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EXPERIMENTAL INVESTIGATION OF THE THERMAL PERFORMANCE OF HELICAL COILED FINNED TUBE HEAT EXCHANGER WITH THE EFFECT OF AIR INJECTION TECHNIQUE

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EXPERIMENTAL INVESTIGATION OF THE THERMAL PERFORMANCE OF HELICAL COILED FINNED TUBE HEAT EXCHANGER WITH THE EFFECT OF AIR INJECTION TECHNIQUE

A THESIS

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(M.TECH)

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بسم الله الرحمن الرحيم

ربِّ أوزعني أَنْ أَشْكُرَ نعمتك الَّتِي أَنْعَمْتَ
عَلَيَّ وَعَلَىٰ وَالِدَيَّ وَأَنْ أعمل صَالِحًا تَرْضَاهُ
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الْمُسْلِمِينَ

صَدَقَ اللَّهُ الْعَلِيُّ الْعَظِيمُ

سورة الأحقاف: آية 15

Disclaimer

I confirm that the work submitted in this thesis is my own work and has not been submitted to other organization or for any other degree.

Signature:

Name: Muna Ali Talib

Date: / / 2021

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MUNA ALI TALIB

2021

Supervisors Certification

We certify that the thesis entitled **" EXPERIMENTAL INVESTIGATION OF THE THERMAL PERFORMANCE OF HELICAL COILED FINNED TUBE HEAT EXCHANGER WITH THE EFFECT OF AIR INJECTION TECHNIQUE**

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Abstract

Transferring thermal energy efficiently necessitates using a heat exchanger capable of producing the full thermal power of the energy supply at the lowest possible cost and time. Traditional surface style heat exchangers have significant disadvantages, such as the high heat transfer resistance of the surface, fouling issue, and increasing cost. So, any attempt to enhance the heat transfer characteristics of a surface-type heat exchanger enhances its thermal performance. The air bubble injection and finned helical coil tube are two of the most promising enhancement techniques recently suggested in a separate image.

So, in the present study, the effect of merged enhancement techniques, air injection and finned (externally) helical coil tube on the thermal performance of a vertical counter-current coiled heat exchanger was investigated experimentally. The air was injected into the shell side of the heat exchanger as air bubbles with an average diameter of approximately 100 μm via a porous sparger (new injection method). Furthermore, the study was conducted to optimize the operational parameters of the void fraction (air and water volumetric flow rates) of the shell side under laminar flow ($316 \leq Re \leq 1223$). The experiments were performed with Qs=2,4,6,8and 10LPM, Qa=0,2,4,and 6 LPM, and working fluid inlet temperature difference ΔT were 20 ^{0}C. The effect of varying the operating parameters on the overall heat transfer coefficient U, effectiveness ε, and number of heat transfer units (NTU) with and without air bubbles injection and compared. Besides, we studied the effect size of pitch for finned tube on the thermal performance of the heat exchanger ..

The experimental results showed that the overall heat transfer coefficient and heat exchanger effectiveness were improved significantly due to air bubbles injection,

the maximum enhancement ratio of (U_{fa}/U_{Sa}) was up to 119 %, $(\varepsilon_{fa}/\varepsilon_{sa})$ was 122% and Ntu_{fa}/Ntu_{Sa} was 116% compared with pure water The aforesaid maximum

enhancement ratios were achieved when $Qs = 10LPM$, $Qa = 2$ LPM, $Qh = 2LPM$, size of pitch for fin = 1cm and $\Delta T = 20^0C$. While the minimum enhancement ratio of of (U_{fa}/U_{Sa}) was up to 39 %, $(\varepsilon_{fa}/\varepsilon_{sa})$ was 18% and Ntu_{fa}/Ntu_{Sa} was 36% Foresaid minimum ratio occurred at $Qs = 2LPM$, $Qa = 6$ LPM, $Qh = 1$ LPM, size of pitch for fin $= 2cm$ and $\Delta T = 20^0C$.furthermore, our experiments conclude that $Qa = 2$ LPM and $Qs = 6$ LPM are the optimum flow rates under the current experimental conditions to get on highest improvement ratio of (U_{fa}/U_{Sa}) , $(\varepsilon_{fa}/\varepsilon_{sa})$ and Ntu_{fa}/Ntu_{Sa}.

On the other hand, for the finned tube, the maximum enhancement ratio of of (U_{fa}/U_f) was up to 83 %, $(\varepsilon_{fa}/\varepsilon_f)$ was 55% and Ntu_{fa}/Ntu_f was 81% is clearly obtained at Qs = 10 LPM, Qa = 2 LPM, and size of pitch for fin = 1cm while the minimum enhancement of $(U_{fa}/U_{f)}$ was 27%, $(\varepsilon_{fa}/\varepsilon_f)$ was 11% and Ntu_{fa}/Ntu_f was 26% is clearly obtained at $Qs = 2$ LPM, $Qa = 6$ LPM, and size of pitch for fin = 2cm and $\Delta T = 20\ ^oC$.furthermore, the optimum flow rates under the current experimental conditions to get on highest improvement ratio of $(U_{fa}/U_{f)}$ and $(\varepsilon_{fa}/\varepsilon_{f)}$ was $(Qs = 10$ LPM$)$ and $(Qa = 2$ LPM$)$.

. Additionally, the all value increases with increased coil side flow rate due to increasing thermal capacity (hot fluid) for the same heat process time.

.Contents

NOMENCLATURE

Symbol	Definition	Unit
As	The surface area of heat transfer	m^2
C	Thermal capacity	J/s.k
c	Specific heat	J/kg.K
D_c	Curvature diameter	mm
d_i	Inner coiled tube diameter	mm
D_o	Outer diameter	mm
H	Height column	mm
h	Height coil	mm
k	Thermal conductivity	W/m.K
De	Dean number	-
L	Length	mm
m^o	Mass flow rate	Kg/s
N	Number of turns	-
p	Coil pitch	mm
p_f	The pitch of the Spiral fin	mm
D_b	Diameter bubble	mm
R_c	curvature radius	mm
Re	Reynolds number	-
t	tubes thickness	mm
T	Temperature	K
q	Heat transfer rate	W
Q	Volumetric Flow rate	L/min
Q_a	Volumetric Air Flow rate	L/min
Q_h	Volumetric Hot water Flow rate	L/min
Q_s	Volumetric Cold water Flow rate	L/min
L/min	Litter per minute	-
W	uncertainty in the measurement	-
U	Overall heat transfer coefficient	$W/m^2.\ k$
U_s	Overall heat transfer coefficient (smooth tube)	$W/m^2.\ k$
U_f	Overall heat transfer coefficient (finned tube)	$W/m^2.\ k$
U_{sa}	Overall heat transfer coefficient (smooth tube with air)	$W/m^2.\ k$
$U_{fa=U}$	Overall heat transfer coefficient (finned tube with air)	$W/m^2.\ k$

U_{fa}/U_f	Enhancement ratio for Overall heat transfer coefficient (finned tube with air) to Overall heat transfer coefficient (finned tube without air)	-
U_{fa}/U_{sa}	Enhancement ratio for Overall heat transfer coefficient (finned tube with air) to Overall heat transfer coefficient (smooth tube with air)	-
Subscripts		
a	Air	-
c	Cold fluid	-
h	Hot fluid	-
i	Inner	-
in	Inlet	-
o	Outer	-
out	Outlet	-
min	Minimum	-
LMTD	Logarithmic mean temperature difference	-
Greek Symbols		
ρ	Fluid Density	kg/m^3
μ	Dynamic viscosity	Pa.s
ε	Effectiveness	-
ε_s	Effectiveness of smooth tube	-
ε_f	Effectiveness of finned tube	-
ε_{sa}	Effectiveness (smooth tube with air)	-
ε_{fa}	Effectiveness (finned tube with air)	-
$\varepsilon_{fa}/\varepsilon_f$	Enhancement ratio for effectiveness (finned tube with air) to effectiveness (finned tube without air)	-
Ntu	Number of heat transfer unit	
Ntu_{fa}/Ntu_f	Enhancement ratio for Number of heat transfer unit (finned tube with air) to Number of heat transfer unit (finned tube without air)	
Ntu_{fa}/Ntu_{sa}	Enhancement ratio for Number of heat transfer unit (finned tube with air) to Number of heat transfer unit (smooth tube with air)	

Chapter One

Introduction

INTRODUCTION

1.1Preface

Heat exchanger is an apparatus that employed to transfer heat between two or more fluids placed at different temperatures and in thermal contact with each other. Heat exchanger is employed either singly or compound with a large thermal system in many domestic, commercial and industrial applications, for example, power plant, ventilation, air-conditioning and refrigeration systems, manufacturing,processing,automotiveandaerospaceindustries,Theheatexchanger almost utilized in such process. The objective design of an efficient energy conversionprocessrequiresasuitableheatexchangertoextractamaximumpossible thermal potential of the system's energy source **[1]**. The heat exchanger can be divided into two kinds:

1.1.1 Indirect contact heat exchanger: this kind of heat exchangers the hot and cold fluids are separated by barriers made from a metallic, thermal energy is exchanged through it. Because of the high heat transfer resistance of the metallic barriers, the ability of this type of heat exchanger to extract the thermal potential is limited. In addition to the high cost which results from the large heat transfer area that required overcoming the low efficiency of the surface type heat exchanger.

1.1.2 Direct contact heat exchanger: this kind of heat exchangers direct contact heat exchange doesn't have such problems such as in indirect contract heat exchangers, it relates to the transportation of thermal energy between two or more fluid streams when they are brought into intimate contact with one another **[2, 3, 4, 5, 6 and7]**.

The process of increasing thermal performance of heat exchangers is called heat transfer enhancement which aiming to reduce size, weight, as well as cost manufacturing of heat exchanger (especially surface type). Therefore heat transfer enhancement is an active area of research. Researchers suggested many heat transfer augmentation methods or enhancement techniques which classified into two groups active methods and passive methods.

1.2 Classification of Heat Transfer Enhancement Techniques.

Heat transfer enhancement techniques can be classified into three groups:

1. Passive Techniques
2. Active Techniques
3. Compound Techniques.

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1.2.1 Passive Techniques:

The principle of heat transfer improvement in this methods is to improve thermal conductance. This technique uses a surface and geometrical modifications for the flow channel. Passive method hold the advantage over the active method as its doesn't require any direct input of external power. Treated surfaces can achieve heat transfer augmentation, this methods comprise **[8, 9, 10, 11, 12 and13]**.

- ❖ Rough surfaces: These are the surface modifications that promote turbulence in the flow field at the wall region, primarily in single-phase flows, without an increase in heat transfer surface area.
- ❖ Extended surfaces: They provide effective heat transfer surface area. Recently,

modern development has led to modified finned surfaces that also tend to enhance the rate of heat transfer coefficient by disturbing flow field as well as to increasing heat transfer surface area.

- Displaced enhancement devices: These are the inserts that are used primarily in confined forced convection. They improve energy transport in directly by

  Promoting turbulence and flow mixing at heat exchange surface by displacing fluid from heate dor cooled surface of duct with bulk fluid from the core flow. for example, using wire matrix inside pipe
- Swirl flow devices: They produce swirl or secondary recirculation for axial flow in a channel. These include helical striper cored screw type tube inserts, twisted taps and various forms for altered flow arrangements. They can be used for a single-phase as well as two-phase flow.
- Surface tension devices: These consist of wicking or grooved surfaces, which improve the flow of liquid to the boiling surfaces and from the condensing surfaces.
- Curved tubes: These lead to relatively more compact heat exchangers. The tube curvature due to coiling produces secondary flow and vortices, which promote higher heat transfer coefficients in single-phase flow as well as in most regions of boiling.
- Use of additives: These contain an addition of soluble-polymers and gas bubbles in a single-phase liquid flow or liquid droplets in a gas flow. Additionally, they include the addition of solid nanoparticles to enhance the thermo physical properties

1.2.2 **Active Heat Transfer Enhancement Technique:**

From the use and design point view, this technique is complex . its requires some external power to facilitate the desired flow modification and the associated improvement in the heat transfer rate. Heat transfer augmentation by this method can be achieved by using mechanical aids**,** such as instruments that can be used to stir the fluid by mechanical means or by rotating surface. These include rotating the heat exchanger tubes. Briefly this enhancement technique involves **[11, 12, 13]**:

1. **Surface vibration:** It is applied primarily at either low or high frequency, in a single-phase flow to obtain a higher convective heat transfer coefficient.

2. **Fluid vibration:** Pulsations are created in the fluid itself instead of applying vibrations to the surface. This method is used primarily in a single-phase flow and it is considered to be perhaps the most practical type of vibration enhancement due to the mass of the heat exchangers.

3. **Electrostatic fields:** They can be in the form of electric ormagnetic field, from direct or alternating current sources. These methods apply mainly in the heat exchange systems involving dielectric fluids. Depending on the application, it can also produce greater bulk fluid mixing and induce forced convection or electromagnetic pumping to enhance heat transfer.

1.2.3 Compound Techniques:

A compound augmentation technique is the one where more than one of the above mentioned techniques is used in combination with the purpose of further improving the thermo-hydraulic performance of a heat exchanger.

1.2.3.1 Helically Coiled Tube Effect on Enhancement Heat Transfer:

According to the above-mentioned heat transfer enhancement techniques, curved tubes, particularly coiled tubes, have been adopted as essential passive methods due to their compact design. In addition, it has a high coefficient of heat transfer in comparison with the straight tubes **[14]**. Therefore, curved tubes are the most commonly used tubes in several heat transfer applications, such as air conditioning cooling systems, chemical reactors, food dairy processes **[14]**, and heat recovery **[15]**. The curved tube types are shown in Fig 1.1.

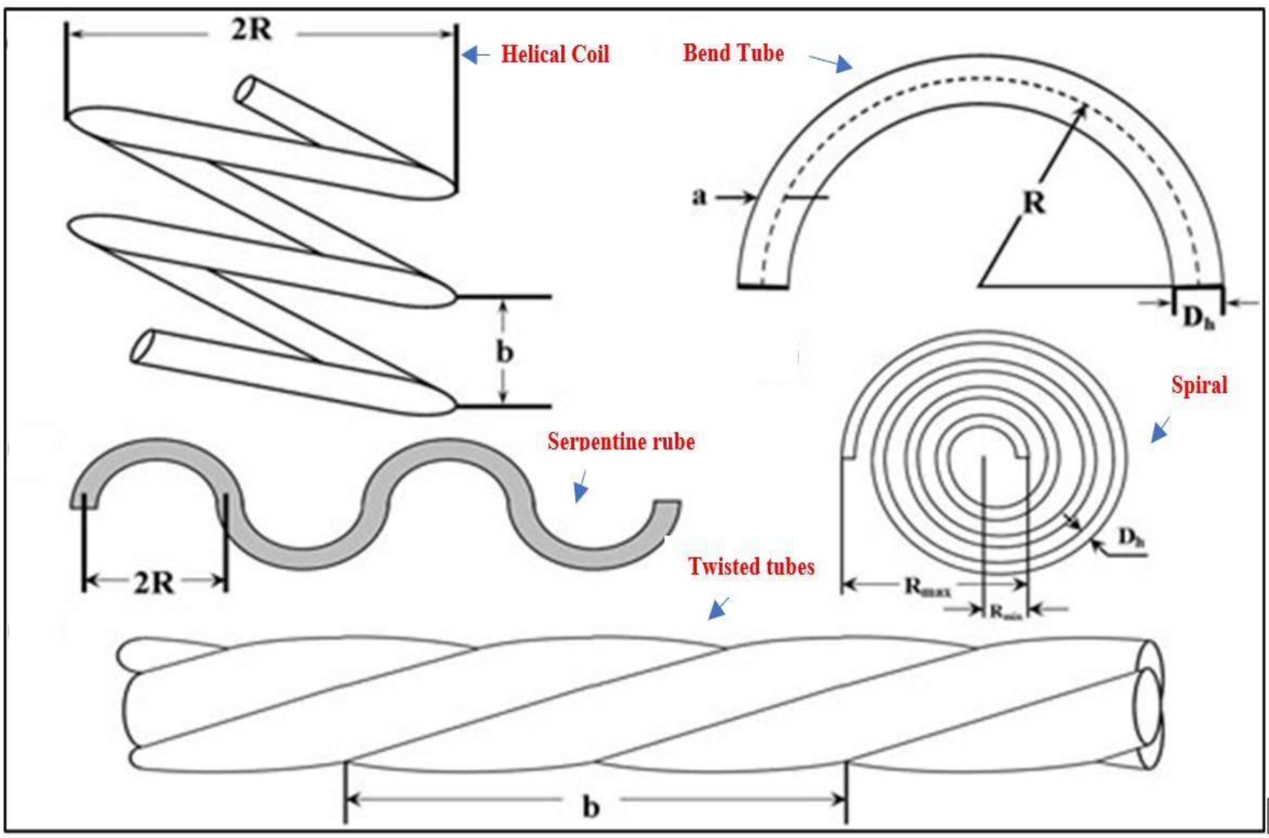

Figure 1.1: Types of curved tube geometries **[14]**.

Helically coiled tubes Fig.1.1 are the well-known type of curved tubes used in a large area of heat transfer applications. It provides a simple and effective means of heat transfer augmentation in a wide variety of industrial applications. The centrifugal forces caused by the curvature of the tube produce a secondary flow field with a circulatory motion pushing the fluid particles toward the core region of the tube. The intensity of the secondary flow field increases as the flow rate increases, and due to the stabilizing effects of this secondary flow, laminar flow persists to much higher Reynolds numbers in the helical coil than in straight tubes **[14]**. Consequently, the heat transfer performance differences between the coils and straight tubes are particularly distinct in the laminar flow region, which received most research attention **[16]**.

1.2.3.2 Secondary Flow in a Helical Coiled Tube:

The geometry of the shell and helically coiled tube heat exchanger is shown schematically in Fig.1.2. When a fluid flows in a helically coiled tube, and due to the interaction between centrifugal and viscous or frictional forces in the curved portion of the flow, a specific characteristic motion is known as secondary flow, as depicted in Fig.1.3. Secondary flow causes displaced fluid of curved pipes outer wall to the curved pipes inner wall. i.e., the fluid in the central region of the pipe moves away from the center of Curvature, and the fluid near the pipe wall flows towards the center of Curvature, but at the same time, this will increase pressure drop (Δp) **[17]**.

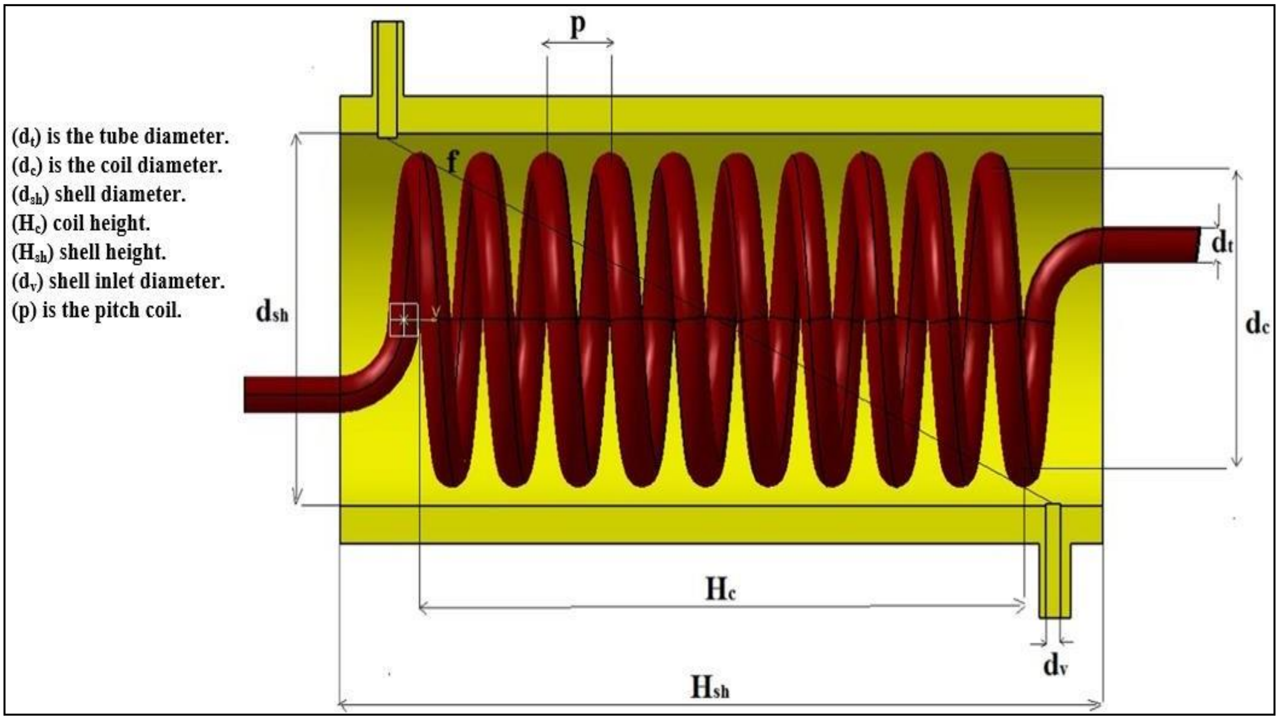

Figure 1.2: Schematic of shell and coiled tube heat exchanger **[17]**.

Dean was experimentally showed that the dynamical similarity of secondary flow depends on a non-dimensional parameter called Dean Number (De) **[18]**:

$$De = \left(\frac{d_t}{2R_c}\right)^{0.5} \left(\frac{d_t V_m}{\nu}\right) \qquad (1.1)$$

Where De is the Dean Number, V_m are the mean velocity along the pipe, ν kinematic viscosity, and d_t is the diameter of the pipe which is bent into a coil of radius R_c **[18]**.

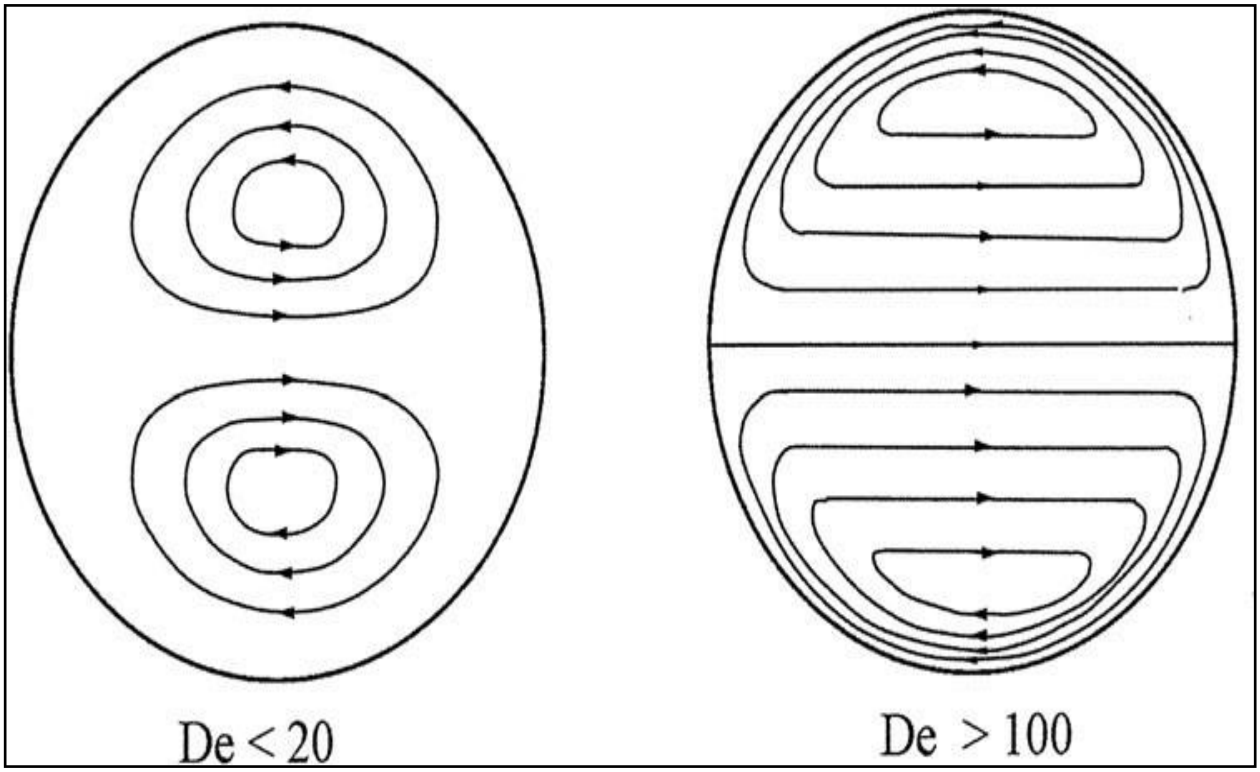

Figure 1.3: Secondary flow areas at low and high Dean Numbers **[19]**.

So, to achieve the maximum thermal exchange in a minimum volume, the ratio of heat exchanger surface area (A_S) to its volume (V) that called area dencity (β) ,should be over($\beta > 700\ m^2/m^3$) to form a compact heat exchanger **[20]**. Therefore, the choice usually fell out on coiled tube heat exchangers to achieve the maximum thermal exchange. since the coil tube is compact; therefore, it requires a smaller space. In addition, it is easy to produce, can operate at high pressure, and is suitable for use under conditions of laminar flow or low flow rates on the shell side. Thus, these heat exchangers can often be very cost-effective **[21]**. Consequently, the focus has begun to enhance further the thermal performance of the helical coiled tube heat exchangers using the different techniques mentioned above.

1.3 Heat Transfer Enhancement by Air Bubbles Injection into Shell Side of Heat Exchanger:

Any desire to improve the heat exchangers thermal performance leads to additional pressure drops, costs, electricity, material, and weight. For example, the turbulators mentioned above are significantly increased the pressure drop along with the heat exchangers. As a result, any attempt to minimize the pressure drop will reduce the increment of heat transfer. Therefore, experts are still looking for a technique that can improve the thermal performance of heat exchangers with a minimum possible pressure drop **[20]**.

Some researchers have investigated sub-millimeter air bubble injection, such as **kitagawa et al. [22]**, as a heat transfer amplification technique for natural convection of static fluids. Several academic experiments have been conducted to explain bubbles fluid-dynamic behavior within liquids **Sadighi et al. [23]** recently suggested the direct application of tiny air bubbles to improve heat exchangers thermal characteristics. Air injection into any liquid contributes to the formation of air bubbles within the fluid. The quantity, shape, distribution, and scale of these air bubbles depend on the method used for air injection. As air bubbles are formed, they tend to float vertically through the liquid due to buoyancy. Air bubbles normal behavior will increase turbulence and break the boundary layer of fluid flow. Thus, tiny air bubbles vertical movement and stability can be used as an appropriate, efficient technique to increase heat exchangers thermal performance **[24]**. Fig 1.4 clearly describes the injection of air bubbles in a vertical shell and a coiled tube heat exchanger.

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1.4 Problem statement

the previous experiments concentrated on a narrow variety of operational conditions, which are insufficient to explain the process by which the bubbles caused the improvement. However, the present work may be adopted as an expansion of the work in question on this methodology and to fill a void that exists in the literature with the following points:

1. Investigate the effect of injected air bubbles on the thermal performance of four shells and helically coiled tube heat exchange.

   - Smooth coiled tube heat exchange.
   - Finned Coiled tube with 1.cm size of pitch .
   - Finned Coiled tube with 1.5 cm size of pitch.
   - Finned Coiled tube with 2 cm size of pitch.

2. Study the influence of variation in the injected air flow rate and pitch of spiral fins on the thermal performance of a shell and helically coiled smooth and (externally) finned tube heat exchanger.
3. Different heat transfer parameters are calculated depending on the measurement; these include:

   - Overall heat transfer Coefficient
   - Effectiveness.
   - The number of thermal unit(NTU)

To achieved these objectives, an appropriate experimental set-up was designed and manufactured especially for this experimental study and equipped with the necessary measurement instruments for temperature, and flow rates.

1.5 Thesis Outline

The present thesis is divided into five chapters and four appendixes, as flowing:

- The current chapter [chapter one] is a general introduction to heat exchanger enhancement techniques, helically coiled (smooth and finned) tube heat exchangers, fluid flow mechanism inside the coiled tube, and finally, the air bubble injection technique.
- Chapter two presents a review of the literature that is relevant to the topic of the present thesis. These studies include the most popular enhancement techniques of the shell and helically coiled tube heat exchanger in chronological order. These include the surface modification technique, compound enhancement techniques.
- Chapter three describes the experimental work. A description of the experimental apparatus used to investigate the effect of air bubbles injection on heat transfer characteristics of the vertical shell and helical coiled (smooth and finned) tube heat exchanger is given. The procedure of the experiments performed and the physical properties of the working fluid are outlined in chapter three.
- Chapter four presents the discussion of the experimental results obtained for various parameters at different operational conditions.
- Chapter five is about the conclusions obtained from the experimental study and provides some recommendations for future work.

1.6 Scope of The Present Work

It can be summarized as follows:

1. The air injection technique seems one of the most promising techniques for improving the thermal performance of a shell and helically coiled tube heat exchangers. Nonetheless, further laboratory experiments and theoretical (analytical, numerical) studies are needed to fully comprehend these essential enhancement techniques.

2. Hard turbulators enhancement techniques mentioned in the literature survey, such as wire coiled, twisted tape, corrugated tubes, wavy stripes, barbed wires, and springs, require additional materials, increasing the heat exchanger's cost, increased pressure drop, and weight**.**

3. It has been shown clearly that no investigation technique has been reported on the air injection as enhancement techniques ways used in coiled finned tube and shell heat exchanger .The present work is introduced as a new knowledge adding for the mentioned works on the coiled finned tube and shell heat exchanger with bubble of air injection technique and to fill the gap that clear shown in the literature survey with the following organization:

Creating a suitable helical external finned tube by wilding copper wire with 1mm diameter with three pitches wrapped 1, 1.5 and 2.0 cm a helical coiled tube (passive technique)

investigate a new porous sparger with outer diameter 5 cm for injection air bubbles into shell side (active technique)

Investigate wide range of parameters that non-investigated before. These parameters include: for example wrapped copper wire around helical tube , a wide range of the shell side flow rate ($Qshell$=2,4,6,8 and 10 LPM) ,the air flow rate

($Qair$=0,2,4,and6LPM), the coil side($Qcoil$ = 1, 1.5 and 2 LPM) and finally the inlet temperature difference (ΔT=20°c)

Chapter Two

Literature Review

LITERATURE REVIEW

2.1 Introduction

More recently, air bubble injection has been suggested as one of the promising enhancement methods of heat exchanger thermal performance**.**Upon air injected into a liquid, an air bubble will form within the liquid. However, these bubbles number, shape, size, and distribution rely on the injection technique and the fluids properties. Once the bubbles form in the liquid, naturally, they float and move upward due to the buoyancy force, which initiates by the difference in the densities of the liquid and air. This random motion of bubbles will mix the liquid bulk and break down the hydrodynamic and thermal boundary layers, enhancing heat exchange in thermal systems, such as heat exchangers. However, the mixing level depends on many factors, such as the air bubbles flow rate, number of bubbles, size of bubbles, and heat exchanger design **[23-27]**

2 .2 Air injection enhancement technique.

The natural behavior of an air bubbles injection into any liquid due to the buoyancy force, can improve the turbulence level and destroy the thermal boundary layer of the fluid flow. So, vertical movement and mobility of small air bubbles can be used as a suitable active technique to improve the thermal performance of heat exchangers. Two phase flow and air bubbles injection studies were expressed as follow.

Firstly, The influence of the injection period on the bubble rise velocity and their shape were experimentally studied by **D. Funfschilling and H. Z. Li [28].** The exprimental performed with three different fluids were employed: a newtonian fluid (99.5% glycerol), an inelastic shear thinning fluid (2% carboxymethylcellulose—CMC in water) and an elastic shear thinning fluid (0.5% polyacryl- amide—PAAm in water). Flow field results around a rising bubble reached by PIV tend to show a disconnection between the shape of the bubble and surrounding flow field.

U. Puli et al [29] experimentally examined behavior of bubbles in subcooled flow boiling of water in a horizontal annulus at mass fluxes ranged from 400 to 1200 kg/m2-s, heat fluxes ranged from 0.1 to 1 MW/m2, and pressures varying from 1 to 4 bar using a high-speed visualization methods and automated image-processing analysis algorithms. From the results, they are found that behavior of bubble is significantly influenced by a mass flux of working fluid and applied heat flux, whereas the pressure of operating fluid influences the bubble generation process indirectly.

Y. Wang et al. [30] Investigated the characteristics of gas-liquid two-phase slug flow in a vertical narrow rectangular channel. They are used a high-speed video camera system. The results demonstrated that the increase of gas flow rate leads to increase the average length of bubbles. The increment of liquid flow rate leads to the reduction of the average length increment of bubbles.

The effects of sub-millimeter-bubble injection on the laminar natural convection of water along a heated vertical plate with uniform heat flux were investigated experimentally by **A. Kitagawa et al. [31],** used thermo- couples and the PTV technique. In the range of volumetric flow rate 34≤ Q ≤57 mm3/s. The results showed that the enhancement of heat transfer coefficient with the sub-millimeter bubble injection is 1.35–1.85 times higher than pure fluid. Moreover the ratio of the heat transfer coefficient with sub-millimeter-bubble injection to that without injection, increases with an increase in the bubble flow rate or a decrease in a heat flux of the wall. Hence, it is expected that sub-millimeter-bubble injection is a proportional technique for enhancing the heat transfer rates for laminar natural convections.

N. M. Nouri and A. Sarreshtehdari [32] experimentally investigated the influence of air bubbles injection into a rotary device on skin friction. They have injected microbubbles with the size smaller than 300 μm. The results that obtained from the study showed that void frictions less than 4% caused a reduction of about 90% in skin friction due to a rotational velocity that creates with small values of void friction.

