CHAPTER 1

Introduction

The subject of enhanced heat transfer has developed to the stage that it is of serious interest for heat exchanger application. There are many applications where the fluid-to-gas heat exchanger is important to the thermal process such as heating and air conditioning, refrigeration, automotive engineering, petrochemical and food processing, and other industries.

1.1 Dimple surface

1.1.1 Heat transfer performance on dimple surface

The conventional heat transfer enhancement approaches for increasing either the heat transfer rate or the turbulence of fluid stream, in general, involve with the incorporation of fins, baffles, turbulizers and etc. Although, these approaches are the effective method for improving the heat transfer performance; however, the increasing of fluid stream pressure drop should be concerned. The dimpled surface shown in Figure 1.1 is one of the effective methods for improving the heat transfer rates without the significant pressure drop. Normally, the dimpled surface generates the vortex flow within its hole as illustrated in Figure 1.2 and the augmentation of heat transfer is obtained.

1.1.2 Literature review

Recently, dimples or concave surfaces have been in focus extensively. In the early investigations, Afansayev et al. (1993) investigated the overall heat transfer and pressure drop for turbulent flow flat surface with staggered array of spherical dimples. Significant 30-40% increasing of heat transfer without appreciable pressure losses were reported.

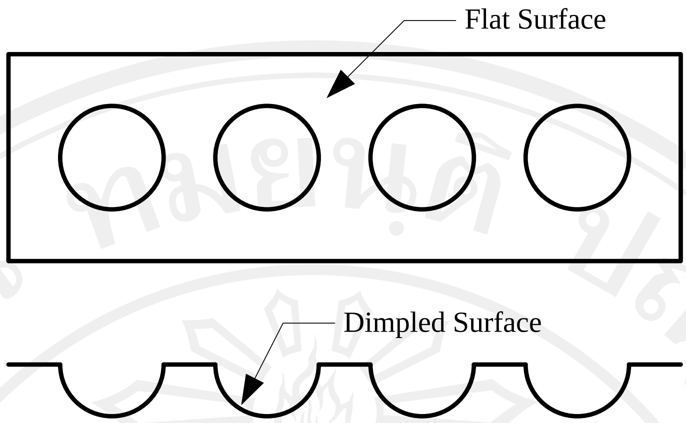


Figure 1.1 A flat plate having dimpled surface.

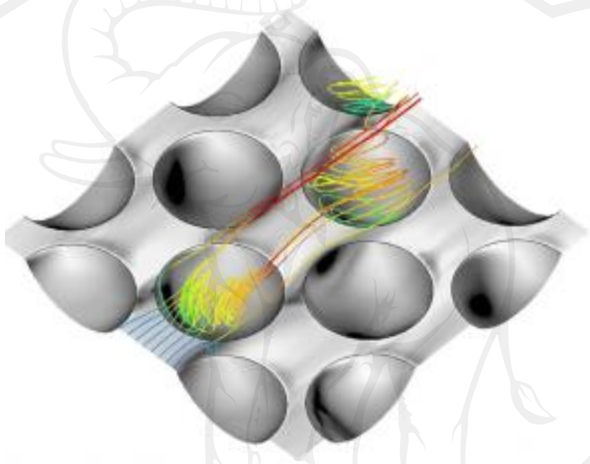


Figure 1.2 Vortex flow within its hole.

(Source; http://intellectualventureslab.com/?p=2567).

Chyu et al. (1997) studied the heat transfer coefficient distributions of air flow in the channel over flat surface indent with staggered arrays of the dimple. Their result shows the local heat transfer coefficients are significantly higher than values in the channels with smooth surfaces. Enhancements of about 2.5 times of smooth surface values, and pressure losses are about half the values produced by conventional rib turbulators.

Mahmood et al. (2001) described the flow structure above the dimpled flat surface. Flow visualizations showed vortical fluid as vortex pairs shed from the dimples. Their results showed the existence of a low heat transfer region in the upstream half of the dimple cavity followed by a high heat transfer region in the downstream half. Additional regions of high heat transfer were identified at the downstream rim of the dimple and on the flat surface of downstream of each dimple

Ligrani et al. (2001) reported that as the H/D (Channel height to dimple diameter) increase the secondary flow structures and flow mixing intensified decreased. Nevertheless, Moon et al. (2000) obtained almost a constant heat augmentation ratio of 2.1 for a dimpled passage with H/D from 0.37 to 1.49.

