African Research Review

An International Multi-Disciplinary Journal

ISSN 1994-9057 (Print) *ISSN 2070-0083 (Online)*

Volume 2 (4) September, 2008 **Special Edition: *Engineering***

Thermal Design and Simulation of a Heat Exchanger for a Nigerian Refinery

(pp. 124-143)

A. O. Oluwajobi -Department of Mechanical Engineering, Obafemi Awolowo University, Ile-Ife. jide_ogunjobi@yahoo.com

A. E. Akpan - Department of Mechanical Engineering, University of Ibadan, Ibadan – Nigeria

Abstract

A heat exchanger was designed to replace the fired heater at the Crude Distillation Unit of the Warri refinery, Nigeria. This would eliminate the problems such as pollutions and the high cost associated with the use of fired heater to heat the preflashed crude. The exchanger was designed with the use of the INPRO Heat Exchanger Software, which would use atmospheric residue at a temperature of 345℃ to heat crude oil from 224℃ to 252℃. The heat exchanger of TEMA AES configuration having a surface area of 620m^2 and tube length of 6.1m was then simulated with the use of software – Thermal Simulator, written to determine the effect of changes in mass flow rate on pressure loss and heat transfer coefficient. Simulation results showed an increase in all output parameters, except the tube side heat transfer coefficient, which remained constant due to the high viscosity and low Reynolds number of the flow. The cost of producing the *exchanger was estimated at about $120,000, with an estimated payback period of about 6 months.*

Keyword: Thermal Design, Simulation, Heat Exchanger, Warri Refinery

Introduction

The Crude Distillation Unit of a refinery is concerned with the initial separation of crude oil into petroleum fractions by the use of distillation columns. Distillation columns operate on the principle that if a mixture of two volatile liquids is heated, the vapour that comes off initially will have a higher concentration of the component with the lower boiling point.

An important factor in the distillation of crude is the temperature of the feed into any of the distillation columns. The temperature of the feed is important because it affects the rate and the quality of the separation of the fraction in the column. So the feed must be raised to the appropriate temperature to ensure optimum separation.

At the Warri refinery, there are four major distillation columns, namely; the Preflash Column (10-C-07), used to separate light fractions, like liquefied petroleum gas and light gasoline at low temperatures; the Fractional Column (10-C-01), which is the main separating column for the different petroleum fractions; the Stabilizing Column (10-C-05), used to separate liquefied petroleum gas from naphtha and the Splitting Column (10-C-06), used to separate light naphtha from heavy naphtha. Two of these columns use fire heaters to preheat feed into the columns. There are problems associated with the use of fire heaters, which include the following: high operating costs (since the fuel used as a heat source is expensive); coking (soot may be deposited on tubes of the burner leading to resistance in heat transfer); high maintenance costs (a fired heater has many expensive parts, which may require frequent replacement, e.g. the burner tips); and environmental pollution (the exhaust gas from the fired heater contains high quantity of carbon dioxide, a green house gas).

An alternative to the use of fired heater for the feed preheat is a heat exchanger. This paper proposes a shell and tube heat exchanger for use in place of a fired heater for preheating of feed entering the preflash column of the crude distillation unit. The proposed heat exchanger would make use of heat from one of the hot streams in the crude distillation unit.

In heat exchanger design, the following equation is used:

$$Q = UA\Delta T_m \quad \text{-------------------- (1)}$$

The quantity of heat, Q and the log mean temperature difference ΔT_m are known. The overall heat transfer coefficient, U is given an estimated value and the heat transfer surface area, A is then determined, based on the above equation. This serves as the basis for the determination of the exchanger's configuration. Based on this configuration, the actual value of U is calculated and then compared with the estimated value and the configuration is then varied until the calculated U correlates with the estimated value. The overall heat exchanger coefficient relative to the outside tube surface U_0 is given by

$$U_o = \frac{1}{[(1/h_i)(d_o/d_i) + r_i(d_o/d_i) + r_w + r_o + 1/h_o]} \quad ----(2)$$

where

h_i, h_o = *heat transfer coefficient – inside and outside tubes*

d_i, d_o = *inside and outside tube diameters*

r_i, r_o = *internal and external fouling resistance*

r_w = *tube wall resistance*

Shell and Tube Heat Exchangers

Shell and tube exchangers are generally suitable for pressures of up to 250 bars and temperatures up to 650°C. They may be classified according to the application in which they are used, namely;

R Petroleum and Related Process Application
C Commercial and General Process Application
B Chemical Process Services

Also, they may be classified according to the construction of the exchanger's stationary head, shell and rear end. This classification is by the Tubular Exchanger Manufacturers' Association (TEMA, 1978).

