Effect of Oscillatory Motion in Enhancing the Natural Convection Heat Transfer from a Vertical Channel

Lecturer: Saad Mohammed Jalil
Mechanical Engineering Department- University Of Anbar
Email: saadmj79@gmail.com

Abstract

This paper reports an experimental study regarding the influence of vertical oscillations on the natural convection heat transfer from a vertical channel. An experimental set-up was constructed and calibrated; the vertical channel was tested in atmosphere at 25°C. The channel-to-ambient temperature difference was varied with the power supply to the electrical heater ranging between 15W to 70W divided into five levels. Data sets were measured under different operating condition from a test rig under six vibrating velocities (VVs) levels ranging from (5-30 m/s) in addition to the stationary state. The results show that the maximum heat transfer enhancement factor (E) occurs at Rayleigh number ($Ra=2.328\times10^3$) and vibrational Reynolds number ($Re_v=6.365\times10^3$); this enhancement reached to (7.685%).The results also illustrated that the temperature gradient along the channel wall length was enhanced by inducing the oscillatory motion to the channel. Rayleigh and vibrational Reynolds numbers were ranging between (2.306×10^3 - 5.564×10^3) and (0.0 - 19.86×10^3) respectively. Finally, A correlation which summarized the effects of both Ra and Rev was determined for the Nusselt numbers.

Keywords :Oscillatory Motion, Enhancing, Natural Convection, Vertical Channel.

تأثير الحركة الاهتزازية في تحسين انتقال الحرارة بالحمل الحر من قناة عمودية

المدرس : سعد محمد جليل
قسم الهندسة الميكانيكية ـ جامعة الانبار

الخلاصة

يهدف البحث الحالي إلى إجراء دراسة عملية حول تأثير الاهتزاز العمودي على انتقال الحرارة بالحمل الحر من قناة عمودية.تم تصنيع ومعايرة المنشأ التجريبي المستخدم في الدراسة ،وفي ظروف جوية وعند درجة حرارة (25°C) للمحيط الخارجي تم اختبار القناة المستخدمة لعدة سرع اهتزازية الفرق في درجة الحرارة بين القناة العمودية والمحيط الخارجي تغير وفقا لمقدار القدرة الكهربائية المجهزة للسطح والتي تراوحت بين (15W - 70W).القراءات المسجلة من التجارب للقناة قيست تحت تأثير سرع اهتزاز عمودية مختلفة تراوحت بين (5-30 m/s) بالإضافة إلى الحالة المستقرة (غير مهتزة).من خلال النتائج تبين ان أعلى قيمة لمعامل تحسين انتقال الحرارة بالحمل الحر (E) حصل عند عدد رالي ($Ra =2.328\times10^3$) و عدد رينولد الاهتزازي ($Re_v =6.365\times10^3$) وبنسبة زيادة (7.685%) مقارنة بالحالة المستقرة.كذلك بينت النتائج ان تدرج درجات الحرارة على طول جدران القناة قد تحسن بتأثير الحركة الاهتزازية المجهزة للقناة. خلال جميع التجارب فان عدد رالي و رينولد الاهتزازي تراوحت بين (2.306×10^3 - 5.564×10^3) و (0.0 - 19.86×10^3) على الترتيب. العلاقة التجريبية المستنتجة من النتائج لخصت تأثير كل من عدد رالي و رينولد الاهتزازي على عدد نسلت عند كل حالة.

الكلمات الرئيسية: الحركة الاهتزازية ، تحسين ،الحمل الحر ، قناة عمودية.

1. Introduction

Natural convection heat transfer with air as the working fluid is a highly important cooling mechanism that is currently implemented across a broad spectrum of systems. It is simple, efficient, dependable, quiet operation and poses no hazards to human users, makes it the preferred method of cooling especially in devices where heat transfer rates are at low levels **(Yeh LT,1995, Incropera FP and DeWitt DP,1996, Sathe S,1998, Incropera FP,1999 and Tou SKW et al.,1999)**.Also it concerns a number of applications, ranging from the cooling of electronic equipment to the heating of buildings. Laminar heat transfer by natural convections has been studied extensively for different geometries **(Ramanathan S and Kumar R,1991, Vafai K and Ettefagh J, 1990a, Vafai K and Ettefagh J,1990b, Daloglu A and Ayhan T,1999, and Khanafer K and Vafai K,2000)**.

Vertical channels are frequently encountered configuration in natural convection cooling where the flows are induced by the buoyancy of thermal diffusion alone. A significant domain of natural convection flow has relatively small velocity magnitudes and contain almost unnoticeable turbulence levels. Spaces cooled by natural convection flows vary from one application to another. Natural convection inside vertical channels had received an increase attention in the past decade.There are only few works that dealed with heat transfer and flow induced by natural convection inside vibrating geometries. As an example,**(Fu and Shieh,1993)** studied the effect of buoyancy and vertical vibrations at the four walls of a square closed cavity.**(Kwak et al.,1998)** discussed the effects of having vibrations on the temperature of one plate of a closed cavity to the natural convection inside the cavity. **(Oronzio Manca et al.,2003)** experimentally introduced air natural convection in an asymmetrically heated channel with unheated extensions. The addition of downstream unheated extensions improves the thermal performance of the channel for some configurations, longer extension and the lower the aspect ratio the lower the wall temperature in the channel.**(X.R. Zhang et al.,2004)** described a numerical study of the laminar natural convection on a periodically oscillating vertical flat plate heated at a uniform temperature. Of particular interest of the paper was the heat transfer characteristics when the oscillatory velocity being close to the flow velocity in the velocity boundary layer under non-oscillation conditions. The results showed that the heat transfer of the problem under consideration significantly depends on the dimensionless oscillation velocity, a relative size between the oscillation velocity and the flow velocity in the velocity boundary layer of a stationary plate. The heat transfer enhancement was found to be increased with the dimensionless oscillation frequency, amplitude, the Prandtl number, but decreased with the Grashof number.

