International Conference on Ideas, Impact and Innovation in Mechanical Engineering (ICIIIME 2017)
Volume: 5 Issue: 6

ISSN: 2321-8169
26 - 32

A Review on Performance Analysis of an Air Cooled Condensers to Improve Heat Transfer Rate

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Abstract: Significant amount of power production comes from thermal power plant and in thermal power plant mostly water cooled condenser is used which consumes significant amount of water and due to limited fresh water availability particularly in water stress countries, application of air cooled condensers (ACCs) growing rapidly but plant with ACC suffers a penalty of 5-6% drop in efficiency. Thus, to mitigate the drop-in efficiency, performance of air cooled condenser can be improved by increasing heat transfer coefficient, back pressure of turbine and fan loading. Back pressure of turbine is dependent on ambient condition, so one of the prominent way is to increase heat transfer coefficient and decrease the fan loading, but in this review paper it is mainly focused on various work carried out till now on ACC to increase the heat transfer which mainly depends on different parameters like fins type, pitch, length been carried out in different fins to increase the efficiency of ACCs, by numerical analysis and experimental procedure. Second part of the paper discusses various work which has been carried out in studying different tube shape and layout and their subsequent effects on pressure drop and their net effect on overall performance of ACC assessed.

Keywords*:* *Air Cooled Condenser, Fins, Tubes* .

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1. Air Cool Condenser Description.

The air-cooled condenser consists of a set of A-frame capacitor cells each equipped with an axial flow fan. A single ACC cell has finned tubes arranged in parallel along the inclined walls of the A-frame unit. The ACC cell models use a single row of finned tubes, each tube consisting of a rectangular steel channel To carbon with aluminium fins. The steam enters the ACC cell through the large vapour duct and condenses as it flows down the inclined tubes forming the walls of the frame A which is then collected in a condensation line at the bottom. A typical ACC cell has a 12 × 12 m cavity, with finned tubes from 9 to 12 m long and a tip angle of 60 degrees. Each finned tube has an approximate dimension of 25 × 190 mm, with fins 25 mm high. The air is drawn through the tubes and fins by large fans with an axial flow of about 9 m in diameter. ACC capacitors are generally located 20-50 m above ground level and surrounded by wind walls to reduce the impact of wind and potential recirculation of air. The diagrams of an assembly (a) ACC, (b) ACC cell, and (c) an individual vane steam tube are shown in Figure 1.

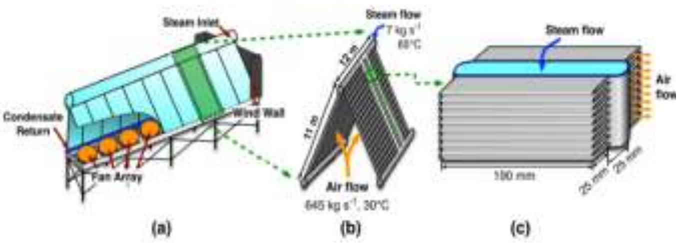

Fig-1. Air Cooled Condenser.

2. Air Cool Condenser Challenges.

Air-cooled condensers generally require very high capital investment than wet-cooled condensers because thermal capacity of air is much lower than water and thus the setup includes heat exchangers of large size, huge fin areas & require support structures. For the above said reason installation and operational costs are around 3.5 to 5 times more than wet cooling systems for ACC systems. In order to minimize the installation, operational cost and to improve efficiency heat transfer rate has to be increased.

Thus to increase the heat transfer rate initial temperature difference ITD = ($T_{steam,in}$ – $T_{air,in}$) has to be increased, increase in steam inlet temperature leads to increase in condensation turbine back pressure in-turn the steam cycle efficiency and power output are reduced drastically by 5-6% in comparison to wet cooling system. Statistical data concluded that increase in ITD about 3 K results in reduction of power generation by 1.1%.