H. Sadighi Dizaji et al. [33] proposed air injection bubbles as a considerable technique to improve thermal performance of shell and helically coiled tube heat exchanger in which a liquid used as a working fluid(especially water)**.** Where they have studied experimentally the influence of air bubble injection on the number of thermal unit (NTU) and performance of vertical shell and coiled tube heat exchanger as well as exergy loss due to air bubbles injection. Volumetric Flow rate and temperature of hot water (coiled side) were kept constant at 1L/m and 40 ºC respectively. Inlet temperature of cold water (shell side) kept constant at 12 ºC with different mass flow rate from (0.0831 to 0.2495) kg/s. The experimental performed with various air bubbles injection conditions at constant air flow rate (1L/m) as predicted in (figure 2.1) . The results were demonstrated that the amount of number thermal unit and effectiveness augmented significantly due to air bubbles injection, maximum enhancement obtained was occurred in case " d ". Dependent on air injection state and cold water mass flow rate (shell side) an increase of up to 1.5–4.2 times in NTU, 1.23–2.59 times in dimensionless exergy loss, and 1.36–2.44 in effectiveness compared to the pure water (without bubble injection).from the results obtained, they are concluded that the ratio (n/d) is an effected parameter in these cases, higher amount of (n/d) mean more enhancement. Where n is number of holes and d is diameter of hole.

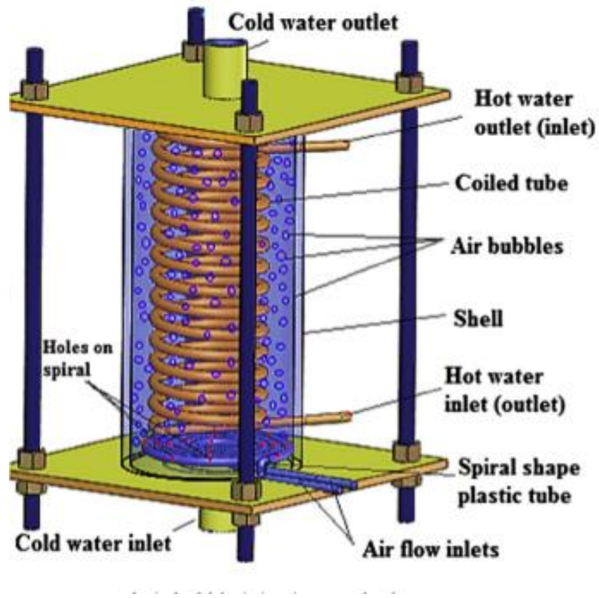

Figure 2.1. A general view of test section include air bubble injection method**[33]**.

Moosavi et al. [34] suggested to increase air flow rate to 5 L/m gradually, they are investigated the effect of air bubble injection with different air flow rates on thermal performance, frictional characteristics and pressure drop of a vertical shell and helical coiled tube heat exchanger , the results showed that the air flow rate has important roles on the effect of bubble injection into the heat exchangers and the increment of air flow rate causes augmentation of the overall heat transfer coefficient. Air injection into the shell side of the heat exchanger enhanced the overall heat transfer coefficient about 6-187% due to increasing of air flow rate. The pressure drop in the shell side due to air bubble was 3.4 kpa.

Khorasani and Dadvand [36] studied experimentally the influence of air bubbles injection on the performance of horizontal shell and helical coiled tube heat exchanger, as well as they evaluated the variations of NTU, effectiveness and exergy loss due to the air bubble injection into the shell side with different air flow rate. The experimental performed with parallel and counter configuration. Temperature and flow rate of hot water (coil side) kept constant with 43 ºC (1 L/m), whereas the volume flow rate of shell side varied from 1 L/m to 5L/m with constant temperature 23 ºC. The obtained results showed that the NTU and effectiveness ε of the heat exchanger were improved significantly due to injection of the air bubbles. An increasing in the value of NTU was 1.3-4.3 times. The maximum enhancement occurred in the counter flow structure at 1 l/m water flow rate in the shell side and air flow rate 5 l/min. The exergy loss improved around 1.8-14.2 times the value of non-injected air bubbles situation. The supreme exergy loss occurred in the counter flow configuration with the shell side water flow rate of 1 l/min and air flow rate of 5 l/min. In addition, the air bubbles injection led to increase in the effectiveness. The maximum effectiveness due to air bubbles injection was 0.815 realized in the counter flow structure with the shell side water flow rate of 5 l/min and the air flow rate of 5 l/min.

H. Sadighi Dizaji et al. [37] clarify experimentally the effect of air bubble injection on heat transfer rate, NTU, and effectiveness in a horizontal double pipe heat exchanger. Air injection method was based on creating tiny holes on a plastic tube to generate air bubbles along the heat exchanger. The mass flow rate and inlet temperate air through the plastic tube were 0.098 × 10-3 Kg/s And 26°C respectively. The flow rate and inlet temperature of cold water (outer pipe) were kept constant at 0.083 kg/s (Re=4000) and 25°C respectively. Hot water (inner pipe) inlet temperature was kept constant at 40° with different mass flow rate variant from 0.0831 Kg/s to 0.2495 kg/s (Re=5000~16000) study are related to bubbles inlet parameters. The results are summarized as follows: Nusselt number improved about 6% - 35% based on air injection condition and Reynolds number. The effectiveness of heat exchanger increased about 10%- 40%. Maximum effectiveness was achieved when air bubbles were injected into the annular space. So, it can be concluded that the ratio of (n/d) is a very important parameter. Design of holes on the plastic tube and their number and diameter can enhance the performance of heat exchangers.

A. Nandan and G. Singh [38] investigated experimentally Overall heat transfer coefficient, exergy losses and dimensionless exergy loss of shell and tube heat exchanger**,** hot water was flowing to the shell side at different flow rates (1, 1.5, 2, 2.5, 3 l/m) and at four different temperatures (30, 40, 50 and 60 C). Cold water was kept at a constant flow rate around 2 l/m and constant temperature. An aquarium pump was employed to injection air. The flow rate and inlet temperature of air injected was 0.05833 kg/s and 15 C respectively. Four different situations with and without air bubbles in a tube side were carried out. The results that obtained from the experiments showed The heat transfer rate with air injection during the tube was increase 25-40% accompanied by air injection at the enhancement of 20-30% and at shell inlet which gives an improvement of 10-20% as compared to pure water at various Reynolds Numbers. the overall heat transfer coefficient with air injection during the tube gives an improvement by 30-45% followed by air injection at the tube inlet which gives augmentation by 20-30% which shows improvement of 10-15% to pure water at various Reynolds Number. Exergy loss with air injection during the tube gives an augmentation of 35-45% air injection at inlet which gives improvement by 20-30% and improvement by 10-15% as compared to pure water at various Reynolds Number. Increase in the hot water temperature gives an augmentation in heat transfer rate, overall heat, and the dimensionless exergy loss.

Energy conservation and thermal management in internal combustion engines has an important role in the overall performance which directly affects the engine exhaust emissions and the total fuel consumption.

H. Ghasemi Zavaragh et al. [39] studied experimentally optimization of heat transfer and efficiency of the engine by employ air bubble injection inside the engine cooling system as a new strategy to improve the performance of the cooling system. The strategy of the air injection has two meaningful advantages. In the warming state, significant air injection into the coolant fluid leads to heating engine components faster by creating air layers (heat insulation layer) around the cylinder. For the cooling state, air injection is limited to a specified level for creating turbulence flow to improve heat transfer by mixing boundary layers. The results showed that the periodic air bubble injection by using the appropriate procedure can increase fuel economy and reduce engine pollutant emissions also.

S. Pourhedayat et al. [40] investigated experimentally the influence of air bubble injection on dimensionless exergy destruction, NTU, and effectiveness of a vertical double-pipe heat exchanger. Each type of heat exchanger requires a specific injection method depending on the heat exchanger structure. The Procedure which employed to create air bubbles was based on make a small holes on a ring tube with different number and diameter placed in the annular space of the heat exchanger, Water fluid was used as a working fluid , operation parameters of the inner tube (hot water) and outer tube(cold water) of the heat exchanger were (40°C with constant Re = 5500 based on water flow rate) and cold water (25°C with 5000< Re< 16000), respectively. An aquarium-air pump was employed to injection air for both sides of the ring tube. The result showed that Bubbles cause an increment of further exergy destruction as well, which is natural. However, exergy destruction by bubbles is less than the other hard type of turbulators, which act as a barrier against the flow movement. Nusslet number, dimensionless exergy destruction and effectiveness were enhanced about 57%, 30% and 45% respectively with n=36 and d= 0.3. The proficient design of perforations on a plastic tube and logical choice of their number and diameter can improve the performance of heat exchangers.

E. M. S. El-Said and M. M. A. Alsood [41] investigated experimentally two new methods were (cross air injection and parallel injection to working fluid) to clarify the effect of air bubble injection its direction on the thermal performance and pressure drop of a shell and multi-tube heat exchanger. The obtained results showed that cross injection method has a significant effect on effectiveness, NTU, overall heat transfer coefficient compared to the parallel method. Pressure drop due to cross injection higher than parallel injection. Moreover, the overall heat transfer coefficient enhanced about

131- 176 % when cross injection employed.

Baqir et al. [42] experimentally investigated the effect of micro bubble in a helically coiled tube heat exchanger's side shell. Porous sparger was used to provide tiny air bubbles as a new strategy that undressed before in the literature surevy. Their findings showed that the increase in the overall heat transfer coefficient due to the air bubble injection was around 130 per cent.

Kreem et al. [43] studied experimentally the effect of air bubble on thermal performance of shell and coiled tube heat exchanger especially temperature distribution along shell side d. the results showed there is a huge effect on thermal performance.

Sokhal et al. [44] experimentally evaluated the effect of air bubble injection on the thermal and exergy of a shell and tube heat exchanger. The findings from experimental data showed that the rise in the injected air bubble's flow rate improves the heat exchanger's efficiency while the loss of exergy increases with the flow rate. The minimum dimensionless exergy loss and Nu were around 27.49 per cent and 13 per cent compared to no bubble injection

Subesh et al. [45] studied experimentally two different air bubble injection techniques (parallel and cross-flow) into the shell side of a shell and tube heat exchanger were. A wide range of shell side and injected airflow rates measured the possible improvement of the heat exchanger's thermal performance, including Nu, the overall heat transfer coefficient and effectiveness due to the bubble injection. Their findings showed that the cross-injection air bubble technique was more effective than the parallel, but with a higher drop in pressure.

Ghashim et al. [46] studied experimentally the effect of air bubble injection on the heat transfer and pressure drop of a helically coiled tube heat exchanger working under turbulent flow conditions has been.

The injected air flow rate range was 1.5, 2.5 and 3.5 l/min, and the hot fluid Re was between 9000 and 50000 and Nu, effectiveness, NTU and exergy losses were measured.

The results showed that the air injected contributed to a 64-126 per cent rise in Nu, while the pressure drop increased from 66-85 per cent for the entire range of the measured Re.

The heat transfer enhancement performance of vertical helically coiled tube heat exchanger was investigated experimentally by **Saif S.Hassan [47] et al**. The experiments were carried out with a 50 cm height heat exchanger and 15 cm internal diameter. The operation parameters were: four different mass flow rate for cold water shell side and three different mass flow rates for hot water coiled tube side and four different airflow rates. The temperature difference betwwen hot and cold side was 20°C. 4 k-type thermocouples used. The experimental data demonstrated that the increase of air injection flow rate have a significant effect which could be responsible for the heat exchanger's thermal enhancement. Moreover, the injected air pressure having a minor effect on thermal performance.

The effect of bubble size on thermal performance in a vertically coiled tube heat exchanger was studied experimentally by **Saif S.Hassan [48] et al**. 0.1, 0.8, and 1.5 mm represent the bubbles size which tested in the present study. The experiments carried out with four shell-side flow rates 2, 4, 6, and 8 LPM, constant hot fluid flow rate 1 LPM, and constant $\Delta T = 20$ °C. The 4 k-type thermocouples used to measured temperature distribution along the height of the heat exchanger. Results showed that air bubbles' injection generally led to a significant augmentation of the heat transfer rate and the optimum bubble size was 0.8 mm. Moreover the results demonstrated that the air bubbles injection and bubbles size have a major role in temperature distribution along height column of the heat exchanger.

Table 2.1: Summary of Important Investigations of literature review

Authors	Year	Fluid	Type of investigation	method	Observation	results
Dizaji [23]	2015	water	experimentally	Injection air bubbles via plastic sparger inside vertical shell and helical coil of heat exchanger.	-number of holes (28-224). -diameter of bubbles (0.3mm). -both the parallel and counter flow were investigation.	- NTU/NTUa (1.5-4.2). - ε/εa (1.36-2.44).
Dizaji [47]	2016	water	experimentally	Injection air bubbles inside horizontal shell and helical coiled tube heat exchanger by two positions inside (tube and shell).	-diameter of holes (0.3 and 0.7mm). -number of holes (40-80). -counter flow. -maximum ε occur when air injection inside annular space.	-NU (6%-35%). -ε (10%-40%).
Nandan [49]	2016	water	experimentally	Injection air bubbles at (tube entrance, shell entrance, and throughout the tube section.	-heat transfer coefficient (h) increased with increased Re (6600-23000), where h is a function of Re.	-NU (15%-20%) when air injection at shell entrance. - NU (20%-35%) when air injection at tube entrance. NU (30%-45%) when air injection throughout the tube section.
Moosavi [24]	2016	water	Experimentally	Injection air bubbles into: -helical coil. -vertical shell.	-number of holes (224). -diameter of bubbles (0.3mm).	-maximum enhancement occur in case air injection to shell. -ε (3%-56%), and U (6%-187%). -ΔP coil (49%-370%), and ΔP shell (13%-225%).
khorasan [48]	2017	water	experimentally	Injection air bubbles via new method (plastic tube beside outer wall of shell).	-number of holes unknown. -diameter of holes (0.3mm). -counter and parallel flow.	-NTU (1.3-4.3), counter flow, Qs=1 Lpm, and Qa=5 Lpm. -ε max (0.815), counter flow, Qs=5Lpm, Qa=5Lpm.
Alsaid [50]	2018	water	experimentally	Injection air bubbles in parallel and cross flow into shell side and multi tube heat exchanger with baffles.	-for cross injection use 18 nozzle with diameter 0.7mm. -for parallel injection use bending tube with diameter holes (0.7mm).	-for cross flow ΔPmax (54.2mmbar), and for parallel flow ΔPmax (45.2 mmbar), -for crossflow U (131%-176%), ε (136%-173%).

Authors	Year	Fluid	Type of investigation	method	Observation	results
Ahmed [51]	2019	water	experimentally	Injection air bubbles via porous sparger inside vertical shell and helical coil tube of heat exchanger in parallel with cold water flow.	-unknown number of holes. -assume 100µ diameter of bubbles. - counter flow were investigation.	- Max NTU (1.93), and ε (0.83). - Min NTU (0.66), and ε (0.63).
Kareem [47]	2019	water	experimentally	Injection air bubbles via porous sparger inside vertical shell and helical coil tube of heat exchanger in parallel with cold water flow.	Study the influence of injection air bubbles on temperature distribution along the vertical shell and helical coil tube heat exchanger.	-high heat transfer was recorded that indicated through sudden rising of the temperature along the heat exchanger. -the study state can be achieved faster as the (Vd) increased.
Panahi [56]	2017	water	experimentally	Injection air bubbles inside vertical shell and helical coiled tube heat, and experiments were performed for both parallel- and counter-flow configurations.	-number of holes (28,56,112, and 224). -diameter of bubbles (0.3mm). -Re (1000-4000).	-NU (50%-328%), and ε (53%-127%) in counter flow.
zavaragh [57]	2017	water	Experimentally	Injection air bubbles inside water jacket of engine.	Air injection inside water jacket has two significant advantages: a- in warm up stage, air bubbles cause rapid heating of engine components. b- In cooling stage will generate vortices that will destroy boundary layer to decrease heat resistance, whereas increase heat transfer rate.	The fuel consumption will decrease by approximately 5.69%, while engine pollutant emission lowering to approximately 16.93.
shukla [59]	2019	water	numerically	Using ANYSYS FLUENT 14.5 CFD software.	Five different air bubble diameters (0.05,0.1,0.3,0.5, and 0.7mm) were tested.	-The air bubbles with diameter 0.7mm have the greatest heat transfer rate. -Maximum NTU 1.39, U 1327, and ε 0.74.

Authors	Year	Fluid	Type of investigation	method	Observation	results
Li Ya [35]	2012	water	numerically	using helical coil tube cooperating with spiral corrugation and studied effect on heat transfer rate.	-using different spiral pitch.	-finding showed that, decreasing spiral pitch will increase heat transfer, and increasing pressure drop. -heat transfer rate (50%-80%), pressure drop (50%-300%).
sivala [36]	2021	water	experimentally	Using helical fin over the inner pipe of double pipe heat exchanger.	-compare the thermal performance of smooth tube heat exchanger with finned tube heat exchanger.	-the heat transfer rate enhancement (38.46%), and effectiveness (35%) at higher flow rate.
Solano [41]	2012	water	experimentally	Using corrugated tube, dimpled tube, and wire coils to study the effect of artificial roughness on thermal performance of heat exchanger.	the comparison was carried out using the three best samples from the various geometries in previous work [41].	Results show that the shape of the artificial roughness exerts a greater influence on the pressure drop characteristics than on the heat transfer augmentation.
Wongchar [43]	2012	water	experimentally	Using Cuo/water nanofluid in corrugated tube fitted with twisted tape.	The nanofluid and twisted tape results are compared with those obtained from nanofluid alone and twisted tape alone.	The finding showed that the thermal performance associated with the combination of Cuo/water nanofluid and twisted tape were higher than those associated with the individual techniques under identical operating conditions.
Muna	2021	water	experimentally	Injection air bubbles via porous sparger inside vertical shell and finned coil tube of heat exchanger in parallel with cold water	-unknown number of holes -assume 100μ diameter of bubbles -counter flow were investigation	

2.3 The summary

After figuring out a number of the previous researches and techniques which were used to improve the shell and coiled tube heat exchanger with air bubble injection technique, many critical points can be outlined. The thermal performance of shell and coiled tube heat exchanger can be affect hardly by inserting special techniques to enhancement heat transfer and thus increase the efficiency of a heat exchanger such as; air bubble injection technique, helical coil, etc. The addition of special materials to the working fluid, such as nanoparticles at different proportions and different types have been used to improve the thermal parameters of shell and tube heat exchanger. Some of the suggestions designed to the sparger by increasing the contacting area between heated coiled tube and cooled fluid, thereby increasing the temperature of the liquid passing through it. Some researchers have proposed a new design by adding a sparger to increase the acquire heat as a design shell and tube. The other parameters of the shell and coiled tube heat exchanger were studied such as increasing the flow rate of air bubbles, temperature difference used, and the type of internally / externally finned coiled tube. The mixture of water and air of two-phase flow systems works to increase the heat transfer coefficient in a specific condition, such as the proportion of water and air, air bubble size, tube diameter used, angle of the inclined tube.

In this study ,the thermal performance of shell and coiled externally finned tube heat exchanger under air bubbles injection conditions and mass flow rate variation will be carried out and fill the gap which appears in the study by performing experiments over the a wide range of operational parameter for fluid and air flow rates inside shell and coiled finned tube.

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Chapter Three

Experimental Work

EXPERIMENTAL WORK

Introduction

This section describes the experience apparatus, method, and experimental conditions used in this study. In this study ,The previous studies gives a clear indication of the possibility to improving the thermal showing of a shell and helically coiled finned tube exchanger of heat by using air bubbles injection technique

The current research has been working and establishing an experimental device to verify the impact bubble of air injection and the effect of the distance between one fin and another of the finned tube on the thermal showing of the perpendicular shell and helical coiled (smooth and finned) tube exchangers of heat .

3.1 Test system part.

Figure 3.1 and Figure 3.2 show the exact views and structure of the test bench. The test bench include of three main elements. The first section of the heating unit was designed to provide hot water in the side of the lined pipe, which included a hot water tank and an electric heater. The second cooling unit is designed to supply cold water to the side of the register. The cooling unit consists of a cooling water tank, rotary compressor, evaporator, condenser, additional valve and thermostat. The third stage heat exchanger (Figure 3.3) consists of a PVC jacket, a spiral wound copper pipe and a porous sparger, which is mounted under the jacket on the side facing the generator. air bubbles. An air compressor (HAILEA ACO-318, 4) was employed to ventilate the air. As in Figs. 3.2 and Fig. 3.3, the system flow rate was the dissipation flow rate, with hot water being boiled into the tube from the upper side and shell side flow rate entering the tube from the lower side. The flow bubbles of air and shell side flow rate is the flow that together flows from bottom to top. 8mm thick ultra-high heat dissipation (0.002W / mK) polyolefin foam wrapping paper was employed as thermal insulation about the hot water reservoir , cooling water reservoir and water heater.part of the test to reduce heat loss to the area as much as possible. The PICOLOG 6 TC-08 data logger contains 8 type K thermocouples to display the temperature of rooms, stores and temperature distribution along the skin immediately on the PC. Three rotameters (panel type) were used to amount air and water in flow rates, both are used to measure hot and cold water with an uncertainty of ± 0.2 LPM and a value equal to the flow rate of air and an uncertainty of ± 0 To taste 2 LPM.

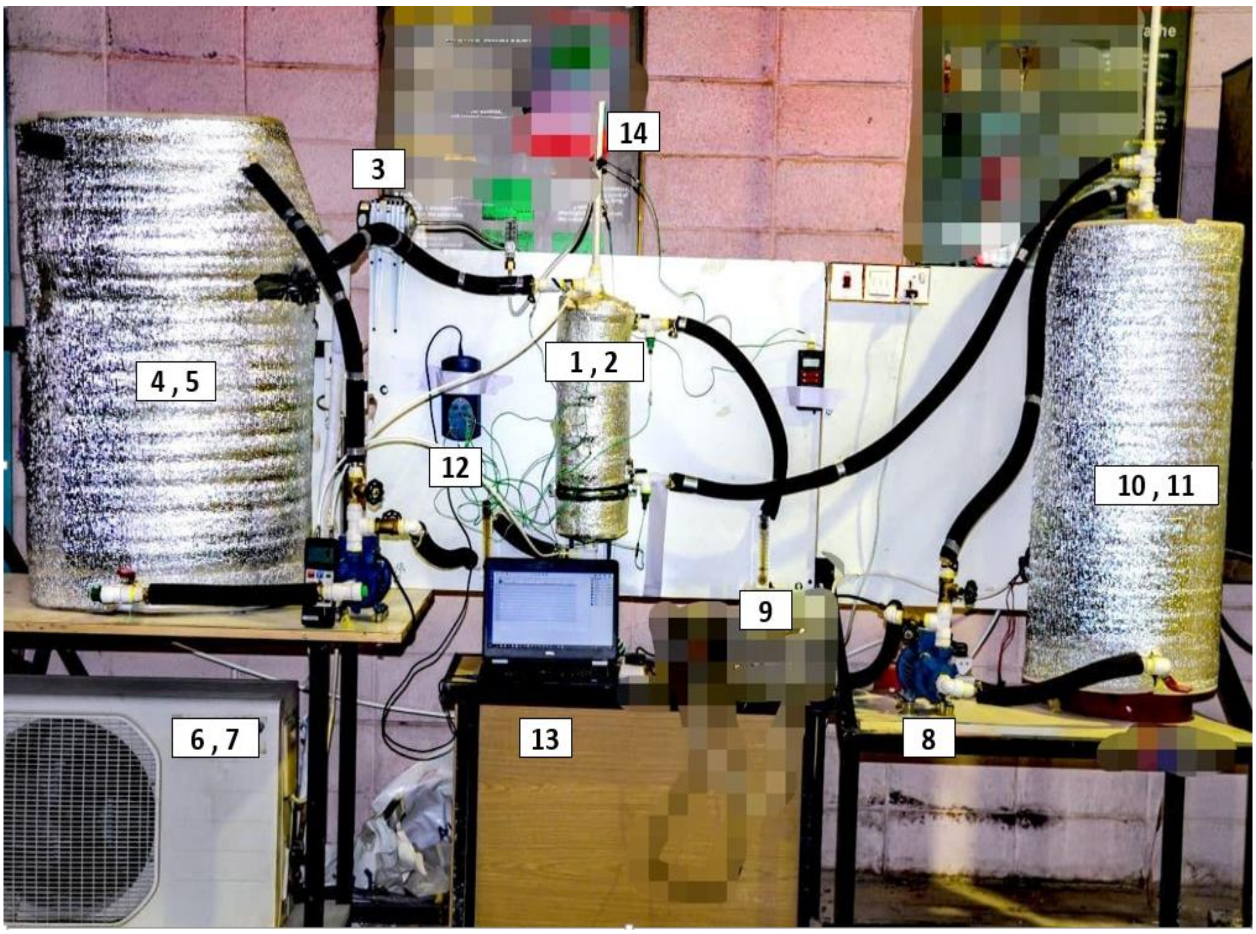


Fig. 3.1: Overview of the experimental configuration: 1.Shell tube, 2. Helical coiled tube, 3. Aquarium air compressor, 4.cold water tank, 5.evaporator, 6. Compressor, 7.condensor, 8. Water pump, 9. rotameter, 10.hot water tank, 11. Electric heater, 12. Data logger 13. PC, 14. Air vent

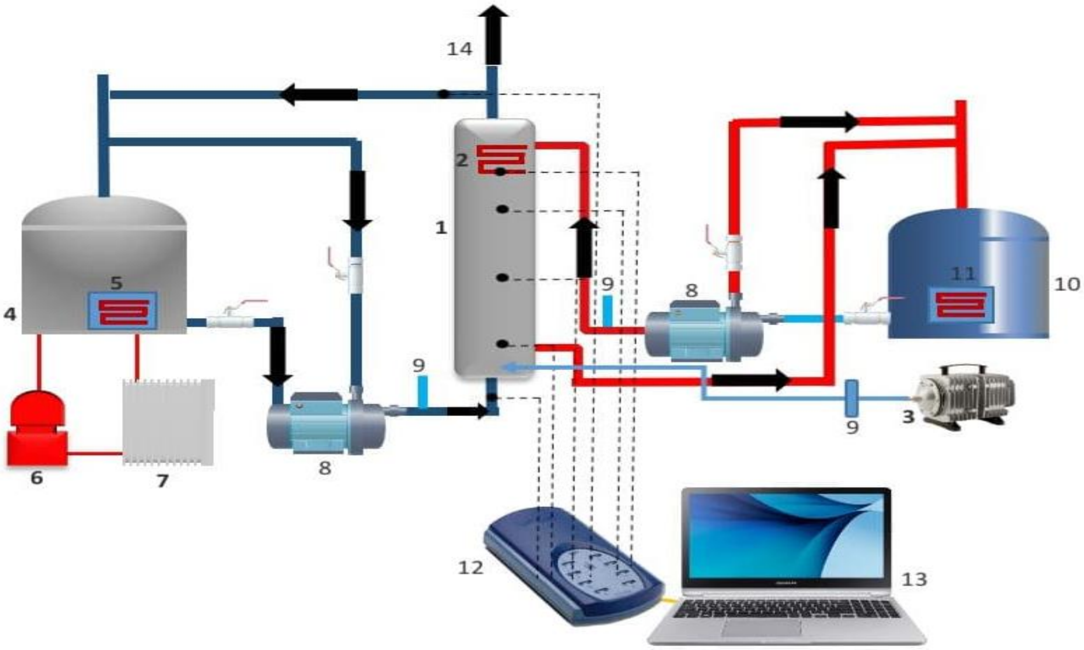


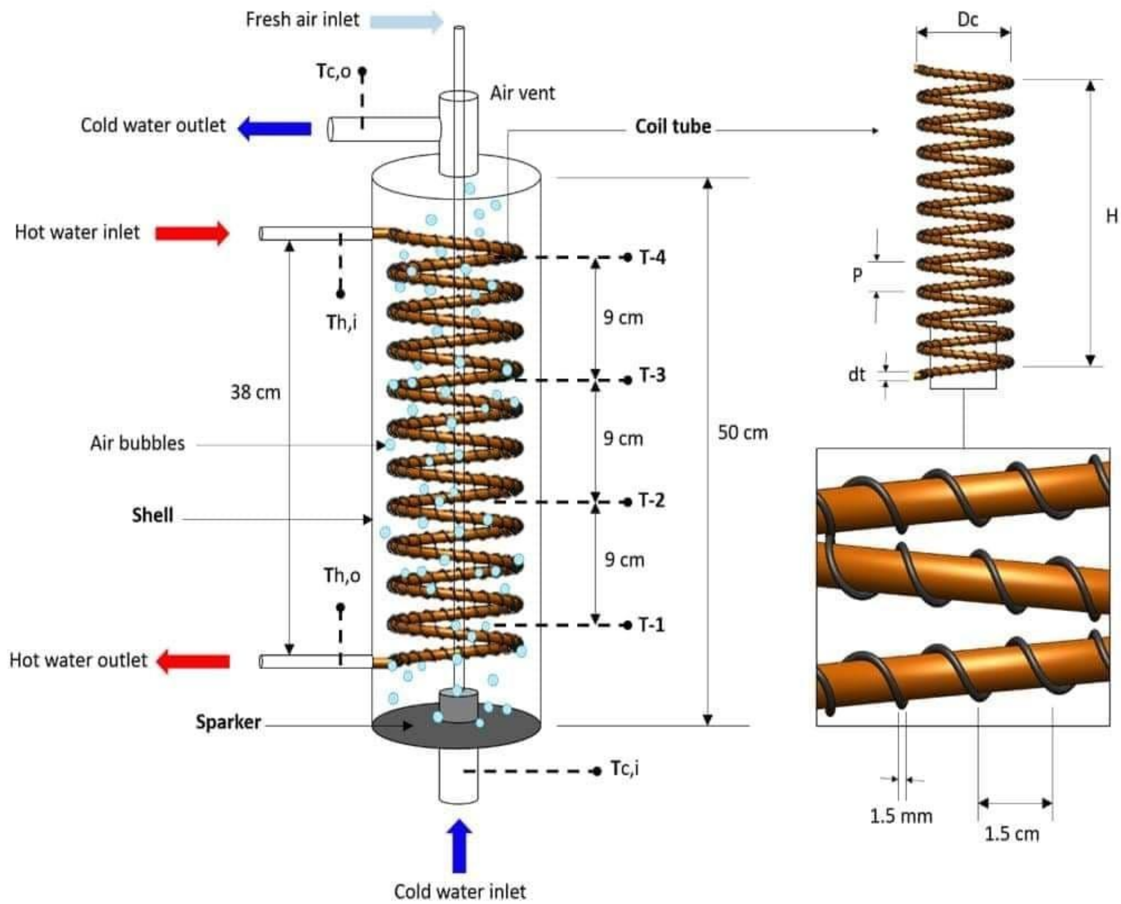
Fig. 3.2: A planning view of the empirical configuration: 1.Shell tube, 2. Helical coiled tube, 3. Aquarium air compressor, 4.cold water tank, 5.evaporator, 6. Compressor, 7.condensor, 8. Water pump, 9. rotameter, 10.hot water tank, 11. Electric heater, 12. Data logger 13. PC, 14. Air vent

3.3 Test Section.

As seen in the schematic view (Figure. 3.3), the test section comprised three main parts are PVC shell tube, Helical coiled tube (a copper straight tube before coile operation with 3.939 m was manually wrapped to form helical coiled tube, see Figure 3.4) and Stone porous sparger.

The geometrical designs of the pipe shell and the pipe shell are presented in Table 3.1. Eight thermocouples are employed to measure the temperature of the heat exchanger. Four of them were employed to measure the flow and flow of hot water ($T_{h.i}$ and $T_{h.o}$), and cold water ($T_{c.i}$ $and T_{c.o}$), and the other 4 thermocouples were fixed over the shell side of the test section to measure the distribution of temperature along the shell tube

(T_1 , T_2 , T_3 *and* T_4) as showed below.



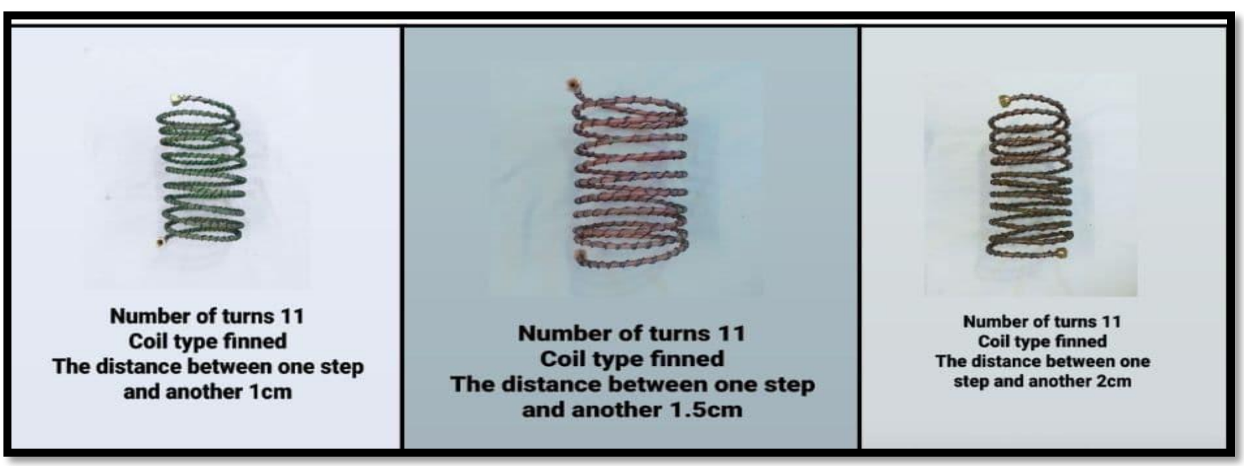


Fig. 3.3: Diagram of the experimental part.

