Optimization of Air Cooled Heat Exchanger Design Using HTRI

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***Abstract*- Commercial software tools for design of air cooled heat exchanger (ACHE) are widely used in chemical engineering departments for equipment design. In this paper an optimization process is carried out by varying tube pitch (transverse pitch), number of tube rows and air velocity using Heat transfer research incorporation (HTRI) software. With the help of graph the optimized result is shown for different values of different parameter. The proposed design was successfully verified by comparing the obtained output result with design taken from the literature.**

***Keywords*- Pitch, Transverse.**

I. INTRODUCTION

Air-cooled heat exchangers are second only to shell-and-tube exchangers in frequency of occurrence in chemical and petroleum processing operations. These units are used to cool and/or condense process streams with ambient air as the cooling medium rather than water. Cooling with air is often economically advantageous, e.g., in arid or semi-arid locations, in areas where the available water requires extensive treatment to reduce fouling, or when additional investment would otherwise be required to expand a plant's existing cooling-water supply. Regulations governing water use and discharge of effluent streams to the environment also tend to favour air cooling. Although the capital cost of an air-cooled exchanger is generally higher, the operating cost is usually significantly lower compared with a water-cooled exchanger. Hence, high energy cost relative to capital cost favours air cooling. Air cooling also eliminates the fouling and corrosion problems associated with cooling water, and there is no possibility of leakage and mixing of water with the process fluid. Thus, maintenance costs are generally lower for air-cooled exchangers [1].

II. TYPES OF AIR COOLED HEAT EXCHANGER

The forced draft ACHE is the most economical and most common style air cooler where axial fans used to force air across the fin tube bundle are mounted below the bundle and therefore the mechanical sections are not exposed to the hot exhaust air flow. Also, another advantage of this arrangement is the fact that provides direct access to bundle for replacement.

The induced draft ACHE involves axial fans to pull air across the fin tube bundle and it is the second most economical arrangement (Fig. 1). In contrast to the previous arrangement, the fans are positioned above the bundle thus offering greater control of the process fluid and bundle protection due to the additional structure [2].

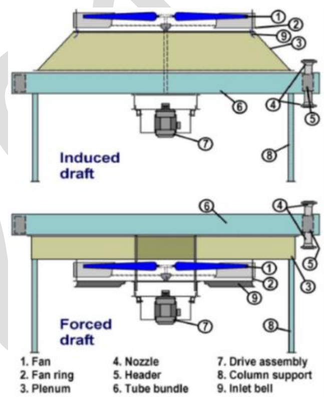

Fig. 1 Basic configuration of induced & forced type air cooled heat exchanger [3]

III. FINS

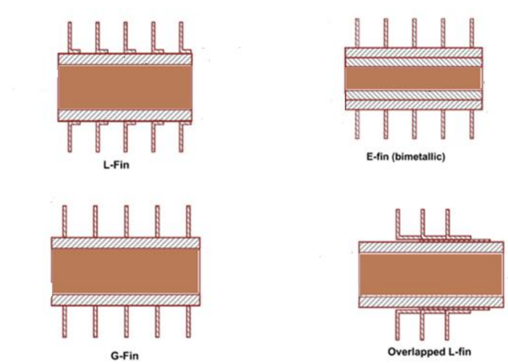

Fig. 2 Different types of Fins [1]

The design procedure of air cooled heat exchanger is very time consuming and in industry common practice is to rely on commercial software. Number of software packages are available to assist designers in producing a satisfactory equipment design .commercial software available for heat exchanger design is HTRI. In software we have option of rating, design & simulation [5].

In commercial software design user defined values of independent design parameters, fixed parameters, process specification and user updated values of independent design parameter sent to design routine from that it is sent to trial solution. From that it goes to design warning. If yes then we need to update the value of independent design parameter else no then to rating mode because it's good for design practice. Then design is satisfactory then exit otherwise you need to update the value of independent design parameter [6].

