# OPTIMAL GEOMETRIC ARRANGEMENT OF UNFINNED AND FINNED FLAT TUBE HEAT EXCHANGERS UNDER LAMINAR FORCED CONVECTION

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## ABSTRACT

This thesis describes the three-dimensional numerical analysis and experimental study of the heat transfer and flow characteristics in the un-finned and finned flat tube heat exchangers for in-line and staggered configurations. Flat tubes are vital components of various technical applications including modern heat exchangers, thermal power plants, and automotive radiators. The objectives of this research are to develop a numerical code to predict the thermal–hydraulic characteristics of laminar forced convective flow, to identify optimal spacing tube-to-tube and fin-to-fin for the maximum overall heat transfer rate and minimum power pumping of the fan between the tube bundle and surrounding fluid at the fixed volume and to develop a new correlation for overall heat transfer rate and power pumping in general and optimum configurations. Conservation equations (mass, momentum, and energy) were solved to develop code utilizing Visual-FORTRAN based on finite volume technique to determine the temperature and velocity fields. Subsequently, the overall heat transfer rate and power pumping among the tubes, fins, and fluid flow were calculated. The algorithm of semi-implicit method for pressure-linked equations was utilized to link the pressure fields with velocity. Finally, the subsequent set of discretization equations was solved with line-by-line method of the tri-diagonal matrix algorithm and the Gauss–Seidel's procedure. Twelve fixed tubes were used in the experimental setup for flat tube configurations were obtained with these uniformly fitted tubes with a fixed volume. The experimental setups with several arrays of tubes and fins were fabricated with the same volume. The results were reported of the external air flow in a range of Reynolds numbers based on the hydraulic diameter of 178 to 1,470. It can be observed from the obtained results that the geometric optimum for tube-to-tube spacing was ($S_t/d_T \cong 1.6$) in the in-line configuration and ($S_t/d_T \cong 2.0$) in the staggered configuration. Meanwhile, fin-to-fin spacing was $\Pi_f = 0.025$, according to general dimensionless variables. Up to 1.48 and 1.11 times (in-line) as well as 2.3 and 1.4 times (staggered) of heat transfer gain were noted in the optimal configuration for the low and high Reynolds numbers. A newly developed correlation of heat transfer rate and power pumping was then proposed. Approximately 87.5 % of the database described the heat transfer correlation within ± 15 % for the in-line configuration. For the staggered arrangement, 82% of the deviations were within ± 15 %. Up to 97.2 % of the database can be correlated with the proposed power pumping correlation within ± 18 % for the in-line arrangement, and 86.2% of the deviations were within ±15 % for the staggered arrangement. In the in-line configuration, the mean errors of the heat transfer and power pumping correlations were 9.5 % and 12.2 %, respectively. In the staggered configuration, the mean deviation errors of heat transfer and power pumping correlations were found to be 9.5 % and 11.1 %, respectively. The predictive correlations developed in this study for in-line and staggered configurations can predict the heat transfer rate and power pumping of both un-finned and finned flat tube heat exchangers, which can be applied to the design of future heat exchangers in the industry.

