Experimental and Mathematical Research of Convection Heat Transfer for a High Integral Finned Tube Heat Exchanger

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Abstract. This research exhibits an experimental and numerical investigation for the characteristics of heat transfer by utilizing an integral and smooth high finned tube and shows how the integral high fins affect in improving the transfer of heat. In the experimental work, the experimental rig consists of a cold water loop, a hot air loop and the trial part which is a concentric parallel flow double pipe heat exchanger. The first test smooth tube made of brass has external and internal diameter (32 mm) and (22 mm), respectively, the second test section made of the same material has external fins, the third test section possesses internal fins, and the fourth test section has internal and external fins. The dimensions of the fin are (2 mm) in thickness, (2 mm) in height and pitch (2.5 mm) center to center. And, the water flow rates are (6, 8, 10, 12 and 14 L/min). The inlet water to the trial tube was at temperatures (15, 25, 35, and 45°C). The investigational results manifested that the air side heat transfer coefficient of the smooth tube was lesser than the integral high finned tube. The ratio of improvement when using the integral high finned tube was (47.5%, 60.5% and 67.5 %). Numerical simulation was applied on the current heat exchanger to study both the transfer of heat and the field of flow by using ANSYS, FLUENT15 package. Steady state, Newtonian flow, incompressible and three dimensional analyses were assumed. The comparison between experimental work and numerical results elucidated a good agreement.

1. Introduction

Attention of the saving of energy and materials in addition to thrifty desires has prefaced to a work to generate more effective instrument as a heat exchanger (H.E). The objectives of the public thermal hydraulic are to decrease the heat exchanger size wanted for a definite heat job, to fall the agreed temperature difference for the streams of procedure or to decrease the driving power and develop the ability of current H.Es. The investigation of the modified heat transfer interpretation is indicated as heat transfer improvement or strengthening. Generally, this explains an increase in the coefficient of heat transfer [1]. The heat exchanger of water to air is considered as one of the significant types of a dual pipe concentric tube. This kind of H.E has different implementation, such as apartment buildings and condominiums, hybrid schemes, domestic heating, systems of dehumidification, and air conditioning.

2. Literature Survey

Sparrow and Ramsey [2] described an outstanding investigational work on the effect of tip permission for a surprised wall-fixed array of cylinders. They acquired data on the coefficients of heat transfer via carrying out the similarity amid mass and heat transfer by the method of naphthalene sublimation. It was found that when the length of the cylinder rises, the coefficient of heat transfer rises
temperately and the tip permission is amid the pin and the covering drops. Oppositely, the array pressure drop rises noticeably with the rising length of cylinder. This performance was clarified with the inter-cylinder velocities for short pins which were fewer than the mean velocity, while for the longer cylinders, the inter-cylinder velocities tended to approximate the mean value.

Rabas et al. [3] offered a significant thoughtfulness that the thermal presentation is practically not relied on the Re number for the bigger density of fin (0.980 fins/mm). It was noted that the density of fin has much resilient effect on the performance for the bigger diameter of tube (d = 3.175 cm). The low fins, which are lower than 0.635 cm at the variety of Re as 13 ≤ Re ≤ 25*10³, were studied.

X. Hu and A. M. Jacobi [4] stated investigational researches of the local features of mass transfer of annularly finned tubes in cross flow. The differences because of boundary layer improvement, forward-edge parting, horseshoe vortices, tube wake, and tip vortices were explained. Furthermore, the commonly situated local maximum in the mass transfer rates related to the horseshoe vortex system was found, and speculations as to their mechanism was obtainable. From the concluding heat transfer performance from the outcomes of mass transfer, it was find that the true fin effectiveness is continuously fewer than that achieved with an expected constant convective heat transfer coefficient. The variance is (3–7%) for the high-conductivity substances like the alloys of aluminum and (9–17%) for the low-conductivity substances such as slight steels.

Ayad et al. [5] carried out an analogous work of eight finned–tube heat exchangers. It was found that there was no influence of fin pitch on the heat transfer presentation for four row coils at Re< 10³. On the other hand, the performance of heat transfer is extremely reliant on the fin pitch at Re > 10³. Reducing the fin pitch for a two-row conformation leads to a rise in the performance of heat transfer.