(N. Bianco,T and S. Nardini,2005) investigated numerically the natural convection in air in a convergent channel. An extended computational domain was adopted, which allowed to take the account of diffusion by both momentum and energy outside the channel. Temperature profiles were strongly affected by the convergence angle at low Rayleigh numbers. At the lower minimum gap, streamlines and isotherms show a low pressure region in the channel due to a choked flow in its upper end. **(Wu-Shung Fu and Chien-Ping Huang,2006)** performed a numerical simulation to study effects of a vibrational heat surface on natural convection in a vertical channel flow. The results showed that for the same Rayleigh number, natural convection of a vibration heat plate with a certain combination of frequency and amplitude was possibly smaller than that of a stationary state.**(L. A. Florio and A. Harnoy,2006)** investigated a dynamic means of locally enhancing laminar natural convection cooling in a vertical channel through the localized application of fluid oscillations numerically. The two-dimensional system considered for these purposes was a vertical channel with a small transversely oscillating plate placed near a constant heat flux channel wall. The flow and heat transfer in the system resulting from the combined effects of the natural convection and the oscillating plate were determined. The results indicated that for displacement amplitudes of at least one-third of the mean spacing and with dimensionless frequencies ($Re/\sqrt{Gr}$) of at least 2π,the local heat transfer coefficient could be enhanced as much as 41%.

(N. Kasayapanand and T. Kiatsiriroat,2007) presented numerically the electrohydrodynamic effect to natural convection inside the vertical channels by the use of computational fluid dynamics technique. The relation between channel aspect ratio and number of electrodes that performs the maximum heat transfer was expressed incorporating with the optimum concerning parameters. **(Subhrajit Dey, Debapriya Chakrborty, 2009)** implemented a numerical methods (finite volume) to study the effects of an oscillating fin in enhancing heat transfer. It is conclude that there are benefits in Nu for an oscillating fin over a conventional static fin because of contraction and disturbances imported to the thermal boundary layers. The benefits improved with higher frequencies and amplitudes of oscillation.

(L. Cheng et al.,2009) proposed a novel approach to enhance the heat transfer by using the flow-induced vibration of a new designed heat transfer device. Thus the flow-induced vibration is effectively utilized instead of strictly avoiding it in the heat exchanger design. The vibration and the heat transfer of these devices were studied numerically and experimentally. It was found that the new designed heat exchanger can significantly increase the convective heat transfer coefficient and decrease the fouling resistance. Therefore, a lasting heat transfer enhancement by the flow-induced vibration can be achieved. **(P. Poskas et al.,2011)** presented the results of experimental investigation of local opposing mixed convection heat. The analysis of the results revealed significant increase in the heat transfer with the increase of air pressure (Gr number). Also a sharp increase in heat transfer was noticed in the region with vortex flow as comprised with the turbulent flow region.

In the above mentioned research it is noted that much of the work have been carried out on natural convection inside different geometries and that related to behavior of vertical channel subjected to the effect of oscillation have not been fully investigated. The present investigation mainly attempt to shed light on the oscillation induced heat transfer enhancement , the base plate of a vertical channel and its walls are allowed to have oscillatory motion in vertical direction. The experiments are carried out at constant heat flux with various frequencies. Accordingly, an investigation is done for better understanding about performance of natural convection inside vibrated vertical channel.

2. Experimental Set-Up And Data Reduction

The main objective of the experimental setup was to assist in development of natural convection requirements in a vertically oriented channel subjected to influence of oscillatory motion. A summary information about the experimental apparatus, devices used and procedure are described below.

2.1 Experimental Set-Up

A schematic view of the experimental setup is presented in fig.1. The essential components of the setup include a wood room of 2×1.5m dimensions with 2m height, experiment testing model was represented by a vertical channel, instrumentation for measurements temperatures (channel and ambient temperatures), power supply to the heater, and the frequencies supply to the experiment model by the exciter. Tests were carried out under constant heat flux condition with various frequencies and amplitude, which lead to various vibrating velocities (VVs) for the channel in ambient temperature controlled at 25^{o}C for all experiments by a thermostat/heater unit as shown in fig.1. Five levels of heat flux ranging between (15-70W) were considered to study the natural convection behavior from the channel. At each level, the channel subjected to six values of VVs ranging between (5-30 m/s) in addition to the stationary state.