## ABSTRAK

Tesis ini menerangkan tiga demensi analisa berangka dan kajian eksperimen pemindahan haba serta ciri-ciri aliran tiub penukar haba bersirip dan tidak bersirip dalam bentuk sususan sebaris dan susunan tak serentak. Tiub rata adalah komponen penting dalam pelbagai aplikasi teknikal termasuk dalam penukar haba moden, janakuasa terma, dan radiator otomotif. Penyelidikan ini bertujuan untuk menbangunkan kod berangka bagi meramalkan ciri-ciri terma-hidraulik lamina aliran olakan paksa, menentukan jarak optima antara tiub-tiub dan juga sirip-sirip, pada maksimum keseluruhan kadar pemindahan haba dan minima kuasa pengepaman kipas diantara susunan tiub dan bendalir sekitar, pada isipadu tetap dan juga membangunkan korelasi baru kadar pemindahan haba dan kuasa pengepaman pada konfigurasi umum dan optimum. Persamaan pengabdian (jisim, momentum dan tenaga) telah diselesaikan untuk membangunkan kod menggunakan Visual-FORTRAN berdasarkan teknik isipadu terhingga untuk menentukan suhu dan halaju lapangan. Seterusnya, kadar keseluruhan pemindahan haba di kalangan tiub, sirip, dan bendalir telah dikira. Algoritma separuh-tersirat untuk persamaan tekanan-berkaitan telah digunakan untuk menghubungkan tekanan lapangan dengan halaju. Akhir sekali, persamaan pendiskretan set berikutnya telah diselesaikan dengan kaedah matriks algoritma garis demi garis tiga-pepenjuru dan tatacara Gauss-Seidel. Dua belas tiub tetap telah digunakan dalam persediaan eksperimen untuk konfigurasi tiub rata,tiub-tiub yang dipasang seragam dengan isipadu tetap.. Eksperimen disediakan dengan beberapa tatasusunan tiub-tiub dan sirip-sirip telah difabrikasi pada isipadu yang sama Keputusan telah dilaporkan aliran udara luar pada julat nombor Reynolds berdasarkan diameter hidraulik antara 178 ke 1,470. Dari keputusan didapati geometri optimum untuk jarak tiub ke tiub adalah ($S_t/d_T \cong 1.6$) pada konfigurasi sebaris dan ($S_t/d_T \cong 2.0$) pada konfigurasi susunan tak serentak. Sementara itu, jarak antara sirip ke sirip adalah $\Pi_f$ = 0.025 mengikut pemboleh ubah umum tak berdimensi. Gandaan pemindahaan haba sehingga 1.48 dan 1.11 kali (sebaris) dan 2.3 dan 1.4 kali (susunan tak serentak) diperhatikan adalah konfigurasi optimum bagi nombor Reynolds yang rendah dan tinggi. Satu korelasi baru dibangunkan daripada kadar pemindahan haba dan kuasa mengepam seterusnya telah dicadangkan. Kira-kira 87.5 % daripada pangkalan data menyifatkan korelasi pemindahan haba dalam lingkungan ± 15 % bagi konfigurasi sebaris. Untuk susunan tak serentak, 82% daripada sisihan berada dalam lingkungan ± 15 %. Sehingga 97.2 % daripada pangkalan data boleh dikaitkan dengan cadangan korelasi kuasa mengepam dalam lingkungan ±18% untuk susunan sebaris, dan 86.2% daripada sisihan berada dalam lingkungan ± 15 % bagi susuanan tak serentak. Dalam konfigurasi sebaris, ralat min korelasi pemindahan haba dan kuasa mengepam adalah masing-masing 9.5 % dan 12.2 %. Dalam konfigurasi tak serentak, ralat min sisihan korelasi pemindahan haba dan kuasa mengepam masing-masing adalah 9.5 % dan 11.1 %. Kaitan ramalan telah dibangunkan dalam kajian ini untuk kedua-dua konfigurasi sebaris dan tak serentak yangboleh meramalkan kadar pemindahan haba dan kuasa mengepam untuk kedua-dua penukar haba tiub rata bersirip dan tidak bersirip yang mana ia boleh diaplikasi untuk merekabentuk penukar haba masa depan di industri.

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# NOMENCLATURES

| **Symbol** | **Description and Unit** |
|---|---|
| 2D | Two dimensional |
| 3D | Three dimensional |
| $A$ | Area, $m^2$ |
| $A_c$ | Cross-sectional area |
| BFC | Body fitted coordinates |
| CFD | Computational fluid dynamic |
| CV | Control volume |
| CVFEM | Control-volume finite element method |
| $c_p$ | Specific heat, J/(kg K) |
| $d$ | Diameter of round tube, m |
| $D_h$ | Hydraulic diameter, m |
| $d_L$ | Longitudinal diameter of flat tube, m |
| $d_T$ | Transverse diameter of flattened tube, m |
| $E$ | Input voltage, Volt |
| $Er$ | Relative error |
| FVM | Finite volume methods |
| $G_1$, $G_2$, $G_2$ | Contravariant velocity components |
| GE | Governing equations |
| GIT | Grid independency test |
| $Hy$ | Array height, m |
| HEM | Heat exchanger module |
| HRSG | Heat recovery steam generator |