C. V. M. Braga et al. [6] studied the parameters of friction and heat mean transfer coefficients for a turbulent flow in annular duct areas with continued longitudinal rectangular fins fixed to the internal wall. The cold air was flowing over the annulus, whereas the warm water was delivered over the internal pipe. The thermal boundary settings were remained fixed temperature through the inner tube surface, the external tube being isolated. There are twenty continued longitudinal fins installed to the inside wall of the annular duct. The tests were accomplished with the operating of H.E in the parallel flow model. The variety of the air temperature difference amid the parts of leaving and entrance was (10 to 25°C). The total tests and determinations were prepared below settings at a steady state. The entrance temperature of water in the experimental rig part was (98°C), and the pressure was upper than (1 atm). The variation of the air flow Re was amid (10⁴ < 5x10⁴). The experimental outcomes for the mean Nu number of the finned annular duct section, in terms of the air flow Re, can be stated according to the subsequent equation:

\[ Nu = 0.00529Re^{0.8680} \]

The mean Nu number of the air flowing over the plain annulus is signified as Nu:

\[ Nu = 0.00314Re^{0.9474} \]

Ali et al. [7] tested the heat transfer improvement along the exterior surface of a pipe which is made of copper with circular fins joined on the external surface heat exchanger in a rectangular passage with the flow of air intermittently. Their attempts were applied by utilizing four kinds of circular fins with (9.2 cm) external diameter, (3.2 cm) internal diameter, and (0.1 cm) thick tied on pipe of copper. Every kind possesses (5) circular fins. The first kind has 5 fins devoid of inclined blades, the second kind possesses 5 inclined vanes per one fin, the third kind has 7 inclined vanes for each 1 fin and fourth kind possesses 9 inclined vanes for each fin. The results showed that Nu is around (11.80%), (20.25%) and (27.50%) for the 2nd, 3rd and 4th kind, respectively, and it is greater than that for the 1st kind. Furthermore, the development of the transfer of heat and the performance of fin produced a considerable decrease in thermal resistance that can be got from the 4th kind in comparison with the 1st kind.

Bashar [8] studied the increase of the convective heat transfer in a single-phase turbulent flow via utilizing triangular fins experimentally and mathematically. In the investigational work, the influence of
space distance (2.2, 2.7 and 3.2 cm) amid each two fins formed, and the volumetric flow rate on the coefficient of heat transfer was observed.

A number of triangular fins are factory-made from Cu. They are connected to the exterior surface of tube. They have (1 cm) length of base, (0.1 cm) thickness, (1 cm) height, (2.2, 2.7, and 3.2) cm space distance amid two fins sequential and (1.5 cm) pitch amid every two of fins. Thermometers and pressure gauges are connected to the tubes and annular sections. The results obtained from the tubes with triangular fins were compared with the smooth tube heat exchanger. It was found that the triangular fins have a significant effect on the heat transfer augmentations. Also, the result presented that the development of heat debauchery for the triangular finned tube is (3.8 to 5.4) times that of smooth tube at space (2.2 cm) distance amid every two sequential fins. While, the spaces 27 and 32 mm will be (3.5 to 4.9) and (3.2 to 4.5).

Ayad [9] investigated the characteristics of the heat transfer of a cross flow air cooled single tube multi passes (integral and smooth low finned tube) and its influence on the heat transfer enhancement. Two test sections from Perspex duct were designated. Every trial part has a test tube (single aluminum tube multi passes) with passes (4 or 8). The outcomes from the experimental work revealed that the air side coefficient of the heat transfer of the smooth tube was lesser than that of the integral low finned tube.

3. Numerical Simulation

Numerical simulation via utilizing ANSYS, FLUENT15 computational fluid dynamic (CFD) set prototypical was accompanied for executing a mathematical simulation across a H.E by utilizing a three-dimensional model. The solution of the equations of momentum, continuity, and energy was used to study the flow domain into H.E. The evaluation of heat transfer was accomplished for a (triangular, smooth) finned tube H.E. Comprising the triangular-finned tube H.E with the parts of entrance and leaving, the internal and external flow was limited via isolating the tube possessing the parts of entrance and leaving as depicted in the figures below. See the figures 1 to 4.