Table-3.1, A general view of the characteristics of the exchanger of heat (mm) **[24].**

Tube	**Di**	**H**	**Dc**	**N**	**t**	**L**	**di**	**P**
Shell	152.4	500	-	-	3	-	-	-
Coiled tube	-	380	114	11	1.6	3939 (bco)	4.4	30

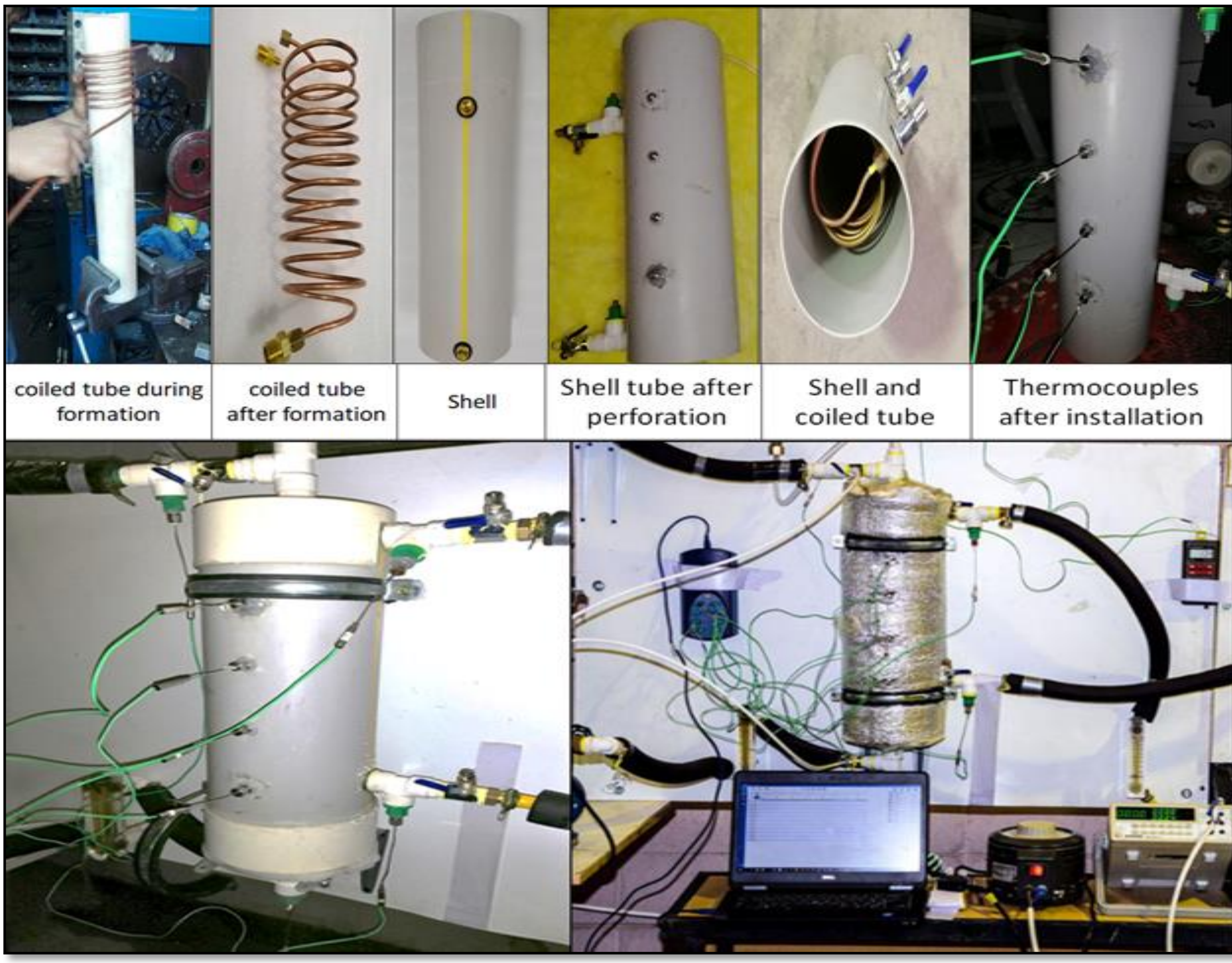


Fig. 3.4: Stages of test section assembly

3.4 Sparger

use Stone pored sparger (Mott Corp.®, Type: 316 L SS ,Farmington, USA , CT) to work small air bubbles in inside a shell tube . (100 μm in avearge). It was placed at the bottom of the shell tube as illustrated in Fig. 3.3. The geometric pattern, over view and planning view of this section are Shown below .

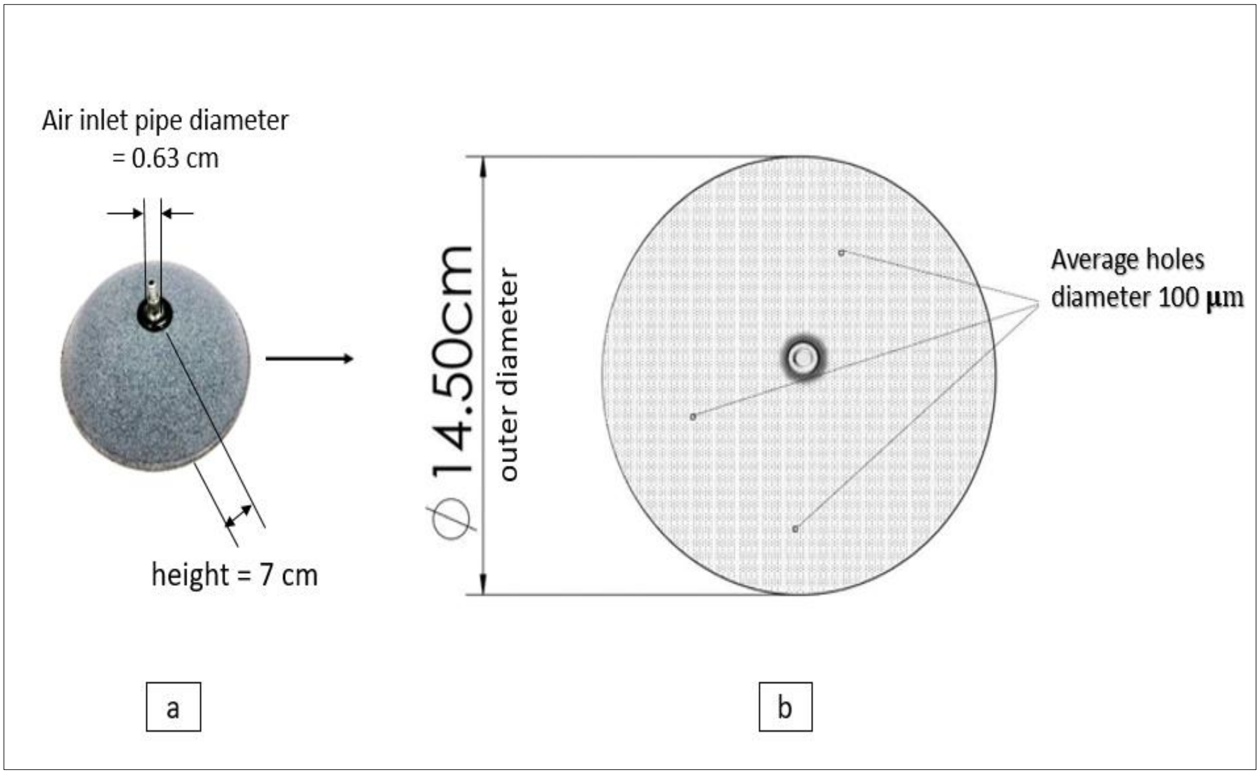


Fig. 3.5: Sparger specification: (a) over view, (b) planning view

3.5 Air supply unit

The ACO-318 electromagnetic air compressor from China (see Fig. 3.6) was used to air-condition the energy-saving device. The compressor status is made-up of top quality ZL 102 alloy of aluminum with customized planning , high heat efficiency, electrically operated magnetic motor, straight-line reciprocating air movement and reasonable shape. Use a new, portable material, label SF3, for the piston and cylinder, with a few energy consumption and an oil-free construction **.** The technical data of the air compressor are listed in **Table-3.2.**

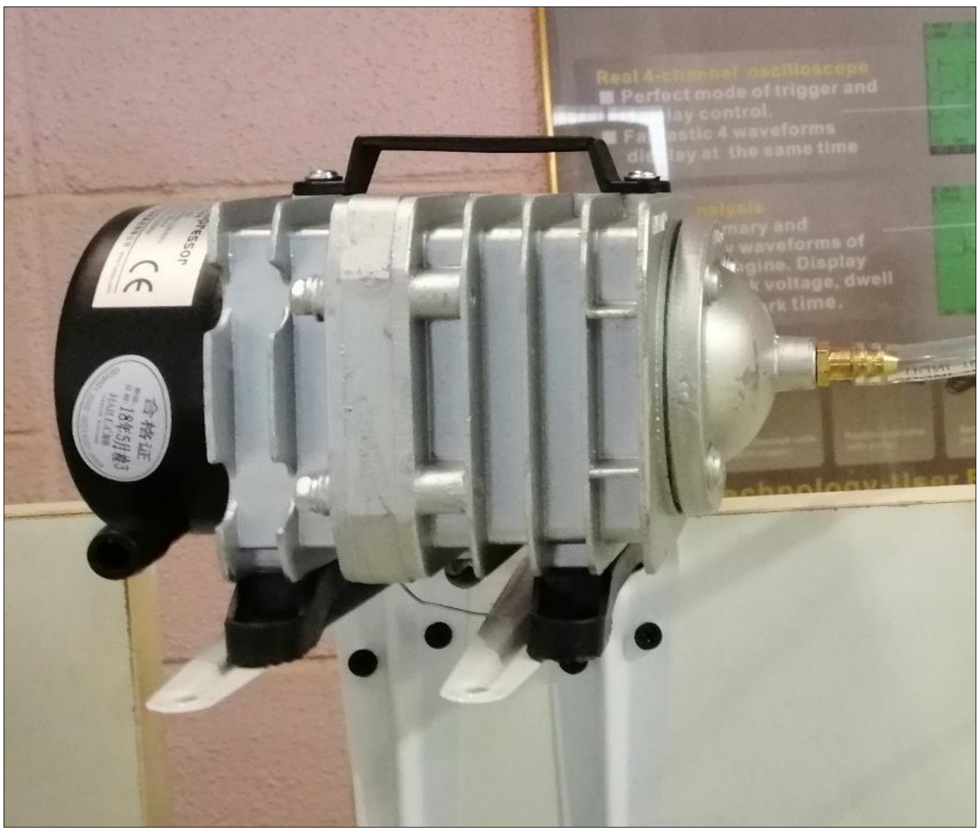


Fig. 3. 6: pneumatic piston

Table-3.2: pneumatic piston properties

part	Comment
Brand name	HAILEA
Module number	ACO-318
Power	45 Watt
Voltage	220-240 Voltage
Frequency	50 Hertz
Pressure	25 kpa
Air flow	70 LPM
Noise	60 dB

3.6 Heating unit

Heating unit used to provide hot water for the coiled tube. Heating unit comprises of water storage tank (120 L), electrical heater,

3.7 Cooling unit

A cooling unit used to provide cold water to shell side. As shown in Fig. 3.1 & 3.2 cooling unit involves water storage resaves (250 L), revolving compressor, extension valve, condenser, evaporator, and thermostat to keep inlet water of the shell side at a constant temperature.

3.8 **Water pumps and rotameters**

Two electric water pumps (KF/0, China) were used, one for the cold water and another for the hot water. Both pumps have the same technical data as tabulated in Table-3. 3. Both pump has a bypass line to control the water inflow rates easily as shown below

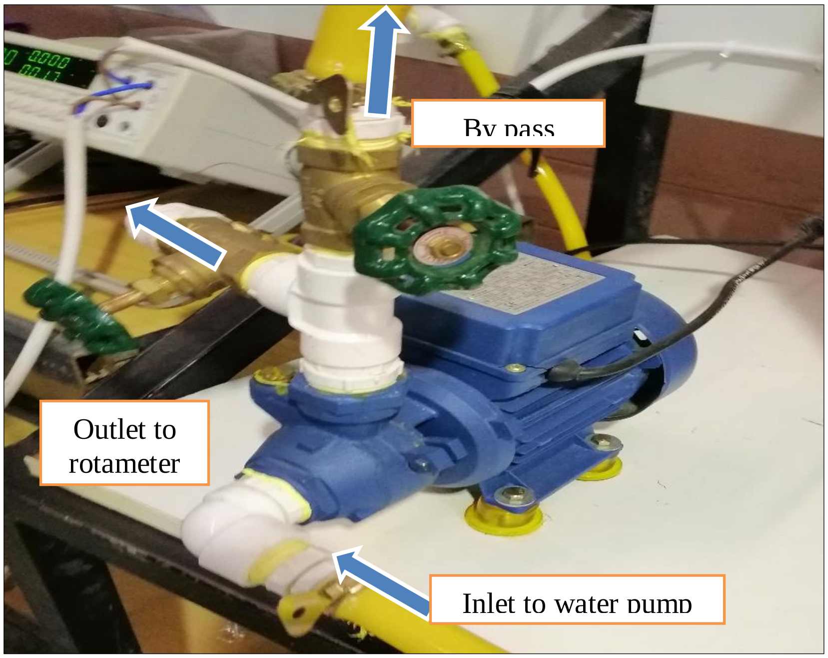


Fig. 3.7: water pump

Table-3.3: water pump technical dada

Parameter	**Comment**
Type	**KF/0**
Max. fluid temperature	**80 ^{0}C**
Power	**0.372 KW**
Voltage	**220 V**
Frequency	**50 Hz**
Pressure	**3 bar**
Max. head	**30 m**
Flow rate	**30 LPM**

The volumetric inflow rates of both kind of water (cold and hot) were measured by rotameters (panel type). Rota-meters with 18 LPM and 4 LPM maximum flow rates were used for the cold and hot water respectively.

3.9 Data Logger

An eight channel data logger, Pico Technology Temperature Data Logger TC08 USB (Fig. 3.8) with the following specifications was used:

1- Eight channel thermocouple data logger
2- Supports all popular thermocouple types
3- Measures from -270 to +1820 °C.
4- High resolution and accuracy (For popular Type K thermocouples the TC-08 can maintain 0.025°C resolution over a –250 to +1370 °C range).
5- Fast sampling rate up to 10 measurements per second.
6- USB interface.
7- PicoLog for Windows data logging software included.

The data logger is connected directly to a computer by a USB connection, where the thermocouple readings displayed directly on the computer.

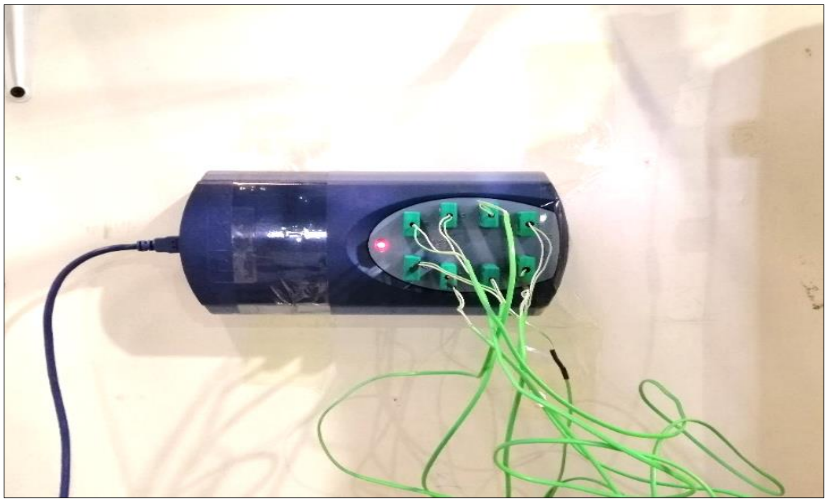


Fig. 3.8: Picolog-6 data logger with 8-thermocouples

3.10 Temperature measurement

The inlet and outlet temperatures of the hot water, cold water, air and temperature distribution along the shell tube were measured by using K-Type (Nickel Chromium, Fig. 3.10) calibrated thermocouples with extension wires (RS Component Ltd, Nothants, UK). All these thermocouples are connected with a data logger, which directly displays their measurements on a PC, excepted of the inlet and outlet air temperature were connected to a digital thermometer as illustrated in Fig. 3.1 and fig 3.2 . The accuracy of the thermocouples are given in Table 3-5, while the calibration curves are given in Appendix A. K-type thermocouple is common used in most thermal applications due to its high corrosion resistance. It has a wide operational temperature range. The specifications of the mentioned thermocouples are:

Temperature Range

•Thermocouple grade wire, −270 to 1260°C

•Extension grade wire, 0 to 200°C

•Melting Point, 1400°C

Accuracy (whichever is greater):

•Special Limits of Error: ± 1.1℃ or 0.4%.

Table. 3-5: Error of temperature measurement in thermocouples

Parameters	Unit	Error
The temperature of the hot water outlet	° C	±0.4
The temperature of the hot water inlet	° C	±0.4
The temperature of the cold water outlet	° C	±0.4
The temperature of the cold-water inlet	° C	±0.5
Cold water along column side T1	° C	±0.4
Cold water along column side T2	° C	±0.4
Cold water along column side T3	° C	±0.4
Cold water along column side T4	° C	±0.5

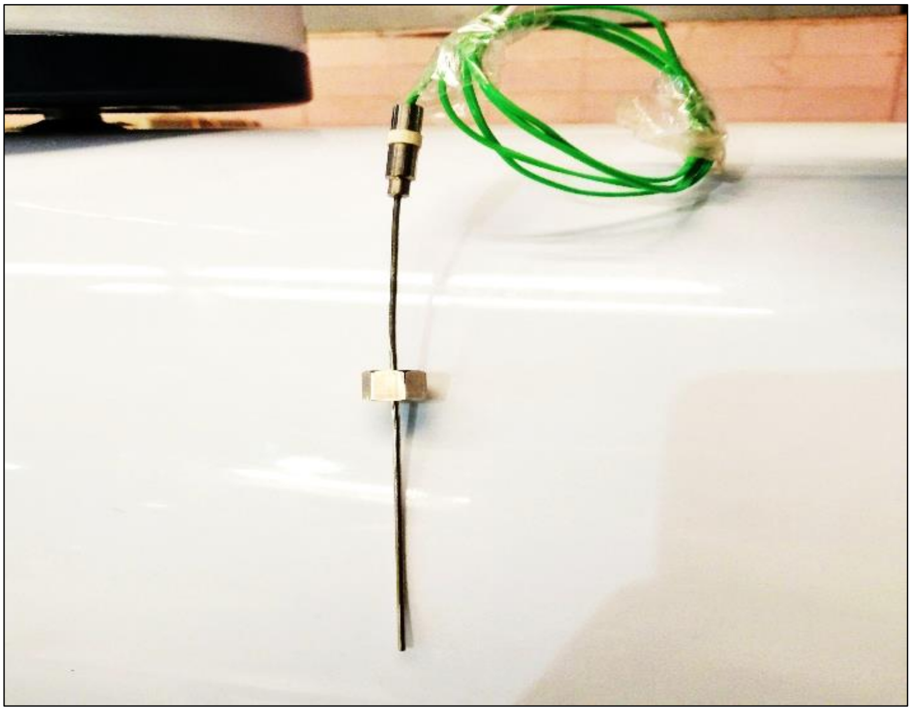


Fig. 3.9: k-type, Nickel-Chromium thermocouple

3.11 Working fluid.

Pure water has several properties that make it useful as a coolant. It has a high specific heat capacity makes it to be an effective coolant. Also, it is fairly common, which means it is readily available. Furthermore, the cost of water is fairly low. Water also has a low viscosity, which represents the liquid's resistance to flow and fits comfortably in at a 7 on the pH scale, meaning it is neither alkaline nor acidic.

3.12 Experimental Procedure

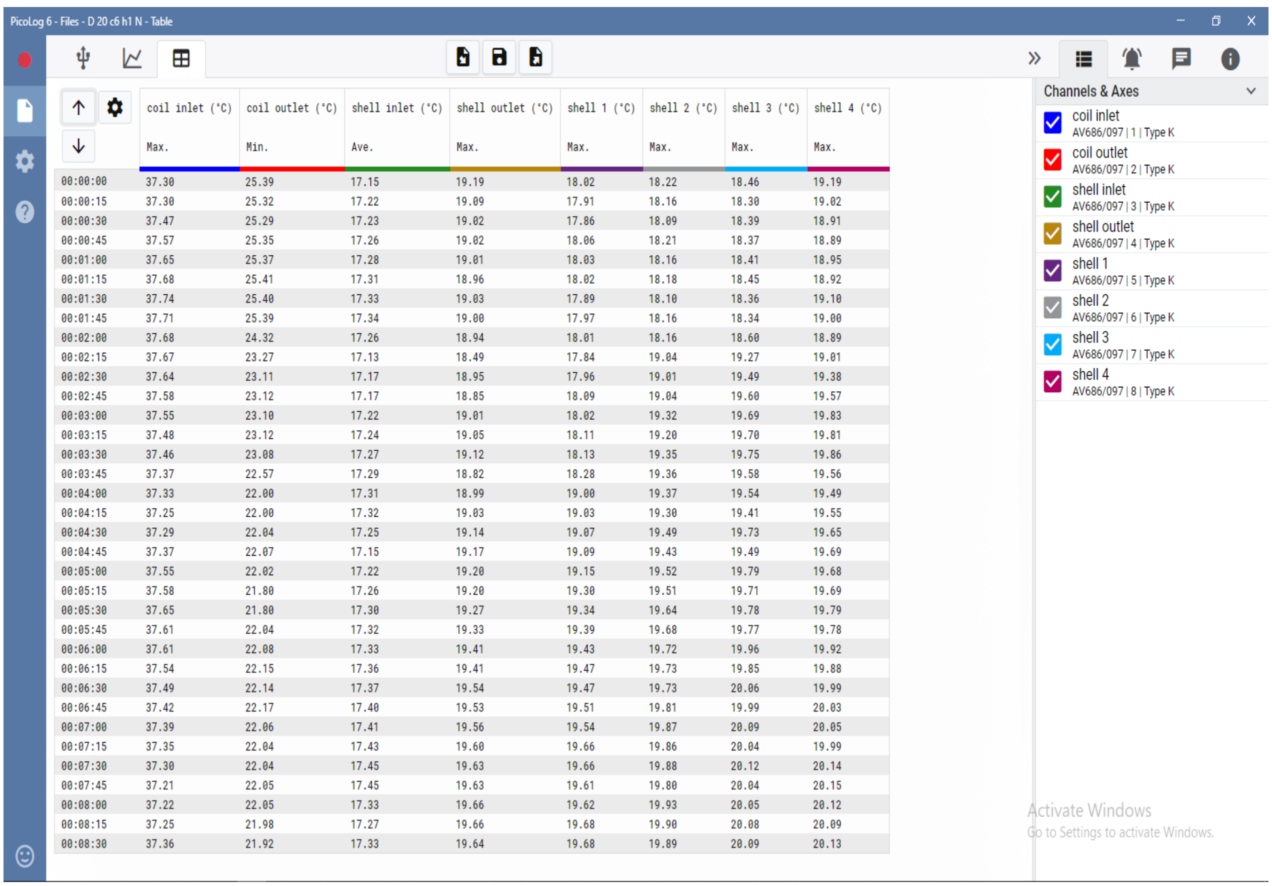
In this work 180 experiments were carried out under many different operating conditions. The samples were divided into three groups (each group consisted of 60 samples) based on the size of the fin for finnted tube. First, 60 samples were reduced to field = 1 cm and the combined temperature of 37 ° C and the shell temperature of 17 ° C (ΔT = 20 °C) were carried out. The flow rates for hot water, cold water and air vary according to Figure-3.6. Two further test units were carried out with the same criteria, which are summarized in Table-3.6, but with different sizes of pitch (1.5 and 2) cm. . To complete this section, the first 60 test methods are discussed as follows:

1. First of all preparing the experimental set-up and checking all the instruments and connections if have leakages, additional to check the electrical power supply.
2. Circulate the hot water (coil side) and the cold water (shell side) by running the pumps and choosing a specific flow rate. This stages objective to maintain a consistent temperature in the experimental system.
3. Prepare hot water and cold water in storage tanks by running the heat and cooling systems and selecting the temperature difference (ΔT=20 ^{o}C). Then specified the hot and cold water flow rates. Firstly the test section is operated without air bubbles injection, and heat is exchanged between hot and cold fluids until reaching a steady-state that is achieved after (20- 25 min) according to the type of coil that used.
4. After reaching steady-state, measurement of flow meters by rotometer of hot and cold water, and temperatures through thermocouples by reading them directly on the PC through the data logger. These readings are the first heat exchanger measurement experiment to run without air injection. This process is repeated for each side of the shell, and the side flow rate of the coil.
5. At these conditions (step 3) after steady state condition, take the measurements of the flow meters and thermocouples by read them directly on the PC via the data

loger. These reading represents the measurement of the first experiment where the heat exchanger working without air injection. This step is repeated for each shell side and coil side flow rates.

6. Select the air flow rate value and start injection of air into the shell side. Once the air being injected, the measurements of the temperatures along the test section are recorded automatically by data loger

7. This procedure (step 1 to 5) is repeated for each individual run. Figure 3.12 represent the screen capture of the measured temperatures along the heat exchanger



	coil inlet (°C) Max.	coil outlet (°C) Min.	shell inlet (°C) Ave.	shell outlet (°C) Max.	shell 1 (°C) Max.	shell 2 (°C) Max.	shell 3 (°C) Max.	shell 4 (°C) Max.
00:00:00	37.30	25.39	17.15	19.19	18.02	18.22	18.46	19.19
00:00:15	37.30	25.32	17.22	19.09	17.91	18.16	18.30	19.02
00:00:30	37.47	25.29	17.23	19.02	17.86	18.09	18.39	18.91
00:00:45	37.57	25.35	17.26	19.02	18.06	18.21	18.37	18.89
00:01:00	37.65	25.37	17.28	19.01	18.03	18.16	18.41	18.95
00:01:15	37.68	25.41	17.31	18.96	18.02	18.18	18.45	18.92
00:01:30	37.74	25.40	17.33	19.03	17.89	18.10	18.36	19.10
00:01:45	37.71	25.39	17.34	19.00	17.97	18.16	18.34	19.00
00:02:00	37.68	24.32	17.26	18.94	18.01	18.16	18.60	18.89
00:02:15	37.67	23.27	17.13	18.49	17.84	19.04	19.27	19.01
00:02:30	37.64	23.11	17.17	18.95	17.96	19.01	19.49	19.38
00:02:45	37.58	23.12	17.17	18.85	18.09	19.04	19.60	19.57
00:03:00	37.55	23.10	17.22	19.01	18.02	19.32	19.69	19.83
00:03:15	37.48	23.12	17.24	19.05	18.11	19.20	19.70	19.81
00:03:30	37.46	23.08	17.27	19.12	18.13	19.35	19.75	19.86
00:03:45	37.37	22.57	17.29	18.82	18.28	19.36	19.58	19.56
00:04:00	37.33	22.00	17.31	18.99	19.00	19.37	19.54	19.49
00:04:15	37.25	22.00	17.32	19.03	19.03	19.30	19.41	19.55
00:04:30	37.29	22.04	17.25	19.14	19.07	19.49	19.73	19.65
00:04:45	37.37	22.07	17.15	19.17	19.09	19.43	19.49	19.69
00:05:00	37.55	22.02	17.22	19.20	19.15	19.52	19.79	19.68
00:05:15	37.58	21.80	17.26	19.20	19.30	19.51	19.71	19.69
00:05:30	37.65	21.80	17.30	19.27	19.34	19.64	19.78	19.79
00:05:45	37.61	22.04	17.32	19.33	19.39	19.68	19.77	19.78
00:06:00	37.61	22.08	17.33	19.41	19.43	19.72	19.96	19.92
00:06:15	37.54	22.15	17.36	19.41	19.47	19.73	19.85	19.88
00:06:30	37.49	22.14	17.37	19.54	19.47	19.73	20.06	19.99
00:06:45	37.42	22.17	17.40	19.53	19.51	19.81	19.99	20.03
00:07:00	37.39	22.06	17.41	19.56	19.54	19.87	20.09	20.05
00:07:15	37.35	22.04	17.43	19.60	19.66	19.86	20.04	19.99
00:07:30	37.30	22.04	17.45	19.63	19.66	19.88	20.12	20.14
00:07:45	37.21	22.05	17.45	19.63	19.61	19.80	20.04	20.15
00:08:00	37.22	22.05	17.33	19.66	19.62	19.93	20.05	20.12
00:08:15	37.25	21.98	17.27	19.66	19.68	19.90	20.08	20.09
00:08:30	37.36	21.92	17.33	19.64	19.68	19.89	20.09	20.13

Fig.3.12: Screen capture for data logger page during recording experimental data

Table-3.6, Test conditions for the first group of the experiments in pitch of finned tube = 1cm

water flow rate of coil side(LPM)	water flow rate of shell side (LPM)	Air flow rate (LPM)
1	2	0, 2, 4 ,6
	4	0, 2, 4 ,6
	6	0, 2, 4 ,6
	8	0, 2, 4 ,6
	10	0, 2, 4 ,6
1.5	2	0, 2, 4 ,6
	4	0, 2, 4 ,6
	6	0, 2, 4 ,6
	8	0, 2, 4 ,6
	10	0, 2, 4 ,6
2	2	0, 2, 4 ,6
	4	0, 2, 4 ,6
	6	0, 2, 4 ,6
	8	0, 2, 4 ,6
	10	0, 2, 4 ,6

Same the experiments that tabulated above were performed for two others sets when the size of pitch for finned tube=1.5cm and 2cm

Table-3.7: Average uncertainties of the performance parameters

Parameters	**Uncertainty**
Overall heat transfer coefficient	±14.21%
Effectiveness	±8.52 %
NTU	±12.26 %

3.14 Calculation Equations of Experimental Work.

3.14.1 overall heat transfer coefficient

The overall heat transfer coefficient can be calculated using the equation shown below **[24]**.

$$U = \frac{q}{A_s \cdot T_{\mathrm{LMTD}}} \tag{3.1}$$

Where q, $A_s$and ΔT_{LMTD} represent the heat transfer rate, surface area of heat transfer, and the log-mean temperature difference, respectively.

As a consequence, using energy balance, the heat transfer rate (q) can be calculated as follows:

$$q = \dot{m}_h C_{p,h}(T_{h,i} - T_{h,o}) = \dot{m}_c C_{p,c}(T_{c,o} - T_{c,i}) \tag{3.2}$$

The log-mean temperature difference for counter flow can be calculated from the equation below **[24]**:

$$\Delta T_{\mathrm{LMTD}} = \frac{(T_{h,i} - T_{c,o}) - (T_{h,o} - T_{c,i})}{\ln\left(\frac{T_{h,i} - T_{c,o}}{T_{h,o} - T_{c,i}}\right)} \tag{3.3}$$

$$Re_{\mathrm{shell}} = \frac{Vd}{v} \tag{3.4}$$

$$V = \frac{Q}{\frac{\pi}{4} D_h^2} \tag{3.5}$$

$$D_h = D = \frac{4\left(\frac{\pi D^2}{4}\right)}{\pi D} \tag{3.6}$$

Where Q, Re, v, D_h and ט represent the water flow rate through the shell side of the heat exchanger, Reynold number of shells, shell velocity, shell hydraulic diameter, and kinematic viscosity, respectively.

3.14.2 effectiveness (ε)

The effectiveness of a heat exchanger explains the ratio of the actual to maximum possible heat transfer **[24]**:

$$\varepsilon = \frac{actual\ heat\ transfer\ in\ a\ heat\ exchanger}{maximum\ possible\ heat\ transfer\ in\ a\ heat\ exchanger}$$

$$= \frac{q}{C_{min}(T_{h,i} - T_{c,i})} \qquad (\mathbf{3.7})$$

Where q , C_{min}, T_{hi}, and T_{ho} represent the heat transfer, thermal capacity, temperature of coil inlet,and temperature of coil outlet.

3.14.3 Number of Heat Transfer Units (NTU).

The number of heat exchangers (NTU), which is a non-invasive method, He was be set at the same time to estimate the temperature of the exchanger heat and the magnitude of the exchanger of heat and to calculate from the following equation**[23]**:

$$\mathrm{NTU} = \frac{\mathrm{U.A_S}}{\mathrm{C_{min}}} \qquad (3.8).$$

Where $Cmin$ represents the value of the smallest heat transfer capacity for heat transfer.

$$C_{min} = m^{\cdot}.c_p \qquad (3.9)$$

- percentage of enhancement ratio was calculated as follows [24]:

$$\text{Percentage of enhancement ratio} = \frac{\text{new value} - \text{orginal value}}{\text{orginal value}} * 100\% \qquad (3.10)$$

Chapter Four

Experimental Results & Discussion

INTRODUCTION

This section shows the results of the surface heat transfer properties such as overall heat transfer (U), effectiveness (ε), number of heat transfer units(NTU) for heat pipes (smooth and finned) in both cases with and without air injection.

As previously shown in refs [24, 25, 28], the use of air injection significantly improves the thermal performance of the heat exchanger. . Therefore, it is expected that the size of heat exchanger that requires the same operation can be reduced even when used without air injection.

4.1 Overall Heat Transfer Coefficient.

In this work, the superheating of the temperature rise (U) as an important indicator of heat work under various operating conditions was extensively studied and adapted

As shown in figure (4.1) the difference of the overall heat transfer coefficient with the shell side in flow rate at three various pitch (p=1,1.5,and 2) cm, and three different air flow rates (Q_air=2,4,and 6 L/min) in addition to case of without air injection(Q_air=0 L/min). The coil side flow rate is held constant at 1 L/min , as well as , two other figures (Figs. 4.2 and 4.3) are presented for various coil side in flow rates (Qcoil=1.5 and 2 L/min), the effect of pitch at 1 cm was clear comparted to 1.5 and 2 cm. Overall heat transfer coefficient increases significantly when air bubbles are injected into the shell tube regardless pitch size. It is obvious that the enhancement of the overall heat transfer coefficient continues to improve for increasing shell side in flow rate.

