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Analysis of a Condenser in a Thermal Power Plant for possible augmentation in its heat transfer Performance

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Abstract - The Surface Condenser in Dr.NTTPS is modeled and numerically analyzed in SOLIDWORKS Flow Simulation. The temperatures obtained from the numerical analysis of the modeled condenser closely match with those of the working temperatures in the thermal power plant, which validates the model. Flower baffles are installed in the design of the shell-and-tube heat exchanger in order to improve the heat transfer performance. Three Sector angles 75^0, 90^0 and 115^0 are considered for the analysis. The temperature distribution of the steam on the shell side is represented in the form of flow trajectories. The correlation of the overall heat transfer coefficient with varying angle is identified and is found that with increasing sector angle of flower baffles, the overall heat transfer coefficient increases. A comparative study of the condenser integrated with flower baffles to that of the condenser with segmental baffles is done. The effectiveness of the condenser is found to have increased with increasing baffle angles due to the induced turbulence. The heat transfer performance of the condenser with segmental baffles is higher than that of the condenser with flower baffles, but segmental baffles cause large pressure drop. The heat transfer coefficient on the shell side is calculated analytically and compared graphically with other condenser configurations.

Keywords: condenser, shell-and-tube heat exchanger, heat transfer performance thermal power plant.

I. INTRODUCTION

A heat exchanger is a complex device that provides the transfer of thermal energy between two or more fluids, which are at different temperatures and are in thermal contact with each other. Heat exchangers are used either individually or as components of a large thermal system, in a wide variety of commercial, industrial and household applications, e.g. power generation, refrigeration, ventilating and air-conditioning systems, process, manufacturing, aerospace industries, electronic chip cooling as well as in environmental engineering. The improvements in the performance of the heat exchangers have attracted many researchers for a long time as they are of great technical, economical, and not the least, ecological importance. Performance improvement becomes essential particularly in heat exchangers with gases because the thermal resistance of gases can be 10 to 50 times as large as that of liquids which requires large heat transfer surface area per unit volume on gas side. The traditional methods of reducing the air-side thermal resistance are by increasing the surface area of the heat exchanger, or by reducing the thermal boundary layer thickness on the surface of the heat exchanger. Increasing the surface area is effective but it results in the increase in material cost and increase in mass of the heat exchanger. One of the methods to reduce boundary layer thickness is by the generation of passive vortices. In this technique the flow field is altered by an obstacle to generate a vortex oriented in the direction of the flow. The resulting change in the flow due to an obstacle alters the local thermal boundary layer. The net effect of this manipulation is an average increase in the heat transfer for the affected area. The present work is undertaken to compute the heat enhancement levels achievable in a plate-fin heat exchanger built-in vortex generators mounted on these fins in the form of rectangular winglets.

II. SYSTEM MODEL

Condensers

Condenser is a heat exchanger in which there is a phase change of vapor into liquid (condensation) and the heat rejected is absorbed by the cold fluid. Condensers may be classified into two types based on whether the condensate and coolant are separated by a solid surface or if they are mixed together. Those Condensers in which the condensate and the coolant remains as two different streams are subdivided into three main types: air-cooled condenser, shell-and-tube condenser and plate condenser. Steam turbine exhaust condensers are also called as surface condensers. Surface condensers are shell-and-tube condensers with cross flow arrangement..

Effectiveness of Condenser:

Effectiveness of a heat exchanger is defined as the ratio of actual heat transferred for the given area of the HX (Qactual) to heat transferred when there is infinite heat transfer area (Qmax).

Or

It is the fraction of the maximum heat transferred through an infinite heat transfer area. Effectiveness of an heat exchanger is given by following expression in terms of number of transfer units(NTU).