Next crucial challenge is in summer condition when the power requirement is high & the ambient condition with rise in temperature in summer increase steam condensation pressure leading to reduced power output. Even in condition with high wind speed results in increase in inlet temperature of the air in the suction side of fan reason behind this is recirculation and thus the heat transfer rate gets decreased, again even with cross flow wind leads to increase in blade loading which leads to high horse power consumption.

3. Literature Review

3.1 Fins

Vanfossen and Brigham [1] studied the heat transfer improvement by using cylindrical pin fin by varying fin height to pin fin diameter ratio and number of rows in a staggered way. The author concluded that with the use of cylindrical pin fin with h/D ratio of 4 heat transfer gets increased by 35%.

Tian et al. [2] carried out the study considering the wavy fin-and-tube heat exchanger with delta winglets to investigate characteristics of the air-side heat transfer and fluid flow. The results show that by using delta winglets, longitudinal vortices were generated which enhances heat transfer performance with pressure drop as penalty. Oblique arrangement of the wavy fins, finned tubes were more prone to fouling in ambient conditions. The fouling is also a key factor which reduces heat transfer performance of finned tube heat exchangers.

Feng et al. [3] studied various fin structure and concluded that the discontinuous short wave fin enhances heat transfer by periodic generation of boundary layer.

Bell and Groll [4] conducted study to test heat transfer performance and air-side pressure drop of plate-fin and coils having micro channel under clean as well as fouled conditions. Results showed that micro channel coil with louvered fin and fin pitch below 2.0 mm were prone to fouling compared to larger fin pitch for louvered heat exchangers.

Lijun Yang et al. [5] modelled new wave-finned flat tube bundle where fin surface was rotated to perpendicular to prevents fouling problem. The characteristics of the pressure drop and heat flow rate of the orthogonal and oblique finned tubes with the air velocity at the minimum flow area were compared which shows that orthogonal finned tube has higher heat transfer rate than the oblique finned tube arrangement but at the same time with the penalty of increase in pressure drop, since the length of the orthogonal finned tube was almost twice that of oblique one and hence pressure drop gets increased. Effects of fin pitch [5] were discussed which shows how friction factor (f) varies with the non-dimensional fin pitch at different Re. It was observed [5] that friction factor decreases with increase in fin pitch at low Reynolds number reason behind these characteristics was as the fin pitch broadens leads to enlargement of the equivalent hydraulic diameter of passage of fluid flow. Approximately the friction factor inversely proportional to the hydraulic diameter which results in the reduced friction factor with increasing the fin pitch at lower Reynolds number. As Nusselt number gets significantly increased with increase in fin pitch that leads to lesser number of fins to be used because of larger fin pitch, which means less heat transfer area. Effects of fin height [5] were also discussed by the author and the results obtained showed that friction factor had no change with increasing fin height at high Reynolds number because friction factor was almost constant for turbulent flow. The buoyancy effect caused due to the natural convection leads to decrease in pressure drop. Finally, the new and original model were compared and results for various fin pitches, fin height were obtained. The friction factor gets decreased but the Nusselt number increases with increased in fin pitch. The total heat flow rate of an entire finned tube was obtained at a maximum value of 1.5 mm.

Mousa [6] studied influence of heat transfer and friction factor in a horizontal rectangular channel containing pin fins. Author concluded that average Nusselt number gets increased as gap rate with pin fin and distance between pin fin was decreased.