The increase in overall heat transfer coefficient can be assign to the influence of the injected air bubbles. Due to the density difference between the air and water bubbles inside the shell, the air bubbles are pushed vertically through the column of the water under the effect of the buoyant force. As a result, the air bubble will pull water from the side of the shell and thus increase the Reynolds number (Re) and turbulence degree. Adding that, the movement of a circulating water within a shell, such as a shell cover, will also cause secondary water flows to occur, increasing mixing level. Thus, both the boundary layers hydrodynamic/thermal on the outer surface of the coil will be ruptured.

This is important when the temperature of the water layers around coil tube surface is significantly higher than the temperature of the rest of the larger section of the water inside the shell.

Moreover, the poor thermal conductivity of air could also decrease the rate of heat loss to the surroundings.

Finally, it should be noted that the actual water velocity inside the shell is greater than the water velocity measured from Eq (3.5) since more mixing flow (water flow + airflow) can rise through the same shell after the airflow injection within the shell side of the heat exchanger. Furthermore, necessarily, the mixing flow would rise at a faster flow rate than pure water. As a result, the actual water velocity and the actual shell side Reynolds number are greater than the Reynolds number calculated from Eq (3.4). As a result, this additional Reynolds number can be added as another positive action of the air injection that increases the overall heat transfer coefficient rate of the heat exchanger.

Generally, the rate of overall heat transfer coefficient (U) substantially increases when the air bubbles are injected into the shell side while the flow rates of the water are at constant. It is clearly shown that the augment of the overall heat transfer coefficient continues to progress with increasing shell side mixing water/air bubbles flow rate. Interestingly, (U) enhancement seems to have a maximum value depending on the air injection flow rate. Thus, the air flow rates of 2 L/min can be represented as the flow rate required to produce the maximum improvement in (U) within the study range for the three hot coil flow rates (1,1.5 and 2 L/min).

The overall heat transfer coefficient (U) increases significantly when air bubbles =2 L/min Q shell=10 L/min, size of pitch =1cm because increasing the surfers area of heat transfer. Both techniques were used in the present study in order to enhance overall heat transfer coefficient.

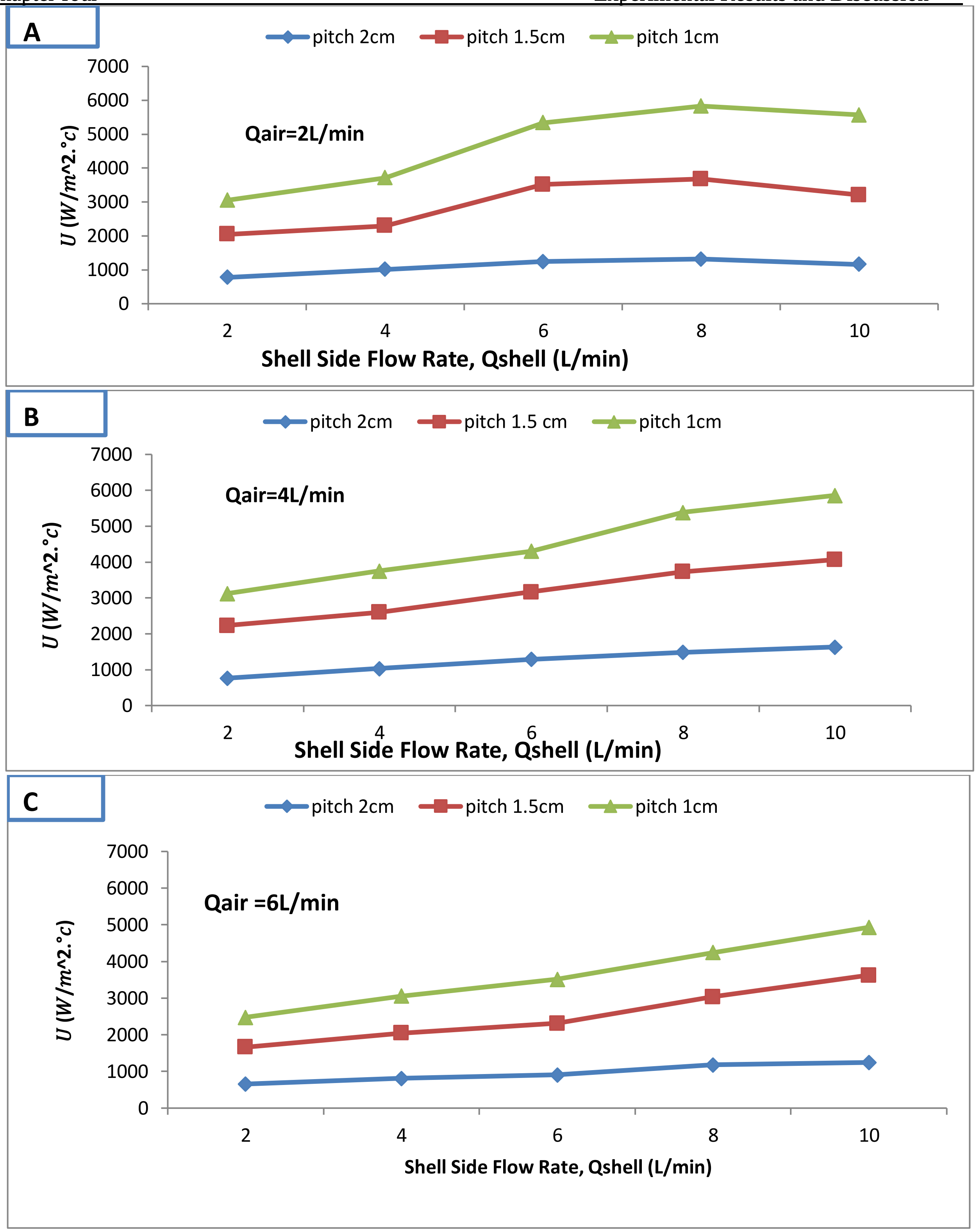


Fig. 4.1. Impact of air bubbles injection on the overall heat transfer coefficient of a three different helical finned coiled tube for a constant coil side flow rate $(Q_{coil} = 1\ l/min)$

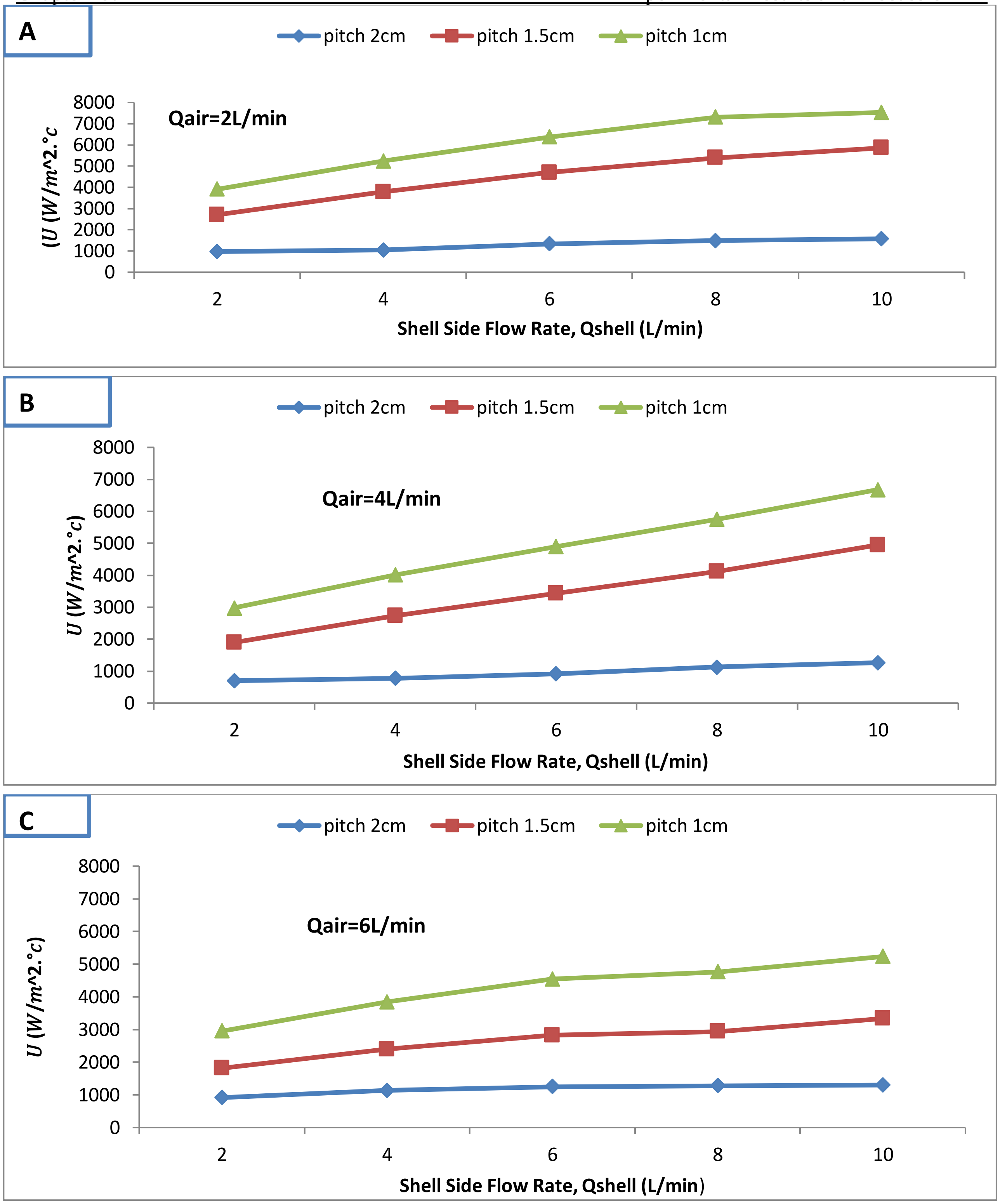


Fig. 4.2: Impact of air bubbles injection on the overall heat transfer coefficient of a three different helical finned coiled tube for a constant coil side flow rate $(Q_{coil} = 1.5\ l/min)$

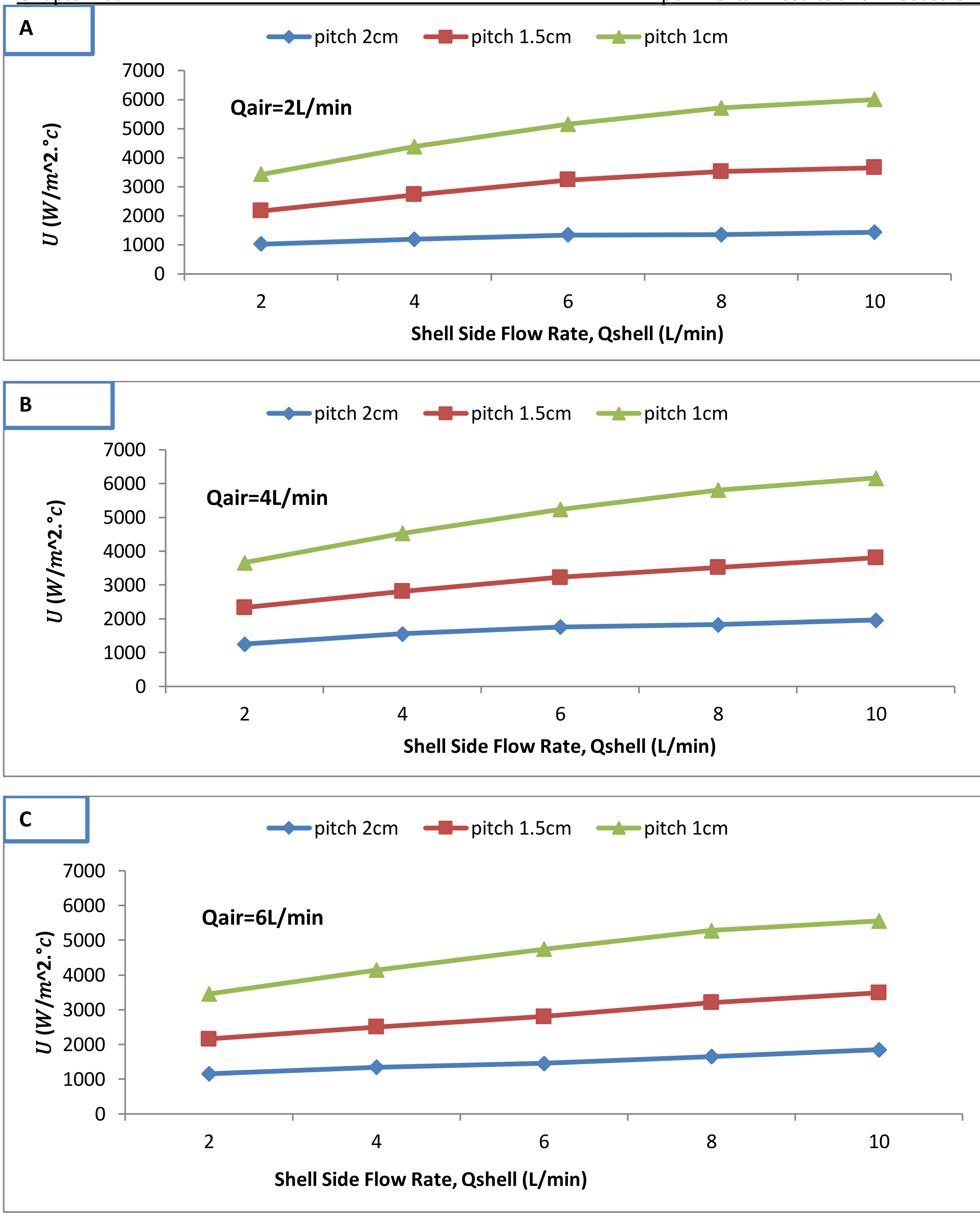


Fig. 4.3: Effect of air bubbles injection on the overall heat transfer coefficient of a three different helical finned coiled tube for a constant coil side flow rate $(Q_{coil} = 2\ l/min)$

Figure 4.4. express the variations of enhancement ratio $\boldsymbol{U_{fa}/U_{sa}}$ with same operation parameters above shell side flow rate (2,4,6,8 and 10) L/min at three different pitch size (P=1,1.5,and 2) cm, and three different constant air flow rates (Qair=2,4 and 6 L/min divided into three figures as shown below (a, b and c) and constant coil side flow rate at 1 L/min. As well as , two other figures (Figs. 4.5 and 4.6) are presented for different coil side flow rates (Qcoil=1.5 and 2 L/min) where $\boldsymbol{U_{fa}}$ refer to the overall heat transfer coefficient of a shell and finned coil heat exchanger with air bubble injecting and $\boldsymbol{U_{sa}}$ refer to the overall heat transfer coefficient of a shell and coil heat exchanger with air bubble injecting . Maximum enhancement ratio was at 1 cm pitch size because raising the number of fin increasing the surfers area of the heat transfer , 1.5 L/min hot water flow rate and 2 L/min air flow rate as a result of air injection and increasing curvature ratio due to reduce pitch size into 1 cm to improve secondary flow configuration.

It can be concluded from figures. 4.4, 4.5, and 4.6 that the highest improvement ratio of the overall heat transfer coefficient ratio $\boldsymbol{U_{fa}/U_{sa}}$ for size of pitch =1 cm and Qshell = 10 L/min is 119 % and the minimum value is 39% at Qshell=2 L/min and size of pitch = 2 cm.

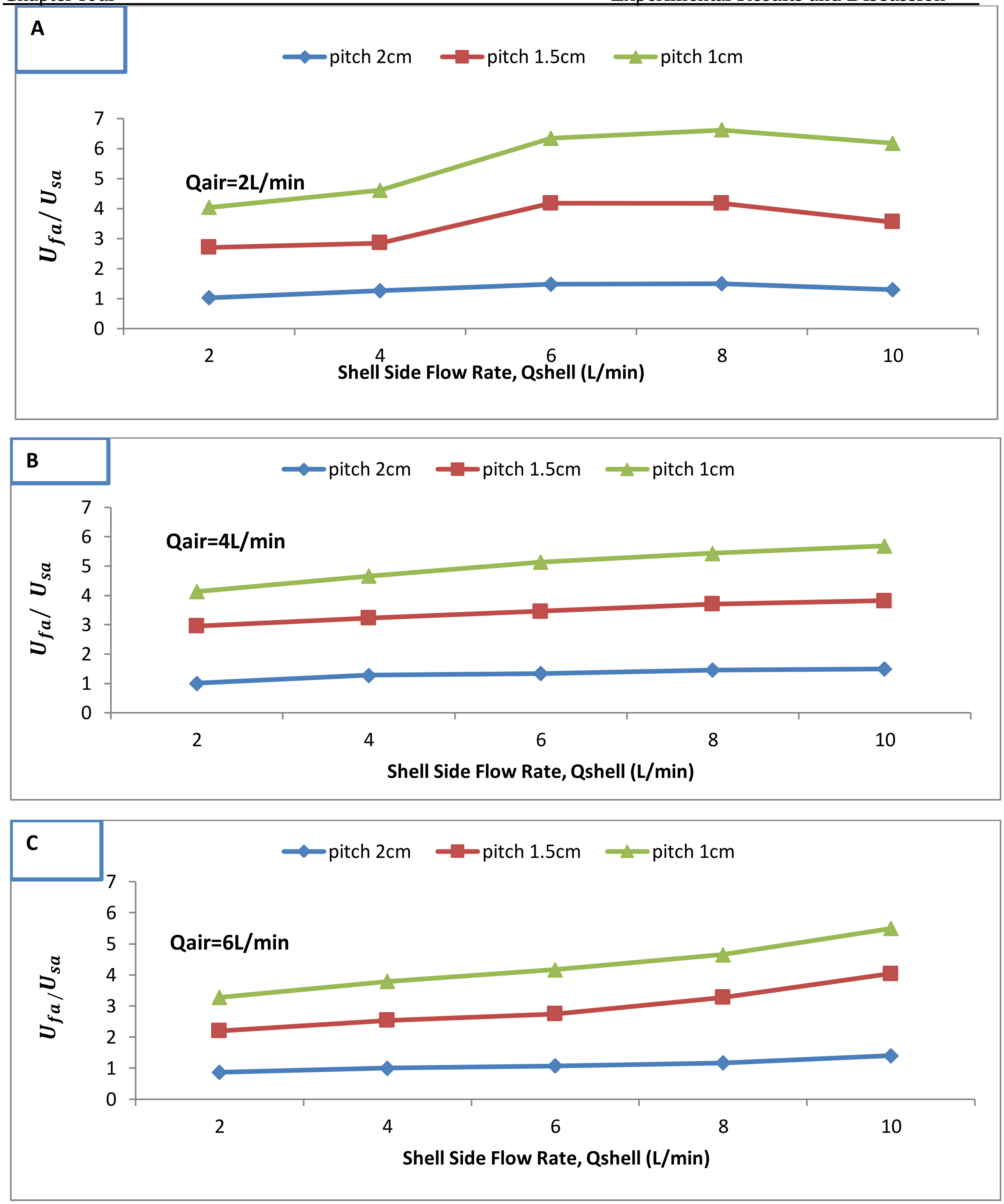


Fig. 4.4: The difference in the enhancement ratio ($\boldsymbol{U_{fa}/U_{sa}}$) with the shell flow rate for three different pitch and constant coil (hot) flow rate ($Q_{coil} = 1\ l/min$)

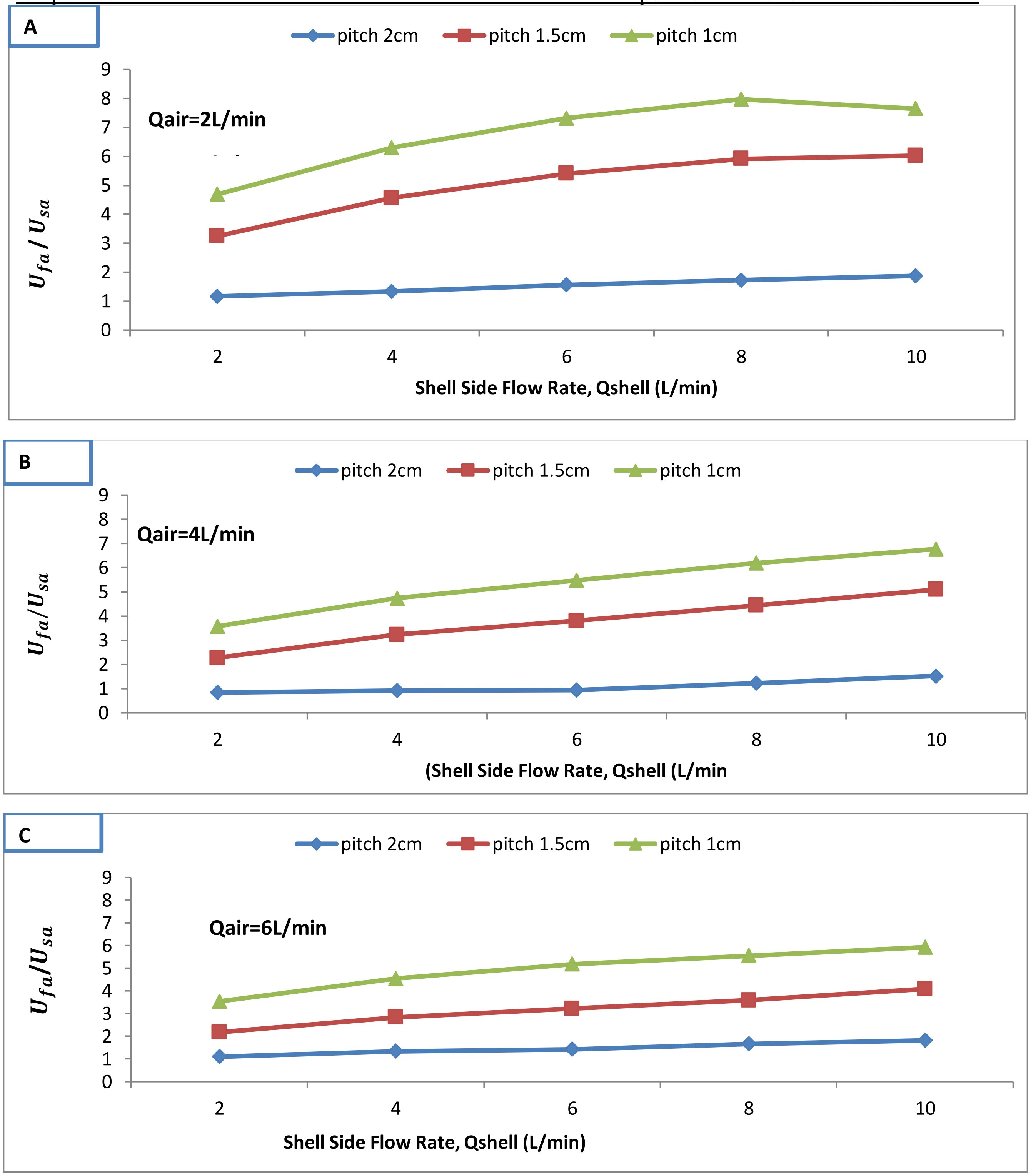


Fig. 4.5: The difference in the enhancement ratio ($\boldsymbol{U_{fa}/U_{sa}}$) with the shell flow rate for three different pitch and constant coil (hot) flow rate ($Q_{coil} = 1.5\ l/min$)

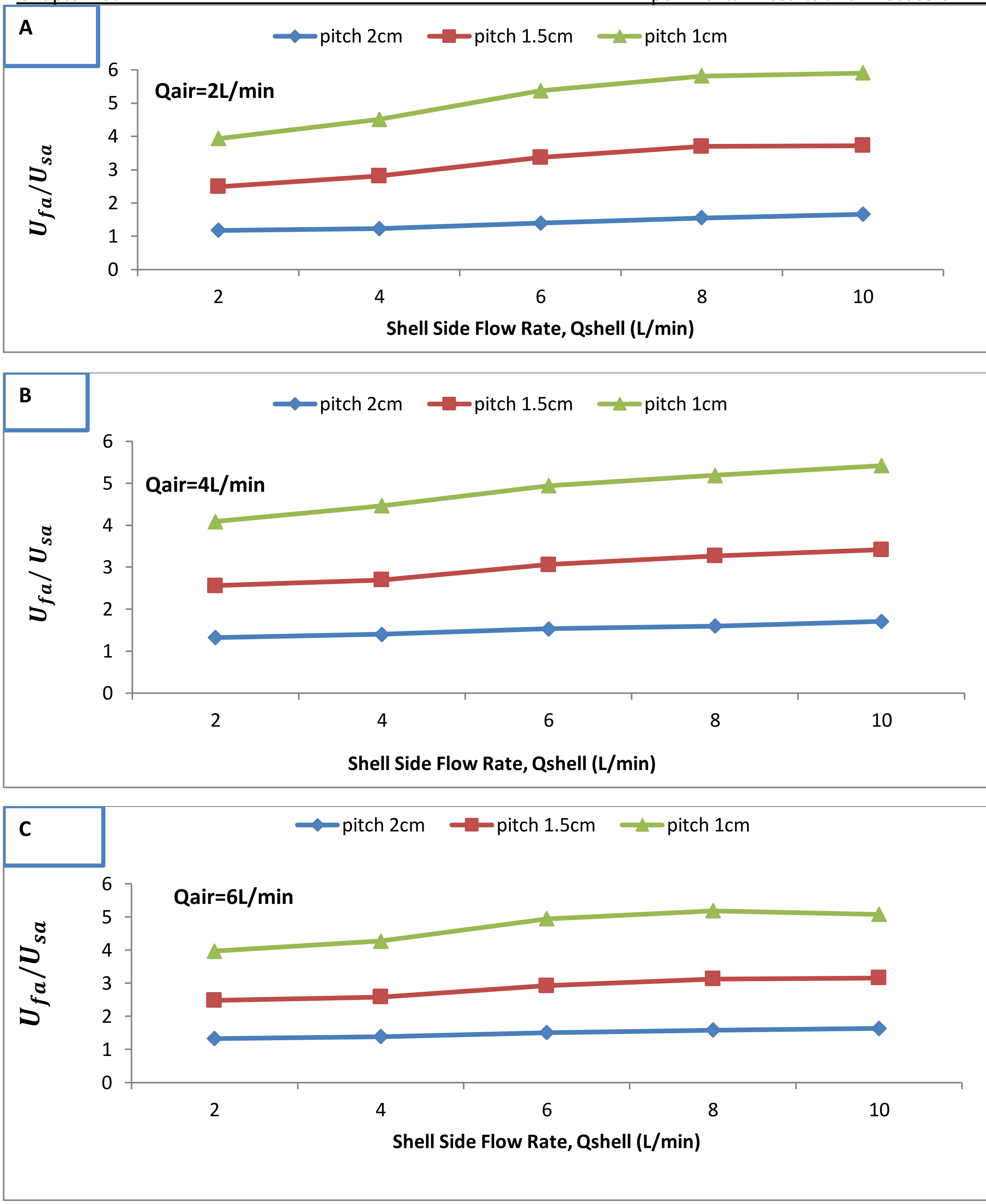


Fig. 4.6: The difference in the enhancement ratio ($\boldsymbol{U_{fa}/U_{sa}}$) with the shell flow rate for three different pitch and constant coil (hot) flow rate ($Q_{coil} = 2\ lmin$)

In addition to the influence on the measured values of U, it is instructive to also consider the dimensionless performance improvement of the heat exchanger. This is done taking into account the gain ratio $\boldsymbol{U_{fa}/U_f}$ which represents the value of the overall heat transfer coefficient for the state with and without air injection for coiled finned tube.

Figure. (4.7) shows the variation of $\boldsymbol{U_{fa}/U_f}$ as the casing side flow rate varies with fixed the flow rate on the coil side ($Qcoil$ = 1 L/min) and three air injection flow rates (2,4 and 6 L/min). For all cases measured here (Figures. 4.8 and 4.9), the ratio of $\boldsymbol{U_{fa}/U_f}$ initially increases significantly as the flow velocity on the shell side increases. This results in a clear maximum value ($Qshell$ = 10 L/min). For the maximum flow rate on the calender side considered, the ratio $\boldsymbol{U_{fa}/U_f}$is always significantly greater than 1, which means that the performance of the exchanger is always higher than in the case without air injection. Also the figures (4.7, 4.8 and 4.9) show that $\boldsymbol{U_{fa}/U_f}$ increases with increasing coil flow rate, but in all state the maximum gain of $\boldsymbol{U_{fa}/U_f}$ is clearly reached at $Qshell$ = 10 L/min, Qair = 2 L/min and size of pitch = 1cm.

It can be concluded from figs. 4.7, 4.9, and 4.10 that the highest improvement ratio of the overall heat transfer coefficient ratio $\boldsymbol{U_{fa}/U_f}$ for size of pitch =1 cm and Qshell = 10 L/min is 83 % and the minimum value is 27 % at Qshell=2 L/min and size of pitch = 2 cm.

