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الملخص العربي

نظرا لحاجة كثير من التطبقات الهندسية الى مبادلات حرارية فعالة ومدمجة قدر الامكان وأقل تكلفة ، تستخدم أشكال خاصة لتلك الاسطح لتولد مسارات متعرجة للهواء او الغاز كأحد الطرق الاقتصادية لزيادة معدل الانتقال الحرارى. إضافة الى زيادة مساحة سطح التبادل الحرارى لنفس الحجم فإن تلك المسارات تغيرمن سلوك المائع المار خلالها حيث يتم تكسير الطبقة الحرارية الجدارية بسبب تولد دوامات وانفصال خطوط السريان مؤدية الى زيادة معامل انتقال الحرارة بالحمل مع زيادة فقد الضغط. لذا فإن دراسة خصائص السريان خلال تلك المسارات جذب انتباه العديد من الباحثين حديثا الذين قدموا دراسات نظرية وعملية كثيرة لبيان مدى فعالية تلك المجارى المتعددة الأشكال وما زالت فى حاجة الى المزيد من الدراسة.

وحيث أن خصائص سريان المائع خلال تلك المجارى يتوقف على العديد من المتغيرات الهندسية لها من حيث زاوية وشكل سطح التعرج، زاوية الطوروخطوة التموج، طول المسار وأيضا ارتفاع المجرى إضافة الى طبيعة السريان ، فإن النتائج التى تم الحصول عليها والمتعلقة بإنتقال الحرارة وفقد الضغط تعتمد على الشكل الهندسى لمقطع المجرى إضافة الى وجود بعض التباين فى النتائج.

في هذا البحث تم إجراء دراسة معملية للسريان فى ممرات متعرجة ذات نسبة تعرج ثابتة ($\gamma=0.2$)، أسطحها عند درجة حرارة ثابتة لبيان تأثيرتغير رقم رينولدز، المسافة البينية وزاوية الطور للمجرى على معامل انتقال االحرارة بالحمل المتوسط وكذلك فقد الضغط وبيان ذلك على تغير معامل الجودة (*j/f*) . أثناء التجارب تم تغيير رقم رينولدز (*Re=3220 up to 9420*) ، المسافة البينية (*spacing, S=4, 6,8, 10-mm*) و زاوية الطور (*phase-shift,* $\varnothing=0^{\circ}, 90^{\circ}, 180^{\circ}$).

تم عرض النتائج كعلاقة بين رقم نوسلت (Nu)، فقد الضغط ΔP_{loss} ، معامل الاحتكاك (f) ، مع رقم رينولدز(Re). وقد أظهرت النتائج زيادة واضحة في معامل انتقال الحرارة المتوسط مع زيادة في فقد الضغط خلال الممرات المتعرجة عنها فى الممرات المتوازية على مدى التغير فى رقم

رينولدز. تراوحت الزيادة في رقم نوسلت المتوسط بين 190 -350% وفى فقد الضغط بين 150 – 180% على التوالى مقارنة بتلك فى الممرات المتوازية تبعا لقيمة المسافة البينية والطور، وأن تأثير تغير المسافة البينية على النتائج أكثر من تأثير تغيرالطور وخاصة عند قيم رقم رينولدز المرتفعة . كذلك تم حساب معامل الجودة j/f لبيان أفضل مدى لكل من $(\varepsilon=S/2A)$ وكذلك زاوية الطور لثلاث قيم لرقم رينولدز.

الرسالة مكتوبة باللغة الإنجليزية وتحتوي على خمسة أبواب وقائمة بالمراجع وتحتوي على 36 مرجعا وثلاثة ملاحق بالإضافة إلى ملخص باللغة العربية موضحةً كالآتي:

الباب الأول:

يحتوي هذا الباب على مقدمة عن المبادلات الحرارية وانواعها، وطرق زيادة فعالية اسطح التبادل الحراري.

