Experimental Investigation Utilizing Thermal Image Technique to the Heat Transfer Enhancement Using Oscillated Fins

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ABSTRACT

Heat transfer around a flat plate fin integrated with piezoelectric actuator used as oscillated fin in laminar flow has been studied experimentally utilizing thermal image camera. This study is performed for fixed and oscillated single and triple fins. Different substrate-fin models have been tested, using fins of (35mm and 50mm) height, two sets of triple fins of (3mm and 6mm) spacing and three frequencies applied to piezoelectric actuator (5, 30 and 50HZ). All tests are carried out for (0.5 m/s and 3m/s) in subsonic open type wind tunnel to evaluate temperature distribution, local and average Nusselt number (Nu) along the fin. It is observed, that the heat transfer enhancement with oscillation is significant compared to without oscillation for low air inlet velocity. Higher thermal performance of triple fins is obtained compared to the single rectangular fin, also triple fins with (height=50mm and fin spacing=3mm) gives better enhancement as compared to other cases. This work shows that the piezoelectric actuator when mounted on the rectangular fins shows great promise for enhancing the heat transfer rate.

KEYWORDS: **laminar flow; forced convection; thermal image; enhancement of heat transfer; oscillating fins.**

دراسة عملية باستخدام تقنية الكامرا الحراريه لزيادة معدلات انتقال الحرارة باستخدام الزعانف المتذبذبة

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الملخص: دراسة تجريبية لانتقال الحرارة حول زعانف مندمجة مع (Piezoelectric Actuator) تستخدم كزعانف متذبذبة في جريان طباقي بالاستفادة من الكامرة الحراريه. انجزت هذه الدراسة لزعانف ثابتة ومتذبذبة وكذلك منفردة وثلاثية. واختبرت نماذج متعددة من الزعانف بارتفاع (35 ملم و 50 ملم) ومجموعتين من النماذج للزعانف الثلاثية مع مسافات بين الزعانف (3 ملم و 6 ملم) وكذلك استخدمت ثلاث ترددات لل (Piezoelectric Actuator) وهي (5 و 30 و 50 هيرتز). واجريت كل الاختبارات ل (0.5 م/ثا و 3 م/ثا) في نفق هوائي نوع مفتوح لحساب توزيع درجات الحرارة وكذلك حساب عدد نسلت الموقعي والمعدل خلال الزعنفة. يلاحظ تحسن مهم في انتقال الحرارة مع الذبذبة مقارنة مع الزعانف بدون ذبذبة للسرع القليلة لدخول الهواء. ونحصل على انتقال حرارة عالي مع الزعانف الثلاثية مقارنة مع الزعنفة المنفردة .

وتعطي الزعانف الثلاثية مع (الارتفاع= 50 ملم والمسافة بين الزعانف=35 ملم) افضل تحسن في انتقال الحرارة مقارنة مع الحالات الاخرى. وتوضح الدراسة انه عند دمج ال (Piezoelectric Actuator) مع الزعانف المستطيلة يعطي تحسن كبير في معدل نقل الحرارة

الكلمات الرئيسية: جريان طباقي، حمل قسري، صور حرارية، تحسين انتقال حرارية، زعانف متذبذبة