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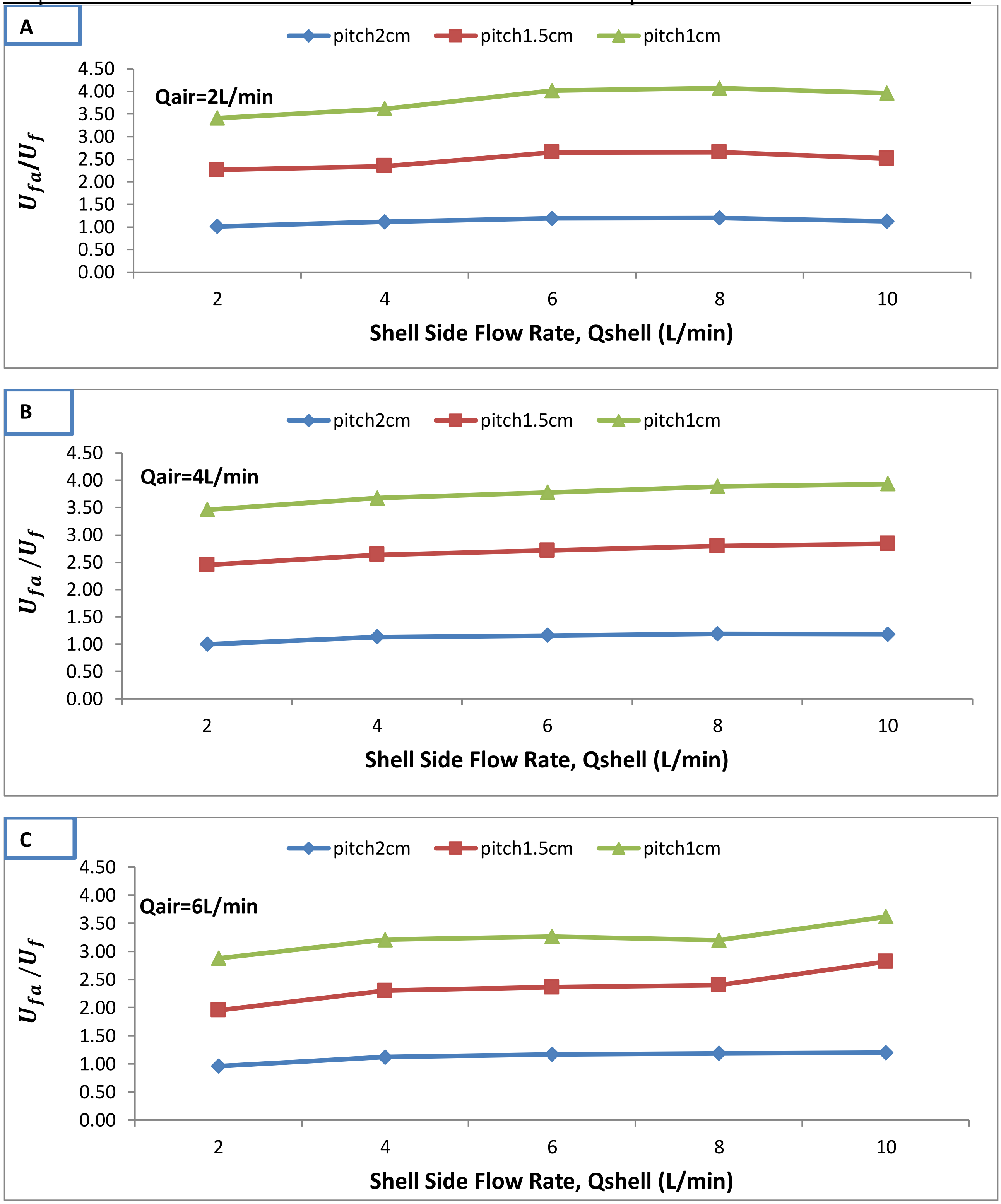


Fig. 4.7: The difference in the enhancement ratio (U_{fa}/U_f) with the shell flow rate for three different pitch and constant coil (hot) flow rate ($Q_{coil} = 1\ l/min$)

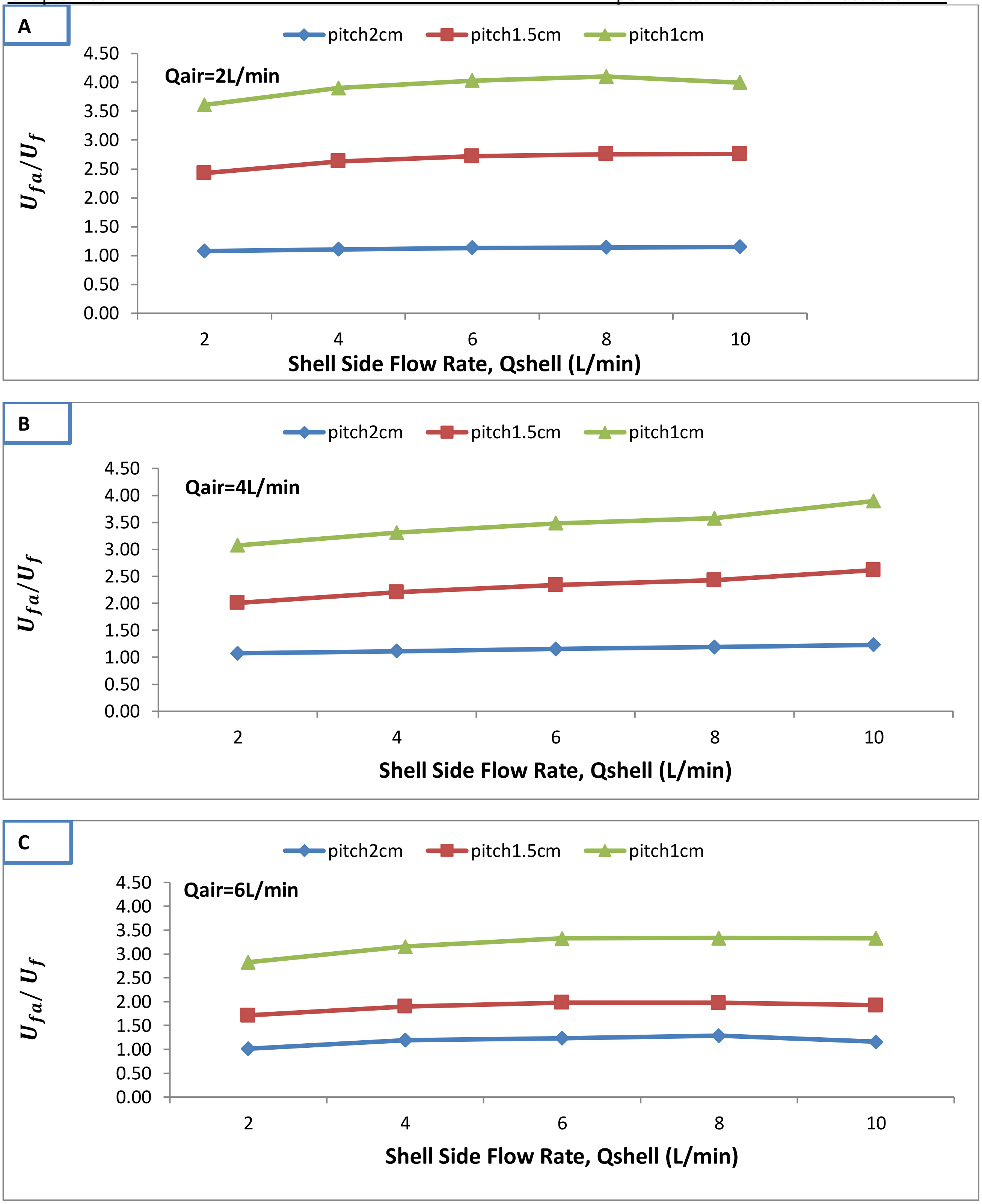


Fig. 4.8: The difference in the enhancement ratio ($\boldsymbol{U_{fa}/U_f}$) with the shell flow rate for three different pitch and constant coil (hot) flow rate ($Q_{coil} = 1.5\ l/min$)

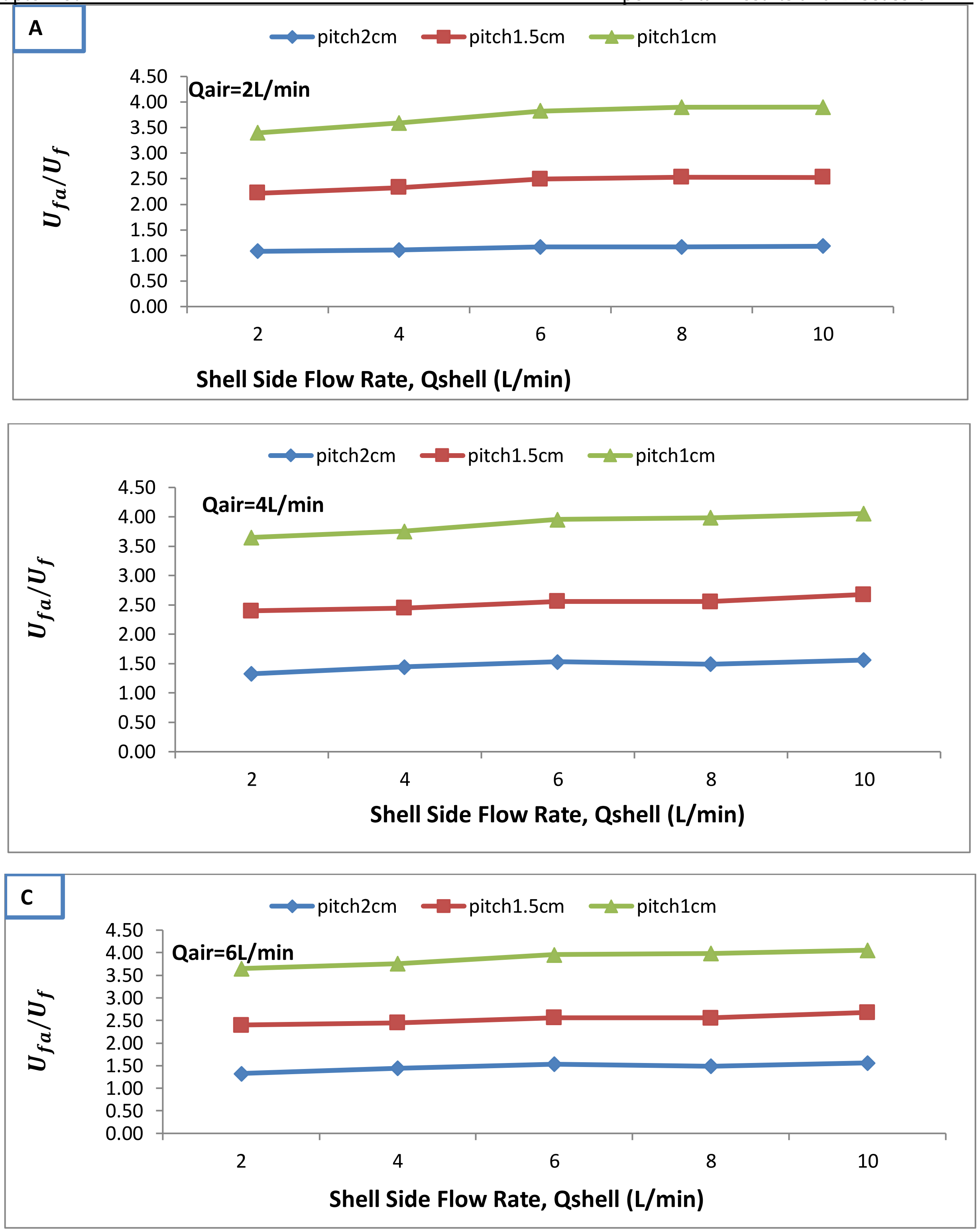


Fig. 4.9: The difference in the enhancement ratio ($\boldsymbol{U_{fa}/U_f}$) with the shell flow rate for three different pitch and constant coil (hot) flow rate ($Q_{coil} = 2\ l/min$)

4.2 Effectiveness

Figure 4.10,shows the effectiveness with the air injected flow rate for a fixing coil side flow rate ($Qcoil$ = 1 $L/$min) and three values of the air rate (Qair = 2,4 and 6LPM). The other two numbers are similar figures (4.11 and 4.12) are presented for variation coil side flow rates ($Qcoil$ = 1.5 and 2 $L/$min).

The increase in output as well as the increase in the shell on the side stream scale can be seen from the figure. Similarly, the impact of the flow rate for shell side on the effectiveness is limit on $Qshell$ = 10 $L/$min. Thus, these observations can lead to the conclusion that $Qair$ = 2 $L/$min and $Qshell$ = 10 $L/$min are the best case flows within the current experimental conditions. This actual search was not noticed in any of the relevant studies. With a regular flow measurement on the coil side ($Qcoil$ = 1 LPM)) the incremental increase in the thermal effectiveness at the shell side flow rate is 10 L/min, air flow rate injection of 2 LPM, pitch = 1 cm ,While the minimum the thermal effectiveness at the flow rate of shell side = 2 LPM, the air flow rate is 6 LPM, the pitch = 2 cm and the temperature difference is 20 ℃

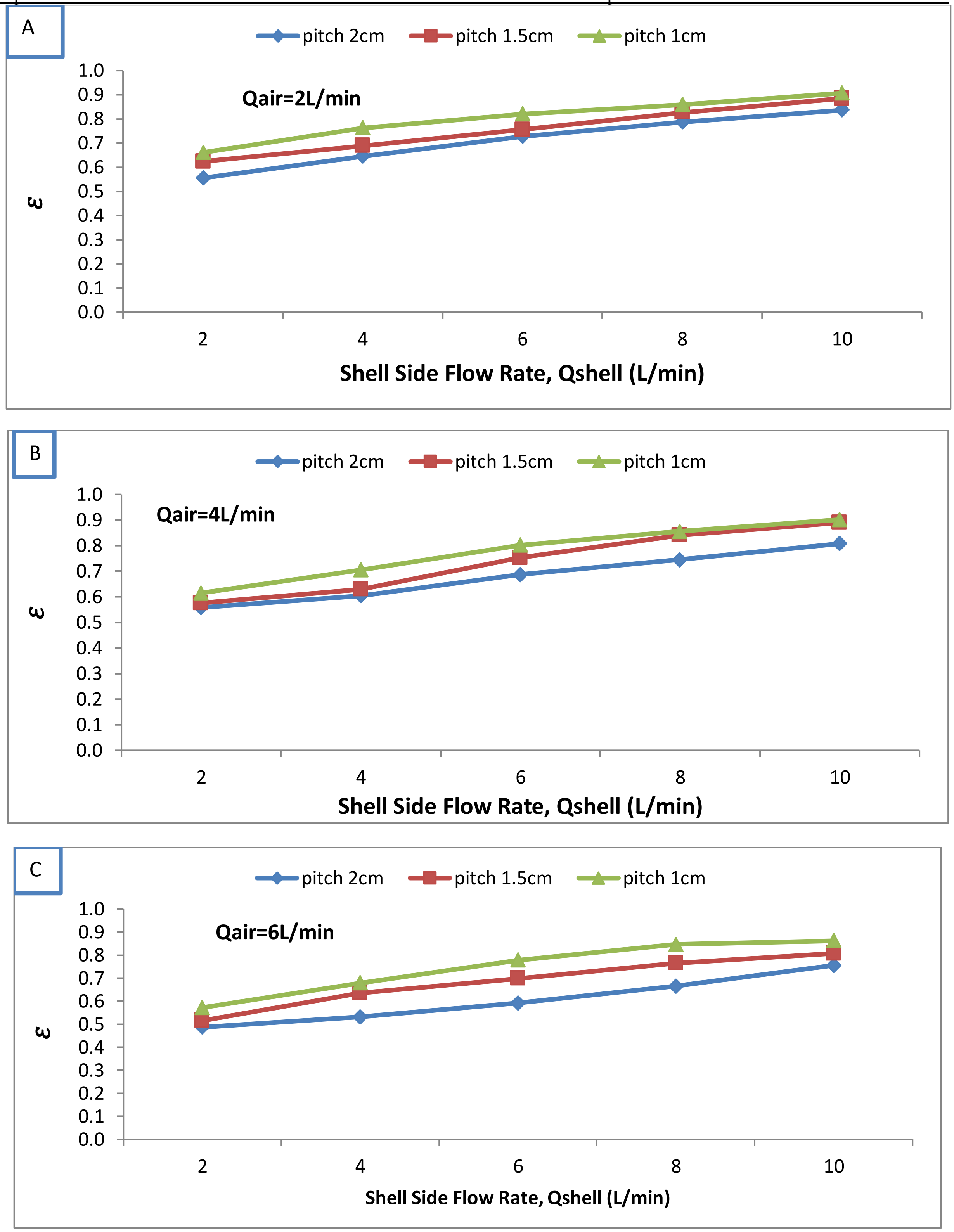


Fig. 4.10: Variation of effectiveness with the flow rate of shell for three different pitch and constant coil (hot) flow rate ($Q_{coil} = 1l/min$)

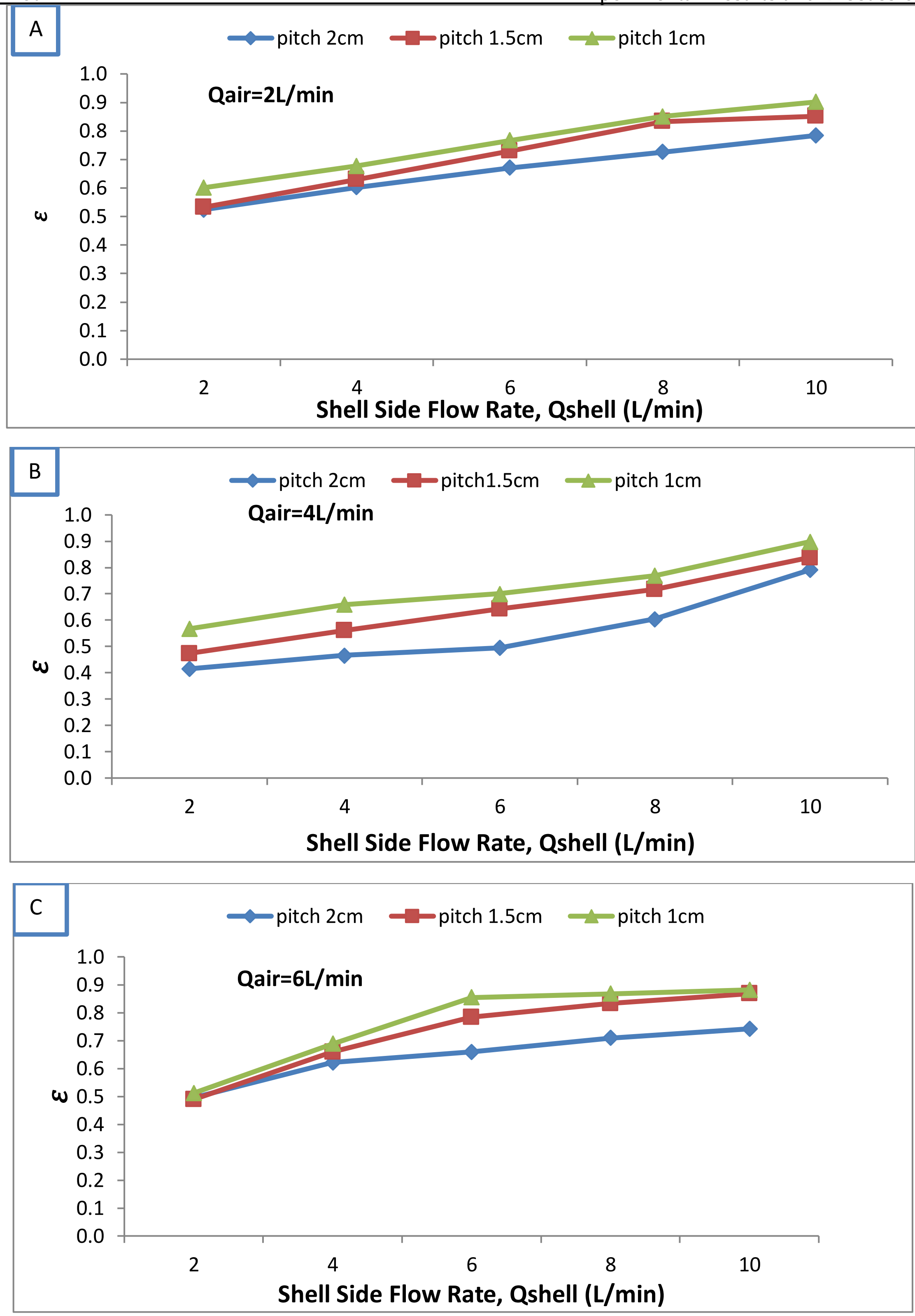


Fig. 4.11: Variation of effectiveness with the flow rate of shell for three different pitch and constant coil (hot) flow rate ($Q_{coil} = 1.5 l/min$)

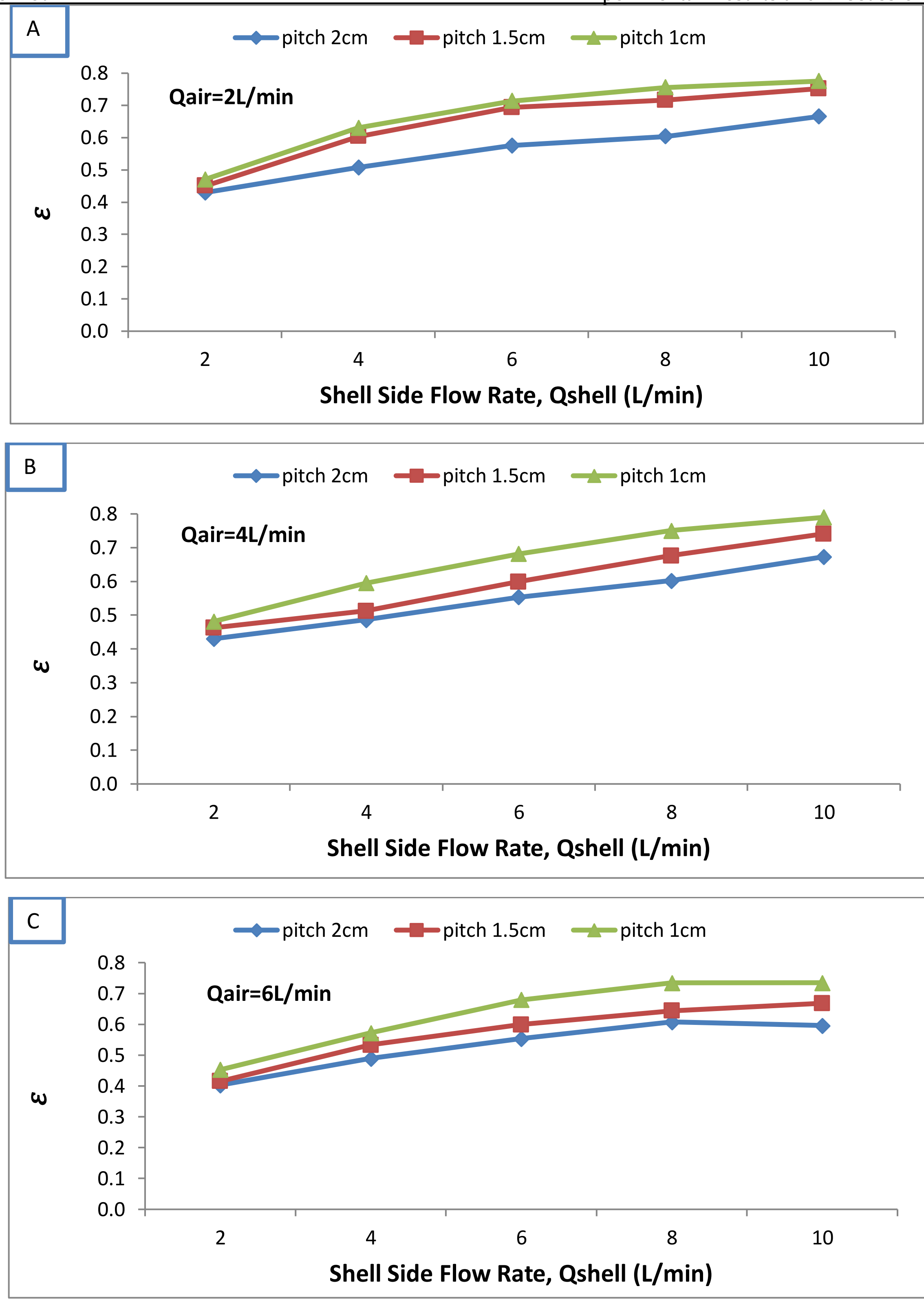


Fig. 4.12: Variation of effectiveness with the flow rate of shell for three different pitch and constant coil (hot) flow rate ($Q_{coil} = 2l/min$)

Figure 4.13 express the variations of enhancement ratio of the effectiveness $\boldsymbol{\varepsilon_{fa}/\varepsilon_{sa}}$ with same operation parameters above shell side flow rate (2,4,6,8 and 10) L/min at three different pitch size (P=1,1.5,and 2) cm, and three different constant air flow rates (Qair=2,4 and 6 L/min divided into three figures as shown below (a, b and c) and constant coil side flow rate at 1 L/min. As well as , two other figures (4.14 and 4.15) are presented for different coil side flow rates (Qcoil=1.5 and 2 L/min) where $\boldsymbol{\varepsilon_{fa}}$ refer to the effectiveness of a shell and finned coil heat exchanger with air bubble injecting and $\boldsymbol{\varepsilon_{sa}}$ refer to the effectiveness of a shell and coil heat exchanger with air bubble injecting. Maximum enhancement ratio was at 1 cm pitch size because raising the number of fin increasing the surfers area of the heat transfer , 1.5 L/min hot water flow rate and 2 L/min air flow rate as a result of air injection and increasing curvature ratio due to reduce pitch size into 1 cm to improve secondary flow configuration.

It can be concluded from figures. 4.13, 4.14, and 4.15 that the highest improvement ratio of the effectiveness ratio $\boldsymbol{\varepsilon_{fa}/\varepsilon_{sa}}$ for size of pitch =1 cm and Qshell = 10 L/min is 122 % and the minimum value is 18% at Qshell=2 L/min and size of pitch = 2 cm.

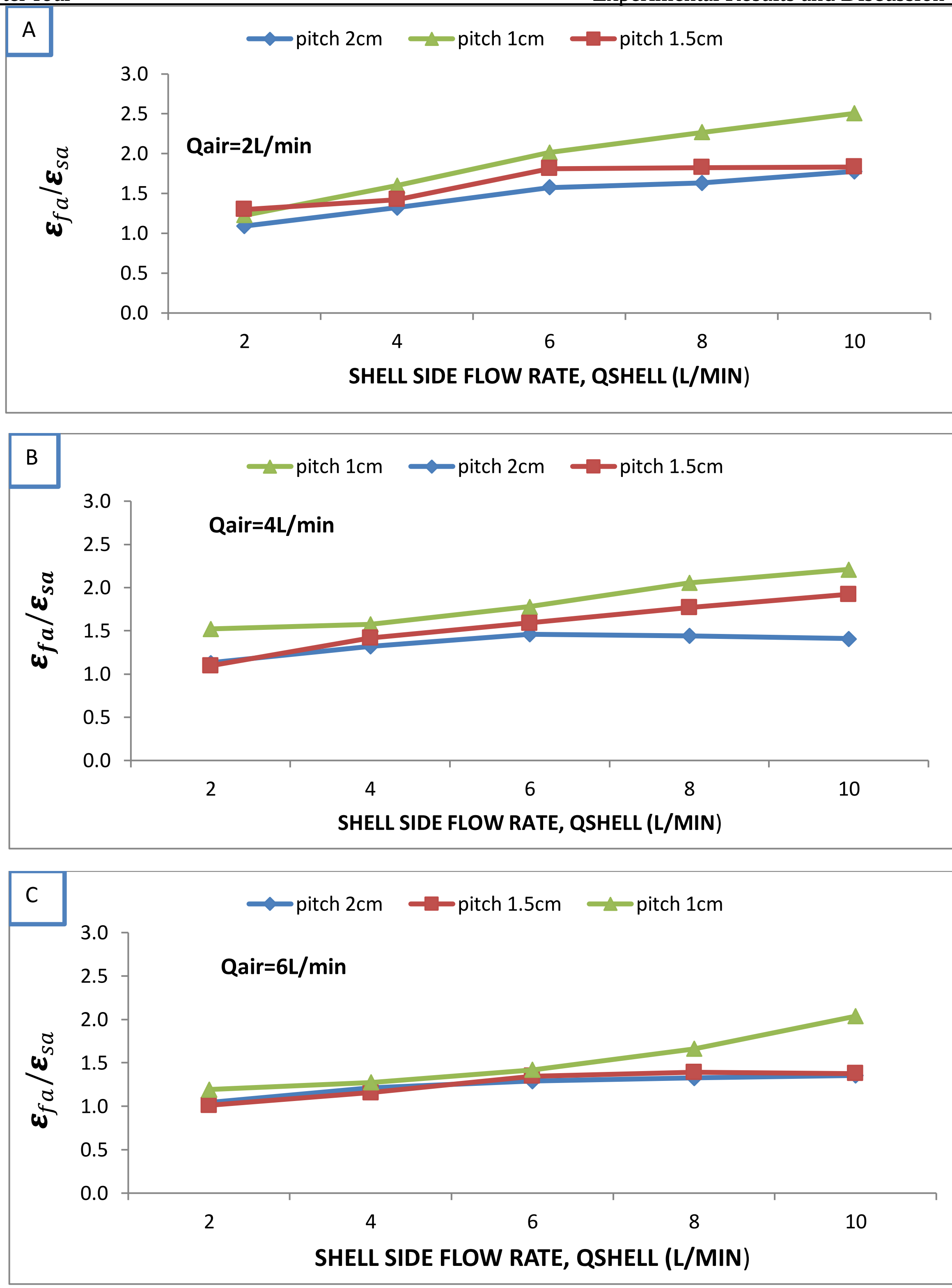


Fig. 4.13: The difference in the effectiveness enhancement ratio with the flow rate shell for three different pitch and constant coil (hot) flow rate ($Q_{coil} = 1\ l/min$)

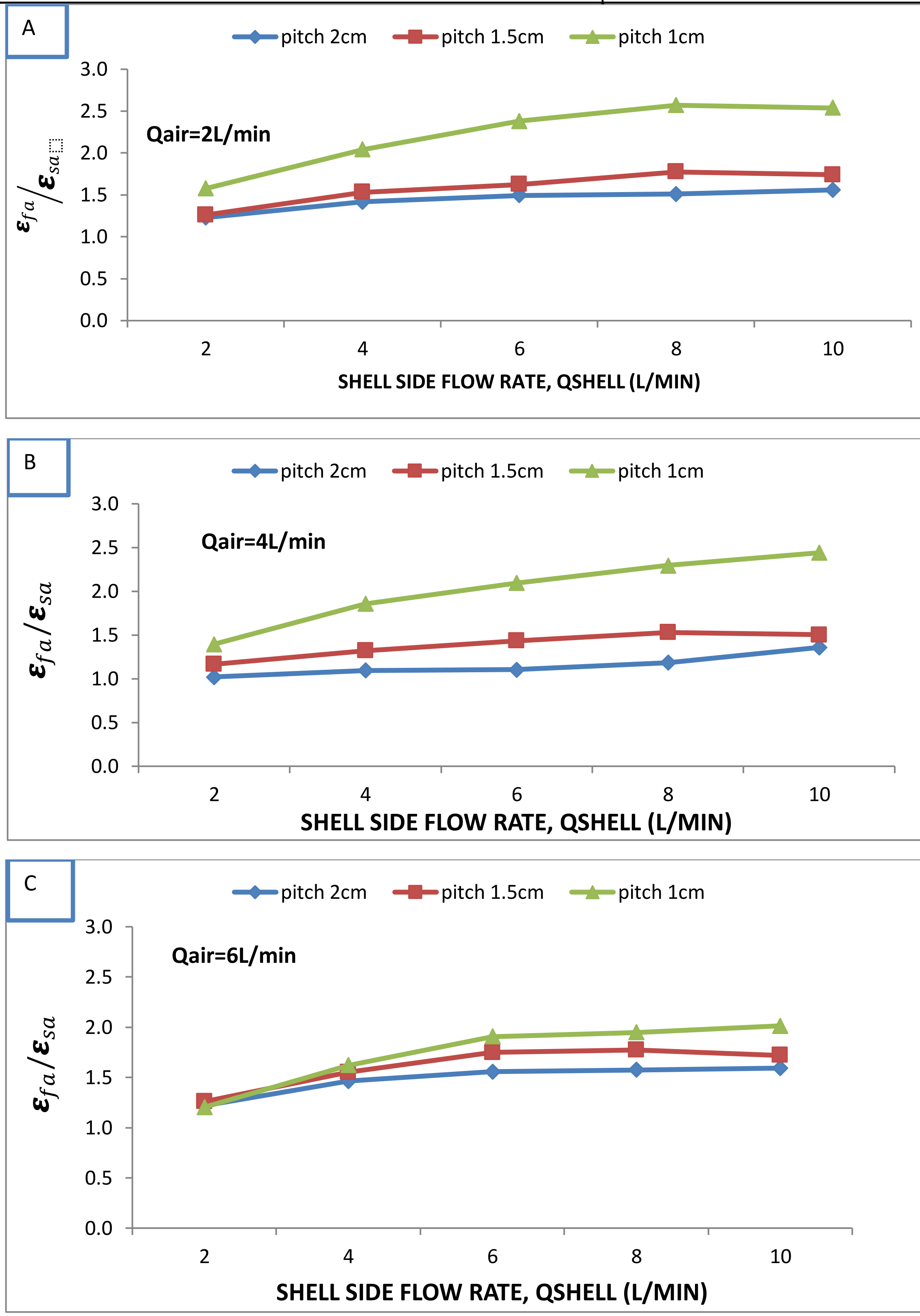


Fig. 4.14: The difference in the effectiveness enhancement ratio with the flow rate shell for three different pitch and constant coil (hot) flow rate ($Q_{coil} = 1.5\ l/min$)

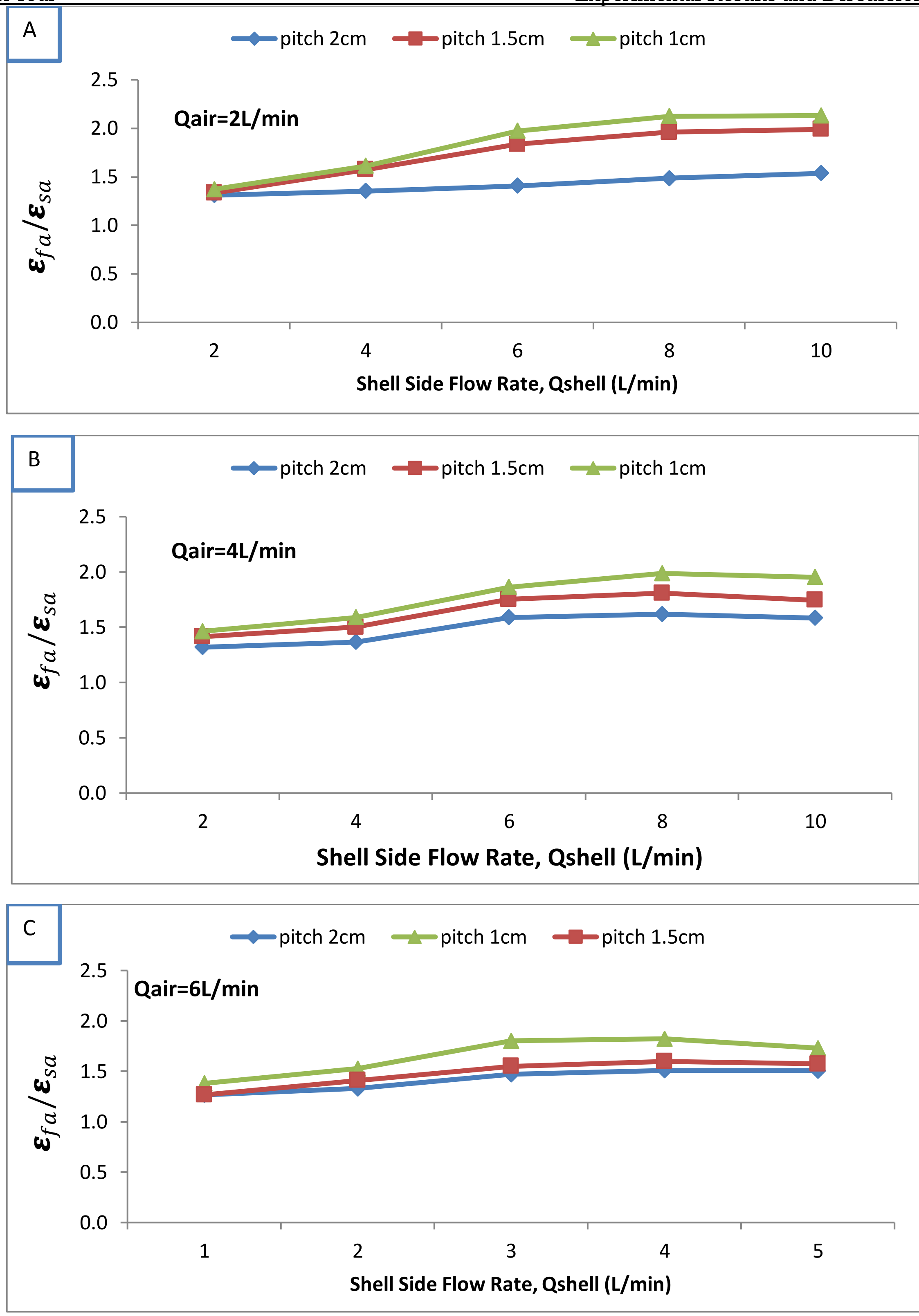


Fig. 4.15: The difference in the effectiveness enhancement ratio with the flow rate shell for three different pitch and constant coil (hot) flow rate ($Q_{coil} = 2\ l/min$)

Figure 4.16, show the relationship between the enhancement ratio of the effectiveness $\boldsymbol{\varepsilon_{fa}/\varepsilon_f}$ and the air injected flow rate for variation shell side flow rate and fixed coil side flow ($Qcoil$ = 1 L/min) . Similarly, two other figures (4.17 and 4.18) are presented for variation coil side flow rates ($Qcoil$ = 1.5 and 2 L/min).

The optimal value to enhance effectiveness of heat exchanger is when the air injected flow rate 2 L/min, $Qshell$ = 10 L/min and pitch=1cm are as seen in Figure 4.16

It can be concluded from figs. 4.7, 4.9, and 4.10 that the highest improvement ratio of the enhancement ratio of the effectiveness ratio $\boldsymbol{\varepsilon_{fa}/\varepsilon_f}$ for size of pitch =1 cm and Qshell = 10 L/min is 55 % and the minimum value is 11 % at Qshell=2 L/min and size of pitch = 2 cm.