IV. OPTIMIZATION OF ACHE USING HTRI

Air cooled heat exchanger are most widely used in petroleum industries. So we have considered the case study of condensation of industrial propane with mass flow rate of 27.78 kg/s at 1751.3 kPa absolute and 70.79 C is considered. The air is available at 35 C. The fluid on the tube side was considered to be a mixture of 97% propane, 2% ethane, 0.5% n-butane and 0.5% i-butane [1]. The fluid property were defined using VMGThermo property package.

Horizontal tube orientation was considered. In this 5 bay were in parallel with single bundle per bay.

Selection of fin:

Type: High fin (Plain round)

Height: 15.875 mm

Base thickness: 0.432 mm

Fin tip thickness: 0.254 mm

MOC: Al 1060 - H14

Tube specification:

Outer diameter: 25.4 mm

Wall thickness: 2.4 mm

Length: 18.288 m

MOC: Carbon steel

Layout: Staggered

Fan Specification:

Arrangement: forced draft

Efficiency: 65% (combined with drive efficiency) [7].

The effect of different parameters on the design of ACHE was studied. These parameters include transverse pitch, face velocity and number of tube rows. The grid design option in HTRI was used to study the pressure drop and different heat transfer coefficient as a function of these parameters.

A. Effect of transverse pitch

To study the effect of transverse pitch, the total heat transfer area was kept constant by keeping the number of tube constant (70 in each row). In this only the distance between the two tubes were changed so the bundle width also varies. The grid design option was used in which the transverse pitch was changed from 50.8 to 76.2 mm with step increment of 2.54 mm. 4 tube rows are used with 2 tube side passes.

Table I

Data of Variation In Parameters Using Grid Design For Different Transverse Pitch

Over Design %	Actual U (W/m2-K)	Bare Outside h (W/m2-K)	Extended Outside h (W/m2-K)	Outside Velocity (m/s)	Outside DP (Pa)	Transverse Pitch (mm)	Bundle Width (m)
2.8	24.2	921.5	42.94	5.18	88.44	50.800	3.607
5.65	23.83	898.00	41.84	4.86	72.83	53.340	3.783
8.19	23.53	878.66	40.94	4.59	61.49	55.880	3.960
10.32	23.25	862.34	40.18	4.38	52.96	58.420	4.136
12.31	23.04	848.56	39.54	4.19	46.42	60.960	4.313
14.15	22.86	836.67	38.98	4.04	41.27	63.500	4.489
15.55	22.67	826.39	38.50	3.91	37.17	66.040	4.666
16.92	22.52	817.23	38.08	3.79	33.81	68.580	4.843
18.1	22.36	809.07	37.70	3.69	31.05	71.120	5.019
19.41	22.26	801.63	37.35	3.60	28.73	73.660	5.196
20.38	22.13	795.08	37.05	3.52	26.80	76.200	5.372

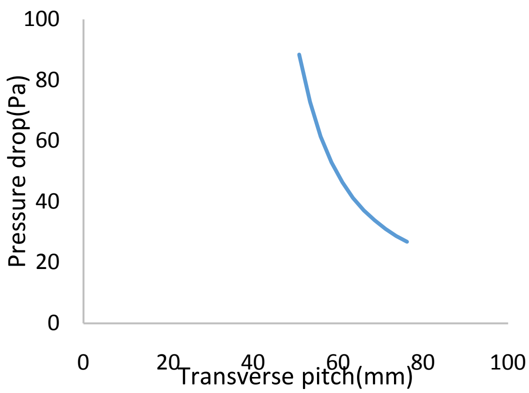

Fig. 3 The Graph of pressure drop vs transverse pitch

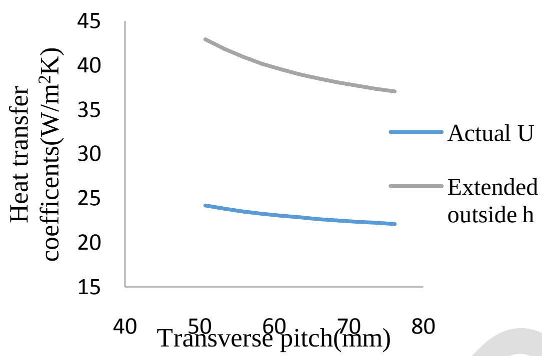

Fig. 4 The graph of different heat transfer coefficient vs transverse pitch

From the graph of figure 3, pressure drop increased with decrease in transverse pitch as the cross flow area was reduced.