الباب الثاني:

يحتوي هذا الباب على عرض ملخصات للأبحاث السابقة و المتعلقة بموضوع البحث وتحديد لموضوع الدراسة المعملية.

الباب الثالث

يتضمن هذا الباب شرح لمكونات جهاز التجارب المعملية الذي تم تصميمه وتنفيذه خصيصاً لموضوع البحث وطريقة عمله وأجهزة القياس المستخدمة وخطوات إجراء التجارب المعملية مع عرض لطريقة تحليل القياسات المعملية للحصول على النتائج في الصورة المطلوبة. (وقد تمت التجارب بمعمل التبريد والتكييف وانتقال الحرارة بقسم هندسة القوى الميكانيكية بكلية الهندسة جامعة المنصورة).

الباب الرابع:

يحتوي هذا الباب على عرض للنتائج المعملية في صورة منحنيات وعلاقات تم تحليلها وعمل مقارنة بين تلك النتائج ونتائج الأبحاث المتوفرة في نطاق موضوع البحث.

الباب الخامس:

يحتوي هذا الباب على عرض لاهم نتائج البحث التي تم الحصول عليها والتوصيات المقترحة للدراسات المستقبلية.

وقد اشتملت الرسالة على ثلاثة ملاحق موضحة كالآتي :

الملحق الأول:

تحليل الخطأ للنتائج المعملية.

الملحق الثاني:

يشمل بعض القياسات المعملية.

الملحق الثالث:

يوضح طريقة تحليل النتائج المعملية.

وتقع الرسالة في 115 صفحة بالإضافة إلى ثلاثة ملاحق تقع في 17 صفحة

ABSTRACT

Extended surfaces are widely used in compact heat exchangers to increase the rate of convective heat transfer of air or gas side flows. Besides increasing the effective heat transfer surface area, geometrically modified wavy passages are applied as turbulence promoters; where breaking and destabilizing of thermal boundary layer are occurred. Using corrugated channels have such properties is one of the most applicable methods to enhance the thermal performance and provide higher compactness of heat exchangers.

Several parameters affect corrugated channel heat and fluid flow characteristics. Among those; the geometrical parameters of the channel and the Reynolds number.

The present study constitutes a comprehensive experimental investigation of heat transfer characteristics and pressure drop in corrugated channels. Measurements on parallel plate channel flow were also performed for the sake of comparison with the results of corrugated channel flows.

During the course of the work, a constant property channel corrugation ratio, ($\gamma=2a/L=0.2$) is used. Three parameters were systematically varied, including the Reynolds number, phase shift, and the channel spacing of the corrugated channel modules.

The investigation encompassed pressure drops, friction factors as well as the heat transfer coefficients. To assess the gained benefits of corrugated channels, a factor of area goodness is predicted for all modules, on which the proper channel geometry and flow region can be pointed out.

The obtained results showed a significant heat transfer enhancement and pressure drop penalty with the corrugated channel flow. The percentage increase in the average Nusselt number ranged between 190 and 350, with compared to those for parallel plate channel flow, depending upon Reynolds number and the corrugated channel geometry with respect to its spacing and phase shift. It is also found that the area goodness factor was better for corrugated flow channels with geometrical parameters of $2 \leq \varepsilon \leq 3$, and $0^\circ \leq \emptyset \leq 90^\circ$. The highest area goodness factor was achieved with low Reynolds number flow regime, $Re \approx 3230$.

Results are compared with those available in the literature, and discrepancies are discussed. These results provide a detailed understanding of the air flow forced convection behavior in corrugated channels in the Reynolds number regime; $3230 \leq Re \leq 9420$.