| | |
|---|---|
| $I$ | Input current, A |
| $J$ | Jacobian of the transformation |
| $k$ | Thermal conductivity of air, W/(m K) |
| $L$ | Array width, m |
| LES | Large eddy simulation |
| $\dot{m}_{Ec}$ | Air mass flow rate entering one elemental channel, kg/s |
| $M$ | Node number of the grid in $\zeta$ -direction |
| MEr | Mean relative error |
| $N$ | Node number of the grid in $\eta$-direction |
| $n_{\mathrm{f}}$ | Fins number |
| $p$ | Pressure, Pa |
| $p_{\mathrm{f}}$ | Fins pitch, m |
| PDE | Partial differential equation |
| PFTHE | Plain fin-and-tube heat exchangers |
| $q_{\mathrm{in}}$ | Input heat, W |
| $Qq$ | Overall heat transfer rate, W |
| $R^2$ | Coefficient of determination |
| RTD | Resistance temperature detector |
| $S$ | Source term |
| SIMPLE | Semi-implicit methods pressure-linked equation |
| SOR | Gauss-Seidel successive over relaxation |
| $S_t$ | Tubes spacing, m |
| $S_{\mathrm{total}}$ | Total source terms |
| $S_{\zeta,\eta}$ | Source term due to nonorthogonality |

| | |
|---|---|
| $S_\phi$ | Source term of $\phi$ |
| $t$ | Tube thickness, m |
| $T$ | Temperature, $^{o}$C |
| $\bar{T}$ | Average temperature, $^{o}$C |
| $T^*$ | Dimensionless temperature |
| TDMA | Tri–diagonal matrix algorithm |
| $u^*,v^*,w^*$ | Dimensionless velocity components |
| $u,v,w$ | Velocity components, m/s |
| $W_x$ | Array length, m |
| $x^*, y^*, z^*$ | Dimensionless Cartesian coordinates |
| $x, y, z$ | Cartesian coordinates, m |
| $\dot{H}p$ | Power pumping, W |

**Dimensionless Groups**

| | |
|---|---|
| $\overline{Nu}$ | Average Nusselt number |
| $\tilde{Q}q$ | Dimensionless overall thermal conductance |
| $\tilde{Q}_q^*$ | Dimensionless overall thermal conductance |
| $S_T^*$ | Dimensionless spacing between rows of tubes |
| $\tilde{m}_s^*$ | Dimensionless mass of solid material (mass fraction) |
| $f$ | Friction factor |
| $j$ | Colburn factor |
| $Nu$ | Nusselt number |
| $Pr$ | Prandtl number |
| $Re$ | Reynolds number |
| $\tilde{H}p$ | Dimensionless power pumping |

**Greek Symbols**

| | |
|---|---|
| $\alpha, \beta, \gamma$ | Coefficients of transformation |
| $\Pi_f$ | Dimensionless fin density in direction $z$ |
| $\alpha_\phi$ | Underrelaxation factor of $\phi$ |
| $\Gamma$ | Diffusion coefficient |
| $\Delta p$ | Pressure drop, Pa |
| $\Delta p^*$ | Dimensionless pressure drop in cross flow |
| $\Delta x, \Delta y, \Delta z$ | Dimensions of the computational cell |
| $\varepsilon$ | Emissivity |
| $\mu$ | Dynamic viscosity, N s/m$^2$ |
| $\rho$ | Density, kg/m$^3$ |
| $\zeta, \eta$ | Curvilinear coordinates |
| $\sigma$ | Slandered deviation |
| $\phi$ | General dependent variable |