The design of a heat exchanger comprises the thermal and the mechanical design.

The thermal design deals with obtaining the correct dimension and configuration of exchanger components in order to obtain the required rate of heat transfer within the limits of a specified pressure drop. On the other hand, the mechanical design deals with the selection of materials for the components of the exchanger, the fabrication of these components and the exchanger assembly details.

Shell Side Flow Models

The ideal shell side flow model has no leakage of the shell side fluid between the adjacent baffle spaces and no by- passing of the tube bundles within a baffle space. Heat transfer coefficient and pressure loss characteristics can be calculated with correlations developed by Engineering Sciences Data Unit International Limited (ESDU, 1973). Here, there is only one flow stream and this situation may be achieved by:

- Welding each baffle to the inside of the shell
- Sealing the annular space around the tube where it passes through the baffles

- Ensuring that tubes completely fill the shell in a uniform manner

The ideal flow model is almost impossible due to the difficulties in construction, cost and the need to have certain tube bundles removable for maintenance purposes.

Operational Problems of Heat Exchanger

Even though heat exchanger designs are based on heat transfer coefficient, poor performance is usually due to factors which have nothing to do with the heat transfer coefficient (Pallen, 1998).

So, for the successful design of a shell and tube heat exchanger, the following potential problems must be addressed, namely;

Fouling:

This is the deposition of insulating material on the heat transfer surface by the process streams. Fouling is an important consideration in design, but it is an extremely complex phenomenon, which lacks repeatability and has been described by Taborek et al (1972) as the major unresolved problem in heat exchanger's heat transfer.

The six processes in which fouling occurs are viz;

- Precipitation of dissolved substances
- Deposit of particulate matter
- Solidification of material through chemical reaction
- Corrosion of the surface
- Attachment and growth of biological organisms
- Solidification by freezing

It was suggested (TEMA, 1978) the fouling factors used to oversize heat exchangers, so as to permit normal operation after fouling has occurred.

Vibration:

Indexed African Journals Online: www.ajol.info

For large heat exchangers, it is common for tubes to fail by vibration. The problems tend to occur generally when the distance between tube baffles is too large. Vibration may reveal itself as a loud noise, increased pressure loss and/or leakage of tubes. Saunders (1988) presented the following ways of preventing vibration.

- Natural frequency of tubes must be greater than twice the vortex shedding frequency
- Natural frequency must be at least twice the turbulent buffeting frequency
- Cross flow velocity must be lower than a critical velocity determined by Connors (1970), above which vibration occurs
- The use of damage numbers adopted by Thorngen (1970), which ensure that baffle shear stress and deflection of tubes are within safe limits. Erksine and Waddington (1973) obtained more reasonable results by relating these damage numbers to fluid density and absolute viscosity

Flow Maldistribution:

On the shell side, the flow may be maldistributed due to fluid bypassing round the tube bundle and leakages between the tubes and baffles.

This leads to reduction in heat transfer coefficient values and lower effective mean temperature difference. This can be corrected by the use of bypass sealing strips and increasing tube pitch.

On the tube side, the dynamic head of the centering fluid can cause higher flow in the central tubes than those on the periphery thus causing backflow in the peripheral tubes. Using an impingement plate on the centerline of an axial nozzle may prevent this.