Xiaoze Du et al. [7] used Punched Longitudinal Vortex generators to increase air side heat transfer on the wavy fin surface of flat tube used in Air Cooled Condenser, using numerical simulation four different types of longitudinal Vortex generators were studied with different attack angles. Particle Image Velocimetry (PIV) technique was used for experimental study in wind tunnel to obtain flow field visualization. Here four types of longitudinal vortex generators were employed, viz. (1) delta wing, (2) rectangular wing, (3) delta winglet pairs and (4) rectangular winglet. Xiaoze Du et al [7] performed the heat transfer and flow characteristics by considering the wavy finned flat tube without longitudinal vortex generator. Temperature distribution for the fin without longitudinal vortex generators at u_{in} = 2 m/s reveals that the at the leading of fin temperature was lower in comparison to that of the other regions. With one delta winglet pair mounted on the middle section of the fin the region of low temperature gets extended on the fin surface that in-turn leaded to the improvement in heat transfer performance. Boundary-layer separation from the fin gets delayed due to the generation of longitudinal vortices developed from one delta winglet which accelerates the local flow velocity and thus intensifies the mixing of cold fluid and hot fluid which leads to increase in heat transfer [7]. The effect of different types of vortex generators and location of delta winglet pairs were obtained in this paper to establish the relationship of their effects on heat transfer coefficient and pressure drop. With increase in attack angle and frontal velocity for different types of longitudinal vortex generator heat transfer gets increased but at the same time pressure drop gets also increased. For each of the four type of vortex generators performance evaluation criteria (PEC) was calculated to establish the fact which show the best performance with minimum pressure drop and found it to be greater than 1 for each of them but main point which was made out of this calculation was that the delta winglet pair having higher value. Thus from the paper [7] it was concluded [7] that the delta winglet pairs were the best longitudinal vortex generators for enhancement of heat transfer on air side. The heat transfer enhancement was influenced by the longitudinal vortex generators which was located on the fin surface and it was concluded [7] that the effectiveness of the delta winged pairs punched in the middle of the fin surface was much higher than those punched ahead and in the rear. With increase in number of delta winglet punched on fin surface air-side heat transfer coefficient tends to increase but the pressure drop also gets increased [7].

Caliskan [8] in his experimental study used triangular and rectangular vortex generators obtained from longitudinal winglet. Both the vortex generators were compared with smooth surface and from the results obtained it was seen that better heat transfer performance was obtained from triangular vortex generators.

Agrawal et al [9] analysed heat transfer and speed changes in a rectangular channel which contains an oscillating surface known as agitator. Oscillation empower improvement in heat transfer by flow mixture and provides more turbulence but in an adverse effect was it led to loss of momentum.

Chao Xu et al. [10] studied three different enhanced fin structures viz. discontinuous fin, staggered fin and fin with vortex-generator were employed and compared with original continuous wavy fin. From the experimental analysis heat transfer features of the enhanced wavy fins were observed by understanding the temperature distribution on surface of fins and compared with that of the original continuous wavy fin. For finding out the temperature distribution infrared thermography technique was used and temperature distributions of fin surfaces were obtained. The different configurations used in this experimental analysis paper [10] such as discontinuous fin, staggered fin, fin with vortex generation and original type continuous wavy fin as base geometry. It was observed that as Reynolds number increases the efficiency of different structured fins tends to decrease. Figures 8(a) and 8(b) show the variation of heat transfer coefficient with respect to Reynolds number and Variation of pressure drop with respect to Reynolds number respectively for different types of fins. It was observed [10] that vortex-generator type and Staggered type both transfers more heat than that of continuous wavy fin.

$$PEC = \frac{(Nu_{enh}/Nu_{ori})}{(f_{enh}/f_{ori})^{1/3}}$$

Equation [10] shown above was used to obtain PEC for different enhanced structures. Figure.9 shows the relation between the PEC and Reynolds Number. It was observed by Chao Xu et al. [10] that for all three-staggered fin, fin with vortex generator and discontinuous fin types the value of PEC was greater than the 1 till the value of Reynolds no was 12000 beyond that for discontinuous fin the value decrease with increase in Reynolds number and from the fig. 9 it was also seen that Fin with vortex generator was having a highest value. As per Chao Xu et al. [10] heat transfer performance increases for discontinuous fin and staggered fin because air flow gets interrupted due to different row structure that causes the flow and thermal boundary layer regeneration. Similarly, for vortex-generator fin type, the vortex disturbs air flow and thus heat transfer gets increased. The best performance was observed nearest location to the vortex-generator but gradually diminishes along direction of air flow. On average, it was observed that discontinuous type, staggered type and vortex-generator type, Nusselt number increases by 11.29%, 28.61% and 56.46%, and the friction factor gets increased by 3.23%, 66.26% and 48.58% [10].