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Mansoura University
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Heat Transfer and Pressure Drop in Corrugated Passages

Thesis

Submitted in Partial Fulfillment for the Degree of Master of Science in Mechanical Power Engineering

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Nomenclature

a	amplitude, mm, for air
A	Surface area, orifice cross sectional area, m^2
C_d	Coefficient of discharge
C_p	Specific heat, J / kg .K
D_h	Hydraulic diameter, m
f	Friction factor
g	Gravity acceleration, m/s^2
W	Channel width, mm
h	Heat transfer coefficient W /m^2 . K
h_o	Manometer height difference across the orifice, m
h_L	Manometer height difference across the test section, m
k	Thermal conductivity, W /m . K
j	Colburn factor; $(Nu/Re\ Pr^{1/3})$
L	Pitch of channel waviness, m
$\dot{m}$	Mass flow rate, kg/s
Nu	Nusselt number; (hD_h/k)
Pr	Prandtl number; $(C_p \mu /k)$
P	Pressure, Pa
q_{conv}	Heat rate, W
Re	Reynolds number; $(u\ D_h / \nu)$
S	Channel spacing, mm
T	Temperature, K
u	Flow velocity, m/s
V	Volume flow rate, m^3/s
Greek symbols	
ε	Channel spacing ratio $(S/2a)$
Ø	Phase shift
γ	Channel corrugation ratio; $(2a/L)$
μ	Dynamic viscosity, kg.m/s

ν	Kinematic viscosity, m^2/s
θ	corrugation angle
ρ	density, kg/m^3
Subscripts	
a	for air
c	cross-section
i	at inlet conditions
h	hydraulic
m	mean value
o	at outlet, orifice area
p	at pipe cross sectional area
s	at surface wall conditions
w	for water
superscripts	
–	average

Chapter 1

INTRODUCTION

1.1 Enhanced Heat Transfer

Studies on heat transfer enhancement have been reported for more than 100 reports **(Webb, 1994).** In recent years, due to the increasing demand by industries for heat exchangers that are more efficient, compact and less expensive, heat transfer enhancement has gained serious momentum. Generally, the study of enhanced heat transfer is focused on two areas: increasing the heat transfer surface area such as extended surfaces and/or increasing the heat transfer coefficient by modifying the flow patterns near the heat transfer surface.

Various methods have been used to improve the performance of heat exchangers, which include the following:

i- Reducing the size of heat exchanger for the fixed heat duty.

ii- upgrading the capacity of an existing heat exchanger.

iii- reducing the approach temperature difference for the process streams, and

iv- reducing the pumping power for fixed heat duty.

These criteria are usually the basis for evaluating the enhanced heat transfer performance in a given application.

Two groups of heat transfer enhancements techniques have been identified: 'passive' and 'active' techniques. Passive techniques use special surface geometries, or fluid additives for heat transfer enhancement, such as coated surfaces or fluid additives, swirl flow devices, surface tension devices, and additives for liquids or gases. The active techniques require external power, such as electric or acoustic fields and surface vibration. Because of more costs involved, active techniques have attracted relatively little enthusiasm in research and practice, and passive enhancement through the use of various special surface geometries tends to be preferred.

Extended surfaces or fins are widely used as passive technique for heat transfer enhancement in heat exchangers. Examples of such extended surfaces or fins include plain fins, louvered fins, offset-strip fins, perforated fins, and wavy or corrugated fins.

Besides increasing the effective heat transfer surface area, extended surfaces also improve the heat transfer coefficient by modifying the flow field. Such fins are particularly attractive for their simplicity of manufacture, and ease of usage in both plate-fin and tube-fin type exchangers.

Corrugated flow channel is formed by placing two corrugated fins side by side and bonded them to the base heat transfer surface. In such a flow channel, the main flow direction is parallel to the fin waviness, but the local flow direction is always changed due to surface waviness; the thermal boundary layer formed in fin surfaces is periodically interrupted by flow separation and reattachment.

At low Reynolds number, the flow is laminar, and stream line flow pattern can be observed; at high Reynolds number, depending on the wavy fin geometry, flow separation, stream-wise and span-wise vortices may occur, and formed a complex flow patterns. These phenomena influence the temperature field significantly, resulting in sizable heat transfer enhancement in comparison to a parallel-plate channel. However, such gains in heat transfer are invariably accompanied by increased pressure drop penalty. Therefore, the primary goal for any enhancement of heat transfer study, as a result of using of corrugated-fin surface, is to increase heat transfer rate as much as possible while minimizing the pressure drop penalty.