Heat Exchanger Network Synthesis

The thermal design of heat exchangers is an integral part of the Heat Exchanger Network Synthesis (HENS). This involves the determination of the number and the configuration of heat exchangers in a process and the operating conditions of each, which would

Indexed African Journals Online: www.ajol.info

conserve energy and reduce expenditure on burning fuel for heating and construction of coolers.
According to Govind et al (1986), HENS may be summarized as follows;
Given N_h, hot process streams (which must give up heat) and N_c, cold streams (which must accept heat), each within a specified supply temperature, target temperature, heat capacity and flow rate, synthesize a network of heat exchanger which brings each to its target temperature and minimizes annual operating and investment cost.

Rabiu (1998) applied the Pinch Analysis Technique (See Linnhoff and Hindmarch (1983)) to analyze the heat exchanger network of the Warri Refinery and recommended that in order to optimize the energy recovery system of the Crude Distillation Unit, the fired furnace, which preheats the feed for the preflash column, should be replaced by a heat exchanger. The heat required to preheat the feed would be obtained from one of the hottest streams of the unit.

Sourcing of Process Data
For the design of a heat exchanger, the following data are needed, viz;
- Flow rates of both streams
- Inlet and outlet temperatures of both fluids
- Allowable pressure drop for both streams
- Operating pressure for both streams
- The heat duty (rate of heat transfer) required for the process should be obtained

Physical properties such as viscosity, density, thermal conductivity, density and specific heat capacity of the two fluids should be determined. Also, the variation of these properties between inlet and outlet conditions should be investigated and the fouling resistance of both streams.

Design of the Heat Exchanger

The INPRO Heat Exchanger Software was used for the design of the exchanger in this study. The heat exchanger is to heat crude oil prior to preflashing in the preflash column. It follows from this that the cold stream is the crude flow entering the preflash column. (See Figure 1).

Process Fluids

The hot stream to be used in preheating the crude is the atmospheric residue. The atmospheric residue is the residue obtained after the flashed crude has been subjected to distillation.

This stream was chosen as the most suitable stream because of its high temperature, specific heat capacity and mass flow rate.

High temperature – a study of the flow chart of the crude distillation unit shows that its temperature of about 345°C is one of the highest in the unit.

Specific heat capacity – At 3600W/m^{2}K, the heat capacity of atmospheric residue shows that it has the capability of giving off large amounts of heat per unit volume.

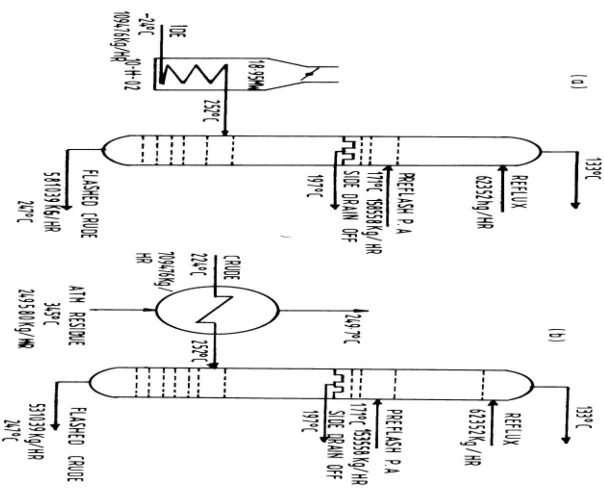

Figure 5 (a) Preflash column with heater.
(b) Column with designed heat exchanger.

Figure 1: (a) Preflash Column with Heate (b) Column with Designed Heat Exchanger

Mass flow rate - Atmospheric residue forms over 40% by mass of all crude oil distilled and so it's in relative abundance and will not require the use of auxiliary pumps to boost the flow.

To obtain the required process data necessary for the design of the heat exchanger, flow conditions of the two fluids were obtained from the process – monitoring unit of the Warri Refinery Complex. The readings were taken over a 10 –day period. The average conditions of flow were used as the basis of the design of the heat exchanger.

Heat Exchanger Configuration

Crude oil was chosen as the tube side fluid, whereas the shell fluid was taken to be the atmospheric residue. This decision was based on the convention that, the shell side fluid is usually the more viscous liquid or the fluid with the lower temperature and or the lower pressure.