Fig.2 illustrates the dimensions and the parts of the experiment model. The channel was made from Aluminum as one part. The channel configuration was produced from rectangular Aluminum bar with dimensions 100×75×20mm by milling a longitudinal groove of dimensions 10×60mm in the center of the bar. The bar constructed by casting an appropriate amount of Aluminum inside a stainless steel ground container and then grained and polished to have the required dimensions for the bar. Aluminum was selected as the channel material because of its high thermal conductivity ,low emissivity ,structural strength, and durability. Nickel-chrome wire heater used for heating the channel base consists of nickel-chrome wire wound round a thin mica plate and insulated on both sides by two mica sheets. A 5mm thick asbestos slab was glued to the bottom surface of the lower mica sheet to reduce heat loss. The rear surface of the asbestos and the lateral surface of the channel base were insulated by a wooden case of 15mm thickness. Hence, all five sides of the test section except the upper surface were insulated. The aim of which was to reduce the heat loss and

To keep the temperature of the boundary around ambient temperature.

The channel base temperature was measured by the average of four temperatures measurements at the back of the base. Four k-type thermocouples were installed at equal spacing in grooves along the bottom surface of the channel. Furthermore, two k-type thermocouples were attached to the rear surface of the wooden case to calculate the conductive dissipated heat transfer. The temperature distribution for the channel walls was collected by installing five k-type thermocouples along the channel wall length with 10mm between them. This indicts a clear temperature gradient along the channel wall length under the effect of the oscillatory motion. All thermocouples were connected to a k-type selector switch and the output of the selector was supplied to a digital thermometer. From the calibration test of the thermocouples used the accuracy of averaged steady state temperature measurements haven't exceed $\pm 0.5^{o}C$.The heater was supplied by stabilized AC power through a regulator which maintains a constant voltage during the data reading. The output of the regulator is fed to a variable transformer (variac) so that the necessary heating level can be selected. The current flow and voltage drop are measured by an ammeter-voltmeter combination. The power input is measured by a calibrated wattmeter. The measurements of wattmeter are verified by the readings of voltmeter and ammeter.

The channel was fixed vertically in a position between two horizontal arms of an iron frame; the frame supported the wooden case, which was connected to the exciter. The oscillatory motion was induced by the exciter-power amplifier unit as shown in fig.1. The oscillatory motion of the channel measured as a vibrating velocity by the vibration meter, which is sensing the signal of vibration sent by the electrodynamics pick-up. After completion the experiments in non-vibrating condition the desired vibrating velocity was set according to the vibration meter reading by adjusting the power amplifier unit to import vibration to the test section by the exciter. The experimental data of temperatures, vibrating velocities, and the electric power, were measured when the system operates under steady state conditions.

2.2 Processing Of The Experimental Data

The experimental data includes the values of temperatures, velocities of vibration, voltage drop across the electrical heater and electrical current. Using these values, the Nusselt number is calculated by definition **(M. Dogan , M. Sivrioglu, 2010)**:

$$Nu = \frac{h.S}{k_{air}} \tag{1}$$

Where k_{air} is the conduction heat transfer coefficient for air at film temperature (T_f) which is calculated as below:

$$T_f = \frac{T_b + T_\infty}{2} \tag{2}$$

Where T_b is the base temperature of the channel which is represented the average temperature of the thermocouples that placed in the base plate of the channel. Then the temperature difference ratio TDR is given by:

$$TDR = \frac{\Delta T_v}{\Delta T_{unv}} \tag{3}$$

It is computed at six locations along the channel wall length represented by the non-dimensional channel length (X=x/L).

The Convection heat transfer from the channel (Q_{conv}) is calculated from an energy balance:

$$Q_{total} = Q_{conv.} + Q_{cond.} + Q_{rad.} \tag{4}$$

The total dissipated energy is determined from Ohm's law, $Q_{total} = VI$ at the heater source. The voltage drop (V) and current (I) are measured during the experiment.

The heat loss through the insulation of test section is calculated from :

$$Q_{cond} = k_{ins.} \times A_{ins.} \frac{\Delta T_{ins}}{\Delta y_{ins.}} \tag{5}$$

Where k_{ins} is the thermal conductivity of the insulation and ΔT_{ins} is the difference between internal and external surface temperatures of the insulation.

For the heated surface represented by the vertical channel, radiative heat transfer was also taken into account using the flowing equation:

$$Q_{rad} = \varepsilon\sigma AF(T_b^4 - T_\infty^4) \qquad (6)$$

Where the view factor F between the channel and its surrounding was taken to be unity, (see e.g. **(C.G. Rao et al.,2002 & E. Kchoc et al.,2003)**), The results of experiments showed that the radiation losses never exceeded 2% of the total power supplied. Now the heat transfer coefficient can be calculated and expressed as:

$$h = \frac{Q_{conv}}{A(T_b - T_\infty)} \qquad (7)$$

Other dimensionless numbers affecting the heat transfer are:

The Rayleigh number

$$Ra = Gr.\Pr = \frac{g.\beta.(T_b - T_\infty).S^3}{\alpha\nu} \qquad (8)$$

Where $\beta = \frac{1}{T_f}$,and the characteristic length was taken as the space between the channel walls **(M. Dogan , M. Sivrioglu, 2010)**.