With this the figure 4 shows that this has substantial effect on bare outside heat transfer coefficient. This result in small increase in overall actual heat transfer coefficient (U). The standard practice is to kept pitch 2.5 times of the pipe outer diameter. But during optimization if pressure drop or the flow regime are to be altered then this parameter can be considered.

B. Air velocity

The heat transfer coefficient of air is very small. So the turbulence due to air velocity become important criteria to work upon. Also the optimization of this parameter is very important as low velocity would result in recirculation of hot air which would reduce the efficiency. For this calculation, transverse pitch of 63.5 mm was considered with 70 tubes per row (total 4 rows

Table II

Data of Variation In Parameters Using Grid Design For Different Face Velocity

Over Design %	Actual U (W/m2-K)	Act. Face Velocity (m/s)	Bare Outside h (W/m2-K)	Extended Outside h (W/m2-K)	Outside Velocity (m/s)	Outside DP (Pa)
14.15	22.86	2.10	836.67	38.98	4.04	41.27
18.25	23.08	2.20	851.33	39.67	4.24	44.53
22.09	23.29	2.31	865.91	40.35	4.44	47.90
25.72	23.51	2.41	880.43	41.02	4.64	51.37
29.06	23.71	2.52	894.85	41.69	4.83	54.96
35.3	24.12	2.73	923.44	43.03	5.23	62.47
40.98	24.52	2.94	951.68	44.34	5.63	70.42
43.72	24.73	3.04	965.86	45.00	5.83	74.56
48.57	25.10	3.25	993.50	46.29	6.23	83.16
50.92	25.28	3.36	1007.20	46.93	6.43	87.62
55.35	25.65	3.57	1034.34	48.19	6.82	96.86
59.59	26.00	3.78	1061.09	49.44	7.22	106.52
63.53	26.34	3.99	1087.48	50.67	7.62	116.61
67.29	26.68	4.20	1113.37	51.88	8.02	127.11

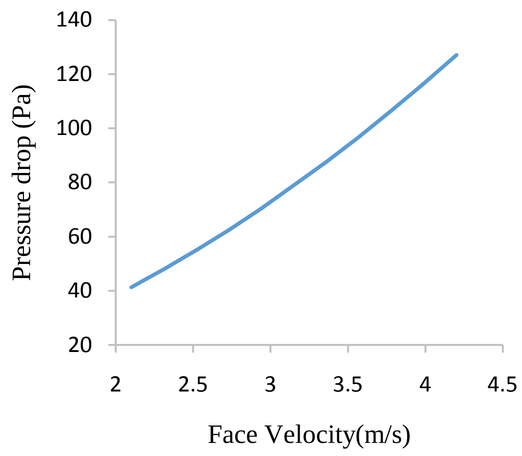

Fig. 5 The graph of pressure drop vs face velocity

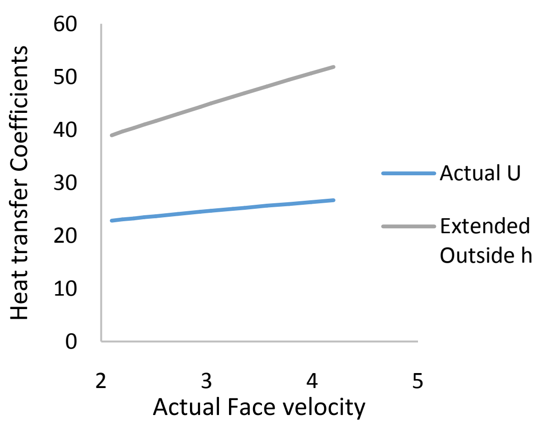

Fig. 6 The graph of heat transfer coefficient vs actual face velocity

The graphs in figure 5 and 6 shows that the pressure drop and the actual U both increased with increase in face velocity. So the optimization was required. The face velocity between 3-3.5 m/s can be an optimized selection.