The outer tube diameter, pitch and pitch angle of 25.4mm, 31.8mm and 30^o respectively are based on the fact that the tube side fluid, crude oil, is viscous and require large tube size. To enhance the compactness of the heat exchanger, the triangular tube pattern was chosen. The overall heat exchanger configuration chosen was the TEMA AES, so the heat exchanger is to have a front end of the stationary head type, one pass shell and a floating head with a split backing ring.

Other information/parameters were obtained from literature.

The tentative design parameters were fed into the heat exchanger design software, - INPRO Heat Exchanger and the final design outputs are the number of tubes, number of baffles, baffle spacing, channel nozzle dimensions, shell diameter, heat transfer surface area. Appendix 1 shows the heat exchanger data sheet and the output.

Figure 2 shows the baffle arrangement and spacing.

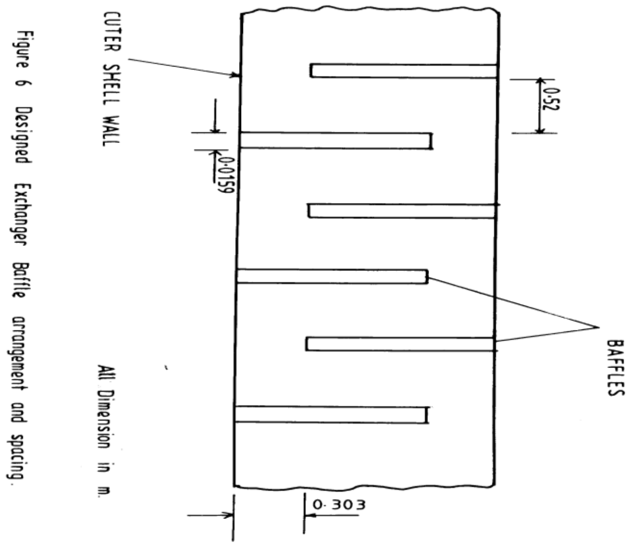

Figure 2: Designed Exchanger Baffle Arrangement and Spacing

Performance Analysis

In order to determine the exchanger performance under varying flow conditions which may occur during service (in the refinery), it is necessary to carry out a process simulation of the exchanger. The analysis is to determine the effect of varying mass flow rates on the pressure loss as well as its effect on heat transfer coefficient on both the shell and the tube sides.

A software was developed in Visual Basic for the simulation. The Visual Basic was employed to create a user-friendly graphical interface. The program is code-named Thermal Simulator.
The program is based on the following relationships.

Tube Side Properties:
Turbulent flow

$$\alpha_I = 0.0225(\lambda/d_i)\Pr^{0.495}\mathrm{Re}^{0.795}[\exp - 0.0225(In\Pr)]\Theta \text{ ----- } (3)$$

Transition Flow

$$\alpha_I = 0.1(\lambda/d_i)(\mathrm{Re}^{2/3} - 125)\Pr^{0.495}[\exp - 0.0025(In\Pr)^2][1 + d_i/L]^{2/3}\Theta \text{ ----- } (4)$$

Laminar Flow

$$\alpha_I = 1.75(\lambda/d_i)[Gz + 0.0083(Gr\Pr)^{0.75}]^{1/3}\Theta \text{ -------- } (5)$$

Pressure Drop

$$\Delta P_i = \frac{4f_i Lm}{2pd_i\Theta}$$

$$where\ f_i = \frac{16}{\mathrm{Re}}\ (la\min ar\ \ flow) \qquad \text{------------- } (6)$$

$$f_i = 0.0035 + \frac{0.264}{\mathrm{Re}^{0.42}}\ (turbulent\ \ flow)$$

$$f_i = 0.0122\ \ (transition)$$

α_I = internal heat transfer co-efficient (W/m^2K)
λ = fluid thermal conductivity (W/m^2K)
d_i = inside tube diameter (m)
Θ = viscosity corection factor
Re = Renolds number
Pr = Prandl number
L = tube length (m)
Gr = Grashof number
Gz = Graetz number
ΔP_i = pressure loss inside tubes (Pa)
f_i = friction factor inside circular tube
m = mass velocity (kg/sm^2)
p = fluid density (kg/m^3)

Shell Side Properties

The shell side calculations were based on a model developed by Saunders [5].