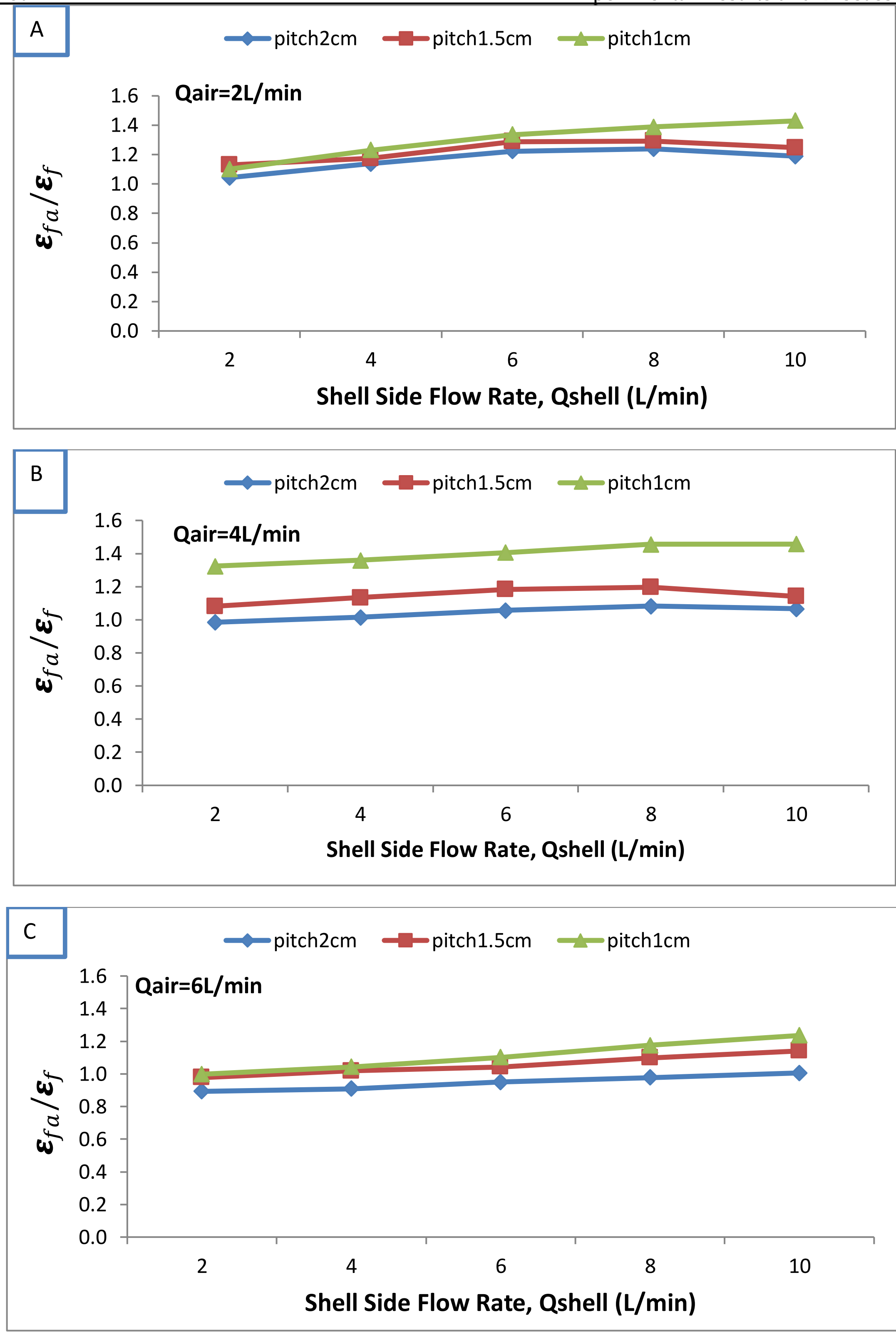


Fig. 4.16: The different in the effectiveness enhancement ratio with the shell flow rate for three different pitch and constant coil (hot) flow rate ($Q_{coil} = 1\ l/min$)

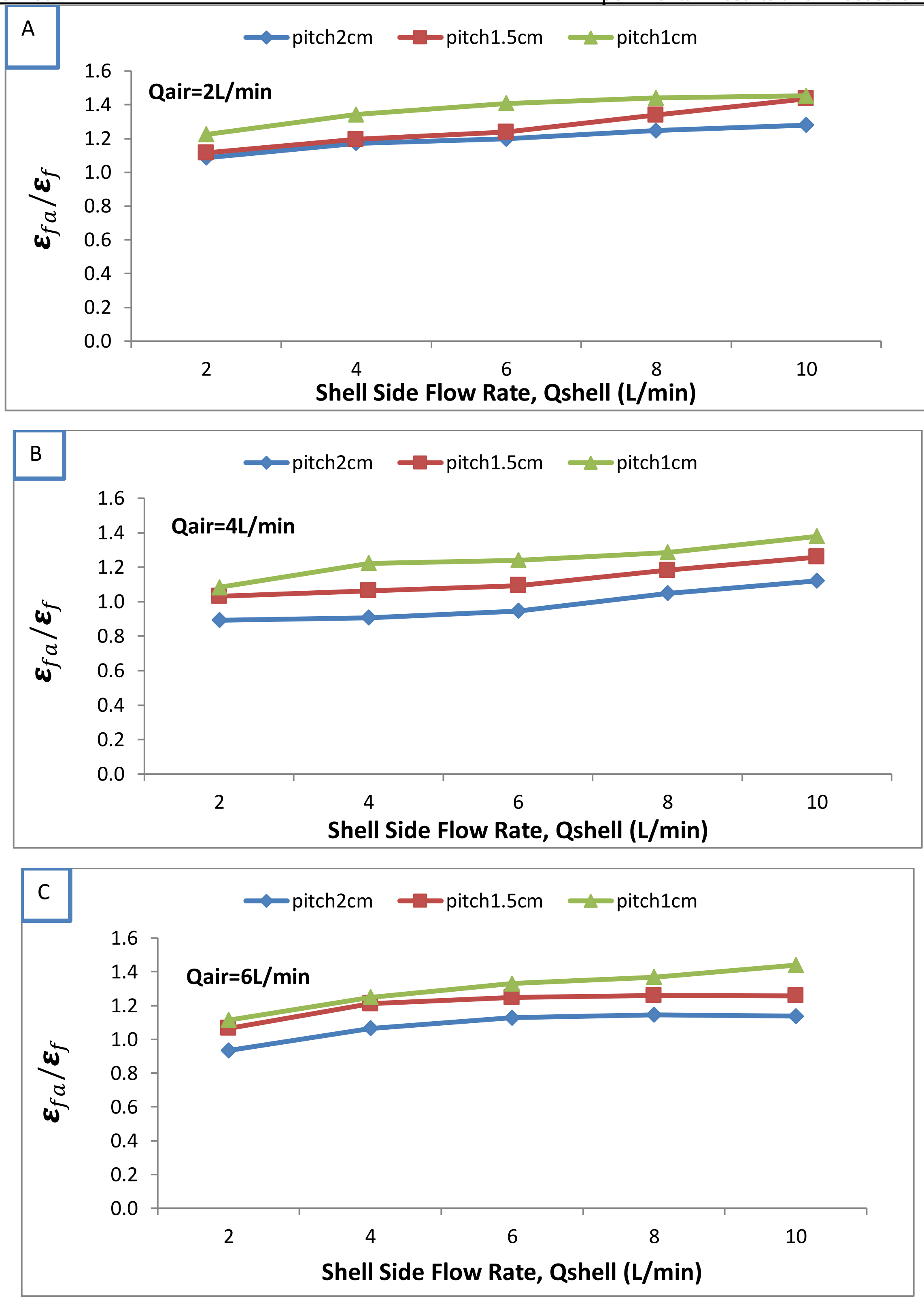


Fig. 4.17: : The different in the effectiveness enhancement ratio with the shell flow rate for three different pitch and constant coil (hot) flow rate ($Q_{coil} = 1.5\ l/min$)

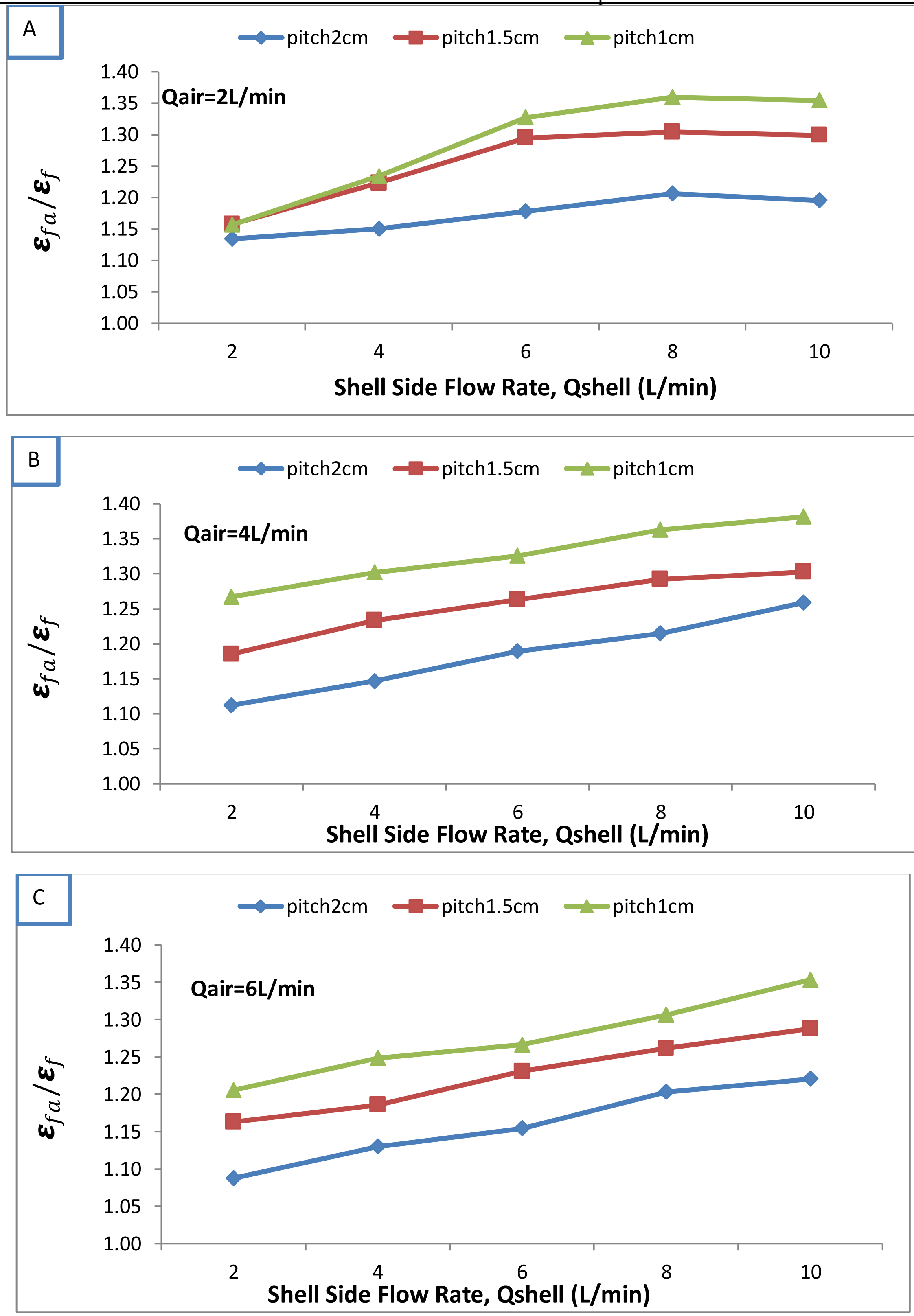


Fig. 4.18: : The different in the effectiveness enhancement ratio with the shell flow rate for three different pitch and constant coil (hot) flow rate ($Q_{coil} = 2\ l/min$)

4.3 Number of Heat Transfer Units (NTU).

. Figure 4.19 shows the NTU difference with the cold water flow rate ($Qshell$) for variation degrees of injected air flow ($Qair$) and the coil side flow rate (= 1 LPM) and the constant temperature change ($\Delta T = 20°C$) . Similarly, the other two (Figures 4.20 and 4.21) are developed in series with different hot water flow rates ($Qcoil$ = 1.5 and 2 LPM), respectively. Of course, the air injected in the form of small waves around the heat gun contributes significantly to the improvement of NTU. The minimum value obtained from NTU was 0.62 at $Qair$ = 6LPM, $Qcoil$ = 2 $L/$min, $Qshell$ = 2 $L/$min ,pitch =2cm. While, the maximum of value of NTU is 6.0 at $Qair$ = 2 $L/$min, $Qcoil$ = 1LPM, $Qshell$ = 10$L/$min ,pitch=1cm

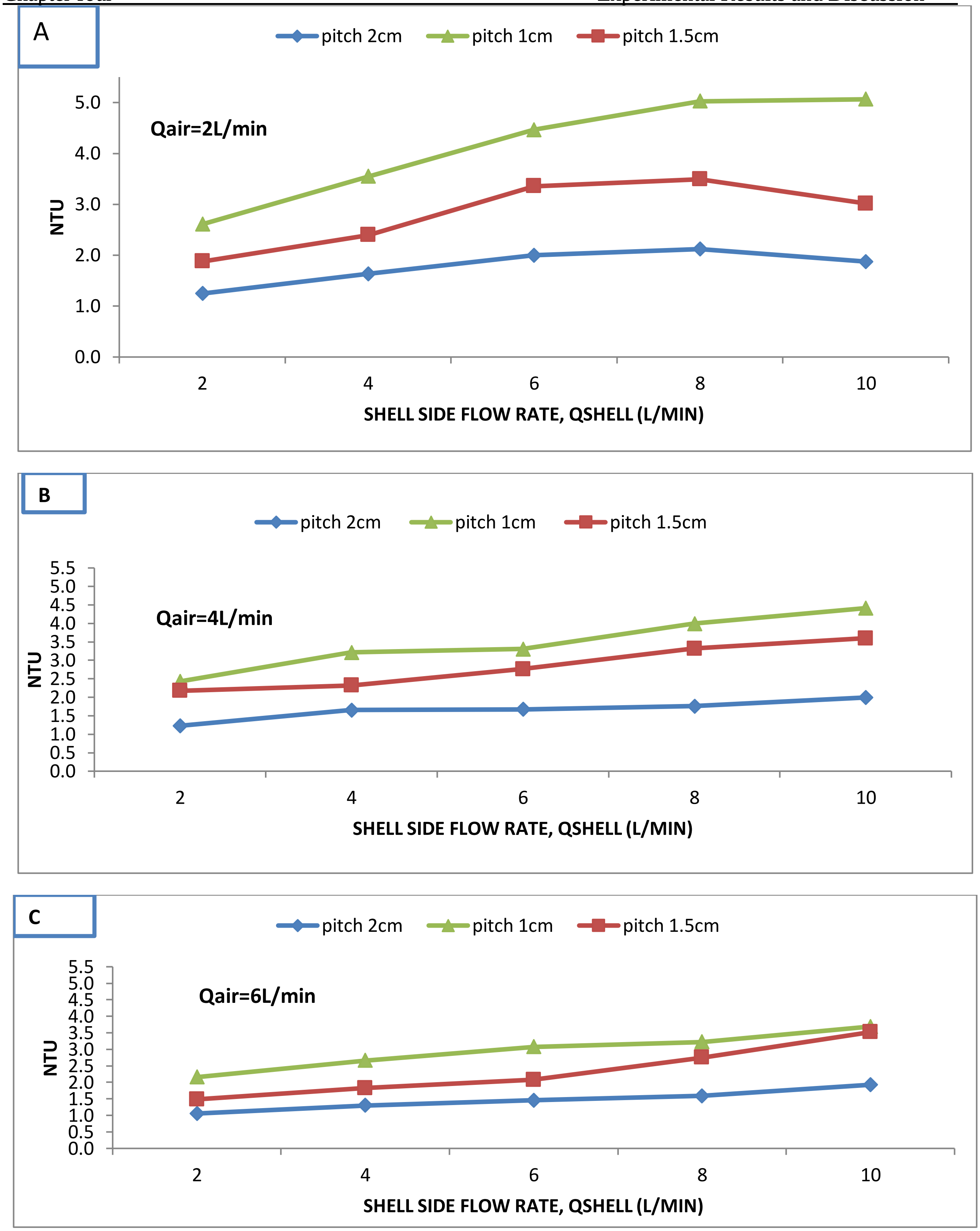


Fig. 4.19.Impact bubbles of air s injection on the NTU of a three variation helical finned coiled tube for a constant flow rate for coil side ($Q_{coil} = 1\ l/min$)

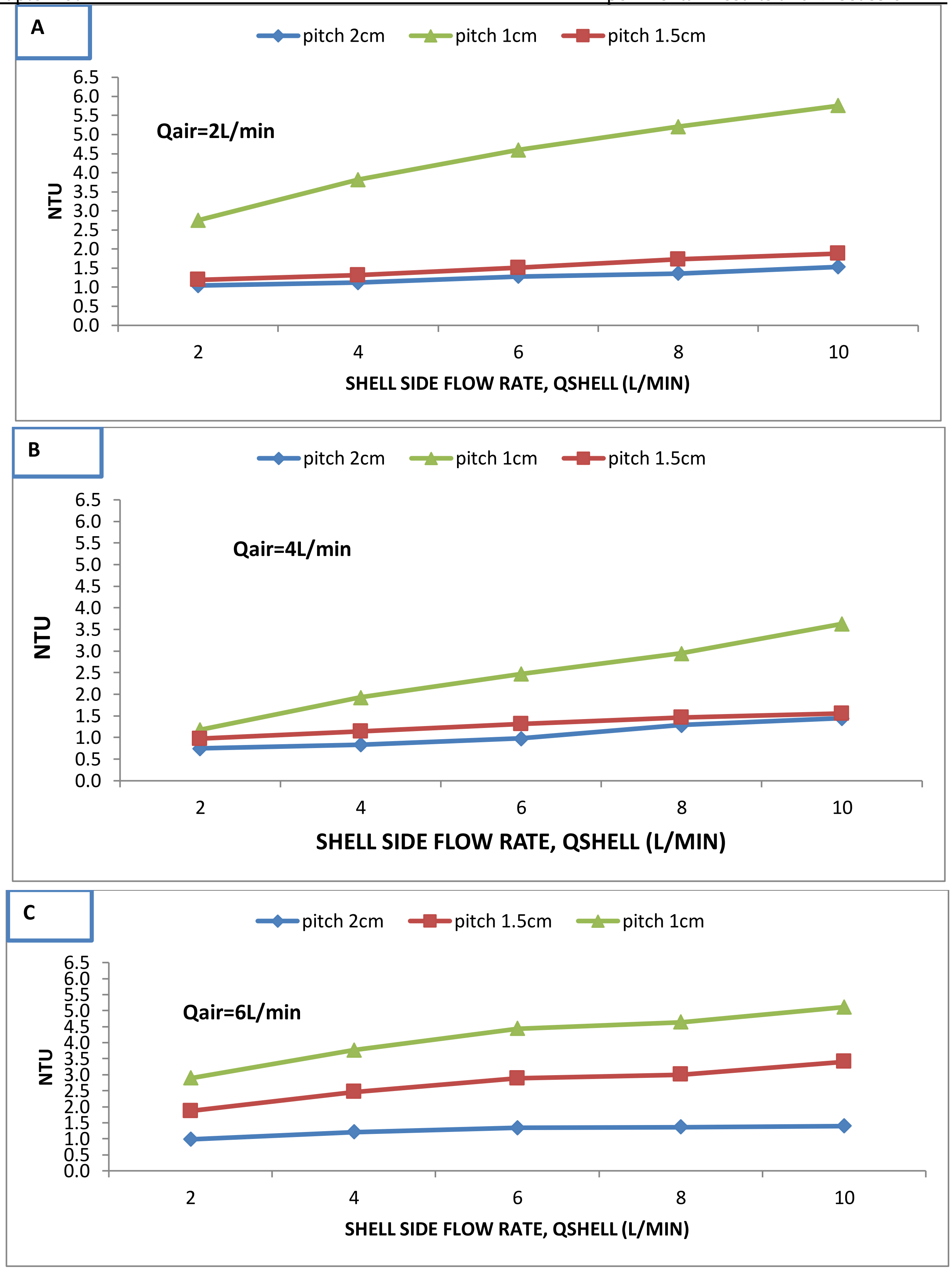


Fig. 4.20 Impact bubbles of air s injection on the NTU of a three variation helical finned coiled tube for a constant flow rate for coil side ($Q_{coil} = 1.5\ l/min$)

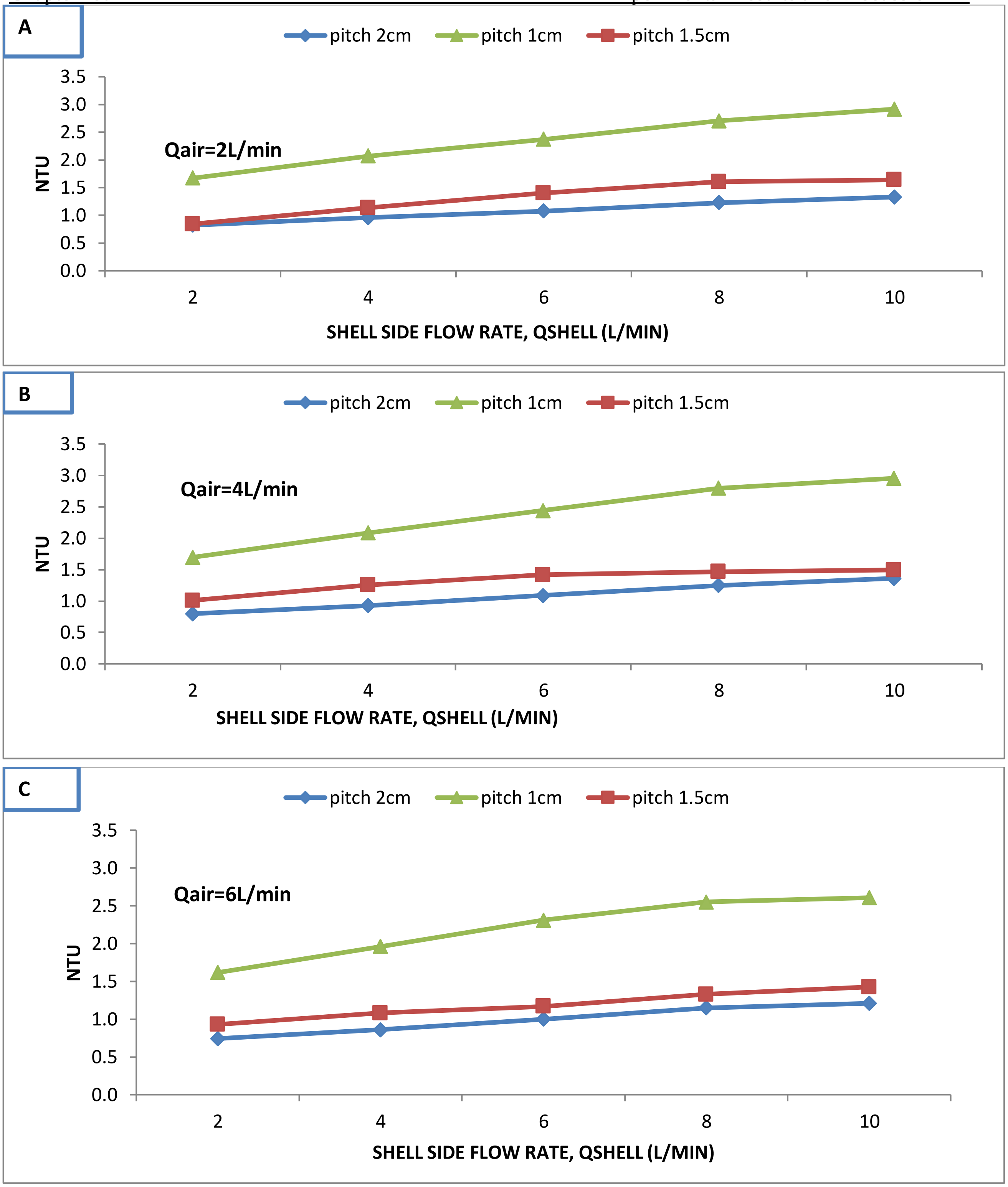


Fig. 4.21.Impact bubbles of air s injection on the NTU of a three variation helical finned coiled tube for a constant flow rate for coil side ($Q_{coil} = 2\ l/min$)

Figure 4.22 express the variations of enhancement ratio $\boldsymbol{Ntu_{fa}/Ntu_{sa}}$ with same operation parameters above shell side flow rate (2,4,6,8 and 10) L/min at three different pitch size (P=1,1.5,and 2) cm, and three different constant air flow rates (Qair=2,4 and 6 L/min divided into three figures as shown below (a, b and c) and constant coil side flow rate at 1 L/min. as well as , two other figures (4.23 and 4.24) are presented for different coil side flow rates (Qcoil=1.5 and 2 L/min) where $\boldsymbol{Ntu_{fa}}$ refer to the number of heat transfer unit of a shell and finned coil heat exchanger with air bubble injecting and $\boldsymbol{Ntu_{sa}}$ refer to the number of heat transfer unit ectiveness of a shell and coil heat exchanger with air bubble injecting . Maximum enhancement ratio was at 1 cm pitch size because raising the number of fin increasing the surfers area of the heat transfer , 1.5 L/min hot water flow rate and 2 L/min air flow rate as a result of air injection and increasing curvature ratio due to reduce pitch size into 1 cm to improve secondary flow configuration.

It can be concluded from Figure s. 4.22, 4.23, and 4.24 that the highest improvement ratio of the effectiveness ratio $\boldsymbol{Ntu_{fa}/Ntu_{sa}}$ for size of pitch =1 cm and Qshell = 10 L/min is 116 % and the minimum value is 36% at Q shell=2 L/min and size of pitch = 2 cm.

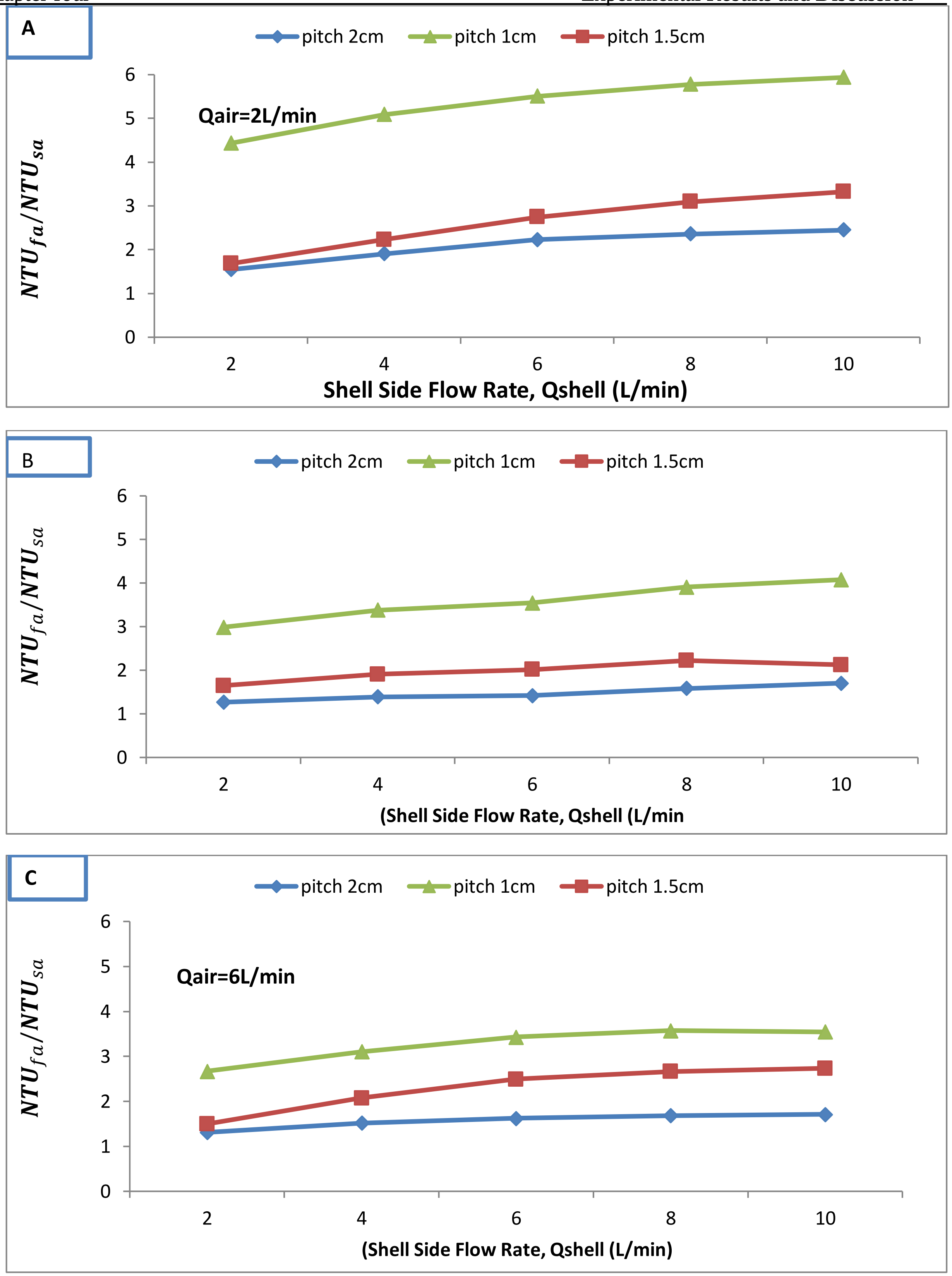


Fig.4.22. the variation in the enhancement ratio ($\boldsymbol{NTU_{fa}/NTU_{sa}}$) with the shell flow rate for three different pitch and constant coil (hot) flow rate ($Q_{coil} = 1\ l/min$)

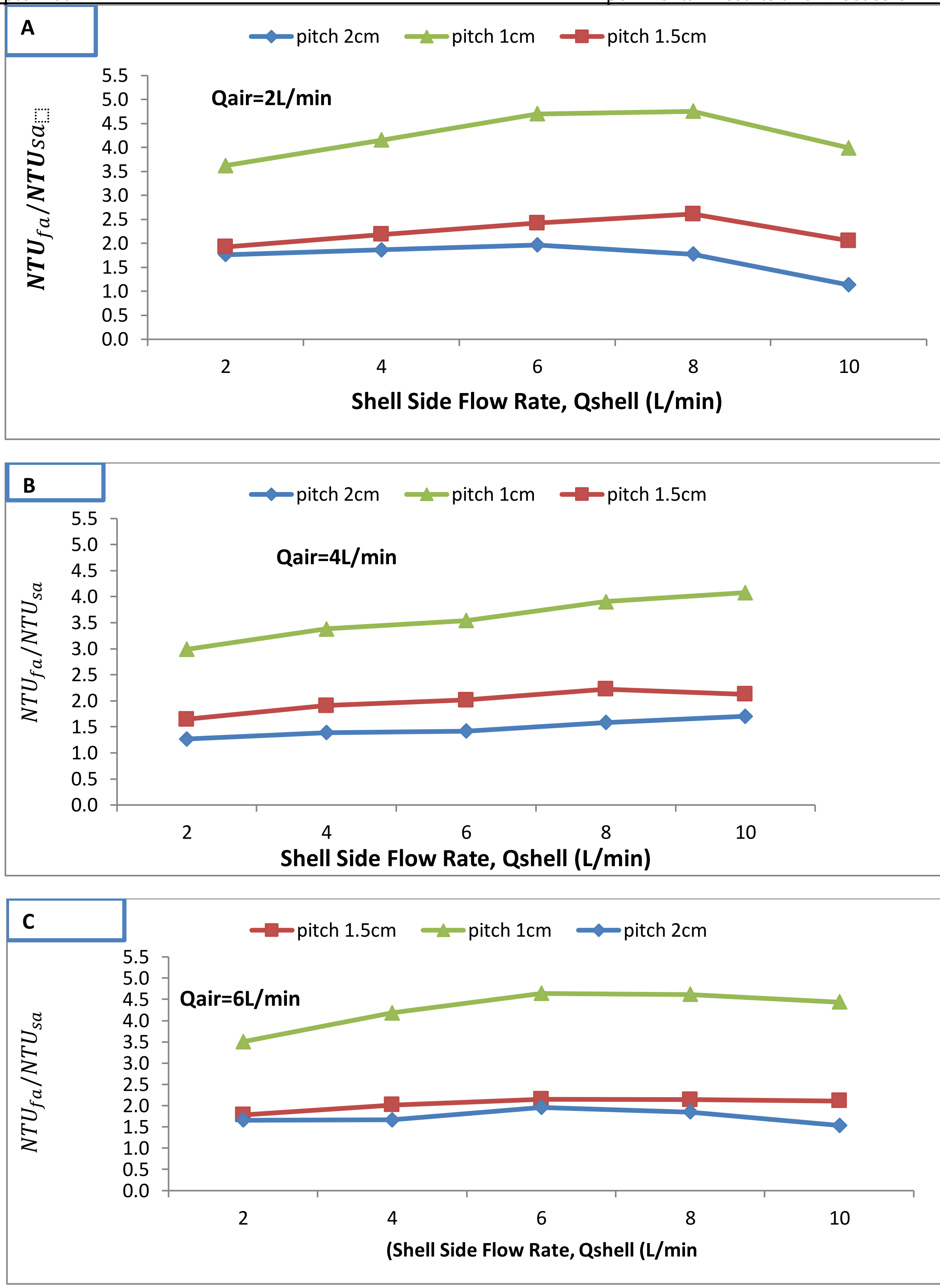


Fig.4.23 the variation in the enhancement ratio ($\boldsymbol{NTU_{fa}/NTU_{sa}}$) with the shell flow rate for three different pitch and constant coil (hot) flow rate ($Q_{coil} = 1.5\ l/min$)

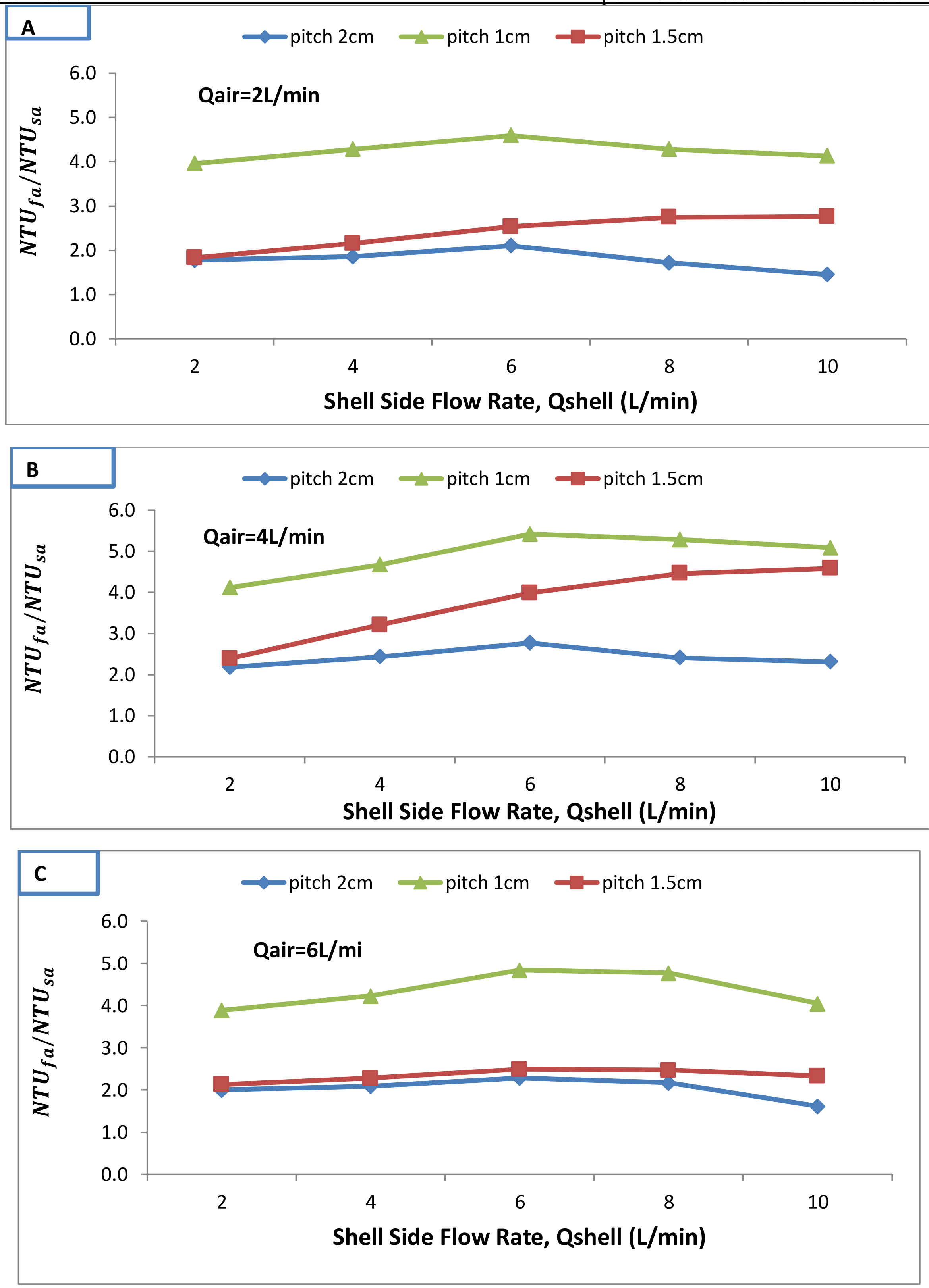


Fig.4.24. the variation in the enhancement ratio ($\boldsymbol{NTU_{fa}/NTU_{sa}}$) with the shell flow rate for three different pitch and constant coil (hot) flow rate ($Q_{coil} = 2\ l/min$)

Figure 4.25, show the relationship between the enhancement ratio of $\boldsymbol{Ntu_{fa}/Ntu_f}$ and the air injected flow rate for variation shell side flow rate and fixed coil side flow ($Qcoil$ = 1 L/min) . Similarly, two other figures (4.26 and 4.27) are presented for variation coil side flow rates ($Qcoil$ = 1.5 and 2 L/min).