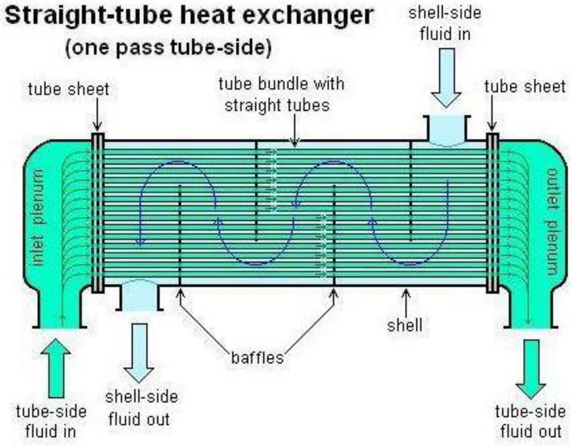

Fig1. Shell and Tube Heat Exchanger

III. PREVIOUS WORK

Xiang-hui Tan et.al[5] has performed numerical simulation on twisted oval tube heat exchanger. The heat transfer performance on the shell side of the heat exchanger under the influence of the geometrical parameters including twisted pitch length(P) and aspect ratio(A/B) is studied. The results reflect that both the Nusselt number and friction factor increase with increasing value of P and A/B. Considering only the influence of P alone, it firstly increases with increasing P and then decreases. It is concluded that the overall heat transfer performance on the shell side increases with increasing A/B. Spiral flows are found on the shell side when A=14mm, B=5mm. The spiral flow intensity becomes drastic with increasing A/B and decreasing P.

Experimental study for enhancing the heat transfer in a heat exchanger installed with helical baffles through blockage of triangle leakage zones is done by Simin Wang et.al.[6]. Fold baffles were used to block the triangle leakage zones between adjacent plain baffles. Although the pressure drop on the shell side increases, the pumping power penalty associated with the pressure drop is very low compared to the increment in heat flux..

The paper presented by Jian-Feng Yang et al.[7] explains about the use of sealing strips to eliminate the bypass stream that exists between the tube bundle and shell. Shell and tube heat exchangers with discontinuous helical baffles (DCH-STHX) and continuous helical baffles (CH-STHX) with the same helix angle of 400 are investigated by employing sealing strips. It is concluded that, larger the width of sealing strips, the better is the heat transfer performance and also the sealing strips are more effective in improving the heat transfer characteristics of CH-STHX than that of the DCH-STHX, particularly at large mass flow rates.