C. No of tube rows

In this case the number of tube rows were changed keeping total number of tube same in order to keep the heat transfer area constant for every case. For this case face velocity used was 2m/s and transverse pitch 63.5 mm.

Table III

Data For Different Tube Rows.

No. of tube rows	Tube in single row	% overdesign	Actual U	ΔP_{air}
2	140	30.76	20.66	22.23
3	93/94	18.61	20.95	31.18
4	70	14.15	22.86	41.27
5	56	-4.28	22.38	51.8
6	46/47	-23.18	22.88	62.65

The pressure drop and actual U both increased with number of tube. The optimization is required to be carried out. The optimize selection is 3-4 rows of tube.

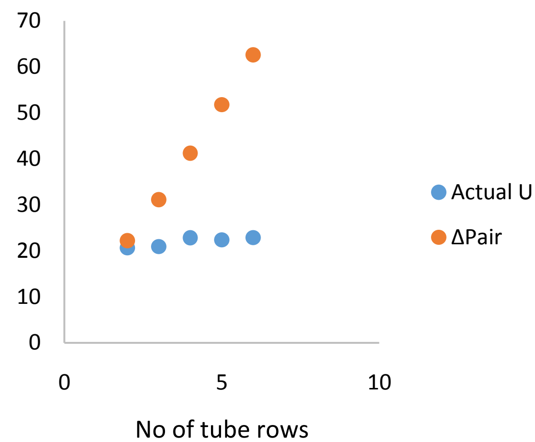

Fig. 7 The graph of actual U and Pressure drop vs no. of tube rows

V. OPTIMIZED CASE BY CONSIDERING ALL THE THREE PARAMETERS

Design Parameters	Design 1
Transverse Pitch	60.96 mm
Air velocity	3 m/s
no. of tubes per row	55
tube rows	4
Output Variables	
Pressure drop tube side	13.239 KPa
Pressure drop air side	92.16 Pa
h tube side	1897.15 W/m^{2}K
h air side	47.93 W/m^{2}K
U	26.414 W/m^{2}K
Overdesign	10.03%

VI. CONCLUSION

A detailed analysis was carried out using HTRI software and optimized by changing different parameters such as transverse pitch, air velocity & number of tubes. The standard pitch was 63.5mm and we optimized it to 60.96mm and optimized face velocity to 2.5m/s, this increased the pressure drop on air side within acceptable range and also the air side heat transfer coefficient. The number of tube row was taken as 4 for the optimized result and double pass was considered to increase the tube side velocity with helped to increase the tube side heat transfer coefficient. These designed is optimized by using grid design option in HTRI.

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REFERENCES

[1] R.W.Serth: "Process Heat Transfer Principles and Application", Elsevier Academic Press, 2007.
[2] Donald Q Kern: "Process Heat Transfer", McGraw Hill Publication, international Edition 1965.
[3] http://www.onsitepowerinc.com/documents/supplierDocs/amercool/Basics%20of%20Air%20cooled%20Heat%20Exchangers%20rev1.pdf. (Accessed 10/09/2015)
[4] Chi-Chuan Wang: Recent Advances in Fin-And-Tube Heat Exchanger, World Scientific, 10.1142/S2010132511000624.
[5] Kakac/ Liu: Heat Exchangers Selection, Rating and Thermal Design, Second Edition, CRC Press, 2002.
[6] Kaveripakam Balapratap, G. Adi Narayana: Optimization the Heat Exchanger Thermal and Pressure Drop using Empirical Formulae as per TEMA and HTRI, International Journal of Scientific Engineering and Technology Research, Vol-4, Issue-01.
[7] Air Cooled Heat Exchangers for General Refinery Services, API Standard 661, Fourth Edition, 1997.