The optimal value to enhance NTU of heat exchanger is when the air injected flow rate 2 L/min, $Qshell$ = 10 L/min and pitch=1cm are as seen in Figure 4.25

It can be concluded from figs. 4.25, 4.26, and 4.27 that the highest improvement ratio of the enhancement ratio of $\boldsymbol{Ntu_{fa}/Ntu_f}$ for size of pitch =1 cm and Qshell = 10 L/min is 81 % and the minimum value is 26 % at Qshell=2 L/min and size of pitch = 2 cm.

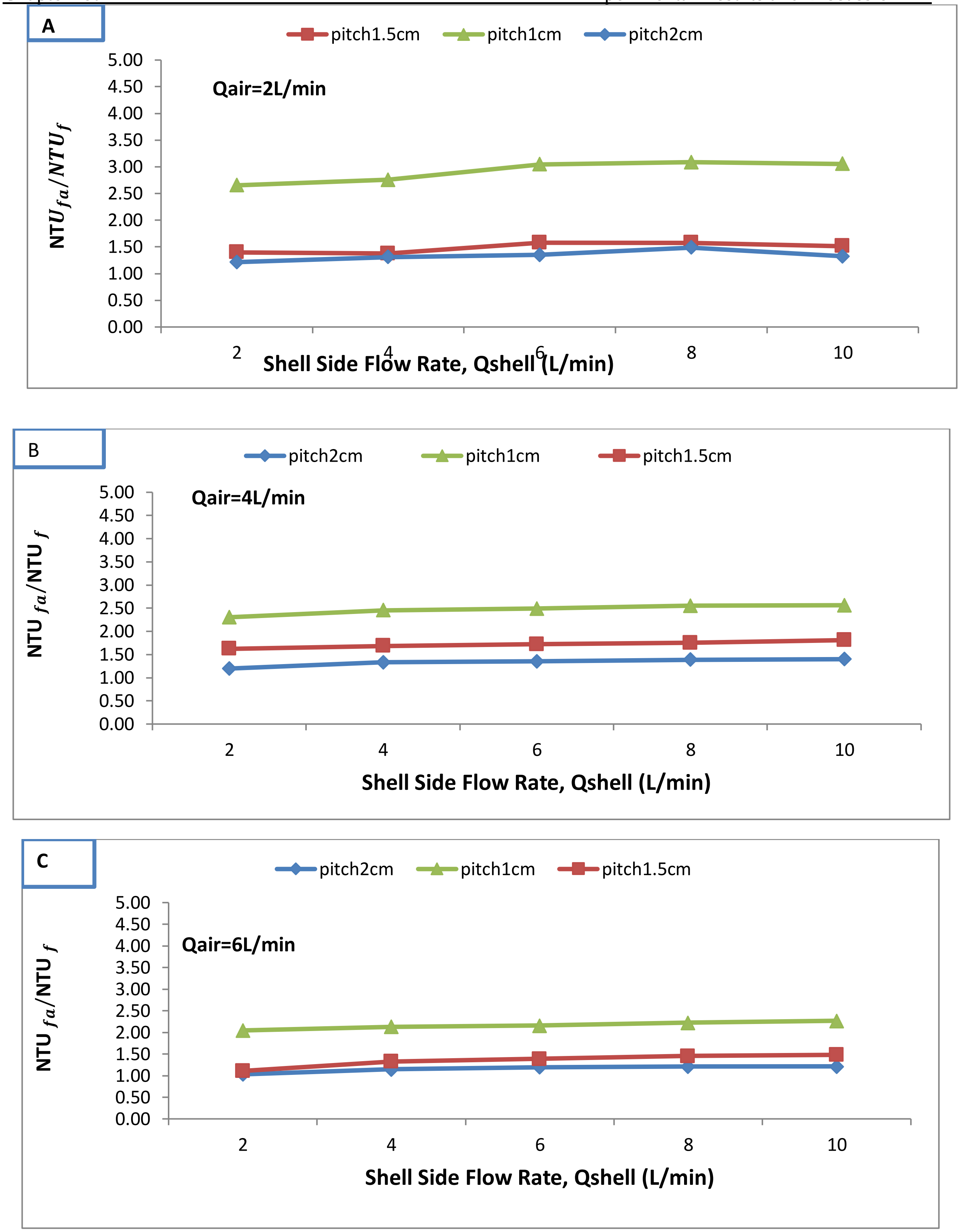


Fig.4.25. the variation in the enhancement ratio ($\boldsymbol{NTU_{fa}/NTU_f}$) with the flow rate of shell for three variation pitch and constant coil (hot) flow rate ($Q_{coil} = 1\ l/min$)

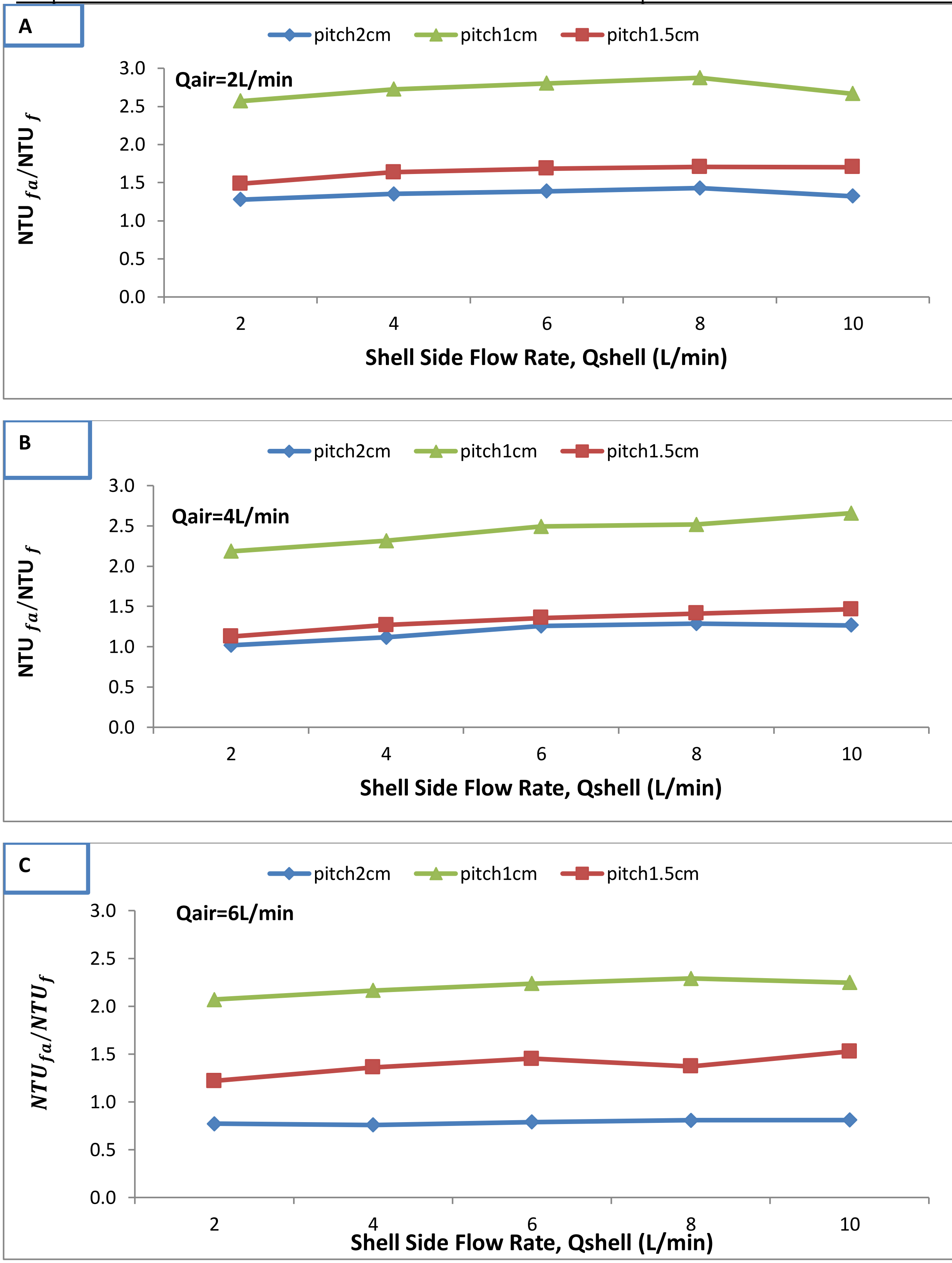


Fig.4.26. the variation in the enhancement ratio ($\boldsymbol{NTU_{fa}/NTU_f}$) with the flow rate of shell for three variation pitch and constant coil (hot) flow rate ($Q_{coil} = 1.5\ l/min$)

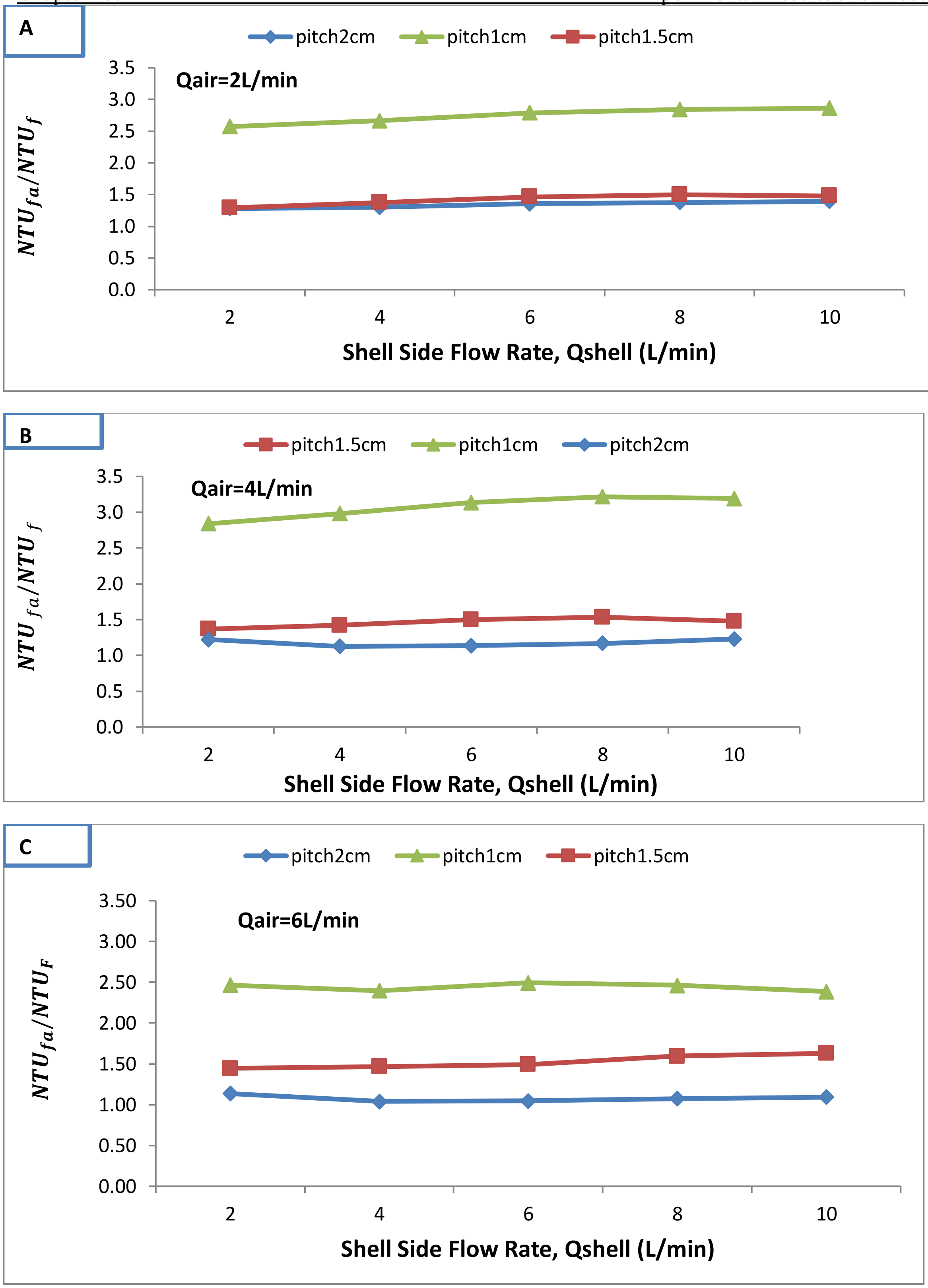


Fig.4.27. the variation in the enhancement ratio ($\boldsymbol{NTU_{fa}/NTU_f}$) with the flow rate of shell for three variation pitch and constant coil (hot) flow rate ($Q_{coil} = 2\ l/min$)

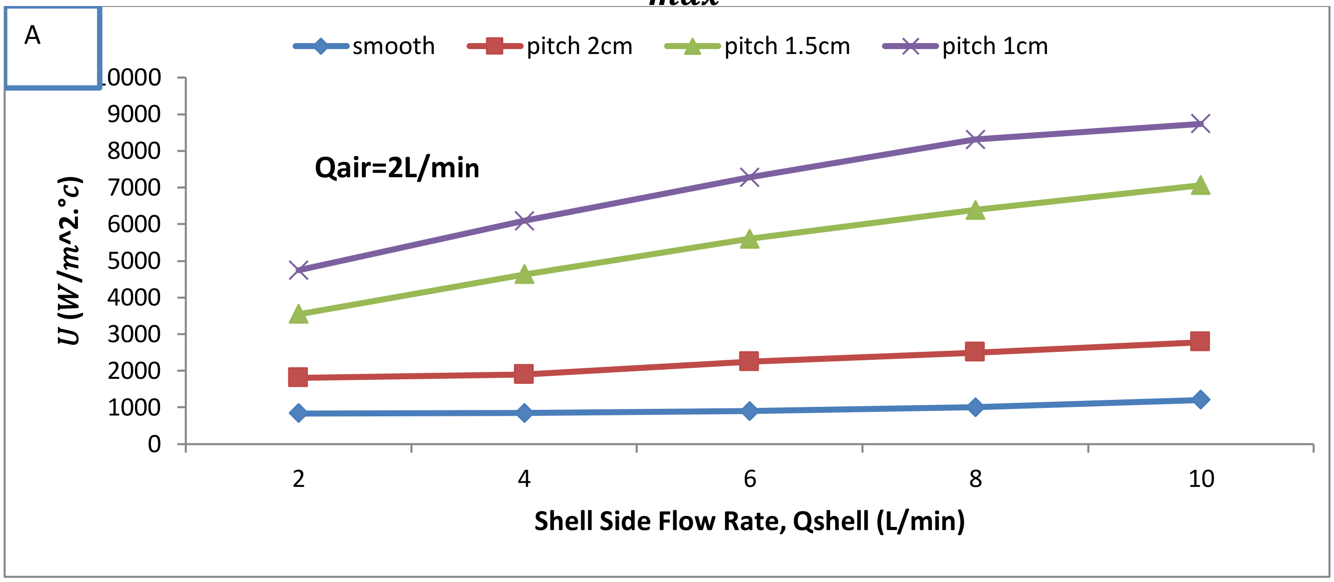
4.4 Comparison of The Overall Heat Transfer Coefficient In the Helical Coil (Smooth and Finned) Tube Heat Exchanger.

Figure 4.28 states the effect of fins and injection air bubbles on the overall heat transfer coefficient (U) at different shell-side flow rates (2,4,6,8 and 10LPM), constant hot water flow rate , stable air flow rate , and (ΔT =20 ^{o}C).

As mentioned previously, the injection of air into the water will generate bubbles inside the cold water. As a result of the difference in density between the water and the air, the bubbles will move vertically to the top. As a result of the movement of the bubbles, the turbulence will increase inside the shell. As a result, the thermal layers formed on the surface of the helical coil tube will be destroyed, which is one reason to increase the transfer of heat from hot water to cold water. The second main reason is the increase in cold water velocity due to injecting the bubbles into a shell, increasing the Reynolds number, subsequently increasing the heat transfer rate.

Similarly, when using spiral fins, the inner fins will cause an additional rotational movement (secondary flow) inside the hot water pipe. In contrast, the outer fins will act as barriers to break down the thermal layer formed on the surface of the helical tube, which will increase the rate of heat transfer between the two liquids

U_{max}



U_{min}

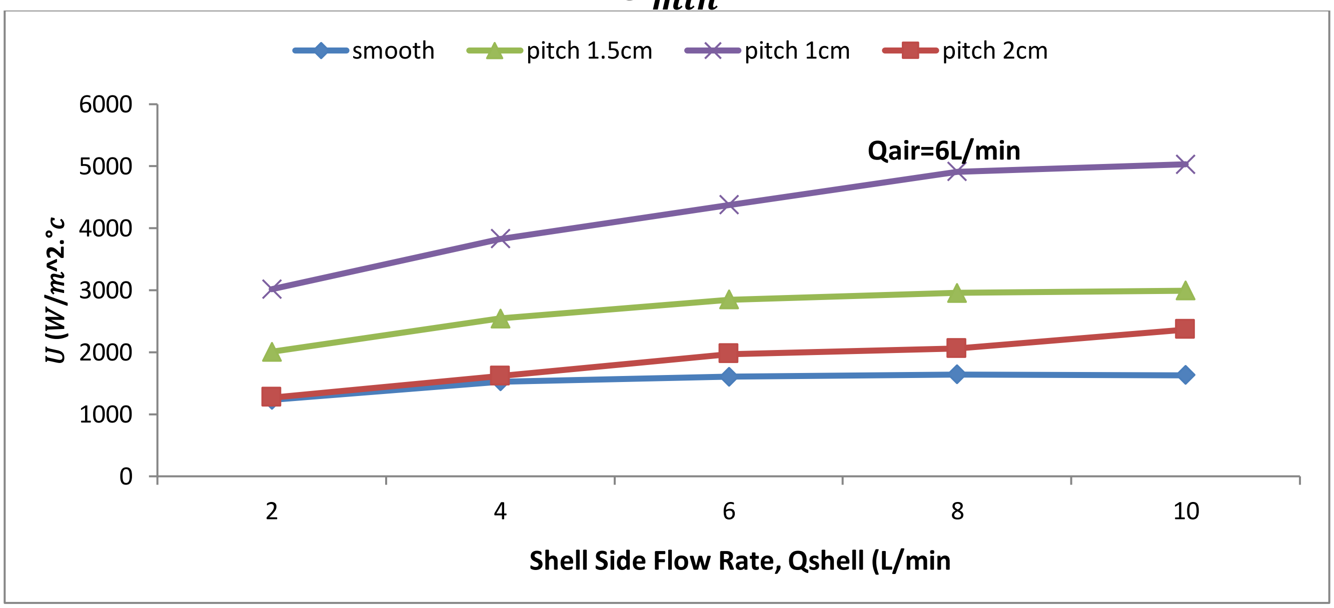
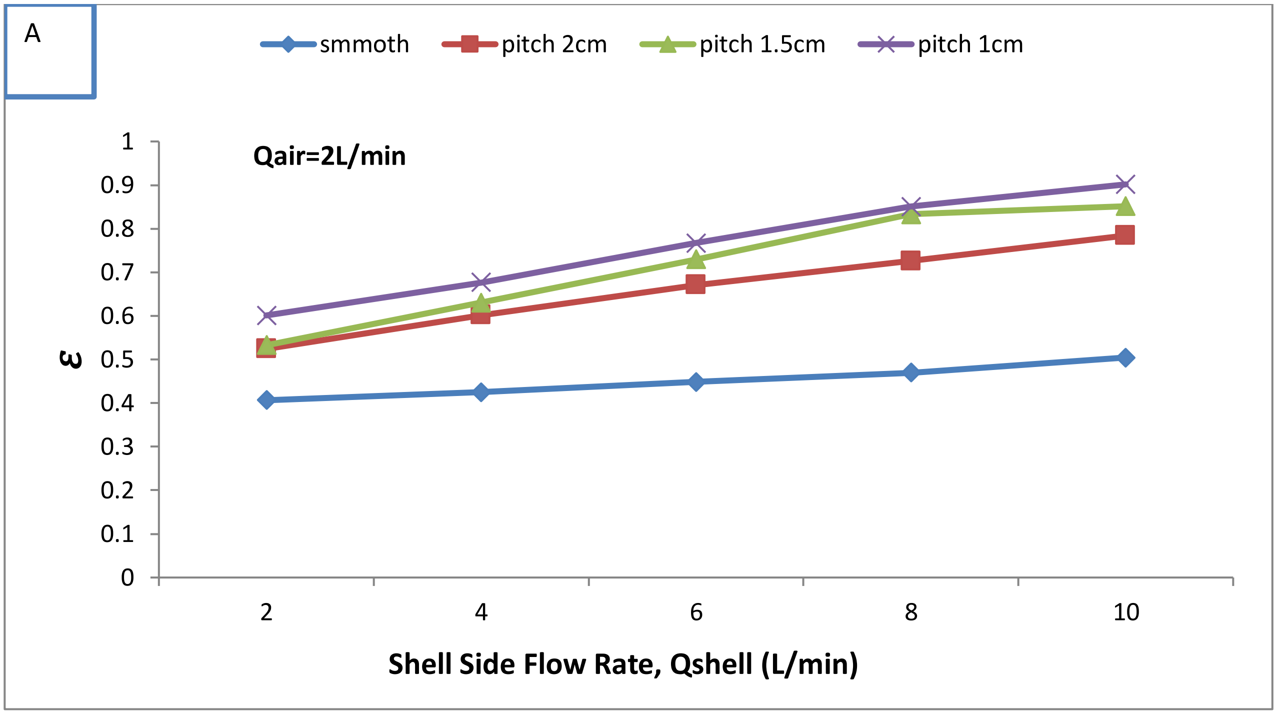


Fig.4.28. Comparison overall heat transfer coefficient in (smooth and finned) helical coil tube heat exchanger at ($\Delta T = 20\ ^{o}C$) for two cases, A) Qcoil=2L/min, B) Q coil=1L/min,

ε_{max}



ε_{min}

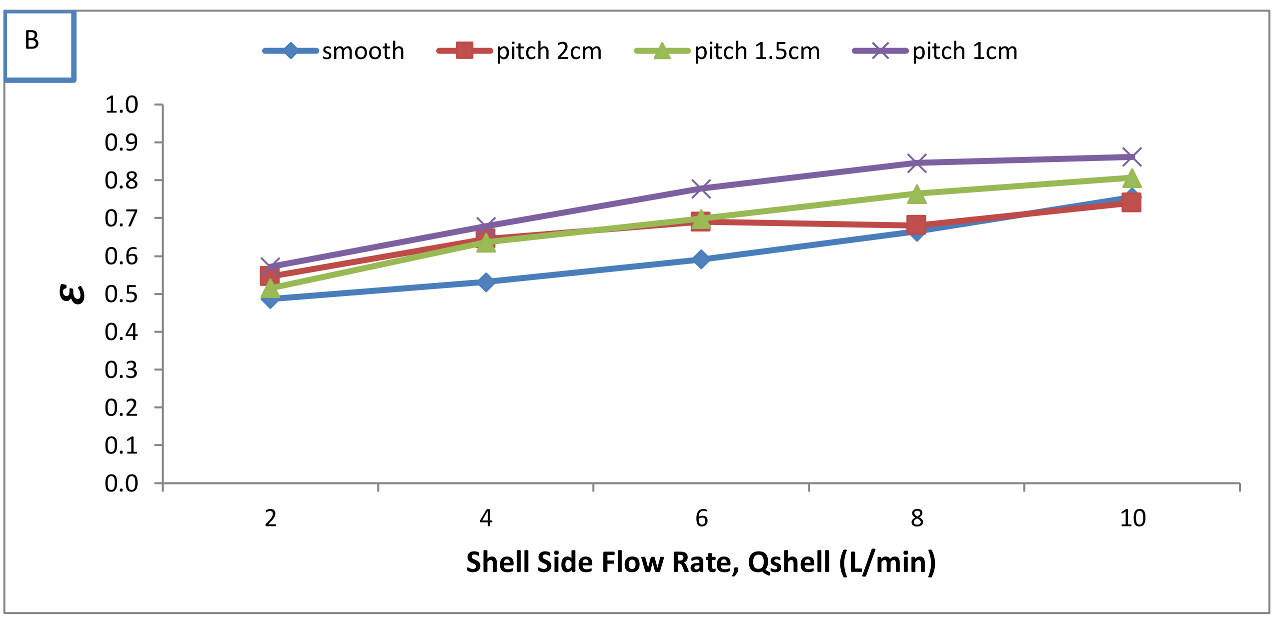
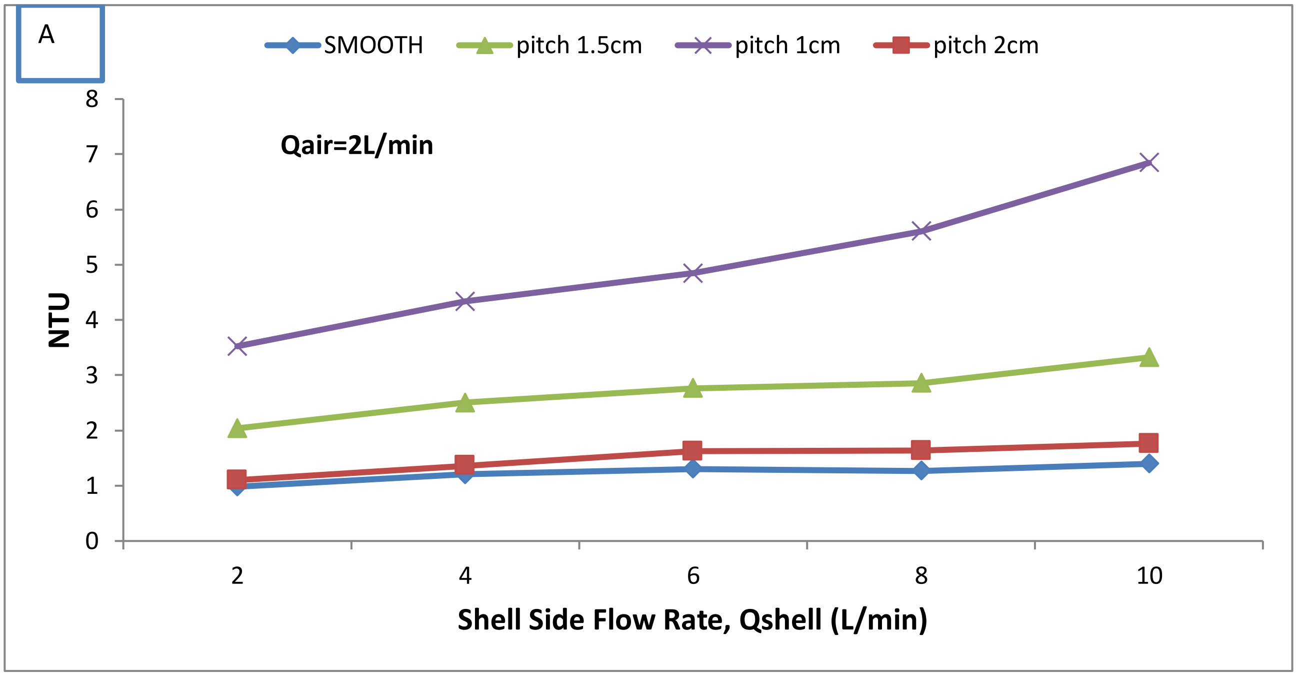


Fig.4.29. Comparison $\boldsymbol{\varepsilon}$in (smooth and finned) helical coil tube heat exchanger at $(\Delta T\ =\ 20\ ^{o}C)$ for two cases, A) Qcoil=2L/min, B) Q coil=1L/min,

NTU_{max}



NTU_{min}

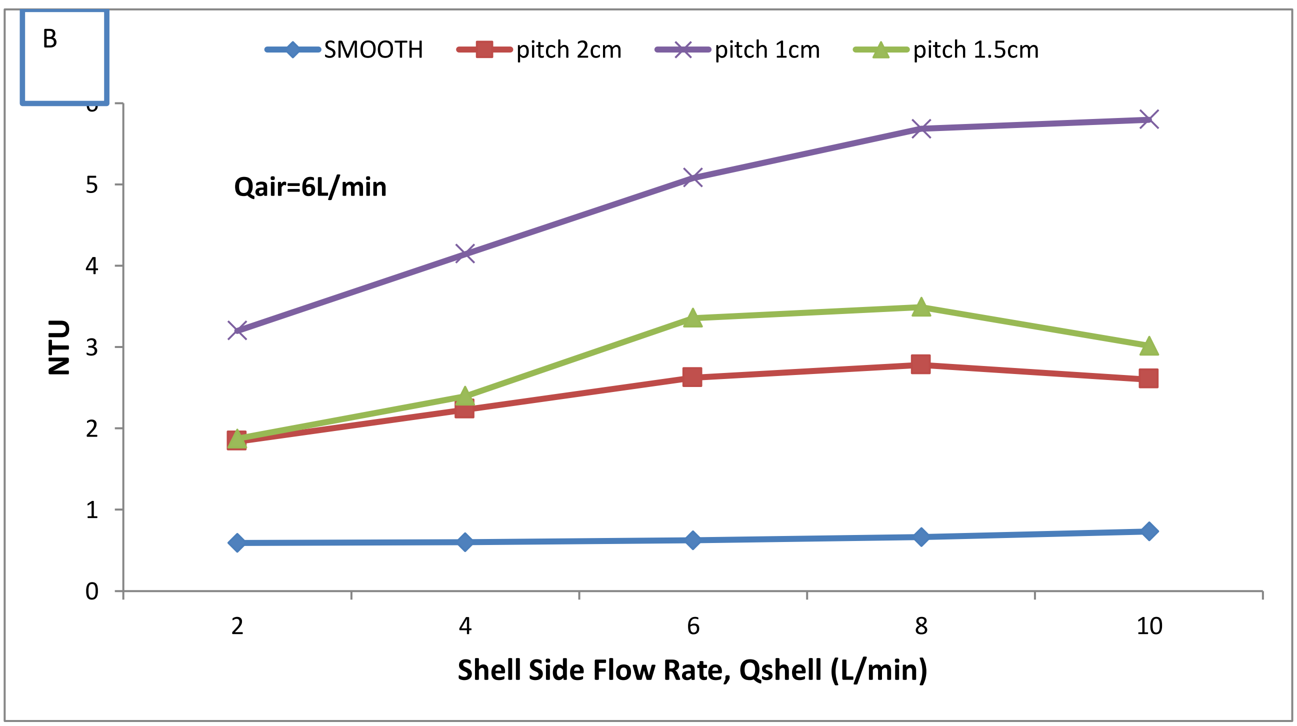


Fig.4.30. Comparison $\boldsymbol{NTU}$ in (smooth and finned) helical coil tube heat exchanger at $(\Delta T = 20\ ^{o}C)$ for two cases, A) Qcoil=2L/min, B) Q coil=1L/min,

4.5 Impact of Inlet Air Bubbles Temperature on Transfer of Heat Rate.

In the current work, the maximum mass of air flow rate was $1.96 * 10^{-4}\ kg/s$ (10 LPM) and the maximum air mass fraction was $5.673 * 10^{-3}$as obtained by equation 4.1 below. However, it is obvious that the mass of air flow rate is a very little compared with water mass flow rate (0.0333 kg/s). In addition to air has very poor thermal conductivity (only 0.02 W/m.K) and poor heat capacity (0.19 J/s.K) therefore, the heat transfer due to the injected air was ignored.

$$MF_{air} = \frac{\dot{m}_{air}}{\dot{m}_{air} + \dot{m}_{water}} = \frac{1.9*10^{-4}}{1.9*10^{-4}+0.0333} = 5.673 * 10^{-3} \qquad \ldots(4.1)$$

However, the contribution of the injected air on the transfer of heat process in the exchanger of heat is tested experimentally since the temperature of inlet and outlet air were measured during the experiments. The maximum temperature difference was found to be having only 0.4°C which could be as a conclusion of the adiabatic expanding of (air volume expand once sprays by the sparger).