Ankur Kumar et al. [11] studied thermal hydraulic characteristics in terms of heat transfer per unit pumping power by considering annular finned tubes considered as the simplest type of fin used. Observation showed that heat transfer coefficient tends to increase with increase in fin spacing but pressure drop tends to decrease about (60-80%) considerably. Parameters considered for study were fin spacing 2-10 mm, fin height 5-10 mm. Effects of fins was studied considering 2 tube rows, at various fin heights 5, 7 and 10 mm at transverse tube pitch = 36.8, 40.8 and 46.8 mm respectively. The effect of fin spacing with respect to (A) heat transfer coefficient (B) pressure drop was studied and it was seen that the value of heat transfer coefficient was lowest at minimum fin spacing (2 mm) for all the fin heights. Reason behind this was that mass flow rate was minimum at lowest fin spacing and thus the heat transfer was low, at the same inlet velocity. It was observed that as the fin spacing was increased, the area requirement also gets increased. So, if fin spacing is increased from 2 mm to 10 mm, space requirement tends to increase by 150% for respective fin diameters. In this study [11] it was concluded that by increasing the fin spacing there was an increase in heat transfer coefficient and pressure drop gets decreased as penalty for different fin diameters. Hence from the above paragraph we can establish a fact that larger fin spacing should be used while designing the air-cooled condenser but at the same time installation cost will go high. Fin efficiency was also calculated and variations was seen with respect to fin spacing. Variation of fin efficiency with fin spacing for various fin heights of (5 mm, 7 mm and 10mm) was done and it was seen that fin efficiency decrease with the increase in the fin height. The main reason behind this was as fin height increases, there was major decrease in conduction heat transfer, which in return lowers the average fin temperature. It was concluded that optimal parameters for fin spacing was 3-5mm and fin height to be (5) mm for better performance of a condenser.

Fei Duan et al. [12] studied how fin spacing, wave spacing, wave amplitude and Reynolds Number had their influence on the heat transfer performance and the flow characteristics for wavy finned flat tube heat exchanger. They concluded in comparison to straight fin intermittent wavy fins got higher heat transfer. It was seen that longitudinal vortices enhance the heat transfer but transverse vortices had adverse effect on the heat transfer. Effect of fin spacing on flow and heat transfer was studied considering the wave spacing as 9.5 mm and fin spacing as 2.4 mm, 2.6 mm, 2.8 mm, 3.0 mm and 3.2 mm. The effect of average Nusselt number and friction factor as functions of Reynolds number were shown in Fig.12. It was observed from Fig 12 that as fin spacing was increased Nusselt number tends to increase. Nusselt number tends to increase as Reynolds number increases but friction factor f decreases with increase in Reynolds number. It was observed that even at lower Reynolds number for different fin spacing intensity of longitudinal vortices (Se_{ml}) were nearly the same. Second point which was observed is that characteristics of longitudinal vortices Se_{ml} at the different cases of fin spacing (T_p) increases with increase in Reynolds number Re. The change in fin spacing had major role on the intensity of transverse vortex in the span direction (Se_{ms}). Author [12] here made some points that good heat transfer performance was obtained if the fin spacing, the wave spacing and the wave amplitude were chosen to be 3.0 mm, 9.5 mm and 2.3 mm, respectively.