Burgess et al. (2005) reported that both the Nusselt number and the friction augmentation increased as the of dimple depth increased. These are attributed to: (i) increase in the strengths and intensity of vortices and associated secondary flows ejected from the dimples, as well as (ii) increase in the magnitudes of three-dimensional turbulence production and turbulence transport. The effects of these phenomena are especially apparent in local Nusselt number just inside, and the downstream edges of the dimples.

In addition, Wang et al. (2010) investigated a new enhanced heat transfer tube with ellipsoidal dimples. The dimples are disposed to form a certain specified angle between the major axis of the ellipsoid and flow direction, and the direction of the major axis of each adjacent ellipsoidal dimple in the same cross-section is alternated. Experimental tests were carried out with heating water on the shell side with a constant flow rate, and cold air in the tube side with flow rates range from 1 to 55 m^3/h. Their computed results indicated that the Nusselt number for ellipsoidal dimpled tube and spherical dimpled tube are 38.6–175.1% and 34.1–158% higher than that for the smooth tube respectively. The friction factors of dimpled tube were increased by 26.9–75% and 32.9–92% for ellipsoidal and spherical dimples compared with the smooth tube respectively. It was perceived that ellipsoidal dimple roughness accelerates transition to critical Reynolds numbers down to less than 1000.

The conclusion of the literature reviews for dimple surface is shown in Table 1.1.

Table 1.1 Conclusion of the literature review for dimple surface.

The author	Study	Flow type	Dimple geometry	Dimple size	Dimple depth	Dimple arrangement
Afansayev et al. (1993)	Study average heat transfer for turbulent flow over flat plate with dimple.	External flow	Spherical	×	×	×
Chyu et al. (1997)	Study flow structure, heat transfer and pressure drop over flat plate with dimple.	Internal flow	Spherical	×	×	×
Mahmood et al. (2001)	Study flow structure, heat transfer and pressure drop over flat plate with dimple.	Internal flow	Spherical	✓	×	×
Ligrani et al. (2001)	Study flow structure, heat transfer and pressure drop over flat plate with dimple.	Internal flow	Spherical	×	✓	×
Moon et al. (2000)	Study the effect of the dimple imprint diameter to transfer of flow over flat plate with dimple.	Internal flow	Spherical	✓	×	×
Burgess et al. (2005)	Study the influences of dimple depth to heat transfer of flow over flat plate with dimple.	Internal flow	Spherical	✓	×	×
Wang et al. (2010)	Study heat transfer and pressure drop in the ellipsoidal and spherical dimples tube dimpled in shell and tube heat exchanger.	Internal flow	Spherical and ellipsoidal	✓	×	×
This study	Study the thermal characteristic of air flow over flat plate with spherical and ellipsoidal dimple surface.	External flow	Spherical and ellipsoidal	×	×	✓

From the investigations, the results show the superior performance of dimple surface. However, the influence of dimple arrangement on the heat transfer performance has not report on the literature.

1.2 Tube heat exchanger

1.2.1 Dimple tube

1.2.1.1 Heat transfer performance on dimple tube

There are many thermal applications where the fluid-to-gas of tube heat exchanger is crucial element. The conventional enhanced heat transfer approaches such as finned tube, rough tubes, riblets tube and etc. These approaches are effective in increasing heat transfer rates; hence, this also results to the significant pressure drops. Dimple indent is the configuration aiming at increasing heat transfer rates without significant sacrifaction in pressure drop or fouling. Dimple tube give the superior to other approaches, as illustrated in Figure 1.3, the proposed dimpled approach achieves higher heat transfer rates at lower pressure drops, because of vortex flow patterns within the dimples; moreover, it is expected the dimpled approach will not increase (and may even reduce) fouling rates. The red line in Figure 1.3 represents the equality of heat transfer gain and pressure drop penalty associated with this gain reached by majority of the heat transfer enhancement techniques. Figure 1.3 shows that VHTE (vortex heat-transfer enhancement) provide a better balance of pressure drop and heat transfer than most other available techniques. As illustrated in Figure 1.4, each dimple works as a vortex generator that intensifies the rate of convective heat transfer and mass transfer to the dimpled surfaces.