**Subscripts and superscripts**

| | |
|---|---|
| " | Corrected values |
| * | Dimensionless quantity |
| ** | Uncorrected values |
| 1 | First node to the wall |
| *a* | Air |
| *e*, *w*, *n*, *s*, *b*, *t* | Adjacent faces to the main point P |
| *E*, *W*, *N*, *S*, *B*, *T* | Adjacent points to the main point P |
| f | Fin |
| *h* | Hydraulic |

| | |
|---|---|
| i | Inside |
| *i, j, k* | Index notations or coordinate direction identifiers |
| *in* | Inlet, Input |
| *L* | Longitudinal |
| *Max* | Maximum value |
| o | Outside |
| *out* | Outlet |
| *T* | Transverse |
| *s* | Surface |
| *t* | Tube |
| *to* | Total |
| *w* | Tube wall |

# CHAPTER I

# INTRODUCTION

## 1.1 INTRODUCTION

Substantial research effort has been exerted to improve the efficiency of heat exchangers because of the widespread use of these devices in industrial, transportation, and domestic applications, including thermal power plants, means of transport, heating and air conditioning systems, electronic equipment, and space vehicles (Incropera et al. 2011). Increasing the efficiency of heat exchangers would greatly reduce the cost, space, and materials required in their use (Bejan and Kraus, 2003). The demand for increased supply of energy continues to increase in all facets of society. The answer to this demand is the intelligent use of available energy. The utilization of available energy to optimize industrial processes has been a popular research topic in recent years because of the extensive use of heat exchangers in industrial applications, such as tube arrangements, un-finned and finned systems, refrigeration, air conditioners, and heaters (Webb and Kim, 2007). The heat exchanging equipment in these devices must be designed such that they can be accommodated by the devices that enclose them. Therefore, an optimized heat exchanger would provide maximum heat transfer for a given space (Bejan, 2000; Bejan and Lorente, 2008). Such equipment strikes a balance between reduction in size or volume and maintenance and enhancement of its performance.

Improving the performance of heat exchangers is important because of the economic and environmental effects of these devices. The loss incurred during operation can be reduced by rationalizing the use of available energy. The volume incurred by the array of tube heat exchangers should be fixed. Heat exchangers must be fitted according to available space (Bejan, 2000) through a process called volume constrained optimization. In this process, the optimal spacing between tubes of known geometry is determined in a manner that maximizes the overall heat transfer (thermal conductance) between the array and the surrounding fluid. The development of cooling techniques for electronic packages is a common example of basic optimization. Considerable effort has been exerted to determine the optimal spacing for various geometric configurations, be it natural or forced convection (Bar–Cohen and Rohsenow, 1984; Bejan, 2004; Bejan and Sciubba, 1992; Kim et al., 1991; Knight et al., 1991, 1992; Ledezma et al., 1996; Matos et al., 2004).

Flat tubes are vital components in various technical devices, including modern heat exchangers and automotive radiators. Although many researchers have studied fluid flow and heat transfer in objects of various shapes, flat tubes have not been fully investigated (Bahaidarah, 2004). Flat tubes have been recently incorporated into automotive air conditioning evaporators and condensers. Their cost has been reduced because of the developments in automotive brazed aluminum manufacturing technology (Min and Webb, 2004; Webb and Kim, 2007). Compared with circular tube heat exchangers, flat tube heat exchangers have lower air–side pressure drop and higher air–side heat transfer coefficients. Given their smaller wake area, flat tube heat exchangers are likely to have lower pressure drop than circular tube heat exchangers. The same reason accounts for the smaller amount of vibration and noise in flat tube heat exchangers than in circular tube heat exchangers (Bahaidarah, 2004). The external heat transfer coefficient of the tube as well as the pressure drop of the fluid flowing externally are the most critical design variables of tubular heat exchangers (Webb and Kim, 2007). From nuclear reactors to refinery condensers, various energy conversion and chemical reaction systems have been installed with tubular heat exchangers in them. Tube configurations have been reported to positively affect heat transfer (Nishiyama et al., 1988; Wung et al., 1986)