1.2 Compact Heat Exchanger

In forced-convection heat transfer between a gas and liquid, the heat transfer coefficient of the gas is significantly low as compared to that of the liquid. One of the ways to increase the heat transfer coefficient and increased effective heat transfer area is to use of specially configured surfaces (such as fins) . For heat transfer between gases, the total surface area of heat exchanger may be 10 times larger than that of liquid-to-liquid heat exchangers in which the total heat transfer rate is comparable. In this case especially, the use of extended surfaces can substantially reduce the size of heat exchanger. These considerations have led to the development of heat exchangers with large surface area density that accommodates enhanced heat transfer rates. Such heat exchangers are referred to as ' compact heat exchangers'. In general, the ratio of total heat transfer surface area over total volume for a compact heat exchanger is greater than 700 m^2/m^3 .

Compact heat exchangers can be classified in two main types: plate – fin type as shown in Fig. 1.1 and tube- fin type or secondary surface heat exchanger as shown in Fig 1.2, and rotary regenerators for gas flows. The hydraulic diameters for most compact heat exchangers are very small and often located in the range of 1 mm to 10 mm. In industrial applications, the flows are often characterized by laminar or low Reynolds flow.

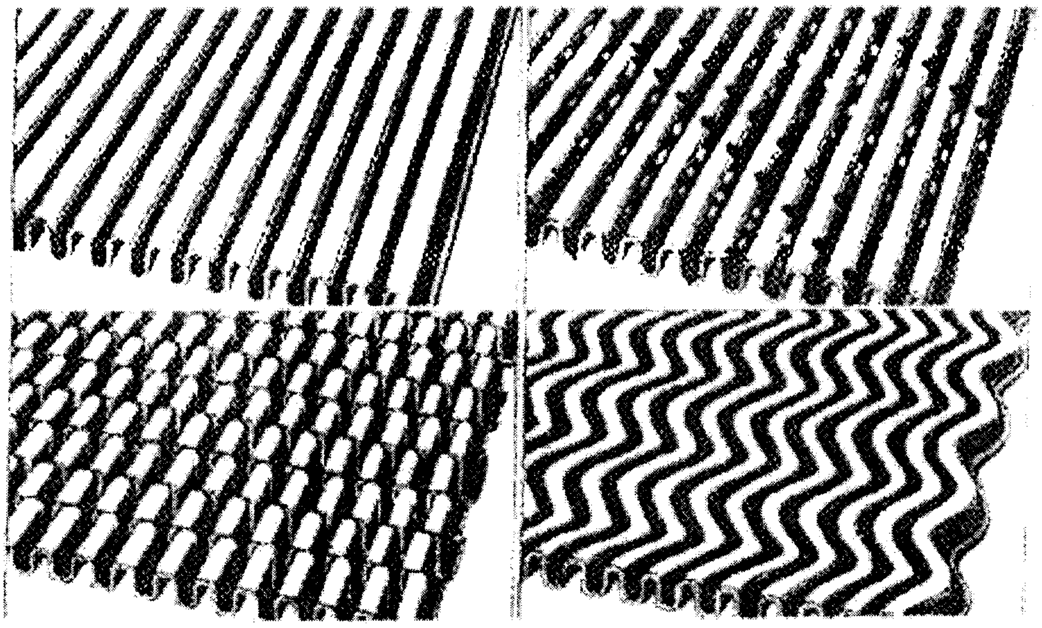


Fig. 1.1 Fin geometries for plate-fin heat exchanger

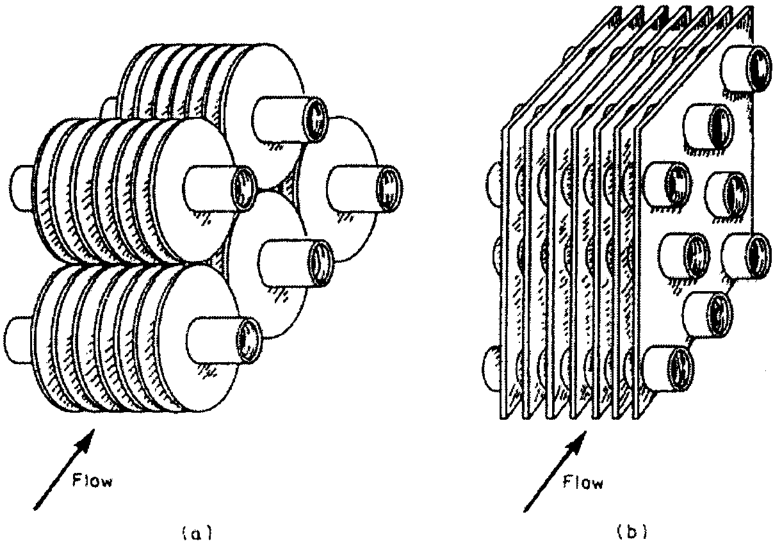


Fig. 1.2 (a) Individually finned tubes, (b) Flat or continuous fins on an array of tubes.

Some advantages are observed in compact heat exchangers compared to the traditional shell-and-tube heat exchanger such as high thermo-hydraulic performance, small size and compact volume.