1.2.1.2 Literature review

They are many researchers have applied the dimple on tube surface, for shell and tube application, Chen et al. (2001) investigated heat transfer enhancement in a coaxial-pipe heat exchanger using dimples as the heat transfer modification on the inner tube. Tube-side Reynolds numbers were in the range of 7,500 – 52,000. They reported yielding inward-facing, raised dimples on the inner tube increased the values of heat transfer coefficient significantly above those for the smooth tube. Heat transfer enhancement ranged from 25% to 137% at constant Reynolds number, and from 15% to 84% at constant pumping power. At a constant Reynolds number, the relative *J* factor had values from 0.93 to 1.16, with four dimpled tube configurations having values larger than unity.

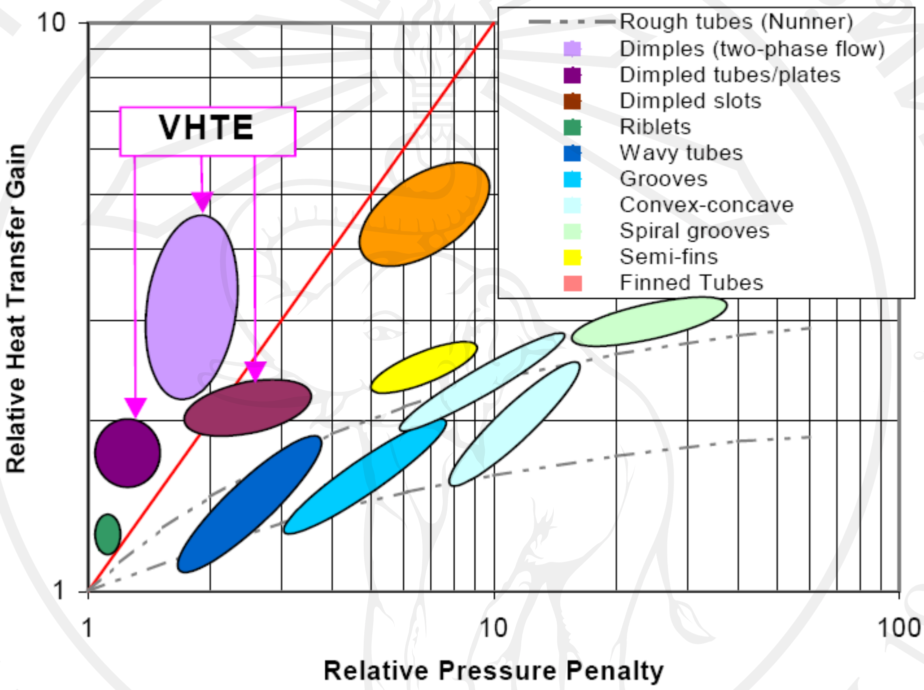


Figure 1.3 Compared to other types of heat transfer enhancement approaches (Source; Chudnovsky and Kozlov (2006)).

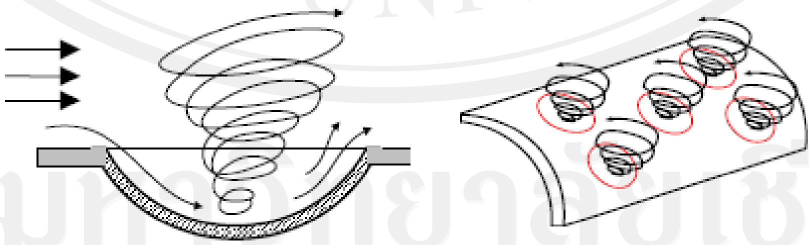


Figure 1.4 Vortex generations on dimple surface (Source; Chudnovsky and Kozlov (2006)).

