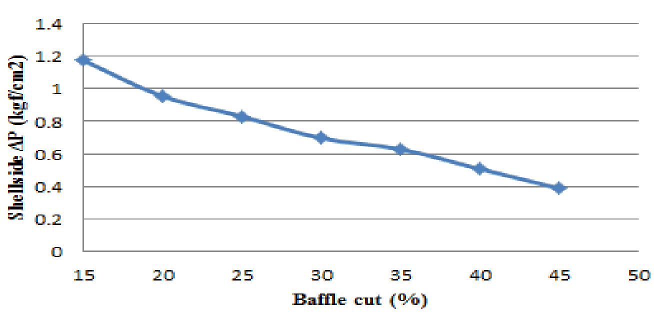

Fig. 8: Shellside ΔP vs % baffle cut

Initially the heat exchanger TEMA type is selected for given fluid service. The above heat exchanger is used for petrochemical application. Both tubeside and shellside fluids are dirty fluid services, so the equipment need to be clean frequently on both sides, therefore recommended head type is A. To allow the differential thermal expansion that occurs due to high temperatures, S type rear head is incorporated in this design. E type shell is the best choice as it can accommodate the shellside pressure drop. So for this specific problem TEMA AES is selected for the given process conditions.

The shellside analysis is more tedious and complex than the tubeside analysis. Thermal design starts with deciding all the major parameters for exchanger. The baffle cut and baffle spacing has predominant effect on the heat exchanger design.

6.1 Case 1

The effect of varying baffle cut on shellside analysis while keeping other construction parameters as constant is illustrated.

The first run is taken with the baffle cut as 15 %, the shellside pressure drop (1.174 kgf/cm2) exceeds the allowable pressure drop (1 kgf/cm2). To obtain a better design, six consecutive baffle cuts 20%, 25%, 30%, 35%, 40% and 45% are taken for the optimization. The results of these baffle cut on stream analysis is plotted on the graph as shown in fig. It is observed that, with increase in baffle cut from 15% to 45%, the main cross flow stream (B) increases progressive, pass partition stream (F) increases slowly while tube to baffle hole (A), baffle to shell (E) fractions decreases steadily.

The effect of baffle cut on the heat transfer coefficient and pressure drop are shown in fig. the shellside flow velocity is the function of baffle cut. With increase in baffle cut flow velocity decreases, which in turn decreases the heat transfer coefficient. The pressure drop is proportional to the flow velocity; hence it decreases with increase in baffle cut but not as fast as the heat transfer coefficient. The peak value of heat transfer coefficient obtained at 25% baffle cut.

6.2 Case 2

The above equipment with all the construction parameters is taken as constants to optimize the design for varying baffle spacing. The 25 % baffle cut is the optimum value for the given problem. In order to optimize the above design with baffle spacing six consecutive runs are taken for baffle spacing of 500 mm to 750 mm. The results obtained from change in baffle spacing on shellside stream analysis, pressure drop and heat transfer coefficient are discussed.

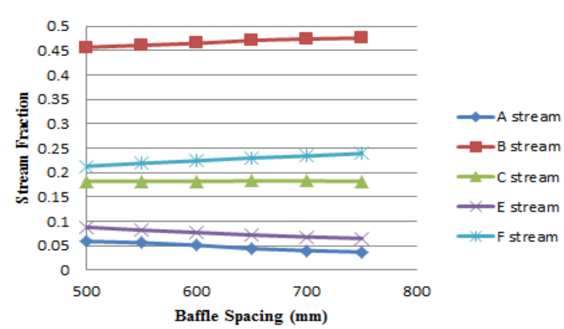

Fig. 9: Stream fractions vs baffle spacing

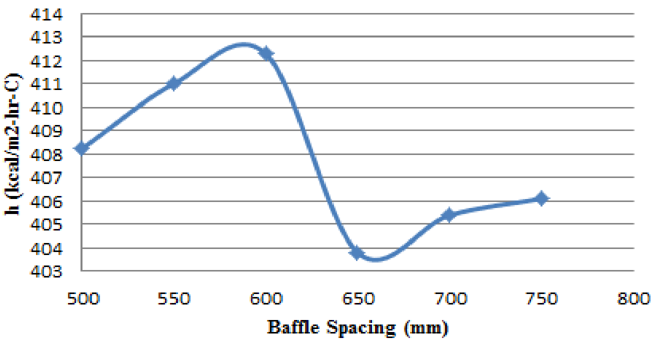

Fig. 10: h vs baffle spacing

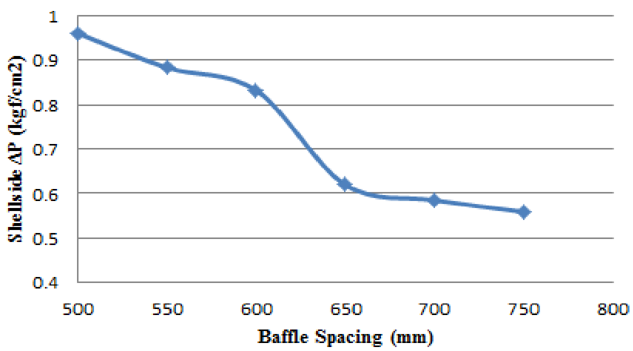

Fig. 11: Shellside ΔP vs baffle spacing.

The graph of baffle spacing verses stream analysis is plotted as shown in fig. It is seen that, as the baffle spacing increased from 500 mm to 750 mm, B stream increases progressively, F stream increases slowly, C stream remains steady while E and A stream decreases steadily.

The overall heat transfer coefficient of the shellside stream increases rapidly and maximise at 600 mm baffle spacing, while shellside pressure drop falls progressively. The allowable shellside pressure drop is 1 kgf/cm2, the design with baffle spacing 500 mm utilises the shellside allowable pressure drop most effectively while baffle spacing from 650 mm to 750 mm not utilise much effectively, hence ruled out. After 500mm baffle spacing, 550 mm and 600 are the other best choices for effective utilisation of allowable pressure drop.

7. CONCLUSION

It is found that the guidelines for TEMA type selection gives the better options to choose the TEMA configuration required in petrochemical application. The TEMA AES is the best choice for the given problem.

The shellside design is complex as there is not just one stream but main stream and four other leakage streams. It observed that, on varying baffle cut from 15% to 45%, 25% baffle cut gives the maximum heat transfer coefficient, better main cross flow stream fractions, and effective utilization of allowable pressure drop. The 25% baffle cut is the best choice for this design. The baffle spacing of 600 mm gives maximum heat transfer coefficient of 412.29 kcal/m2-hr-C, better main cross flow fractions and effective utilization of shellside allowable pressure drop. The design is optimize at 25% baffle cut and 600 mm baffle spacing.

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