Heat Transfer Co-efficient

$$h_o = (F_F\ F_P)(F_M\ F_C)(F_E\ F_A)\Theta_{nl} \quad \text{----------------- (7)}$$

Pressure Drop

ΔP_c = cross flow pressure loss in one pass

$$= \frac{(F_F\ F_P)(F_M\ F_C)}{\Theta_{nl}} \quad \text{----------- (8)}$$

ΔP_w = pressure loss in one window

$$= (F_F\ F_P)(F_M\ F_C)$$

$\Delta P_e = pressure\ loss\ in\ inlet\ or\ outlet\ cross\ pass$

$$= \frac{\Delta P_c X_e}{L_r^e}$$

$$where\ L_r = \frac{inlet\ or\ outlet\ space}{central\ baffle\ spacing}$$

$$e = 1.75\,(turbulent\ flow, triangular\ and\ rotated\ square\ pitch)$$

$$= 2.00\,(turbulent\ flow\ square\ pitch)$$

$$= 1.00\,(laminar\ flow)$$

$\Theta_{nl} = viscosity\ correction\ factor\ assuming\ no\ leakage\ at\ baffle\ plates$

$X_e = correction\ factor\ determined\ from\ tables$

$F_E, F_F, F_P, F_M, F_C, F_A$ *are all design factors which account for various aspects of shell side flow which are determined from tables developed by Saunders*(1988)

Total pressure side drop,

$$\Delta P_s = (N_b - 1)\Delta P_c + N_b \Delta P_w + \Delta P_{ei} + \Delta P_{eo} \qquad \text{----(9)}$$

$where\ N_b = number\ of\ segmental\ baffles$

$\Delta P_{ei}, \Delta P_{eo} = pressure\ loss\ at\ inlet\ and\ outlet\ cross\ pass$

The Algorithm

- Input shell side and tube side mass flow rates
- Input increment in mass flow rates required for the calculation
- Calculate upper and lower limits of mass flow rate variation on both shell side and tube side

For each increment in the tube side mass flow rate,

- calculate the Reynolds and Prandtl Numbers
- calculate the tube side heat transfer coefficient

- calculate the tube side pressure drop

For each increment in shell side mass flow rate

- calculate the cross flow mass flow rate and hence the Reynolds number
- calculate the shell side heat transfer coefficient
- calculate the shell side pressure drop

Using the respectful values for the mass flow rates on both the shell and the tube sides of the heat exchanger and selecting the increment, the thermal simulator then determines the heat transfer coefficients and the pressure drop for each of the increasing values of the mass flow rates between 80% and 120% of the design value input.

Simulation Results

The thermal simulator was used to analyze the heat exchanger designed by the INPRO Heat Exchanger Software. The shell side flow rate of 69.3kg/s and a tube side flow rate of 197kg/s were used. The results are shown in Appendix 2.

The results show an increase in shell side heat transfer coefficient and pressure drop with increasing flow rates. On the tube side, although the tube pressure drop increased with mass flow rate, the overall heat transfer coefficient remained the same.

Cost Analysis

Basic Costs

Using a shell internal diameter of 1295mm, the shell side design pressure of 30 bars and
the tube side design pressure 20 bars.

Indexed African Journals Online: www.ajol.info

$$C_b = 0.65C_{bt} + 0.35C_{bs}$$
$$where\ C_b = combined\ basic\ exchanger\ \cos t$$
$$C_{bt} = basic\ tube\ side\ \cos t = \$51500$$
$$C_{bs} = basic\ shell\ side\ \cos t = \$61500 \qquad \text{-------- (10)}$$
$$C_b = 0.65(51500) + 0.35(61500)$$
$$= \$55000$$

The above costs were based on a heat exchanger with tube outer diameter of 19.05mm and 25.4mm pitch with 6096mm long tubes. The designed heat exchanger varies slightly from this configuration (outer tube diameter of 25.4mm, 31.5mm pitch and 6096mm tube length) and as such, a cost factor is introduced to cater for the variation.