Determining the optimal geometry is important to maximize the volumetric heat transfer under the volumetric constraint and reduce the pressure drop (power pumping). In this study, the geometric parameters were first identified to initiate the optimization of the overall heat transfer rate among the tubes and the air free stream. The following two geometric parameters were identified in the arrangement.

i) tube-to-tube spacing, $S_{\mathrm{t}}$

ii) fin-to-fin spacing, $p_{\mathrm{f}}$

## 1.2 PROBLEM STATEMENT

The increase in heat transfer gain and decrease in pressure drop in any tube bank configuration using a heat exchanger are important considerations for the design regardless of the tube shape (Canhoto and Reis, 2011; Knight et al., 1992). The reduction in volume for the thermal system is also important. The main problem in these applications (thermal system) is how to disperse heat and reduce pumping power. The principal parameters involved are tube and fin spacing. Many studies have investigated heat transfer and fluid flow in cross flow over several tube shapes. Furthermore, the optimal spacing (optimum design) in circular and elliptic tubes has been extensively studied. However, the optimal spacing in flat tubes has not been fully studied. This study aims to determine the optimum spacing (tube-to-tube and fin-to-fin) for un-finned and finned flat tube heat exchangers with both in-line and staggered configurations. Several parameters affect the rate of heat transfer and pressure drop; these parameters include external fluid velocity, tube diameter, tube spacing, and fin spacing (Hsieh and Jang, 2012; Jin et al., 2013). Therefore, the present study attempts to address the optimal arrangement of un-finned and finned flat tube heat exchangers for laminar forced convection under fixed volume. Both in-line and staggered configurations are considered. The maximum overall heat transfer rate and minimum pumping power are presented at the optimal spacing of tubes and fins. The correlations of heat transfer density rate and pumping power for general and optimal arrangements are also calculated. This study contributes to the technical aspect of heat exchanger applications.

## 1.3 OBJECTIVES OF THE STUDY

The objectives of the present work are summarized as follows:

i) To develop a numerical code to predict the thermal–hydraulic characteristics of laminar forced convective flow over 3D un-finned and finned flat tube bank heat exchangers with in-line and staggered configurations.
ii) To analyze the heat transfer and flow characteristics in un-finned and finned flat tube bank heat exchangers.
iii) To identify optimum arrangements (tube and fin spacing) to maximize the overall heat transfer rate and minimize the pumping power in a specific fixed-volume configuration.
iv) To develop a new correlation for overall heat transfer rate and pumping power in general and optimum configurations.

## 1.4 SCOPE OF THE STUDY

This thesis attempts to analyze the heat transfer and flow characteristics in un-finned and finned flat tube bundles and to evaluate the optimum design. The optimum design should yield maximum heat transfer (cooling or heating) under a fixed volume while reducing the pressure drop and pumping power of flow. The optimum design can be established by determining the optimum spacing between tubes and fins. The key steps of study are as follows.

i) Analysis is performed on the air side of the un-finned and finned flat tube heat exchangers (external flow).
ii) The cross-flow arrangement is considered.
iii) The flow is considered a steady, laminar forced convection incompressible flow in the 2D and 3D domain.
iv) Both in-line and staggered configurations are considered in this study.
v) Experimental work is conducted to provide a set of data for the validation of the simulation model as well as previous numerical and experimental studies under several geometrical and flow parameters.

## 1.5 ORGANIZATION OF THE THESIS

This thesis is arranged in a manner that clarifies the subject from all aspects; it provides details on the facts, observations, arguments, and procedures to achieve the objectives. The thesis contains five chapters. The thesis introduction presents the following in the specified order: the importance of using flat tubes, the problem statement, the main objectives, and the scope. Chapter II describes and discusses studies related to the objectives of the present study. Chapter III is divided into two main sections: experimental work and computational models. The first section of this chapter presents the experimental equipment, followed by the details of the setup of all equipment, test modules, and instrument calibration, including an open-circuit low-speed wind tunnel and a thermistor. The second section lists the suppositions and describes the mathematical model for the physical problem of un-finned and finned flat tube banks. This section includes problem description, the governing equations, the generated grid, and the numerical solution. The results and discussion of the 2D and 3D models for both in-line and staggered un-finned and finned flat tube banks are presented in Chapter IV. This chapter also contains the validations and comparisons of the experimental and simulation results. The simulation results are evaluated and validated in light of the empirical data. The development of new correlations for overall heat transfer rate and pumping power is also presented in this chapter. The main findings and discussions are presented in Chapter V. Some recommendations for future research are also provided in this chapter. The last section of this thesis contains the references and appendices related to instrument calibration, experimental uncertainty, and tabulation of the experimental results.