![Figure 1. CFD field of the smooth tube H.E](image-url)
Figure 2. CFD field of the exterior triangular-finned tube H.E

Figure 3. CFD field of the interior triangular finned tube H.E

Figure 4. CFD field of the assorted triangular finned tube H.E
4. The Prevailing Equations
The main differential equation for the fluid flow is specified as: Mass conservation or continuity equation, Momentum conservation or Navier Stokes equation, and equation of Energy conservation [11].

4.1. Equation of mass conservation:
\[
\frac{\partial (\rho u)}{\partial x} + \frac{\partial (\rho v)}{\partial y} + \frac{\partial (\rho w)}{\partial z} = 0
\]

4.2. Equation of momentum conservation:
\[
\rho \left( u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} + w \frac{\partial u}{\partial z} \right) = \rho X - \frac{\partial p}{\partial x} + \frac{1}{3} \mu \left( \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} \right) + \mu \nabla^2 u
\]
\[
\rho \left( u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} + w \frac{\partial v}{\partial z} \right) = \rho Y - \frac{\partial p}{\partial y} + \frac{1}{3} \mu \left( \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} \right) + \mu \nabla^2 v
\]
\[
\rho \left( u \frac{\partial w}{\partial x} + v \frac{\partial w}{\partial y} + w \frac{\partial w}{\partial z} \right) = \rho Z - \frac{\partial p}{\partial z} + \frac{1}{3} \mu \left( \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} \right) + \mu \nabla^2 w
\]

4.3. Equation of Energy:
\[
\rho c_p \left( u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} + w \frac{\partial T}{\partial z} \right) = \left( u \frac{\partial p}{\partial x} + v \frac{\partial p}{\partial y} + w \frac{\partial p}{\partial z} \right) + K \nabla^2 T + \mu \phi
\]

Where:
\[
\phi = 2 \left[ \left( \frac{\partial u}{\partial x} \right)^2 + \left( \frac{\partial v}{\partial y} \right)^2 + \left( \frac{\partial w}{\partial z} \right)^2 \right] + \left[ \left( \frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} \right)^2 + \left( \frac{\partial v}{\partial z} + \frac{\partial w}{\partial y} \right)^2 + \left( \frac{\partial w}{\partial x} + \frac{\partial u}{\partial z} \right)^2 \right] - \frac{2}{3} \left[ \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} \right]
\]

5. Boundary Conditions
The velocity at the entrance was defined for the internal and annuli sides throughout this research. Conversely, the temperature of the entrance of the internal tube was (15, 25, 35, 45°C), whereas in annuli it was (70) °C.

6. Mesh Resolution
In the current search, a free mesh was utilized to approximate the computational field inside a limited number of control volumes via utilizing the pattern or the model of finite–volume. The organized mesh is governed unavailable as it is advantageous for simple states and it converts to inadequate and expended time for difficult geometries. The pattern was meshed via utilizing ANSYS, FLUENT15 (CFD) set. The production and of the modification of mesh scheme are so vital to expect the H.T in the advanced geometries. See the figures 5 to 8.
Figure 5. CFD meshing-part of the external finned-tube heat exchanger (pitch 2.5 mm)

Figure 6. CFD meshing-part of internal finned tube heat exchanger(pitch 2.5mm)

Figure 7. CFD meshing-part of mixed(internal and external) finned tube heat exchanger(pitch 2.5m)
7. Apparatuses of the Experimental Trial