The vibrational Reynolds number

$$\mathrm{Re}_v = \frac{U_v S}{\nu} \qquad (9)$$

The correlation between Nusselt number and both above dimensionless numbers eq.(8&9) is defined as below:

$$Nu = cRa^m \mathrm{Re}_v^n \qquad (10)$$

The constants (c,m,n) are calculated and listed in Table (1) for each case .Also all thermophysical of air appearing in equations above were calculated at film temperature eq.(2)

3. Results and Discussion

The convection heat transfer from vertical oscillated channel under constant heat flux conditions has been experimentally investigated. Tests were conducted under various heat flux and oscillating velocities conditions. the stationary state of the vertical channel was investigated to assess the validity of the experimental results with previous work and used as a reference to compare heat transfer results which generates under six vibrating velocities condition ranging (5-30 m/s) . the air around the base and walls is oscillating because of channel vibration.. Oscillatory motions generally enhance the heat and mass transfer in fluid.

The oscillatory motion of the channel is defined as:

$$d_x = constant \qquad (11)$$

$$d_y = B\sin(2\pi f t) \qquad (12)$$

Where B and f are the amplitude and the frequency of oscillation respectively. The experimental data are analyzed by the procedure that described previously in section 2.2. the temperatures distribution along the channel wall length, heat transfer coefficients and the non-dimensional numbers are calculated at different velocities of vibration under various heat flux levels.

The experimental results obtained in the current study indicate that the oscillation in general enhanced the heat transfer coefficient but in a fluctuating manner. For temperatures distribution the behavior was expected with cooling the base plate by decreasing the base temperature difference.

3.1 Temperature distribution

Plotting the Temperature Difference Ratio (TDR) versus the vibrating velocities (VVs) for non-dimensional channel wall length helps to understand the effect of vibrating velocity variation ,for each heat flux value, on the temperatures distribution along the channel wall length and gives a good indication about the percentage of enhancement.

Figs. 3-8 present the variations of TDR with (VVs) at all power levels and for each non-dimensional channel wall length (X= x/L= 0.0, 0.2, 0.4, 0.6, 0.8 and 1.0). As can be seen from these figures, there is a positive effect on TDR with (VVs) increase for all values of (X) These figures show that the temperature difference ($\Delta T_{unv}=T_x-T_\infty$) at each location along the channel wall length for stationary case is higher than the temperature difference ($\Delta T_v = T_x - T_\infty$) for vibrating state. Evidently, the ratio of the temperature difference between vibrating and stationary condition doesn't exceed (1) in most of the states.

Fig.3 shows the effect of increasing VV on TDR at X=0, which represents the base plate location, for the five power levels . From this figure, it can be seen that the heat transfer enhancement at the channel base depends on increasing or decreasing VV at each power level relative to stationary case. The best enhancing appears at the lowest power level (15W) and VV equal to (10m/s).The TDR ratio for this case decreases by (7.14%), that means there is enhancement in heat transfer. But, when the vibrating velocity is more than 10m/s; the heat transfer enhancement decrease and the TDR ratio at the maximum VVs decreases by (1.43%), but never take over the stationary state. This variation occurs for all power levels else the last point of the maximum power which is increased over the stationary state by (1.94%) that means an increase in channel base temperature at these conditions.

As can be seen from figures (3-8), the behavior of the relation between TDR and VV almost remains constant along the channel wall length. This variation in enhancing TDR along the channel wall length as can be seen from these figures is due to the fact that the buoyancy driven forces become more strong or weak with increasing in VVs and heat flux. For example, at power (50W) it can be seen that the enhancing increases with VVs and then decreases at the maximum VV. That is a result of the effect of oscillating in an increase of the buoyancy driven forces and therefore these phenomenon of enhancing at the beginning increases the enhancement magnitudes by decreasing TDR along the channel length at a specific VVs ranges.

On the other hand it can be seen from the figures that the transfer from decreases to increasing in TDR with VVs depends on the power level . These changes in its path is attributed to the buoyancy driven forces increase with slow oscillations and then becomes weak by the impeded in boundary layer with variation of the air density which is bounded the channel. It leads to slow the hot air circulation and rise but stay all points along the channel wall and its base cooler then the non-vibrated state. Also the figures show that the decreases in TDR for the maximum power occurs at minimum VVs and return to increase with VVs until to be greater than the stationary state only at maximum VVs for all location except the channel edge (X=1). That is occurs as a result of the rising temperature around the walls and the impeding created by the oscillation at these conditions. This behavior considers non-favorite results so it is best to working in previous ranges only with appropriates application for this ranges of temperatures or heat flux.

The same figures, illustrate that there is fluctuating enhancement in TDR with VVs represented by the two power levels (30W & 40W), it is clear that the variation in TDR decreased and then increased slowly and return to decreased at high VVs in opposite manner of the other powers. That is mean the buoyancy force has an effect on the hot air depended on the amount of heat supplied and VVs provided to the channel.