Chapter Five

Conclusions & Recommendations

Conclusions and Recommendations for Future Work

Introduction.

The aim of this work is now to test the injection of small air bubbles into the air and the size of of pitch for finned tube of the direct heat exchanger and the helical tube heat exchanger in the laminar flow. ($316 \leq Re \leq 1523$). This study included an analysis of overall heat transfer coefficient (U), effectiveness (ε), number of heat transfer units (NTU)

5.1 Conclusions.

The experimental findings of the current work, presented in Chapter 4, demonstrated that the injection of air bubbles dramatically improves the thermal performance of a vertical shell and helically (smooth and finned) coiled tube heat exchanger. According to the obtained results, the following conclusions can be drawn.

1. Air bubbles injection into the shell side of the helically coiled tube heat exchanger results in a significant improvement in the overall heat transfer coefficient (U_{sa}) and effectiveness (ε_{sa}) for all cases under study. However, depending on Qa and Qs, the maximum enhancement ratio of (U_{fa}/U_{sa}) was up to 119 %, and ($\varepsilon_{fa}/\varepsilon_{sa}$a) was 122% compared with pure water. The aforesaid maximum enhancement ratios were achieved when Q_s = 10 L/min, Q_a = 2 L/min, and size of pitch =1cm, While the minimum enhancement ratio of (U_{fa}/U_sa) was 39%, and ($\varepsilon_{fa}/\varepsilon_{sa}$) was 18%.

2. From our experiments, we conclude that Qa = 2 LPM and Qs = 10 LPM are the best flow rates (U_{fa} /U_{f}) , ($ε_{fa}/ε_{f}$)and (NTU_{fa}/NTU_{f})to achieve higher levels under current conditions.
3. The value of (Usa, U_{fa}) increases by increasing the flow rate on the coil side due to the increase in thermal conductivity (water temperature) of the single heat treatment. On the other hand, the value ($ε_{fa}, ε_{f}$) decreases with increasing flow velocity on the recoil side due to increasing air pressure (water temperature) with treatment simultaneous thermal
4. The maximum enhancement ratio of (U_{fa}/U_{f}) , ($ε_{fa}/ε_{f}$) and Ntu_{fa}/Ntu_{f} is clearly obtained at Q_s = 10 L/min, Q_a = 2 L/min, and size of pitch =1cm is 83 % ,55% and 81% recepactlly and the minimum enhancement of (U_{fa}/U_{f}) , ($ε_{fa}/ε_{f}$) and Ntu_{fa}/Ntu_{f} is clearly obtained at Q_s = 2 L/min, Q_a = 6 L/ min, and size of pitch =2cm is 27 % ,11% and 26%
5. The maximum enhancement ratio of (U_{fa}/U_{f}) is clearly obtained at Q_s = 10 L/min, Q_a = 2 L/min, and size of pitch =1cm is 83 % , and the minimum enhancement of (U_{fa}/U_{f}) is clearly obtained at Q_s = 2 L/min, Q_a = 6 L/ min, and size of pitch =2cm is 27 % .
6. The maximum enhancement ratio of ($ε_{fa}/ε_{f}$) is clearly obtained at Q_s = 10 L/min, Q_a = 2 L/min, and size of pitch =1cm is 55% and the minimum enhancement of ($ε_{fa}/ε_{f}$) is clearly obtained at Q_s = 2 L/min, Q_a = 6 L/ min, and size of pitch =2cm is 11% .
7. The maximum enhancement ratio of Ntu_{fa}/Ntu_{f} is clearly obtained at Q_s = 10 L/min, Q_a = 2 L/min, and size of pitch =1cm is 81% and the minimum enhancement of Ntu_{fa}/Ntu_{f} is clearly obtained at Q_s = 2 L/min, Q_a = 6 L/ min, and size of pitch =2cm is 26% .

5.2 Recommendations

The present work deals with the experimental study of enhancing the thermal performance of a vertical shell and coiled (smooth and finned) tube heat exchanger by using the air bubbles injection technique. However, the following recommendations for future work should be useful:

1. It is advised to do a special numerical analysis using CFD.
2. Measure the timing and velocity of air bubble generation, which affects the hydrodynamics and heat transfer in the heat exchanger. A clear Perspex tube and a high-speed camera may be used to accomplish this.
3. Study the economic feasibility of our study using the method of merged the passive and active methods, represented by using the air bubble injection technique and helical coil finned (internally - externally) tube.
4. Study the effect of merged (passive and active method) technices through using injection air bubbles inside shell and helical coil finned (internally - externally) tube heat exchanger.
5. Study the effect of merged (passive and active method) techniques on nusselt number, through using injection air bubbles inside the shell and helical coil finned (internally - externally) tube heat exchang

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Indexes.

Appendix (A): Calibration curves

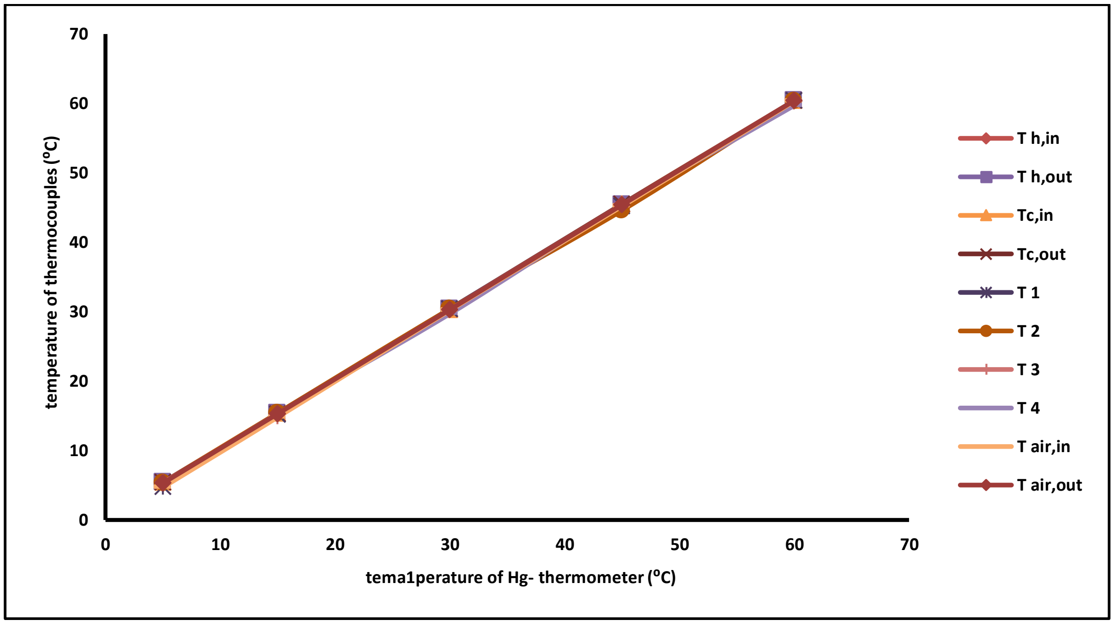

Fig. A-1: Calibration curve of thermocouples

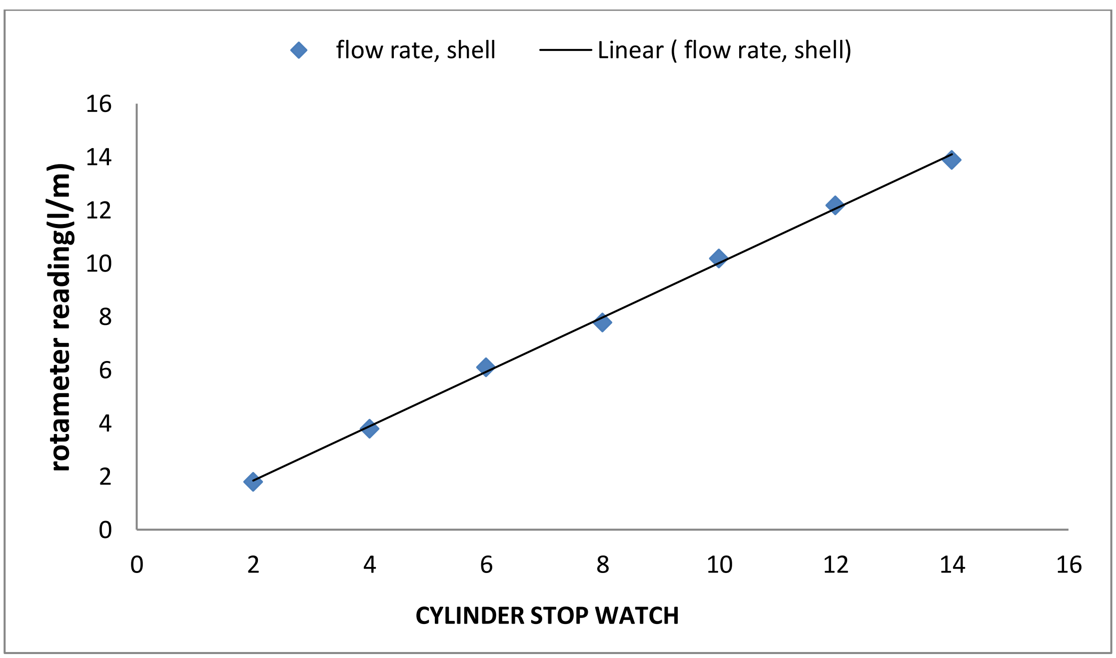

Fig. A-2: Shell side rotameter calibration curve

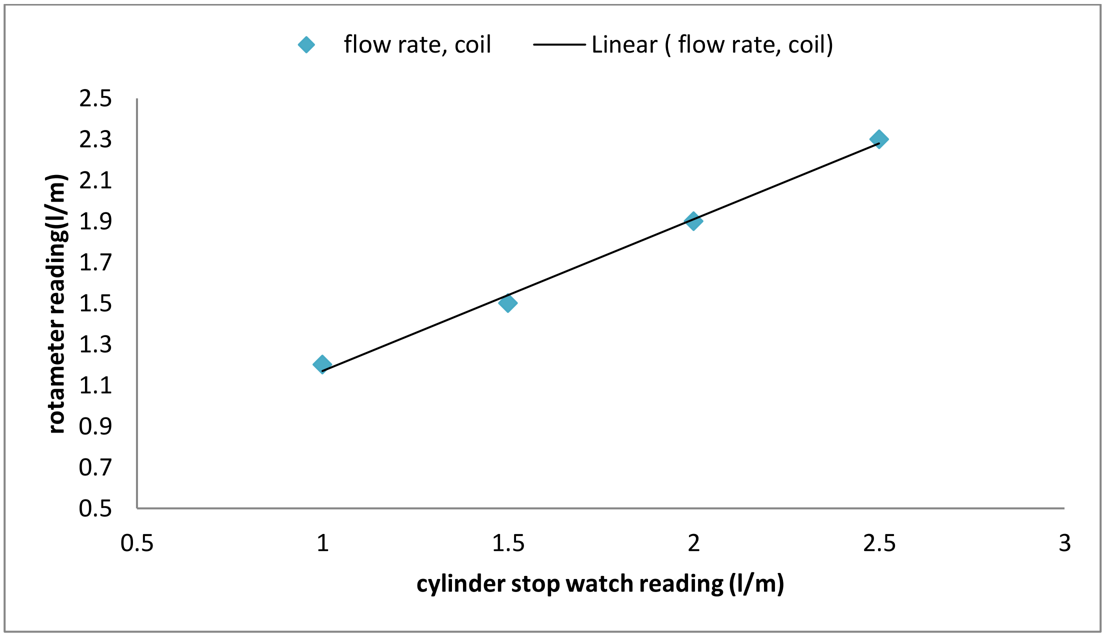

Fig. A-3: coil side rotameter calibration curve

Appendix (B): Uncertainty

The following equation can be used to calculate the uncertainty of experimental data **[64]**:

$$\mathbf{W_R} = \sqrt{\sum_0^{i}\left(\frac{\partial \mathbf{R}}{\partial \mathbf{X_i}}.\mathbf{W_{Xi}}\right)^2} \qquad \textbf{(B.1)}$$

And the result is given as:

$$\mathbf{R} = \mathbf{f}\,(\mathbf{X_1}, \mathbf{X_2}, \dots, \mathbf{X_n}) \qquad \textbf{(2.2)}$$

Where:

W_R: Uncertainty in the result

$X_1, X_2, \dots, X_n$: Independent variables, and

$W_1, W_2, \dots\dots, W_n$: Uncertainty in the corresponding variables.

The uncertainty of the performance parameters that have been experimentally obtained in the present study will calculate sequentially in the following sections using the general formula (B.1).

Overall Heat Transfer Coefficient (U)

The overall heat transfer coefficient (**U**) was calculated by Eq. (4.1) and rewritten below:

$$\mathbf{U} = \frac{\mathbf{q}}{\mathbf{A_s}\,.\,\Delta\mathbf{T_{LMTD}}} \qquad \textbf{(B.3)}$$

According to Eq. (C.1), the uncertainty of **U** is due to the errors of q**,** A_sAnd the temperatures ($T_{h,i}, T_{h,o}, T_{c,i}$and $T_{c,o}$). However, the following general expression can be made:

$$\mathbf{W_U} = \sqrt{\left(\frac{\partial \mathbf{U}}{\partial \mathbf{q}}\mathbf{W_q}\right)^2 + \left(\frac{\partial \mathbf{U}}{\partial \Delta\mathbf{T_{LMTD}}}\mathbf{W_{\Delta T_{LMTD}}}\right)^2 + \left(\frac{\partial \mathbf{U}}{\partial \mathbf{A_S}}\mathbf{W_{A_S}}\right)^2} \qquad \textbf{(B.4)}$$

This leads to:

$$\mathbf{W_U} = \sqrt{\left(\frac{\mathbf{W_q}}{\mathbf{A_s}.\Delta\mathbf{T_{LMTD}}}\right)^2 + \left(-\frac{\mathbf{q}.\mathbf{W_{\Delta T_{LMTD}}}}{\mathbf{A_s}.(\Delta\mathbf{T_{LMTD}})^2}\right)^2 + \left(-\frac{\mathbf{q}.\mathbf{W_{A_S}}}{\mathbf{A_S^2}.\Delta\mathbf{T_{LMTD}}}\right)^2} \qquad \textbf{(B.5)}$$

Where ΔT_{LMTD} is the log-mean temperature difference, and it can be found by:

$$\Delta\mathbf{T_{LMTD}} = \frac{(\mathbf{T_{h,i}} - \mathbf{T_{c,o}}) - (\mathbf{T_{h,o}} - \mathbf{T_{c,i}})}{\ln\left(\frac{\mathbf{T_{h,i}} - \mathbf{T_{c,o}}}{\mathbf{T_{h,o}} - \mathbf{T_{c,i}}}\right)} \qquad \textbf{(B.6)}$$

Hence, the general uncertainty of ΔT_{LMTD} is **[64]**

$$\mathbf{W_{\Delta T_{LMTD}}} = \frac{\sqrt{\begin{array}{c}\left(\left\{\left(\frac{\left[\left(T_{h,i}-T_{c,o}\right)-\left(T_{h,o}-T_{c,i}\right)\right]}{\left(T_{h,i}-T_{c,o}\right)}\right)-\ln\left(\frac{\left(T_{h,i}-T_{c,o}\right)}{\left(T_{h,o}-T_{c,i}\right)}\right)\right\}^{2}\left[W_{T_{h,i}}^{2}-W_{T_{c,o}}^{2}\right]\right)+\\ \left(\left\{\left(\frac{\left[\left(T_{h,i}-T_{c,o}\right)-\left(T_{h,o}-T_{c,i}\right)\right]}{\left(T_{h,o}-T_{c,i}\right)}\right)-\ln\left(\frac{\left(T_{h,i}-T_{c,o}\right)}{\left(T_{h,o}-T_{c,i}\right)}\right)\right\}^{2}\left[W_{T_{h,o}}^{2}-W_{T_{c,i}}^{2}\right]\right)\end{array}}}{\left(\ln\left[\frac{\left(T_{h,i}-T_{c,o}\right)}{\left(T_{h,o}-T_{c,i}\right)}\right]\right)^{2}} \qquad \textbf{(B.7)}$$

Now, the second factor that affects the uncertainty of U is the heat transfer rate q which can be found by:

$$\mathbf{q = \dot{m}_h\, Cp_h\, \left(T_{h,i} - T_{h,o}\right)} \qquad \textbf{(B.8)}$$

Similarly, the uncertainty of q can be calculated as:

$$\mathbf{W_q} = \sqrt{\left(\frac{\partial q}{\partial m_h} W_{m_h}\right)^2 + \left(\frac{\partial q}{\partial Cp_h} W_{Cp_h}\right)^2 + \left(\frac{\partial q}{\partial T_{h,i}} W_{T_{h,i}}\right)^2 + \left(\frac{\partial q}{\partial T_{h,o}} W_{T_{h,o}}\right)^2} \qquad \textbf{(B.9)}$$

Using Eq. (B.8) results in:

$$\mathbf{W_q} = \sqrt{\begin{array}{c}\left(Cp_h.\left[T_{h,i}-T_{h,o}\right].W_{m_h}\right)^2 + \left(m_h.\left[T_{h,i}-T_{h,o}\right].W_{Cp_h}\right)^2 + \left(m_h.Cp_h.W_{T_{h,i}}\right)^2\\ +\left(-\,m_h.Cp_h W_{T_{h,o}}\right)^2\end{array}} \qquad \textbf{(B.10)}$$

Multiply equation (C.10) by $\left[\frac{m_h}{m_h}\right]\left[\frac{Cp_h}{Cp_h}\right]\left[\frac{T_{h,i}-T_{h,o}}{T_{h,i}-T_{h,o}}\right]$**,** results in:

$$\mathbf{W_q} = \sqrt{\begin{array}{c}\left(\frac{m_h.Cp_h.\left[T_{h,i}-T_{h,o}\right].W_{m_h}}{m_h}\right)^2 + \left(\frac{m_h.\left[T_{h,i}-T_{h,o}\right].W_{Cp_h}}{Cp_h}\right)^2 + \left(\frac{m_h.Cp_h.\left(T_{h,i}-T_{h.o}\right).W_{T_{h,i}}}{T_{h,i}-T_{c,i}}\right)^2\\ +\left(\frac{-\,m_h.Cp_h.\left(T_{h,i}-T_{h,o}\right).W_{T_{c,i}}}{T_{h,i}-T_{c,i}}\right)^2\end{array}} \qquad \textbf{(B.11)}$$

where: $\mathbf{q = \dot{m}_h\, Cp_h\, \left(T_{h,i} - T_{h,o}\right)}$

Eq. (B.11) becomes:

$$\therefore \mathbf{W_q = q}\sqrt{\left[\frac{W_{m_h}}{m_h}\right]^2 + \left[\frac{W_{Cp_h}}{Cp_h}\right]^2 + \left[\frac{W_{T_{h,i}}}{T_{h,i}-T_{h,o}}\right]^2 + \left[-\frac{W_{T_{h,o}}}{T_{h,i}-T_{h,o}}\right]^2} \qquad \textbf{(B.12)}$$

The uncertainty of A_S and Cph can be ignored due to their minor effect **[64]**.

Now the uncertainty of U can be obtained by Substitute Eq. B.7 and B.12 into Eq. B.5.

Effectiveness (ε)

The general equation that can be used to calculate the heat exchanger effectiveness is:

$$\boldsymbol{\varepsilon = \frac{q}{Q_{.max.}}} \qquad \textbf{(B.13)}$$

Using Eq. B. 1 above, the general uncertainty formula of ε is:

$$\mathbf{W_{\varepsilon}} = \sqrt{\left(\frac{\partial\varepsilon}{\partial q}\mathbf{W_q}\right)^2 + \left(\frac{\partial\varepsilon}{\partial\, Q_{.max.}}\mathbf{W_{Q.max}}\right)^2} \qquad \textbf{(B.14)}$$

Using Eq. B.13, yields:

$$W_{\varepsilon} = \sqrt{\left(\frac{W_q}{Q_{.max.}}\right)^2 + \left(-\frac{q.W_{Q.max}}{(Q_{.max.})^2}\right)^2} \qquad \text{(B.15)}$$

Uncertainty of $W_{Q_{.max.}}$ can be similarly obtained:

$$\mathbf{W_{Q_{.max.}}} = \sqrt{\left(\frac{\partial Q_{.max}}{\partial m_h}W_{m_h}\right)^2 + \left(\frac{\partial Q_{.max}}{\partial Cp_h}W_{Cp_h}\right)^2 + \left(\frac{\partial Q_{.max}}{\partial T_{h,i}}W_{T_{h,i}}\right)^2 + \left(\frac{\partial Q_{.max}}{\partial T_{c,i}}W_{T_{c,i}}\right)^2} \qquad \textbf{(B.16)}$$

where

$$\mathbf{Q_{,max.} = C_{min.}\ (T_{h,i} - T_{c,i})} \qquad \textbf{(B. 17)}$$

and $C_{min} = m_h * Cp_h$

Using Eq. B. 16 above, yield:

$$\mathbf{W_{Q_{.max.}}} = \sqrt{\begin{array}{c}(\mathbf{Cp_h.[T_{h,i} - T_{c,i}].W_{m_h}})^2 + (\mathbf{m_h.[T_{h,i} - T_{c,i}].W_{Cp_h}})^2 + (\mathbf{m_h.Cp_h.W_{T_{h,i}}})^2 \\ +(-\,\mathbf{m_h.Cp_h W_{T_{c,i}}})^2\end{array}} \qquad \textbf{(B.18)}$$

Multiply Eq. C.18 by $\left[\frac{m_h}{m_h}\right]\left[\frac{Cp_h}{Cp_h}\right]\left[\frac{T_{h,i}-T_{c,i}}{T_{h,i}-T_{c,i}}\right]$**,** results**:**

$$\mathbf{W_{Q_{.max.}}} = \sqrt{\begin{array}{c}\left(\frac{m_h.Cp_h.[T_{h,i}-T_{c,i}].W_{m_h}}{m_h}\right)^2 + \left(\frac{m_h.[T_{h,i}-T_{c,i}].W_{Cp_h}}{Cp_h}\right)^2 + \left(\frac{m_h.Cp_h.(\,T_{h,i}-T_{c,i}).W_{T_{h,i}}}{T_{h,i}-T_{c,i}}\right)^2 \\ + \left(\frac{-\,m_h.Cp_h.(T_{h,i}-T_{c,i}).W_{T_{c,i}}}{T_{h,i}-T_{c,i}}\right)^2\end{array}} \qquad \textbf{(B.19)}$$

But:

$$\mathbf{Q_{max.} = m_h.Cp_h.(T_{h,i} - T_{c,i})}$$

Hence, Eq. (B.19), becomes:

$$\therefore \mathbf{W_{Q.max.}} = \mathbf{Q_{.}max}\sqrt{\left[\frac{W_{m_h}}{m_h}\right]^2 + \left[\frac{W_{Cp_h}}{Cp_h}\right]^2 + \left[\frac{W_{T_{h,i}}}{T_{h,i}-T_{c,i}}\right]^2 + \left[-\frac{W_{T_{c,i}}}{T_{h,i}-T_{c,i}}\right]^2} \qquad \textbf{(B.20)}$$

Finally, the uncertainty of ε can be obtained by substituting Eq. B.12 and Eq. B. 20 into Eq. B.15.

NTU

The number of heat transfer units (NTU) is obtained by:

$$NTU = \frac{U.A_s}{C_{min}} \tag{B.21}$$

Using Eq. (C.5), the uncertainty of NTU can be developed as:

$$W_{NTU} = \sqrt{\left(\frac{\partial NTU}{\partial U} W_U\right)^2 + \left(\frac{\partial NTU}{\partial A_s} W_{A_s}\right)^2 + \left(\frac{\partial NTU}{\partial m_h} W_{m_h}\right)^2 + \left(\frac{\partial NTU}{\partial cp_h} W_{cp_h}\right)^2} \tag{B.22}$$

Which leads to:

$$W_{NTU} = \sqrt{\left[\frac{A_S.W_U}{m_h.Cp_h}\right]^2 + \left[\frac{U.W_{A_S}}{m_h.Cp_h}\right]^2 + \left[-\frac{U.A_S.W_{m_h}}{m_h^2.Cp_h}\right]^2 + \left[-\frac{U.A_S.W_{cp_h}}{m_h.Cp_h^2}\right]^2} \tag{B.23}$$

Similarly, the uncertainty expression of NTU can be developed by substituting the final expression of EqB.5 above into Eq. B.23.

Appendix (C): List of Publications

Air bubble injection technique for enhancing heat transfer in a coiled tube heat exchanger: an experimental study.

University of AlKafeel

Publication Acceptance Letter

Manuscript ID: ENG-460

Authors: Muna Talib1, Ahmed Ali2, Hameed Mahood3, Ali Baqir4

Email of Correspoinding Author: munaalitalib@gmail.com

Affiliation: Al-Furat Al-Awsat Technical University1,2,4, University of Surrey3

Dear (author) we are plased to inform you that, after peer-review process, your manuscript entitled

Experimental Study on the Heat Transfer Enhancement in Externally Finned Helical and Shell Tube Heat Exchanger

has been accepted for presentation in the

3rd INTERNATIONAL SCIENTIFIC CONFERENCE OF ALKAFEEL UNIVERSITY(ISCKU 2021)

which was held at the University of Alkafeel on the 22nd - 23rd of March 2021

Your paper will be published by **American Institute of Physics** (ISSN: 1551-7616) after pass all the journal requirements.

With Best Regards

AIP Publishing Scopus

Prof. Dr. N. Al-Dahan

Rector of Alkafeel University

الخلاصة

يتطلب نقل الطاقة الحرارية بكفاءة استخدام مبادل حراري قادر على إنتاج الطاقة الحرارية الكاملة لمصدر الطاقة بأقل تكلفة ووقت ممكنين. المبادلات الحرارية ذات النمط السطحي التقليدي لها عيوب كبيرة ، مثل المقاومة العالية لانتقال الحرارة للسطح ، مشكلة القاذورات ، وزيادة التكلفة. لذلك ، فإن أي محاولة لتحسين خصائص نقل الحرارة لمبادل حراري من النوع السطحي تعزز أدائه الحراري. يعد حقن الفقاعة الهوائية وأنبوب الملف الحلزوني الزعانف من أكثر تقنيات التعزيز الواعدة التي تم اقتراحها مؤخرًا في صورة منفصلة.

لذلك ، في هذه الدراسة ، تم فحص تأثير تقنيات التحسين المدمجة ، حقن الهواء وأنبوب الملف الحلزوني (خارجيًا) على الأداء الحراري لمبادل حراري ملفوف بتيار معاكس عموديًا تجريبيًا. تم حقن الهواء في جانب الغلاف للمبادل الحراري على شكل فقاعات هواء يبلغ متوسط قطرها حوالي 100 عبر رشاش مسامي (طريقة حقن جديدة). علاوة على ذلك ، أجريت الدراسة لتحسين المعلمات التشغيلية لجزء الفراغ (معدلات التدفق الحجمي للهواء والماء) لجانب القشرة تحت التدفق الصفحي (Re ≤1223 ≥ 316). تم إجراء التجارب باستخدام Qs = 2، 4، 6، 8 و LPM 10، Qa = 0، 2، 4 ، و LPM 6 ، وكان فرق درجة حرارة مدخل السائل العامل Δ20 T درجة مئوية. تأثير تغيير معاملات التشغيل على معامل انتقال الحرارة الكلي U ، والفعالية ε ، وعدد وحدات نقل الحرارة (NTU) مع وبدون حقن فقاعات الهواء ومقارنتها. إلى جانب ذلك ، درسنا حجم تأثير الملعب للأنبوب المزعنف على الأداء الحراري للمبادل الحراري.

أظهرت النتائج التجريبية تحسنًا ملحوظًا في معامل انتقال الحرارة الكلي وفعالية المبادل الحراري نتيجة حقن فقاعات الهواء ،

كانت نسبة التحسين القصوى لـ (U_fa / U_Sa) تصل إلى 119٪ ، كانت ⟦(ε⟧_fa / ε_sa) 122٪ و Ntu_fa / Ntu_Sa كانت 116٪ مقارنة بالمياه النقية.

تم تحقيق نسب التحسين عندما Qs = 10LPM، Qa = 2 LPM، Qh = 2LPM ، حجم الملعب للزعنفة = 1 سم و Δ20 = T درجة مئوية. بينما كان الحد الأدنى لنسبة التحسين لـ (U_fa / U_Sa) يصل إلى 39٪ ، ⟦(ε⟧_fa / ε_sa) كان 18٪ و Ntu_fa / Ntu_Sa كان 36٪ الحد الأدنى للنسبة المقدرة حدث عند Qs = 2LPM، Qa = 6 LPM، Qh = 1 LPM ، حجم الملعب للزعنفة = 2 سم و T = $20^{o}C$ علاوة على ذلك ، استنتجت تجاربنا أن Qa = 2 LPM و Qs = 6 LPM هي معدلات التدفق

المثلى في ظل الظروف التجريبية الحالية للحصول على أعلى نسبة تحسين ((U_fa / U_Sa، ⟦(ε⟧ _fa ⟧
(/ ε_sa و Ntu_fa / Ntu_Sa.

من ناحية أخرى ، بالنسبة للأنبوب المزعنف ، كان الحد الأقصى لنسبة التحسين لـ (U_fa / U_f) يصل إلى 83٪ ، ⟦(ε⟧ _fa / ε_f)⟧ كان 55٪ و Ntu_fa / Ntu_f كان 81٪ تم الحصول عليه بوضوح عند Qs =10 LPM ، Qa = 2 LPM ، وحجم الملعب للزعنفة = 1 سم بينما كان الحد الأدنى من التحسين (U_fa / U_ (f)) 27٪ ، ε_f / ⟦(ε⟧ _fa)⟧ كان 11٪ و Ntu_fa / Ntu_f كان تم الحصول على 26 ٪ بوضوح عند Qs = 2 LPM، Qa = 6 LPM ، وحجم الملعب للزعنفة = 2 سم و T = 20 درجة مئوية

علاوة على ذلك ، معدلات التدفق المثلى في ظل الظروف التجريبية الحالية للحصول على أعلى نسبة تحسين ((U_fa / U_ (f) و ((ε_fa / ε_ (f) كان (Qs = 10 LPM) و (Qa = 2 LPM).

. بالإضافة إلى ذلك ، تزداد القيمة الإجمالية مع زيادة معدل التدفق الجانبي للملف بسبب زيادة السعة الحرارية (السائل الساخن) لنفس وقت المعالجة الحرارية.

جمهورية العراق
وزارة التعليم العالي والبحث العلمي
جامعة الفرات الاوسط التقنية
الكلية التقنية الهندسية- نجف

دراسة تجريبية لتحسين نقل الحرارة باستخدام حقن فقاعات الهواء داخل مبادل حراري من نوع الصدفة وانبوب مزعنف خارجيا

رسالة مقدمة الى
قسم هندسة تقنيات ميكانيك القوى
كجزء من متطلبات نيل درجة الماجستير في تكنلوجيا الحراريات
في هندسة تقنيات ميكانيك القوى

تقدم بها
منى علي طالب
بكلوريوس هندسة تقنيات السيارات

اشراف

الاستاذ الدكتور	**الدكتور**
علي شاكر باقر	**احمد حمودي**

ذي القعدة / 1442