These advantages make compact heat exchangers very attractive in various industrial applications. Some examples of such enhanced surface compact cores include offset-strip fins, louvered fins, perforated fins, and corrugated or wavy fins as seen Fig 1.3 .

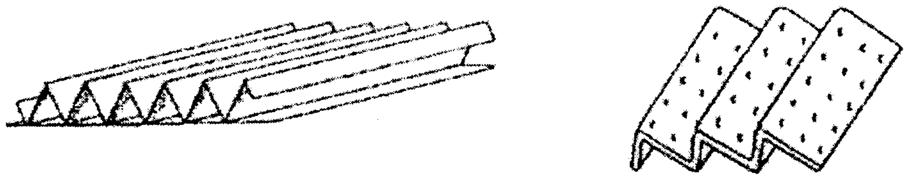
Because of the smaller hydraulic diameter fluid flows through such inter-fin passages are often laminar or low Reynolds number flows in nature. To be effective, the enhancement technique must be applicable to the low-Reynolds-number regime, and is based on the flowing two basic concepts:

- Special channel shapes, such as the corrugated channels in current study, which provide mixing due to secondary flows due periodic boundary layer modulation, separation or disruption.
- Repeated disruption and growth of boundary layers with bluff-body downstream wake generation. This concept is employed in the offset-strip fin, louvered fin, and perforated fin.

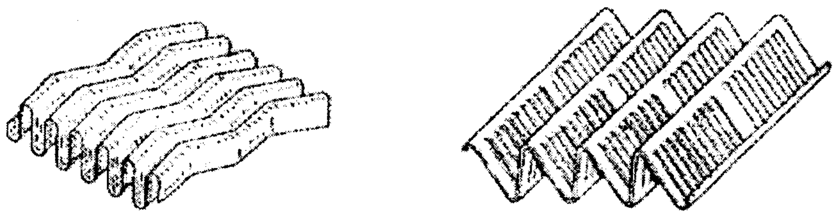
Of these, corrugated or wavy fins are particularly attractive for their simplicity in manufacture, potential for enhanced thermal-hydraulic performance, and ease of usage in both plate-fin and tube-fin type exchangers. The wavy-fin surfaces are also high-performance surfaces comparable to louvered and strip fin surfaces, in which the fin surface waviness causes the flow direction to change periodically. Consequently, the boundary layer separates and reattaches periodically around the trough regions to promote enhanced heat transfer. Increased pressure drop penalty is also accompanied. At high flow rates, swirl flows made up of counter-rotating vortices are observed in the main flow direction.