3.2 Heat transfer enhancement factor (E)

The rate of heat transfer at the surface of stationary vertical channel is controlled by the natural convection boundary layer created by the density difference due to temperature change near its surface. When an oscillatory motion is created between the channel and the adjacent fluid; the heat transfer coefficient at the vibration state increases with increasing VVs in most of studied conditions. Fig. 9 shows the effects of oscillation on average heat transfer rate expressed in terms of the enhancement factor (E) which is defined by the following form:

$$E = \frac{Nu_v - Nu_{unv}}{Nu_{unv}} \tag{13}$$

Where Nu_v and $Nu_{unv.}$ represents Nusselt numbers under oscillating and stationary conditions respectively. As seen from the figure, the factor E increased in various rates depending on the power levels and VVs. Here we couldn't refer to Grashof number instead of the power supplying to the channel because we have at each power level different values of Grashof number depending on the VVs supplied. The oscillation plays an important role in enhancing Nu at various VVs.

Also we can notice that the maximum enhancement occurs at the first power level under vibrating velocity equal to (10m/s), from the results its reached to (7.685%) and then decreases with VVs but remain over the non-vibrated state. It is appear in the figure for the maximum power supplying the enhancement factor increases to (2.835%) and then it is decreased to reach under the stationary state only at the maximum VVs causing reducing in heat transfer rate by (2.42%). From Fig. 9 which shows the variation of E with VVs, it can be seen that the behavior of its variation to yield to the power level in addition to the supplying oscillations. As seen the oscillations increases E with extent of enhancement being most pronounced from small values of power and VVs.

3.3 Influence of Ra and Re_v on Nusselt number

Fig. 10 represents the effect of Ra on Nu for the cases where the channel is vibrating vertically with various VVs. As the Ra increases the Nu increased for all velocities. From the figure it is clear that the non-vibrating case has a lowest values of Nusselt number for the most rang of Ra if it is contrasts with the oscillated cases else the last point at maximum power and VVs. Fig. 11 displays the influence of Rev for all Ra ranges (2.306×10^3 - 5.564×10^3).It is noticed that the changing in Nusselt numbers are greater when the vibrational Reynolds increase at low VVs and then fluctuating with Rev in a different manner with that of Ra.

This increasing in Nu is due to the effect of the vertical vibrations in both buoyancy forces and the slow flow induced by the motion of the channel yet vertical vibrations affected mainly the working fluid adjacent the channel which is heated as a result of temperature differences between the channel and air. The interaction between vibrations and the oscillatory buoyancy forces resulted from vibrations in the channel cause the trend of Nu to change as Rev increases. Also the temperature of adjacent fluid inside the channel are expected to decrease as Ra increase due to the enhancements in thermal convections. Therefore, Nusselt numbers at vibrated channel increase as Rev increases but when the fluctuating increases depending on the Rayleigh number.

Based on experimental results, an empirical equation relates Nusselt number, vibrational Reynolds number and Rayleigh number has been obtained. The software (DGA-V1) is used to synthesize above equation. The percentage error between the theoretical and empirical equation is not exceed (3%). Based in stationary state with all power levels and oscillated states at each power supplying for all vibrating velocities and by using the above software the equation is deduced different coefficients for empirical equation. This will donated us more than equation as specified in table (1).

4. Conclusions

Natural convection from vertical channel under vertical vibration motion was investigated experimentally. The variation of Rev and Ra were considered and the results were examined in detail. The main conclusions could be summarized as follows.

1. For the same Ra natural convection from a vertical channel subject to a vibration motion under various VVs is larger than that of the stationary state.
2. The temperature gradient along the channel wall length under vibrating state in general decreases to more cooling surfaces.
3. The increasing in Nusselt number changes with vibrational Reynolds number in fluctuating manner depending on the Rayleigh number.
4. A maximum value of the heat transfer enhancement factor (E) is (7.685%) and then decreases to minimum positive value by (2.835%)
5. Only one point leads to non-favorite results represented by decreasing E to a negative value by (2.42%) at maximum Ra and Rev.

5. References

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NOMENCLATURE

LATIN SYMBOLS

Symbols	Description	Units
a	Channel wall thickness	m
A	Exposed area of the channel	m^2
A_{ins}	Area of the insulations	m^2
B	Oscillation displacement amplitude	m
d_x	Displacement of oscillating in x direction	m
d_y	Displacement of oscillating in y direction	m
c	Constant	
E	Enhancement factor	
f	Frequency	Hz
F	View factor	
g	Acceleration due to gravity	m^2/s
Gr	Grashof number	
h	Convective heat transfer coefficient	$W/m^2.{}^oC$
k_f	Fluid thermal conductivity	$W/m\ {}^oC$
K_{ins}	Insulation thermal conductivity	$W/m\ {}^oC$
L	Channel wall length	m
m	Constant	
n	Constant	
Nu	Nusselt number	
Pr	Prandtl number	
Q_{total}	Total power supply	W
$Q_{cond.}$	Heat transfer by conduction	W

Symbols	Description	Units
$Q_{rad.}$	Heat transfer by radiation	W
$Q_{conv.}$	Heat transfer by convection	W
Ra	Rayleigh number	
Re_v	Vibrational Reynolds number	
S	Characteristic length	m
t	Time, (s)	
T_f	Film temperature	${}^oC$
T_b	Base temperature	${}^oC$
T_x	Temperature at position (x)	${}^oC$
T_∞	Ambient temperature	${}^oC$
U_v	Vibrating velocity	m/s
W	Channel width	m
x	Coordinate along the channel length	m
X	Dimensionless channel length (x/L)	
y	Coordinate along the channel width	m