IV. PROPOSED METHODOLOGY

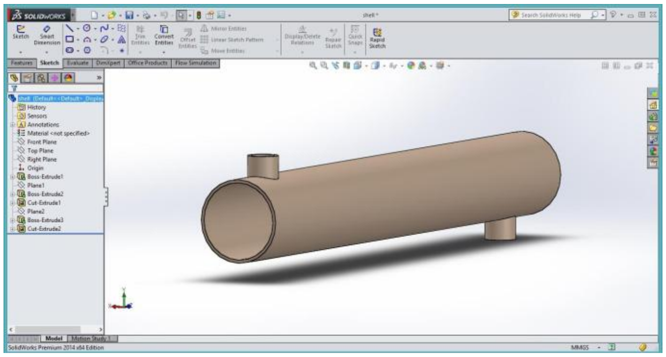

Fig 2 : Shell of the Condenser

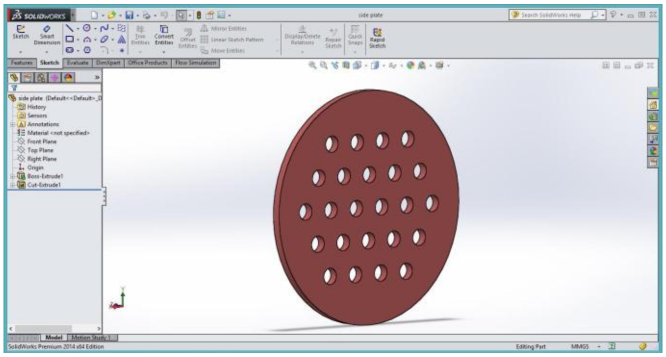

Fig 3: End Plate which acts as tube sheet

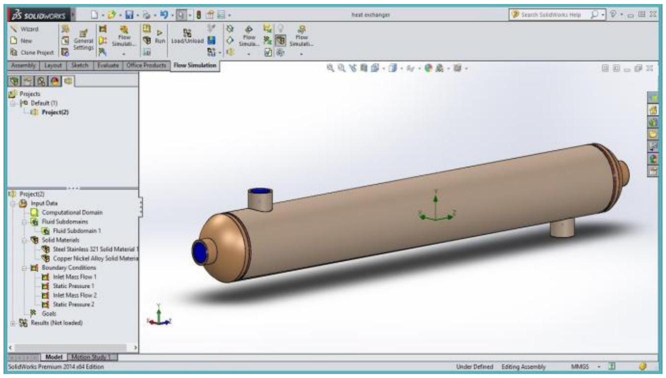

Fig 4: Condenser Model Overview

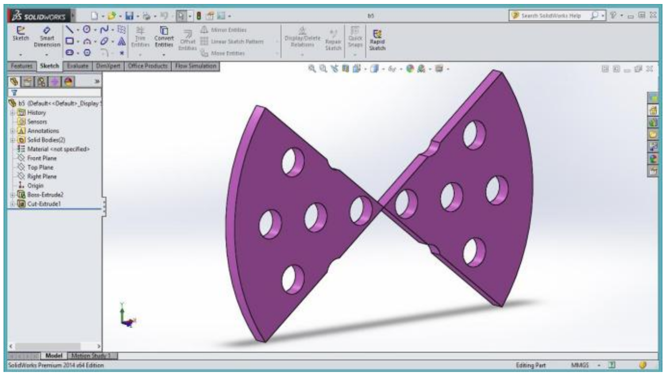

Fig 5: Baffle with 75^0 sector angle

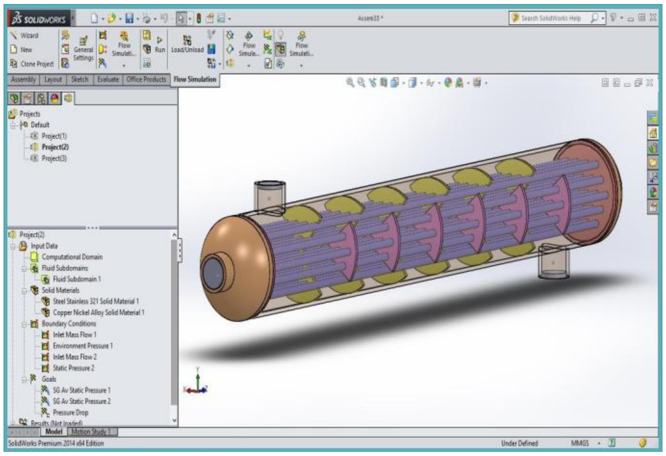

Fig 6: Condenser installed with 75^0 baffles

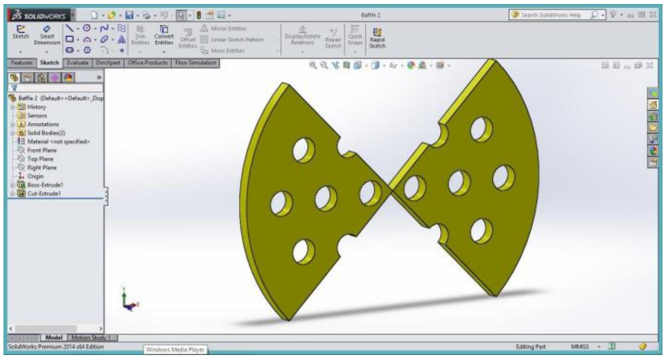

Fig 7: Baffle with 90^0 sector angle

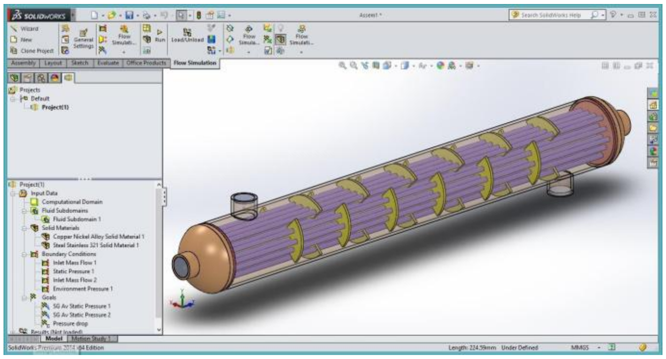

Fig 8: Condenser installed with 90^0 baffles

V. SIMULATION/EXPERIMENTAL RESULTS

Meshing:

The initial mesh settings are set to automatic. The initial mesh size is then set to 6. The generated mesh consists of 3D tetrahedral solid elements. Solidworks adopts autorefinement of the cells based on the complexity of the problem during analysis. The generated mesh of the condenser model is shown in figure.