Since then Vicente et al. (2002) presents the experimental results carried out in dimpled tubes for laminar and transition flows and completes a previous work of the authors focused on the turbulent region. The flow range was Ra = 10^6–10^8. The experimental results of pressure drop for laminar flow showed dimpled tube friction factors between 10% and 30% higher than the smooth tube. Moreover, it was perceived that roughness accelerates transition to critical Reynolds numbers down to 1400. Results showed that at low Rayleigh numbers, heat transfer is similar to the smooth tube one whereas at high Rayleigh, enhancement produced by dimpled tubes can be up to 30%.

For cross flow heat exchanger application, Chudnovsky and Kozlov (2006) designed, constructed, and tested to investigate the effects of adding deep dimples, or shallow dimples to 6 tubes, contained in a total bank of 14 tubes. Significant heat transfer augmentations (compared to a bank of smooth tubes) are produced by using dimples on the surfaces of the tubes employed in the tube bank. The increases in form drag and pressure losses provided by the dimples are small because of they do not protrude into the flow. From their results, it is estimated that overall friction factor ratios range from 0.97 to 1.03, and overall Nusselt number ratios are estimated to range from 1.35 to 1.57 if all 14 of the tubes in the tube bank contain shallow dimples. If all 14 of the tubes in the tube bank contain deep dimples, overall friction factor ratios are estimated to range from about 1.08 to 1.12, and overall Nusselt number ratios are estimated to range from about 1.25 to approximately 1.40.

1.2.2 Oval and flat tube

1.2.2.1 Heat transfer performance on oval and flat tube

Fin-and-tube heat exchangers with flat tubes are widely used as automotive radiators or condensers of automotive air conditioning systems. Usage of oval or flat tubes instead of round tubes will enhance the air-side performance. The refrigerant charge will also be reduced compared with that in round tube. Figure 1.5 shows an automotive radiator geometry having louvered plate fins on flat tube.

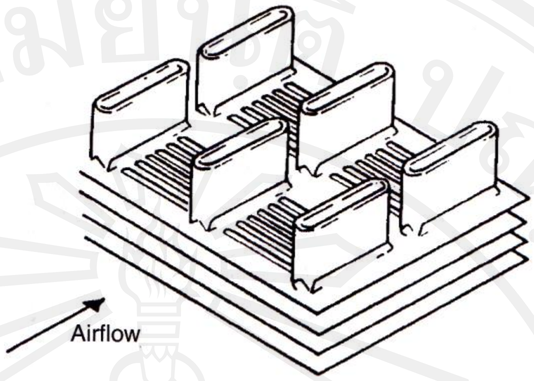
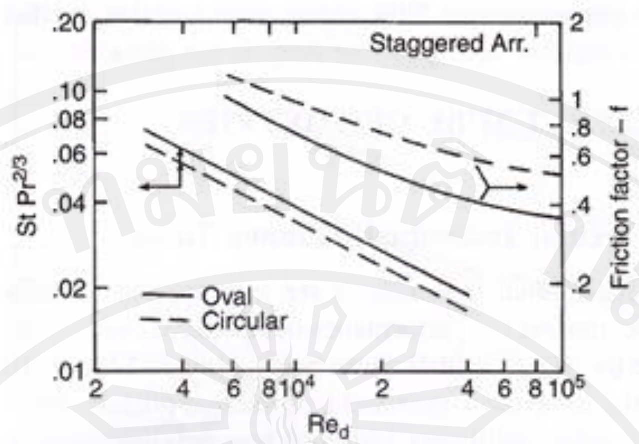


Figure 1.5 Illustration of the louvered plate fin automotive radiator with in line tubes. (Source; Achaichia and Cowell (1988)).

1.2.2.2 Literature review

Oval and flat cross-sectional tube shapes are also applied to individually finned tubs. Figure 1.6 compares the performance of staggered banks of oval and circular finned tubes tested by Brauer (1964). Both banks have 312 fin/m, 10-mm-high fins on approximately the same transverse and longitudinal pitches. The oval tubes gave 15% higher heat transfer coefficient and 25% less pressure drop than the circular tubes. The performance advantage of the oval tubes results from lower form drag on the tubes and the smaller wake region on the fin behind the tube. The use of oval tubes may not be practical unless the tube-side design pressure is sufficiently low.