$$C_X = C_f L$$
$$where\ C_X = extra\ \cos t$$
$$C_f = \cos t\ \ factor\ \ per\ unit\ length = \$1550 \big/ m \qquad \text{---------- (11)}$$
$$L = tube\ length = 6.1m$$
$$C_X = (1550)(6.1)$$
$$= \$9455$$

The cost of the tubes per m^2 is based on whether the tubes are seamless or not. The tubes to be used are seamless. This is to enable the tubes withstand high pressures without rupturing. Also, because the tube ends are not to be welded, but force fitted to the tube sheet.

$$C_T = C_S A$$
$$where\ C_T = \cos t\ of\ tubes$$
$$C_S = \cos t\ of\, seamless\ tubes\ {}^{per}\!/_{m^2} = {}^{\$25}\!/_{m^2} \text{ -------- (12)}$$
$$A = heat\ transfer\ surface\ area\,(m^2) = 620m^2$$
$$C_T = (25)(620)$$
$$= \$15500$$

The total exchanger costs based on estimates is given as follows;

$$C_{1984} = C_b + C_X + C_T$$
$$= 55000 + 9455 + 15500 \quad \text{-------------------- (13)}$$
$$= \$79955$$

Assuming an inflation rate of 3% per year, the current cost, C would be around 150% of the 1984 estimates.

$$C = (1.50)(79955)$$
$$= \$119932.5 \quad \text{---------------------- (14)}$$

The cost of implementing this design was evaluated and compared with the annual operating costs of the current furnace set up and the payback time calculated as follows;
The annual cost of operating the fired furnace has been estimated at roughly \$250000 per year.

Payback time = $\dfrac{\cos t\ of\ constructing\ the\ heat\ exchanger}{annual\ \cos t\ of\ operating\ the\ furnace}$ ------ (15)

$$= \frac{119932.5}{250000}$$
$$= 0.48\ year$$

Conclusion

The possibility of replacing fired furnaces with heat exchangers has been shown for the Warri Refinery, provided that there are flow streams with enough thermal energy to give off heat that would normally be supplied by the furnace. The designed heat exchanger for the Refinery would have a heat transfer area of $620m^2$, a tube length of 6.1m and an outer shell diameter of 1.34m. The exchanger has the TEMA AES configuration with crude oil on the tube side and atmospheric residue on the shell side.

The heat exchanger configuration, when subjected to thermal simulations by the Thermal Simulator software, showed that increases in flow rate lead to increases in pressure losses on both the shell and the tube sides of the exchanger, as well as increases in the heat transfer coefficient on the shell side. It was also found that the heat transfer coefficient on the tube side remained constant. This could be attributed to the low Reynolds number of crude oil in the pipe flow.

Cost analysis showed that the heat exchanger would have an estimated cost of fabrication of about $120000 as compared with the $250000 yearly cost of fuel required to run the fired furnace. This gives an estimated payback period of close to six months.
The actual cost would be definitely higher considering the cost of shipping parts from overseas, cost design consultancy and labour. Also, there could be an extra cost for replacements and for rerouting pipelines.

References

Connors H.J. (1970). Fluid Elastic Vibration of Tube Arrays Excited by Cross-Flow. Proceedings of Symposium on Flow-Induced Vibration in Heat Exchangers. *American Society of Mechanical Engineers Annual General Meeting*, pp 42-56.

Engineering Sciences Data Unit International Limited (ESDU) (1973). *Data Item 73031*.

Erksine J. and Waddington W.(1973). A Review of Some Tube Vibration Failures in Shell and Tube Exchangers and Failure Prediction Methods. *International Symposium on Vibration Problems in Industry*, Keswick, UK

Govind R., Mocsny D., Corson P. and Klei J. (1986). Exchanger Network Synthesis on Microcomputer. *Hydrocarbon Processing* , pp 53-57.

Linhoff B. and Hindmarch E. (1983). The Pitch Design Method of Heat Exchanger Networks. *Chemical Engineering Science. Vol. 38, No5*, pp 745-763.