GREEK SYMBOLS

β	Coefficient of thermal expansion	K^{-1}
υ	Kinematic viscosity	m^2/s
ε	Emissivity	
σ	Stefan-Boltzmann constant	$W/m^2.K^4$
α	Thermal diffusity	m^2/s

Table 1 Coefficients and ranges of governing empirical equations of the studied cases

Case		c	m	n	Ra	Re_v
Stationary (15-70)W		0.08335	0.4688	0	2.437×10^3 - 5.421×10^3	0
Oscillatory Case (5-30) m/s	15 W	49.761	-0.3726	-0.01685	2.306×10^3 - 2.420×10^3	2.918×10^3 - 1.986×10^4
	30 W	0.01311	0.6427	0.01206	3.469×10^3 - 3.536×10^3	2.788×10^3 - 1.9108×10^4
	40 W	104.851	-0.4285	-0.00164	4.052×10^3 - 4.146×10^3	2.743×10^3 - 1.8751×10^4
	50 W	2.8024	0.001143	0.01055	4.241×10^3 - 4.423×10^3	2.627×10^3 - 1.7991×10^4
	70 W	20.8042	-0.19155	-0.02205	5.174×10^3 - 5.564×10^3	2.544×10^3 - 1.7212×10^4
	(15-70) W	0.0975	0.4161	-0.00515	2.306×10^3 - 5.564×10^3	2.544×10^3 - 1.9860×10^4

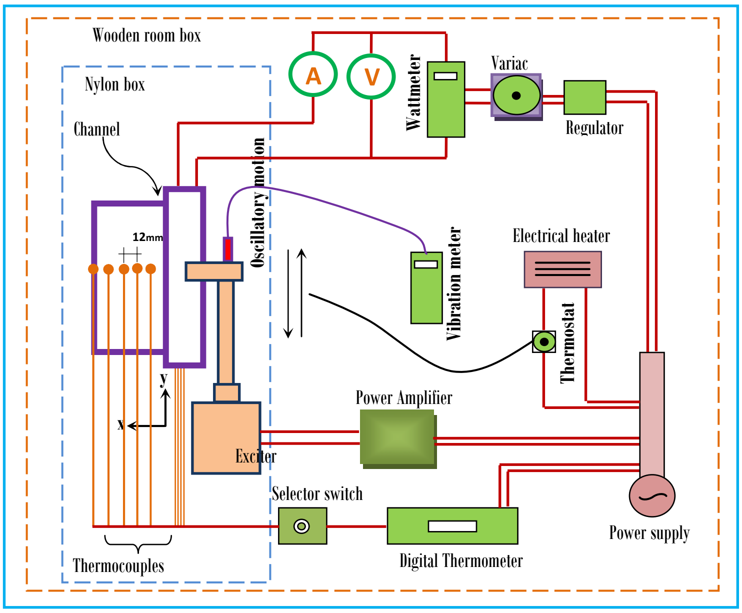

Fig. 1 Schematic diagram of the experimental setup

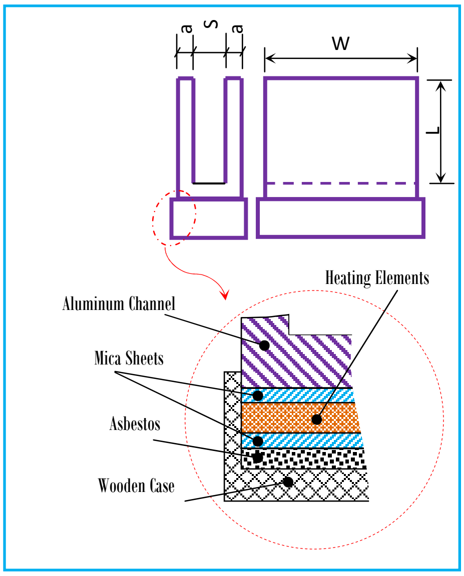

Fig. 2 Dimensions and components of the channel in (mm) (W=100, L=60, a=5, S=10)

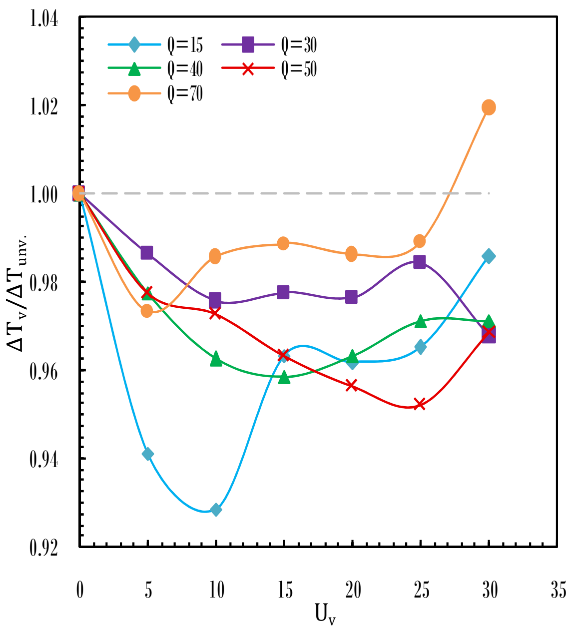

Fig.3 Variations of temperature difference ratio with vibrating velocity at X=0.0