Min and Webb (2004) numerically investigated the effect of tube aspect ratio of an oval tube on the air-side heat transfer and pressure drop characteristics of an infinite row heat exchanger having herringbone wavy fins. The numerical calculations were performed in three dimensions for two frontal air velocities 2.54 and 3.39 m/s yielding hydraulic diameter Reynolds numbers of 770 and 1150. Investigated were five tube geometries including a round tube, three elliptical oval tubes, and a flat tube. The tube



Tube Bank Dimensions (mm)

	Circular	Oval
Tube dia. (d)	29	19.9/35.2
Fin. ht. (e)	9.8	10/9.3
Fin. thk. (t)	0.4	0.4
Face pitch (S_t/d)	1.03	1.05
Row pitch (S_l/d)	1.15	1.04
Fins/m	312	312

Figure 1.6 Heat transfer and friction characteristics of circular and oval finned tubes in a staggered tube layout as reported by Brauer (1964).

configurations are shown in Figure 1.7, and the geometric details we provided in Table 1.3. The 15.88-mm round tube served as the baseline tube. The longitudinal and transverse tube pitches were 31.75 and 30.48 mm. The corrugation height and the projected corrugation pitch of the fin was 1.67 and 10-58 mm. The fin pitch was 2.117 mm. The four oval tubes were made by reforming the baseline round tube and had the same perimeter as that of the round tube. The aspect ratios of the three elliptical tubes were 2.0, 3.0, and 4.29, respectively, and the aspect ratio of the flat tube was 3.0. As the tube aspect increased, the air-side heat transfer coefficient and pressure drop decreased. At 2.54-m/s frontal velocity, the 3.0-aspect-ratio elliptical tube yielded a 9.0% lower air-side heat transfer coefficient and a 48.7% lower air-side pressure drop than the round tube. As compared to the 3.0-aspect-ratio elliptical tube, the same aspect ratio flat tube had a 2.3% higher air- side heat transfer coefficient and a 6.0% higher pressure drop.

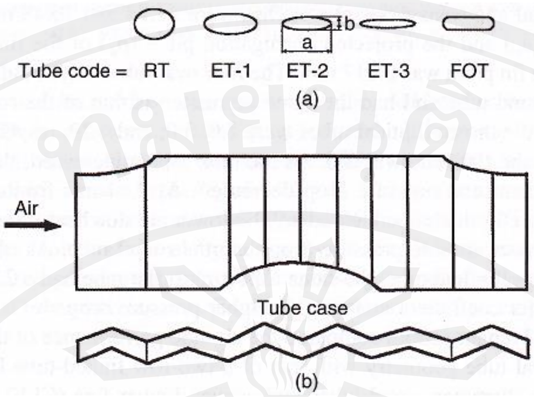


Figure 1.7 Tube geometries considered by Min and Webb for numerical calculation: (a) cross-sectional shape, (b) computational domain for the ET-2 oval tube case. (Source; Min and Webb, (2004)).

Table 1.2 Tube dimensions considered by Min and Webb (2004) for numerical calculation.

Tube Code	a (mm)	b (mm)	a/b	D_h(mm)	$D_h/(D_h)_{RT}$
RT	15.88	15.88	1	14.86	1
ET-1	20.59	10.3	2	12.44	0.82
ET-2	22.37	7.45	3	9.26	0.62
ET-3	23.34	5.44	4.29	6.55	0.45
FOT	20.95	6.98	3	9.54	0.64

Webb and Iyengar (2000) compared the air-side performance of the 1024 fins/m oval tube geometry as shown in Figure 1.8 with that of a two-row finned-tube heat exchanger having 8.0-mm-diameter round tubes and convex louver fins at 807 fins/m, tested by Wang et al. (1996). Operating at the same frontal velocity, the comparison shows that oval tube heat transfer coefficient value is 10% higher and the pressure drop is 17% lower than that of the 8.58-mm-diameter round tube design. The oval tube design has approximately 20% greater surface area ratio than the 807 fins/m convex louver fins design. Hence, the oval tube design offers 32% greater hA/A_{fr} (1.10 × 1.20) than the round tube design. The 32% greater hA/A_{fr} and 17% lower pressure drop offers a significant performance improvement.