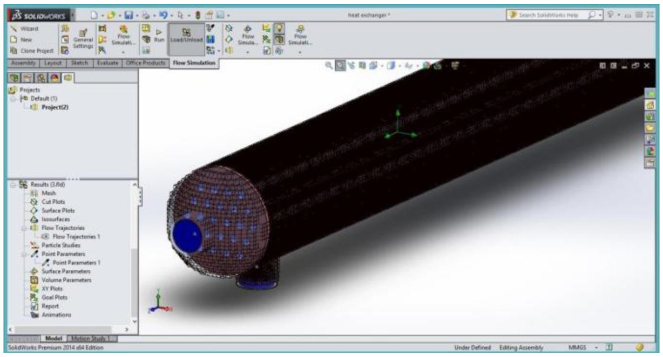

Fig 9: Tetrahedral Solid mesh of the condenser model

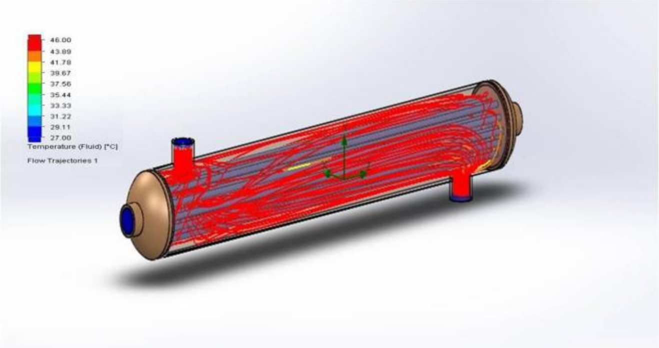

Fig 10: Temperature distribution in the condenser with no baffles

It can be seen from the above figure that the flow remains turbulent with no variation of temperature along the flow path of the steam. The red colored stream lines indicate that the steam at 46^{o}C loses its latent heat of vaporization and changes its phase. The condensate obtained is also at 46^{o}C. The cooling water which enters the tubes at 27^{o}C absorbs all the heat rejected by the steam with a temperature rise of 10^{o}C.

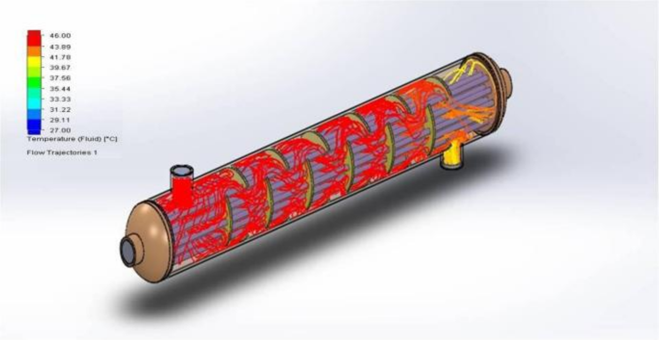

Fig 11: Temperature distribution in the condenser with 75^0 baffles

The flow trajectories representing the temperature distribution of the steam shows that the temperature remains constant, i.e. 46^0C until the steam loses its latent heat of vaporization and the condensate further undergoes sub cooling due to the available heat transfer area. The effect of heat rejected during sub cooling is reflected on the rise in temperature of the cooling water.

Condenser with 90^0 baffle Angle:

The condenser with 90^0 baffle angle has a peculiar characteristic from the design point of view. As the baffle angle is 90^0, the horizontal and vertical baffle set together covers the entire 3600 flow area.

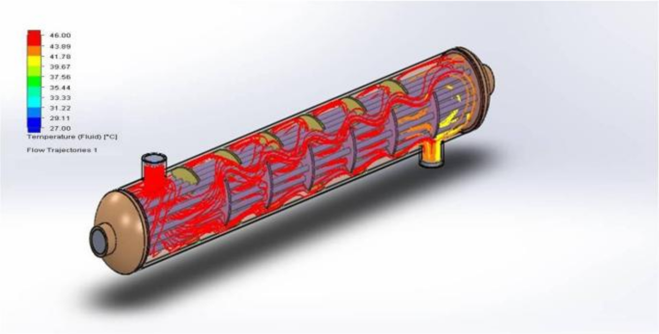

Fig 12: Temperature distribution in the condenser with 90^0 baffles

Analytical Results:

For 75^0 baffle angle:

All the formulae are taken from the reference.

The amount of heat exchange between steam and cooling water can be determined from the conservation of energy principle.