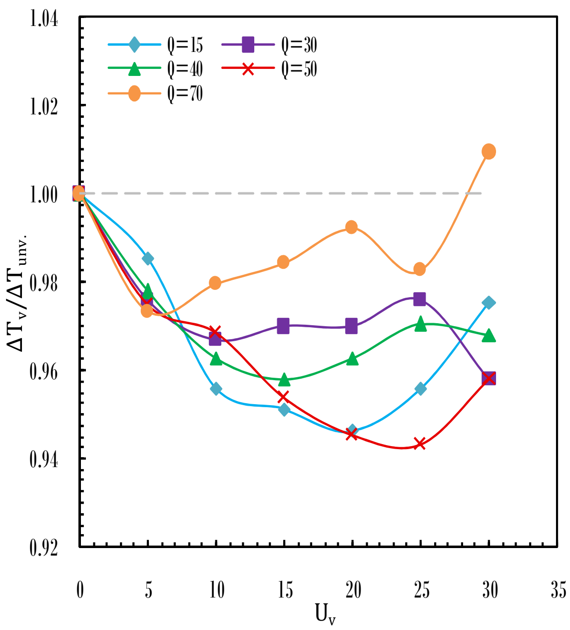

Fig.4 Variations of temperature difference ratio with vibrating velocity at X=0.2

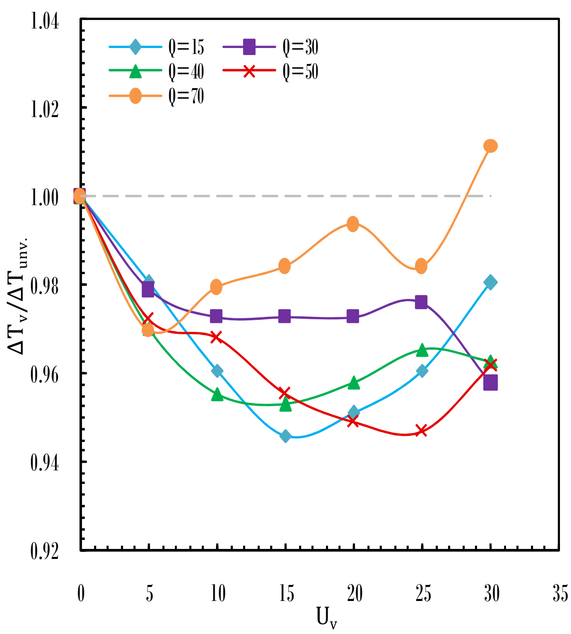

Fig.5 Variations of temperature difference ratio with vibrating velocity at X=0.4

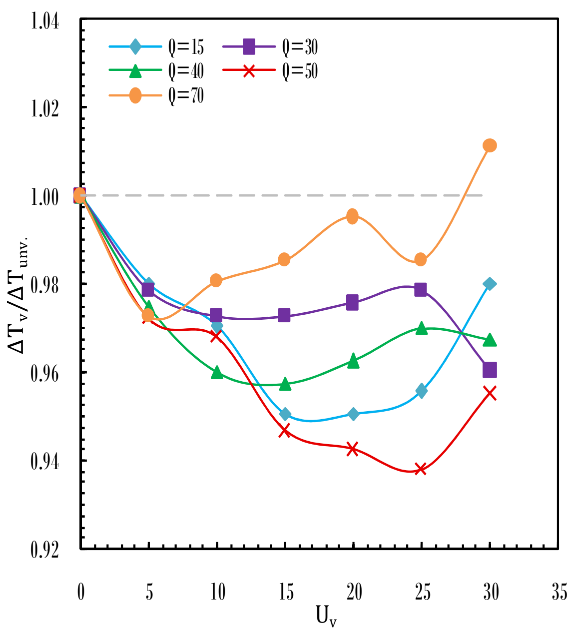

Fig.6 Variations of temperature difference ratio with vibrating velocity at X=0.6

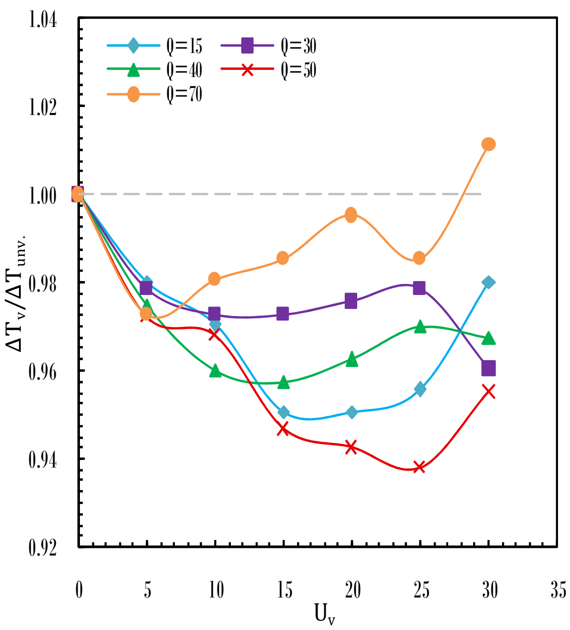

Fig.7 Variations of temperature difference ratio with vibrating velocity at X=0.8