Pallen J. (1998). Heat Exchangers, Vaporizers and Condensers, *Mechanical Engineering Handbook*, Kutz, M. (eds)., John Wiley and Wiley and Sons Inc.

Rabiu M.A. (1998). Optimization of Energy Recovery of Crude Distillation Unit of Warri Refinery. Unpublished M.Sc. Thesis, Obafemi Awolowo University, Ile-Ife.

Saunders E.A. (1988). *Heat Exchangers: Selection*, *Design and Construction*. Longman Scientific and Technical, UK

Taborek J., Aoki T., Ritter R.B., Palen, J.W. and Krudsen J.G. (1972a). Fouling: The Major Unresolved Problem in Heat Transfer Part I. *Chemical Engineering Progress, Vol. 68, No. 2*, pp 59-67.

Taborek J., Aoki T., Ritter R.B., Palen, J.W. and Krudsen J.G. (1972b). Fouling: The Major Unresolved Problem in Heat Transfer Part I. *Chemical Engineering Progress, Vol. 68, No. 7*, pp 69-78.

Thorngen J.T. (1970). Prediction of Heat Exchanger Tube Damage. *Hydrocarbon Processing* pp 129-131.

Tubular Exchanger Manufacturers Association Inc(1978). *Standards of Tubular Exchanger Manufacturers' Association*, 6th Edition. TEMA, New York.

Appendix 1: Heat Exchanger Data Sheet

Fluid Properties	Shell Side	Tube Side
Fluid Material	Atmospheric Residue	Crude Oil
Flow Rate (Kg/s)	69.3	197
Inlet Temperature (deg.C)	345	224
Outlet Temperature (deg.C)	269.6	252
Average Specific Heat Capacity (J/Kg k)	3410	3600

Exchanger Physical Attributes

Type	Floating Head (Split Backing Ring)
Service	Preflash Column Feed Preheater
Heat Duty (MW)	18.95
Log Mean Temprature Difference (deg.C)	66.5
Surface Area (sq. meters)	620
Shell Internal Diameter (mm)	1295
Number of Shells	1
Number of Tube Passes per Shell	2
Baffle Type	Segmental
Baffle Spacing (m)	0.52
Number of Tubes per Shell	1274
Tube Length (m)	6.1
Tube Outer Diameter (mm)	25.4
Tube Pitch (mm)	31.5
Tube Pattern	Triangular (30 degrees)
Material Of Construction	Carbon Steel
TEMA Class	AES

Appendix 2: Performance Results for Design Flow Rates.

Tube Side

Mass Flow Rate Kg/s	Heat Transfer Coefficient W/m2 K	Pressure loss N/m2
157.6	1039.73	25469.23
160.6	1039.73	25954.05
163.6	1039.73	26438.87
169.6	1039.73	26923.69
172.6	1039.73	27408.51
175.6	1039.73	27893.33
178.6	1039.73	28378.15
181.6	1039.73	28862.97
184.6	1039.73	29347.79
187.6	1039.73	30317.79
190.6	1039.73	30802.25
193.6	1039.73	31287.07
196.6	1039.73	31771.89
199.6	1039.73	32524.18
202.6	1039.73	33509.21
205.6	1039.73	34508.94
208.6	1039.73	35523.35
211.6	1039.73	36552.47
214.6	1039.73	37596.27
217.6	1039.73	38654.77
220.6	1039.73	39727.96
223.6	1039.73	40811.86
226.6	1039.73	41918.44
229.6	1039.73	43035.72
232.6	1039.73	44167.69
235.6	1039.73	45314.37

Shell Side

Mass Flow Rate (Kg/s)	Heat Transfer Coefficient (W/m2 K)	Pressure Loss (N/m2)
55.44	1652.91	20539.66
58.44	1707.4	23625.54
61.44	1760.78	25788.64
64.44	1813.17	28052.64
67.44	1864.63	30356.01
70.44	1915.21	31758.41
73.44	1964.97	35238.8
76.44	2013.97	37796.17
79.44	2062.21	40430.36
82.44	2109.76	43140.02