# CHAPTER II

# LITERATURE REVIEW

## 2.1 INTRODUCTION

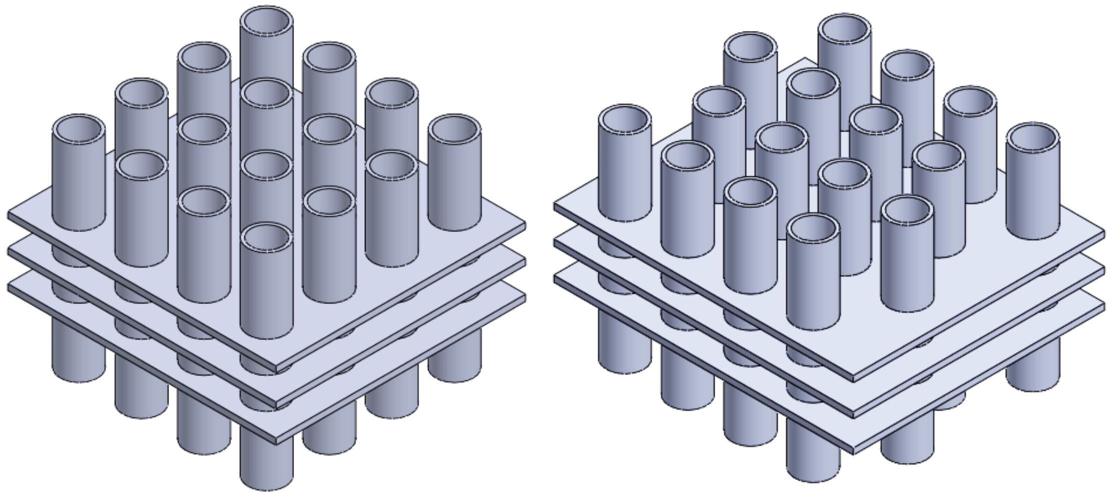
There has been a significant amount of research work carried out to improve the efficiency of heat exchangers. The reason for these efforts is that heat exchangers have a widespread use in industrial, transportation as well as domestic applications such as thermal power plants, means of transport, heating and air conditioning systems, electronic equipment and space vehicles (Manglick, 2003). Because of their extensive use, increase in their efficiency would consequently reduce cost, space and materials required drastically (Brauer, 1964; Manglick, 2003). The aforementioned research work includes a focus on the choice of working fluids with high thermal conductivity, selection of their flow organization and high effective heat transfer surfaces constructed from high-conductivity materials.

This chapter shows a general review of the heat transfer and fluid flow characteristics of a tube banks heat exchanger and discusses the effect on the thermofluid characteristics of several parameters: the frontal velocity of fluid, tube diameter, tube configuration, tube rows, tube spacing, fin spacing, and tube shape. The optimum tube-to-tube and fin-to-fin spacing with the maximum heat transfer rate and minimum pressure drop are also presented. A highlight of the most important correlations for heat transfer and fluid flow in a tube banks heat exchanger is provided. The other specific shapes (flat tube) and confinement of the tube between parallel plates are outlined were reviewed. The chapter also shows and describes the gaps in the research which may be considered by new studies and suggests future work. Finally, this chaper presents the significant conclusions.