Heat Lost by Steam = Heat gained by cooling water

Heat Lost by Steam, **Qs** = m* x L.H + m* x Cp x (Ts – To)

m* = 0.18 kg/s ;

Latent Heat of Steam at 10 kPa = 2392.5 kJ/kg

Cp = 4.2 kJ/kg-K;

Saturation Temperature at 10 kPa = 46 0C

Qs = 0.18 x 2392.5 + 0.18 x 4.2 x (46 – 43)

= **432. 918 kW**

Heat gained by cooling water, **Qcw** = m* x Cp x (to – ti)

m* = 10.25 kg/s; Cp of Water = 4.2 kJ/kg-K

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Qcw = 10.25 x 4.2 x (37.056 – 27)

= **432.9 kW.**

Overall Heat Transfer Coefficient(OHTC), U:

The heat transfer in the heat exchanger can also be expressed in terms of mean heat transfer coefficient and mean temperature difference by the following relation.

Q = Uo x Ao x LMTD = Ui x Ai x LMTD

Heat Transfer Area on Shell Side, Ao = n x Π x do x L

where

n, number of tubes = 24 ;

do, Outer diameter of tube = 19 mm = 0.019 m ;

L , Length of tube = 10 m

ΔT1 = Th,in – Tc,out = 46 – 37.056 = 8.944 0C

ΔT2 = Th,out – Tc,in = 43 – 27 = 16 0C

LMTD = (8.944 – 16)/ ln(8.944/16)

= 12.132 $^{\mathbf{0}}$**C.**

Q = Uo x (24.Π.0.019.10) x (12.132)

432.918 = Uo x 14.326 x 12.132

Uo = 2.491 kW/ m^2 - K

The effectiveness of counter flow heat exchanger is calculated from the following relation

Effectiveness for Condenser,

NTU, Number of Transfer Units = UA/ Cmin

NTU = 2.491 . 14.326/ 10.25. 4.2 = 0.828.

Substituting NTU in Effectiveness Equation, we obtain

ϵ = 0.56

Shell-side heat transfer coefficient:

Reynolds Number for the flow on tube side:

(4.1) ρ**,** density of cold water = 1000 kg/m3 .

v, velocity of cold water = 1.88 m/s. d, inner diameter of tube = 0.017 m.μ, dynamic viscosity of cold water = 0.862 cP

Re = 37,107.9 Hence, the flow is turbulent.

From the Dittus-Boelter Equation Pr, Prandtl number,

Cp , Specific heat of water = 4.2 kJ/kg-K

K, Thermal Conductivity of Water = 0.58 W/m-K

Pr = 0.862 * 10-3 * 4200 / 0.58 = 6.2

Therefore, substituting the respective values in the Dittus-Boelter Equation

ht = 6137 W/m2-K .The effectiveness of the condensers with different baffles is tabulated as below:

Table 4.6: Effectiveness of different condenser models

Condenser Model	**Effectiveness**
With no baffles	0.55
Flower baffles with 75^0 angle	0.5567
Flower baffles with 90^0 angle	0.6071
Flower baffle with 115^0 angle	0.6413
With segmental baffles	0.6618

The effectiveness of the condensers with baffles is found to be more than that of the condenser with no baffles. This improvement in the performance of the condenser is due to the turbulence created by the baffles in the fluid flow. The baffles provide wavy flow to the fluid stream on the shell side, which causes more heat transfer between the cooling water and the steam.

The variation of effectiveness with varying sector angle of flower baffles is plotted along with effectiveness of condenser without baffles and with segmental baffles. The comparison of effectiveness of the condenser with flower baffles, segmental baffles and with no baffles can be observed on the plot between effectiveness Vs baffle angle.

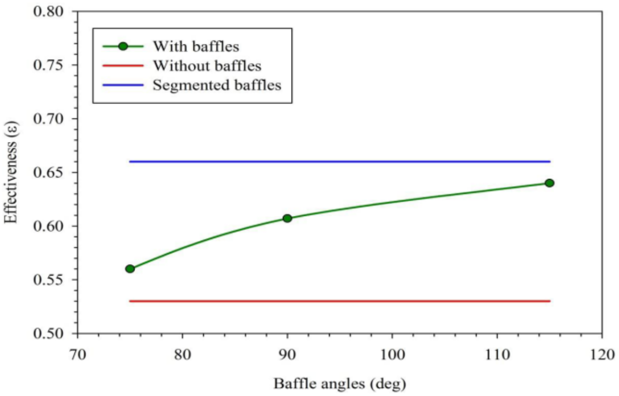

Fig Variation of effectiveness of condenser with baffle angle

From the plot, it is observed that the effectiveness of the condenser without baffles is the lowest and the effectiveness of the condenser with segmental baffles is the highest. Thus, these two values remain as the limiting cases for the flower baffles. The effectiveness of the condenser with flower baffles lies in midway between the limits. As the baffle sector angle increases, the effectiveness increases.