Figure 1 evinces the graphical drawing of the experimental system. The test section consists of two portions. The first portion is an isolated tube which has factory-made from (PVC) of (5.08 cm) internal diameter, (10 cm) length and (0.5 cm) thickness. The second portion is an interior brass tube devoid of or with triangular brass fins. The integral fins tube consists of three types. The first type is an internal fins tube, the second type is an external fins tube and the third type is a mix (external and internal) fins tube, the all types of fins tube have a pitch of (2.5 mm). The smooth copper tube is (10 cm) long and (2.2 cm, 3.2 cm) internal and external diameter, respectively. There were outlet and input section in the top and bottom section of the shell respectively suited at a (20 cm) distance of the edge annular, each nozzle has inside diameter and thickness. Nozzle was made from the same material of the annular. The annular was drilled at a distance (20 cm) from the edge ends, on the top and bottom to weld the nozzle. Several of triangular fins are factory-made from copper, having (2 mm) length of base, (2 mm) height, and (2.5, 3, and 3.5 mm) pitch between each two of fins. Thermometers and pressure gauges are connected to the tubes and annular sections. Thermocouples and pressure gages were mounted on the two ends of the shell, to determine the temperature at the entrance and leaving on the two exchanger sides and the pressure on both shell sides. The tried heat exchanger comprises a single-pass kind concentric tube. See figures 9 and 10.

Figure 8. CFD meshing of smooth tube heat exchanger

Figure 9. Smooth tube of the enhanced of heat exchanger
The annular and the tube heat exchanger are planned for a parallel flow conformation, in which the cold water passes in same path in the tubes to the hot air which passes in the annular side. These portions have installed on the border construction. The frame structure is planned for repairs during the test and making a tranquil for operation.

8. Validation

For the assessment of Nu, the investigational information are in a good acceptance with the current relationships, which are the relationship of Dittus–Boelter. The dependability and accurateness of investigational determinations were examined by utilizing a smooth tube alone. The investigational outcomes were related to the expectation of the relationship of Dittus-Boelter:

\[
Nu = 0.023Re^{0.8}Pr^n
\]

Where:

- \( n = 0.33 \) for cooling
- \( Nu \): The Nusselt number
- \( Pr \): The Prandial number
- \( Re \): The Reynolds number

The Nusselt number was found to increase by increasing the \( Re \) number. A good agreement was obtained amid the experimental data and the outcomes acquired from the Dittus-Boelter correlation. The maximum deviation between the experimental data and the Dittus-Boelter correlation was about 5%. See figures 11, 12 and Table 1.
Figure 11. Representation of the experimental apparatuses

Figure 12. Picture of the experimental apparatuses

\[ Q_h = m \dot{h} c_p h (T_{ho} - T_{hi}) \]

Where:
- \( m \dot{h} \): The mass flow rate (kg.s\(^{-1}\))
- \( Q \): The heat dissipation (w)
- \( T \): The temperature (°C)
- \( C_p \): The specific heat

The Subscripts i, o and h is referring to the internal, external and hot, respectively.

\[ LMTD = \frac{\Delta T_2 - \Delta T_1}{\ln\left(\frac{\Delta T_2}{\Delta T_1}\right)} = \frac{\Delta T_1 - T_2}{\ln\left(\frac{T_1}{T_2}\right)} \]

Where:
\[ \Delta T_1 = T_{hi} - T_{co} \]
\[ \Delta T_2 = T_{ho} - T_{ci} \]

Where: \( \Delta T_{LMTD} \) is the log mean temperature variance (°C).

**Table 1.** Dittus-Boelter and CFD outcomes for various values of Reynolds number

| Re   | Nu Dittus-Boelter | Nu CFD |
|------|-------------------|--------|
| 3019.892 | 17.217             | 27.071 |
| 4026.522 | 21.668             | 31.7113 |
| 5033.152 | 25.907             | 35.8527 |
| 6039.733 | 39.6343            | 29.976 |
| 8035.044 | 39.733             | 46.4237 |

\[ h_i = \frac{Q_h}{A_i(T_m - T_s)} \]

Where:
- \( A \): The tube surface area (m²)
- \( h_i \): The heat transfer coefficient (W·m⁻²·K⁻¹) of the tube

\[ Nu = \frac{h_i d_i}{K_h} \]

Where:
- \( k \): The thermal conductivity (W·m⁻¹·K⁻¹)
- \( D \): The tube diameter (m)

9. Discussions and Results

The finned tube with a base smooth tube as three different cases (external, internal and mixed) integral finned tube at five different volume flow rates of water (6, 8, 10, 12 and 14 liter/min) and one volume flow rate of air under the turbulent flow regime were investigated in a concentric parallel flow heat exchanger to study the improvement of heat transfer owing to finned tube, as shown in figure (13). The Nu number was found to increase by increasing the Re number. A good acceptance was obtained amid the experimental data and the outcomes attained from the Dittus-Boelter correlation. The maximum deviation between the experimental data and the Dittus-Boelter correlation was about (5%).