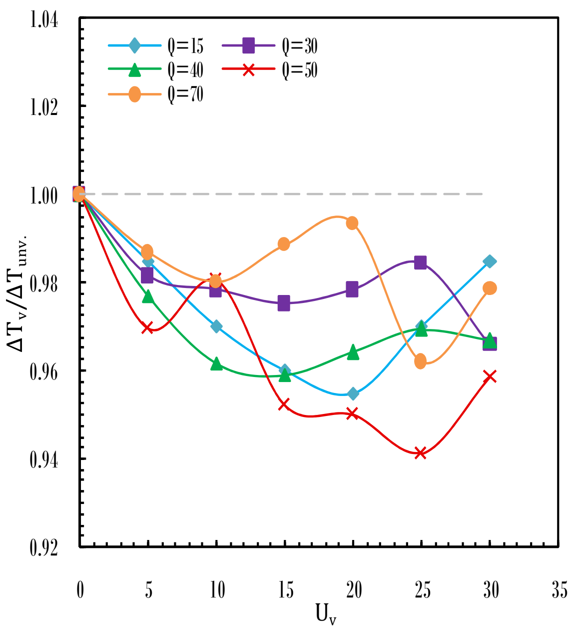

Fig.8 Variations of temperature difference ratio with vibrating velocity at X=1.0

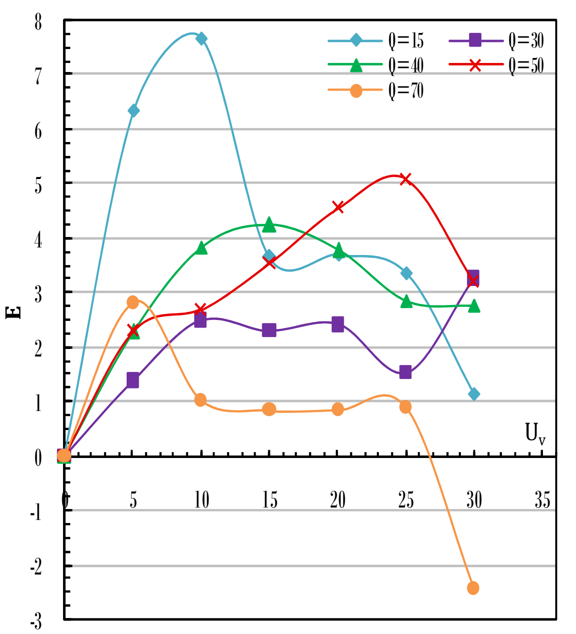

Fig.9 Effect of vertical oscillations on heat transfer enhancement factor (E)

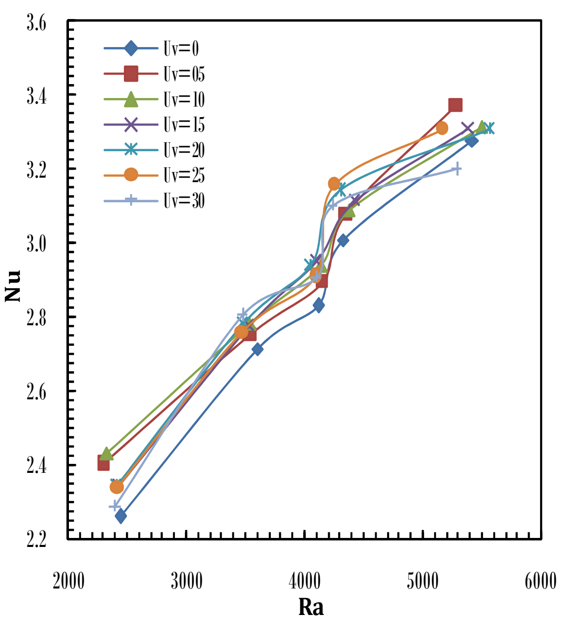

Fig.10 Variation of Nu with Ra for different vibrating velocities

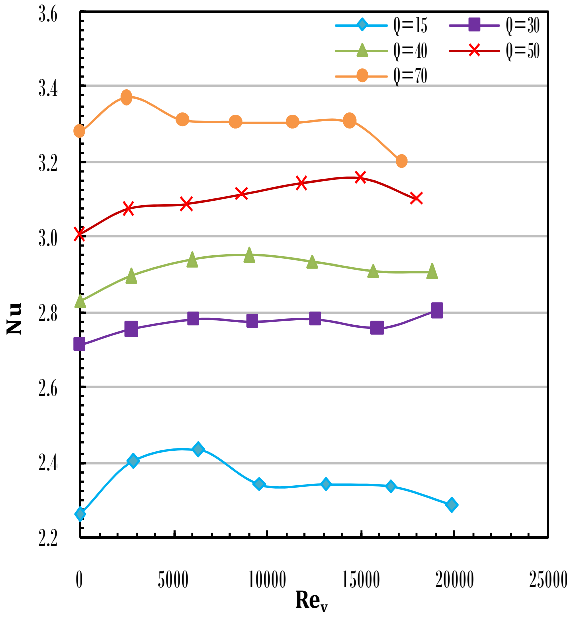

Fig.11 Variation of Nu with Re$_v$ for different power levels