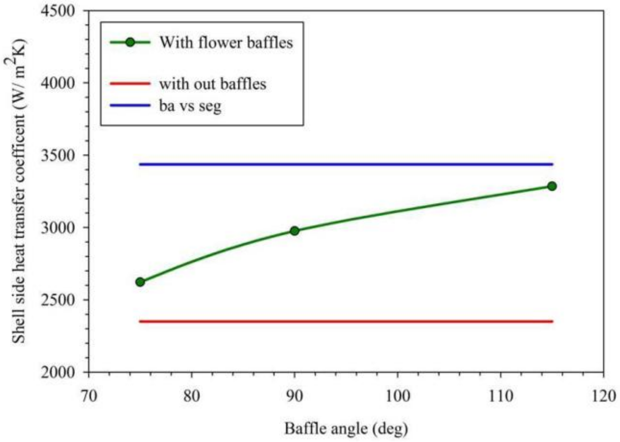

Fig 13 Variation of shell-side heat transfer coefficient with baffle angle

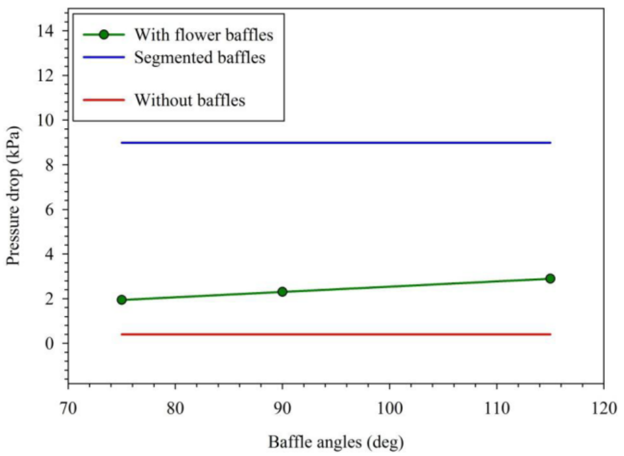

Fig 14 Variation of shell-side fluid pressure drop (Δp) with baffle angle

VI. CONCLUSION

The mass flow rate of cooling water is much higher than that of the exhaust steam flow rate. A mere observation of the above results suggests that after the steam has been condensed, due to high mass flow rate of the cooling water, the condensate is expected to sub cool enormously, but however the maximum drop of temperature obtained is 100C. This is because of the change in specific volume of the steam. As the specific volume of the condensate water is much less than that of steam, condensate is collected at the bottom and is not exposed to all the tubes for heat exchange. Therefore the change in temperature of condensate water is less than expected. A condenser is designed only to condense steam by absorbing the latent heat of vaporization. The insertion of baffles causes the condensate to be sub cooled, which is not desirable. In order to avoid the sub cooling, the mass flow rate of the cooling water can be reduced so that the steam condenses at its saturation 58^0c temperature. Heavy duty pumps are required to pump the water at high flow rates from the river. As the flow rates can be reduced, less pumping power is required and hence the use of baffles also adds to an economical advantage.

The pressure drop on the shell side of the condenser with flower baffles is found to be much less than that in condenser with segmental baffles. Also, the thermal performance of condenser with flower baffles is close to that of the condenser with segmental baffles. Hence, flower baffles can be preferred to be installed in a condenser than the segmental baffles.

The manufacture of the flower baffles is also easier and in fact, less material is required for the manufacture. Hence, the material cost can be reduced. As the baffle angle increases, the flower baffles offer more flow resistance, which may even reduce the heat transfer performance. This may cause more pressure drop just as in case of segmental baffles. The flower baffles with higher sector angles can be considered as segmental baffles. Therefore, with increasing sector angle of flower baffles, the thermal performance of the condenser gets reduced.

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