M.Eren et al [13] studied the effect of grooved pin fins in rectangular channel on heat transfer and friction factor using Taguchi method. Study considered experimental investigation the effect of cylindrical grooved pin fins and triangular grooved pin fins into rectangular channel on heat transfer, friction factor and Nusselt number. Design parameters considered in this study were Reynolds number, grooved distance rates, array types, pin fin geometry and flow direction. Working fluid used in this experimental analysis was air [13]. In this study grooved fins were placed inline as well as staggered arrays arrangement. There were many ways of improving heat transfer, they were active and passive methods. Active method generally consumes more power, so passive methods was the most preferable. Passive method includes modification of flow structure by creating surface roughness, vortex generators or adding pin fins thus it increases surface area and turbulence in the flow which enhances the heat transfer. Here in this paper emphasizes mainly on increasing fluid solid contact area, generation of secondary flow, disruption of thermal boundary layer development and to improve flow mixing. The experimental setup consists of a flow regulator, a fan, an entrance section, thermal camera and devices which were used to measure flow rate, temperature and pressure difference. Variable speed fan was used to force the air through test section. The dimensions of channel inner cross section were 100 mm (wide) and 25 mm (height). The length of the channel was 2500 mm which had three sections: calm section (1723 mm), test section (277 mm) and exit section (500 mm). Plexiglas material with thickness of 10 mm was used for the channel. The dimensions of the heating plate were 100 mm (wide), 277 mm (length). The heating surface was made up of stainless steel foil having thickness 0.02 mm. Two types of pin-fins used were C-GPFs and T-GPFs made up of an aluminium material having much higher thermal conductivity. Both types of pin fins having the same height (h) and bottom area were used. Experiments were carried out considering different ratios of b/e, where ‘‘b” was the convergent section of the pin-fins, ‘‘e” was the divergent section of the pin-fins and with different arrangements of the pin-fins. Grooved fin led to a significant increase in heat transfer because of grooved distances. Effect of grooved pin fins was studied and it was seen that the average Nusselt number gets increased with increasing b/e ratio reason behind this characteristic was because of more flow mixture and vortex generation. The average Nusselt number for triangular was more than cylindrical. It was observed from boundary layer generation that greater heat transfer was obtained in T-GPFs in comparison with C-GPFs. Finally, conclusion from the above paper [13] that Nusselt number tends to increase with increasing b/e ratio which led to increase in flow mixture and vortex generation. Inline arrays have higher Nusselt number compared to staggered layout and the next point which was made from this was T-GPFs has got higher heat transfer than C-GPFs.

3.2 Tubes

Pagliarini [14] concluded that by using corrugated tube swirl and vorticity strength gets increased which leads to better thermal properties.

Matos et al. [15] carried out study for optimization of three-dimensional geometry of circular and elliptic tubes having staggered arrangement for maximizing the total heat transfer rate which showed that compared to circular tube, the elliptical tube arrangement has relatively better overall performance and of low cost

Talaat A. Ibrahim et al [16] studied nature of thermo fluid behaviour for the elliptic tube bundle arrangement in the cross flow. The investigation compromises the consequence of design parameters such as Reynolds numbers (5600–40,000), minor-to-major axis ratios (0.25, 0.33, 0.5 and 1) and flow angles of attack 00–1500. For the circular and the elliptic tube bundle velocity vector plots were plotted for the angle of attack 0^o, 60^o and 90^o and it was observed that the elliptic tube at an angle of attack 0 degree suffers less drag as it has better aerodynamic shape which in turn offers lower hydraulic resistance compared to circular tube layout. The separation of airflow at the circular tube surface starts at an early stage than that of the elliptic tube. With increase in angle of attack air leaves the first row as jets and thus the second tube row act as a guiding medium which accelerates and redirect the jet streams to the inter-passage of the next rows. The former will increase the level of turbulence along the tube array passage and forming vortex shedding which will increase heat transfer coefficient. From the results obtained from the experimental and numerical for Nusselt number versus Reynolds number at different tube axis ratios; for an axis ratio of 0.5, the average Nusselt number at an angle of attack 90^o was found to be greater than that of 0o and 30 degree by 30.5% and 18% respectively. For an angle of attack 60^o and 90^o Nusselt number of the elliptic tubes was having higher values in comparison to circular tubes by 17% and 19% respectively. The out-turn of flow angle of attack on the friction factor at different values of the axis ratio (A_r = 0.25 and 0.5) were shown in Fig. 21. For an axis ratio of 0.5 and angle of attack at 90o friction factor was much higher than that of 0^o and 30 degree by 92% and 80% respectively. From the results obtained in this paper [16] it was seen that in comparison with circular tubes array friction factor of elliptic tubes at different angle of attack 60^o and 90^o was greater by 55% and 65% respectively. But when the angle of attack 0^o the friction factor was 79% lower than the circular tube array because of better aerodynamic shape of the elliptic tube which contributes to less drag. From their study [16] it was concluded that thermal performance under a fixed pumping power obtained at zero angle of attack was maximum and thermal performance occurred at 90^o angle of attack was minimum. It was also concluded that thermal performance of the elliptic tube heat exchanger with the low Reynolds number, axis ratio and at angle of attack 0^o was the best.