The assessment among the Nu numbers for the water side of finned tube at three various states (internal, external and mixed) hinge integral finned tube with smooth tube alone is shown in figures (14) to (16).

The investigational outcomes refer to that the Nu number in finned tube at three different states (case 2 to case 4) is greater by (48.7%, 61.7% and 68.1%) than that of case 1 (smooth tube), respectively. The maximum increment in Nu numbers for the triangular fins is roughly (68.1%) higher than those for the plain tube (case 1). At Reynolds number in the range (212769 - 496391) in mixed (internal and external) integral finned tube at the temperature of water is (15°C). The enhancement in the coefficient of heat transfer is owing to decrease the distance between two fins which increase the grossness of the surface of tube, resulting in destroying the boundaries laminar sub layer which coated the tube. The increasing of turbulence causes a better heat transfer between the tube wall and air. These factors caused the higher values of heat transfer coefficient and as a result the higher Nu number values. The Nu number of air side for the finned tube is bigger than that for smooth tube because of the rise in the area of surface via fins and due to the turbulence generated.

The increase in Nu number might be offered via computing the total ratio of improvement \((Nu_{fr}/Nu_{s})(f_{fr}/f_{s})^{1/3}\) as manifested in figure (17). When utilizing the states from case 2 to case 4 (external, internal and mixed), the improvement ratio \((Nu_{fr}/Nu_{s})(f_{fr}/f_{s})^{1/3}\) is reduced from (2.684) to (1.806) at (6 L/min) from mixed, external to internal from (2.585) to (1.677) at 8 L/min, from (2.681) to (1.696) at 10 L/min, from (2.937) to (1.849) at 12 L/min, and from (2.716) to (1.701) at 14 L/min. See also figures 18-23.
Figure 13. The comparison between experimental data and empirical correlations.

Figure 14. Influence of the water Re number with the air side Nu number at smooth tube.

Figure 15. Influence of air side Nu number with the water Re number at the external integral finned tube.
Figure 16. Influence of water Re number with the air side Nu number at the internal integral finned tube.

Figure 17. Influence of flow rate on the total ratio of improvement for air side.

Figure 18. Influence of integral fins on the Nu number for different flow rates
Figure 19. Variation of mass flow rate with the efficiency for finned tube

Figure 20. Velocity contours for shell side with air flow over; (a) smooth tube, (b) finned tube with pitch (2.5 mm).
Figure 21. Velocity contours for tube side with water flow within; (a) smooth tube, (b) internal finned tube with pitch (2.5 mm).

(a) Flow over smooth tube [G04].
(b) Flow over external finned tube with pitch 2.5 mm [G01-P1].

Figure 22. Temperature contours for the shell side with air flow at (4 m/s at inlet).
10. Conclusion

The increasing of H.T is obvious when using fins on the surface of (outer and inner) of internal tube in the current H.E. The appeared improvement is evident about (67.5%) than that of smooth tube. The use of the integral finned tube increases the Nu with the decreased water temperature. The smooth tube offers lesser Nu than the finned tubes. The less pitch ($p = 2.5$ mm) matched with additional pitch. The outcomes of the numerical simulation elucidated that the addition of fins will improve the dispersal of heat over the heat exchanger, and give numerical results in a good agreement with the results from the experimental work, with a maximum deviation of (10%). The outcomes from the experimental work matched with results obtained from Dittus-Boelter. Good agreements were got with an extreme deviation about (5%). The significant improvement is found in H.E via accepting fins on the mixed (internal and external) integral fins of inner tube. This enhancement appeared clearly in the finned tube with three cases (internal, external and mixed) is (47.5%, 60.5% and 67.5%) than that of smooth tube (case 1), respectively.

11. Future Works

1. Study the effect of another design of an integral finned tube on the flow and heat transfer enhancement.

2. Investigating the effect of using different materials tubes on the heat transfer rate.

3. Study the effect of utilizing various types of fluid on the heat transfer rate.

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