For any such enhancement technique to be considered as a viable alternative for practical heat exchanger application, they must exhibit performance that is at least substantially better than plain fins and comparable to some other existing extended surfaces. Brief descriptions of application and associated mechanism of the extended or fins are depicted in Fig. 1.3.

a. Rectangular **d. Offset Strip Fin**



b. Triangular **e. Perforated**



c. Wavy **f. Louvered**

Fig. 1.3 Surface geometries of plate-fin exchanger: (a) plain rectangular fins, (b) plain triangular fins, (c) wavy fins, (d) offset strip fins, (e) perforated fins, (f) louvered fins.

Because of the difficulty of investigating the actual finned coil with different fin passage configuration, the present work will be carried out on channel, with corrugated walls formed in a plate heat exchanger instead, where both fin spacing and phase shift can be varied. This simulates and gives more information about the characteristics of heat and fluid flow through plate-fin and frame and finned coils of such corrugated passages.

1-3 Aim of the present work

Enhancement of convective heat transfer in heat exchangers with moderate increase in pressure drop needs more investigations. In the present work, experimental measurements of both heat transfer and pressure drop in corrugated channels are carried out. Measurements are made with four corrugated channels of different spacing, three different phase shift, and constant corrugation angle over a range of Reynolds numbers. Tests were also conducted with a parallel-plate channel to establish a base-line for comparison with the corrugated-channel results.

To evaluate the net benefits of using corrugated channels instead of straight or parallel channels, the area goodness factor for the tested corrugated channels at different flow regions will be predicted.

CHAPTER 2

LITERATURE REVIEW

2.1 Forced Convection in Wavy or Corrugated Channels:

2.1.1 Experimental Studies

A large body of literature related to wavy flow channels or corrugated flow channels is available. The fluid flow and heat transfer behavior have been studied experimentally and numerically. The typical flow channels related to current study are listed in Fig. 2.1, and they include: (a) wavy flow channel with phase angle 0° (or in-phase wavy flow channel), (b) wavy flow channel with phase angle 180° (or converging-diverging wavy flow channel), (c) channel with single wavy wall, (d) channel with single triangular corrugation wall, (e) channel with in-phase triangular corrugations, (f) triangular corrugation converging-diverging channel.

(a) Wavy Flow Channels with phase angles

Focke et al, (1985, 1986) conducted flow visualization in channels with wavy walls formed in a plate heat exchanger. The channel plates have corrugation angles of 0°, 45°, 80°, and 90° relative to the channel axis. In their study, the complex flow patterns over a flow range of $10 \leq Re \leq 1000$ were reported. For the channel of 90°

corrugation, they found that the main flow was observed undulating in the direction of channel axis, and at low flow rate, no flow separation was observed. The flow separation first took place at Re 0f 20 , and with the increase of Re, the separated region increase in size , the flow became unsteady after Re $\approx$ 260. The Japanese research group led by Nishimura et al, (1985 – 1998) has conducted a number of flow visualization and mass transfer experiments regarding sinusoidal wavy channels with phase angle of 0^0 , 90^0 , and 180^0 . The channel with 180^0 phase angles is also called converging-diverging channel in other investigations. **Nishimura et al, (1986)** investigated the flow and mass transfer enhancement in wavy channels with phase angles of 0^0 , 90^0 , and 180^0. The experiments covered laminar and turbulent flow regime. Experimental results showed that, the pressure drop of wavy channels is larger than that of the straight channel at the same flow rate. With the increasing of phase angle, this enhanced pressure drop became more remarkable. From flow visualization, it is reported that the flow pattern for all channels changed remarkably at about $Re = 350$. Steady vortices are observed in the crest and trough region in wall wavy channels for $Re < 350$, and as the Reynolds number increase, the vortices became unsteady. Mass transfer augmentation were also reported in flow range of $Re > 350$.

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