Juan Wen et al. [17] studied using Large eddy simulation of flow and heat transfer for flat finned tube used in air cooled condensers. Flat tube heat exchangers have air side pressure drop much lower and air side heat transfer coefficient relatively higher on comparison with circular tube heat exchanger. Wake region was smaller for flat tube heat exchanger when compared to circular tube heat exchanger, so less noise and vibration were expected in heat exchangers with flat tube. In this paper, 3 D calculations based on Large Eddy Simulation (LES) model for flat finned-tube were reported, where prediction of flow around tubes were more précised. When the flow moves around the flat tubes various unsteady effects like separation, vortex shedding and shear-layer instability were induced. Generation of Vortices resulted in shear-layer instability between the wakes of the flat tubes and induces behaviour of transient periodic nature. Figure 23 shows the computational domain and boundary conditions for the two side-by-side flat finned-tubes. The two assistant computational zones as inlet and outlet were constructed which reduced distortion of flow at the entrance and exit level of passage. From the results obtained they concluded [17] that using LES modelling for the direct air-cooled condensers, which helped in capturing detailed complex turbulent motions and resolved turbulent eddies and precise prediction of the transient flow fields than Reynolds average Navier stroke equation RANS approach. Also, it was concluded in this paper that the three dimensional LES method will give better picture for designing ACC cause in designing ACC one should always take into account the transient fluid flow, heat transfer behaviour, wake interaction and vortex-shedding.

Najla El Gharbi et al [18] carried out study on Numerical optimisation of heat exchangers with circular and non-circular shapes. The main purpose of their work was to know the characteristics of heat transfer and fluid flow with different arrangements of tube bundles such as circular, ellipsoidal and wing shaped. Their Study considered a longitudinal section of cross flow tube bundle of various shapes. The tube banks were uniformly spaced tubes with an internal diameter of 21.7 mm and their centres were separated by distance of 22.5 mm in x direction and 45 mm in y direction. It was observed as Reynolds number increases the pressure decreases. It was further observed that elliptical and wing shaped tubes has lower pressure loss in comparison to circular shape reason behind this is smaller wake regions in non-circular shapes. Finally, they concluded that because of large wake region created in circular tubes leads to high pressure drops in comparison to non-circular tubes which are of streamlined shapes. The last conclusion which was made in this paper [18] from their numerical results that there was as such no best geometry available for all flow conditions, i.e. means across all Reynolds number. It was observed that the circular tube shape gives the worst results when Re >1.5*10^4 than the other two geometries, and for Re >2.3 *10^4 the elliptic shape gave the best, but the difference between the two (elliptic tubes and wing-shaped tubes) was almost negligible.