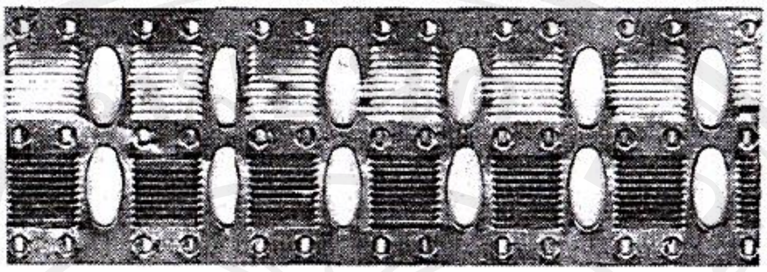


Figure 1.8 Photo of oval tube fins used in analysis of Webb and Iyengar.
(Source; Webband Iyengar (2000)).

The conclusion of the literature reviews for dimple tube, and oval and flat tube are shown in Table 1.3 and Table 1.4 respectively.

Table 1.3 Conclusion of the literature review for dimple tube.

The author	Study	Flow type	Dimple geometry	Tube arrangement
Chen et al. (2001)	Study the heat transfer performance and pressure drop of flow inside the double pipe heat exchanger	Internal flow	Spherical	×
Vicente et al. (2002)	Study the heat transfer performance and pressure drop of flow inside the dimple tube of shell and tube heat exchanger	Internal flow	Ellipsoidal	×
Chudnovsky and Kozlov (2006)	Study the heat transfer performance and pressure drop of flow across the round tube bank heat exchanger	External flow	Spherical	✓
Wang et al. (2010)	Study the heat transfer performance and pressure drop inside the dimple tube of shell and tube heat exchanger.	Internal flow	Spherical and ellipsoidal	×
This study	Study the heat transfer performance and pressure drop of flow across the flat-tube bank heat exchanger	External flow	Ellipsoidal	✓

Table 1.4 Conclusion of the literature review for oval and flat tube.

The author	Study	HEX. type	Research Method	Tube geometry
Brauer (1964)	Study the heat transfer and pressure drop of air flow over tube bank.	Fin-and-tube	Experiment	Oval and round
Min and Webb (2004)	Study the heat transfer and pressure drop of air flow over tube bank.	Fin-and-tube	Numerical	Oval, flat and round
Web and Iyengar (2000)	Study the heat transfer and pressure drop of air flow over tube bank.	Fin-and-tube	Experiment	Oval
This study	Study the heat transfer and pressure drop of air flow over tube bank.	Dimple tube w/o fin	Experiment	flat

From the literature review, heat exchanger equipped with oval or flat-finned tubes give the higher heat transfer coefficient and less pressure drop than those with circular-finned tube. Applying the dimples on flat tube is possibly beneficial approach due to its potential in wake induction. In this research, flat-dimpled tube bank without fins is in focus to study the heat transfer enhancement and pressure drop in comparison with flat finned-tube bank. Ellipsoidal dimple geometry will be used primarily.

1.3 Research objectives

1) To determine the dimple shape which offers the optimal heat transfer performance.

2) To determine the dimple arrangement which offers the optimal heat transfer performance.

3) To determine the thermal characteristics of air flow over flat-dimple tube bank with different tube rows arrangement.

4) To study the flow structure on the dimple surface and study the drag coefficient of air flow over flat-dimple tube.

1.4 Potential benefits

1) The novel design of flat tube heat exchanger having dimples surface is proposed. The new design will have better performance than the conventional type.

2) The empirical data will benefit for designers to design heat exchanger.

1.5 Scope of the study

1) Numerical work on heat transfer performance analysis and pressure loss of air flow over flat surface with different dimples shape.
2) Experimental work on heat transfer performance analysis of air flow over the spherical and ellipsoidal dimples on flat surface with different dimples arrangement.
3) Experimental work on heat transfer performance analysis of air flow over flat-dimple tube bank with different tube arrangement.
4) Numerical work and flow visualization to describe the flow structure and drag coefficient on the dimple tube and dimple surface.