## 2.2 BACKGROUND OF TUBES BANK

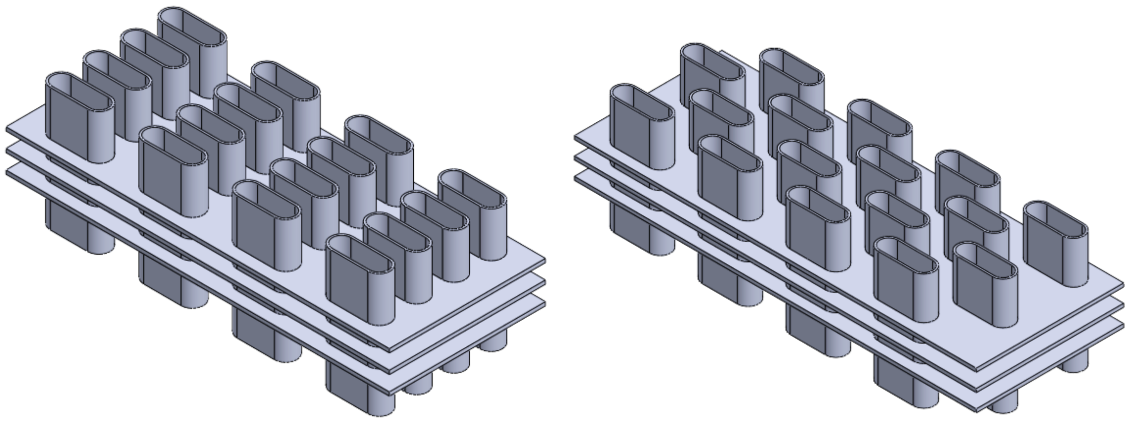
The general configurations of un-finned and finned tube banks heat exchangers were presented in Figures 2.1 and 2.2. Both in–line and staggered configurations of tube as well as the circular and flat tubes shape are illustrated. In general, one fluid flows over the tubes array, while a other fluid at the different temperature moves through the tubes. The rows of tube at the in-line and staggered arrangements in the flow direction of fluid (i.e., inlet velocity of air $u_\infty$) are as shown in Figures 2.2(a) and (b). The characteristics of configuration by the diameter of tube such as $d$, for circular tube and tansverse tube diameter of flat tube as well as by the longitudinal pitch, $P_1$ and transverse pitch, $P_2$ the distance between centres of tube. Beale and Spalding (1999) carried out a numerical investigation of transient incompressible flow in in-line square, rotated square, and staggered tube banks for the $Re$ number range of $30 \leq Re \leq 3{,}000$ and ratio of pitch to diameter of 2:1. The drag lift, pressure drop, and heat transfer coefficient were calculated. A calculation procedure for a 2D elliptic flow is applied to predict the pressure drop and heat transfer characteristics of laminar and turbulent flows of air across tube banks. The theoretical results of the model are compared with previously published experimental data (Wilson and Bassiouny, 2000). A 2D numerical study of the laminar steady state flow in a circular tube banks heat exchanger was carried out for low Reynolds number numbers (Li et al., 2003; Odabaee et al., 2012). The flow in a bundle of elliptical cylinders was investigated both numerically and experimentally (Tang et al., 2009; Yianneskis et al., 2001).

The momentum and energy equations have been solved by using a finite difference method. The effect of the Nusselt number on the surface of the tube was recorded by Juncu (2007). The importance of heat transfer and fluid flow appearances of tube banks in the design of heat exchangers is well known (Žukauskas, 1972). Comprehensive experimental (Kang et al., 1994; Khan et al., 2004; Yao and Zhu, 1994), numerical studies (Jayavel and Tiwari, 2009; Marchi and Hobmeir, 2007; Rahmani et al., 2006; Wilson and Bassiouny, 2000) and both experimental and numerical studies (Nishimura, 1991; Rodgers et al., 2008; Sumner, 2010) of circular tube banks have been done previously. The numerical analysis of laminar forced convection in a 2D steady



(a) In-line classic tube shape

(b) Staggered classic tube shape



(c) In-line flat tube

(d) Staggered flat tube

**Figure 2.1**: The configurations of finned round and flat tube heat exchanger (Adopted form).

Source: Webb and Kim (2005)