Ankur Kumar et al [11] carried out study to check the influence of thermal-hydraulic performance on different tube shapes such as elliptical and circular with diameters ranging from (7-24) mm for circular and 30*10-30*20 mm for elliptical. Geometrical parameters considered were number of tube rows (2-10) and transverse tube pitch (36.8-44) mm. The velocity vectors were plotted and obtained results showed that the flow separation occurs later on the surface of elliptical tubes. The study also concluded that the wake region was found to be minimum for round tubes having smaller diameter tube, while elliptical tube had considerably lower wake region for smaller ellipticity of the tube. Therefore they [11] selected tube design with 30*10 mm for elliptical tube. Study [11] considered the effect of tube rows at two frontal velocities 4.46 and 6.32 m/s with varying number of tube rows from 2 to 10. Results showed that the heat transfer coefficient increases with increase in the row number from 2 to 4 but it tends to decrease beyond Nr = 4. Results also showed that pressure drop tends to increase with increase in tube row number. Observations also showed that pressure drop tends to increase at higher frontal air velocity which results into requirement of high pumping power. Effects of transverse tube pitches were obtained which showed that the heat transfer coefficient was decreasing with the increase in transverse tube pitch. Similarly, pressure drop tends to reduce with the increasing tube pitch because of less obstruction to the flow. Variation of fin efficiency was also observed with the increasing number of tube rows. Initially fin efficiency increases for Nr < 4 but with increase in the number of rows fin efficiency gradually decreases. Increase in turbulence promotes mixing in the flow so fin efficiency was increasing with ranging tube row number 2< Nr < 4. The fin efficiency reduces beyond Nr > 4 as temperature difference between air and fin is reduced due to higher temperature of incoming air. Other important points to increase the overall efficiency of power plant are by controlling back pressure which is dependent on ambient condition and other way to increase the efficiency is by reducing the fan power consumption, which has not been discussed in this review.

4. CONCLUSION

From literature review the following points may be derived.

- Fouling was one of the key factor to reduce heat transfer performance as per the discussion above it was seen that oblique arrangement of the wavy fins, finned tubes were more prone to fouling in ambient conditions. Also, it is seen that orthogonal finned tube has higher heat transfer rate than the oblique finned tube arrangement. designed wave-finned flat tube bundle where fin surface was rotated to perpendicular prevents fouling problem.
- For each of the four type of vortex generators viz. 1) delta wing, (2) rectangular wing, (3) delta winglet pairs and (4) rectangular winglet, performance evaluation criteria (PEC) was calculated and found it to be greater than 1 for each of them but main point which was made out of this calculation was that the delta winglet pair having higher value, with increase in number of delta winglet punched on fin surface air-side heat transfer coefficient tends to increase but simultaneously the pressure drop also gets increased.
- Based on the experimental evidence of Caliskan [8] between triangular and rectangular vortex generators, it was seen that better heat transfer performance was obtained from triangular vortex generators.
- Heat transfer performance increases for discontinuous fin and staggered fin because air flow gets interrupted due to different row structure that causes the flow and thermal boundary layer regeneration.
- Fin efficiency decreases with the increase in the fin height. It was concluded that optimal parameters for fin spacing to be 3mm and fin height to be (5) mm, the wave spacing and the wave amplitude to be 9.5 mm and 2.3 mm for better performance.
- Nusselt number tends to increase with increasing b/e ratio which led to increase in flow mixture and vortex generation. Inline arrays have higher Nusselt number compared to staggered layout and the next point which was made T-GPFs has got higher heat transfer than C-GPFs.
- Optimization of three-dimensional geometry of circular and elliptic tubes having staggered arrangement for maximizing the total heat transfer rate showed that compared to circular tube, the elliptical tube arrangement has relatively better overall performance and economical.
- Elliptic tube at an angle of attack 0o suffers less drag as it has better aerodynamic shape which in turn offers lower hydraulic resistance compared to circular tube layout. The separation of airflow at the circular tube surface starts at an early stage than that of the elliptic tube.
- Since LES modelling for the direct air-cooled condensers helps in capturing detailed complex turbulent motions, resolved turbulent eddies and precise prediction of the transient flow fields. Thus, LES method will give better picture for designing ACC cause in designing ACC one should always take into account the transient fluid flow, heat transfer behaviour, wake interaction and vortex-shedding.

ACKNOWLEDGEMENT

Authors would like to express their gratitude to Royal Academy of Engineering (RAE), Parul University (PU), Larsen &Tubro (L&T) and University of Surrey.

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