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INTRODUCTION

Heat transfer enhancement has become a target in the miniaturization of electronic components. Enhanced heat transfer surfaces can be designed through a combination of factors that include increasing fluid turbulence, generating secondary fluid flow patterns, reducing the thermal boundary layer thickness and increasing heat transfer surface area (David 2010). One of the most difficult challenges in modern power electronics is obtaining sufficient cooling for the components. The operating temperature of the components is an extremely important factor affecting their reliability. The heat sink typically consists of a base plate and a stack of fins. In addition, as the dissipated heat of the components grows, a fan or a pump is needed to obtain higher rates of heat transfer by forced convection. The design process of a heat sink for a given set of electronic components is a very complicated task involving many contradicting optimization criteria (Antti 2005). Numerous methods have been proposed to enhance heat transfer rate of a heat body. These methods can be classified into the passive and active methods. Examples of the former one are treated surfaces and swirl flow devices, while examples of the active methods are surface vibration, fluid vibration, injection and suction (Wu-Shung 2007).Active methods require external power to enhance heat transfer on the contract to passive methods. Extended surfaces or fins are example of passive methods that are commonly used in variety of industrial applications to enhance the rate of heat transfer between primary surface (heat sink) and ambient fluid (M.R.Shaeri 2009). In addition to numerous methods have been proposed to enhance heat transfer rate by using block moving back and forth on a heated surface in a channel flow, this phenomena causes the heat transfer rate of the heat surface to be enhanced because it destroys and suppresses the velocity and thermal boundary layers on the heat surface periodically, studied by (Wu-Shung 2000, 2001, 2007, 2010). (Tae 2008) introduced a novel heat sink with moving fins inserted between cooling fins for electronics cooling application. In this heat sink, they replace the fan module by moving fins inserted in the intervals of fixed fins (cooling fins) for generating fluid flow. The relative motion of the moving fins to the cooling fins, causes a heat dissipation from the cooling fins to the coolant and discharge of the heated coolant. Experimental results show that measured flow rates of the coolant are almost linearly proportional to the rotating speed of the moving fins. The thermal performance of the scroll heat sink is compared to that of an equivalent plate-fin heat sink under the same Reynolds number. The scroll heat sink has at least 14% lower thermal resistance than that of the equivalent plate-fin heat sink in the range 0.135m/s $\le$ V $\le$ 0.56m/s. (H.K. Ma 2009) proposed innovative vibrating fins and successfully built three-dimensional transitional models to investigate design performance. The basic fins, with dimensions of 30mm in length, 1mm in thickness, and 100mm in width, are designed on a 2mm thick finned base. The displacement ratio and frequency of the vibrating fins are 0.9 and 10 HZ. The results showed that the performance of the vibrating fins is strongly affected by dimensions, vibrating frequency, pitch, and amplitude of the fins. As gravity parallels the surface of the finned base, the vibrating fins can perform better than traditional fins on heat dissipation due to the lower convective obstacle. (Subhrajit 2009) carried out a numerical investigation using finite volume method for a 3D oscillating fin to disturb the thermal boundary layer to enhance forced convection heat transfer from conventional heat sink. The fin dimensions were (0.5" *1"*0.01") and substrate was (1"*1"). The power supplied to the substrate was 20W. It was found that such oscillations lead to tip-leakage vortices from the fins. A local up wash is presented on the fin lateral surfaces, hence increasing the overall heat transfer rate. This enhancement in heat transfer was demonstrated through an increase in the time-averaged Nusslet number (Nu) on the fin surface. The objective of this work is to design, manufacture and test a new cooling system. This system is formed of piezoactuated plate fins integrated with a hot substrate subjected to forced convection. The cooling of hot electronic devices integrated with fixed fins may be failed for high heat generation rate, so oscillating these fins will disturb the development of boundary layer and enhances the heat dissipation effectively and reliably from hot substrate to the stream wise flowing fluid. The dependence of temperature distribution, and local Nusselt number of fixed or oscillated fin is quantified on parameters like free stream velocity, fin height, fin spacing and frequency of oscillation of the fin. The impact of the presence of single or triple fixed and oscillated fins has also been highlighted.

EXPERIMENTAL TEST RIG

A designed manufactured experimental model used in this study is consist of: square copper substrate of (100mm, long, 100 mm wide and 11 mm thickness); copper flat plate fin of (0.075mm) thickness and (100mm) long, and heating unit. A schematic diagram of the experimental model is shown in Fig.(1). The effect of single fin and triple fins is investigated for: fins height (H=35mm and 50 mm), fins spacing (W=3mm and 6mm). The piezoelectric actuators in the form of a rectangular shape pair are mounted on each end of the fin surfaces by a heat tape and supported from bottom. The fin oscillation disrupts the growth of the thermal boundary layer and serves to bring about heat transfer enhancement between the fluid and the fin surfaces. A four cartridge electrical heaters are built in the substrate to obtain constant heat flux. The input power to the heater is controlled by a variac transformer and measured by an in-line digital power and energy monitor. A PID controller type WATLOW EZ-ZONE was used to adjust the electrical current supplied to the cartridge heater. The heater output power is 1200 W at (voltage of 250V and current of 4.8A). The substrate-fin assembly is tested for different velocities in a subsonic open type wind tunnel of rectangular cross sectional test section. The test section walls are made of 5mm thickness of plexi-glass and its' dimensions are (340mm height, 340mm wide and 600mm long) is shown in fig. (2) . The wind tunnel used in this work consists of: filter, nozzle with flow straightners, test section, damping chamber, diffuser, and an AC fan. A thermal image camera (H2600) series has the infrared detector of 640*480 pixels is used to view and record the temperature distribution on both the fin and the substrate is shown in figs. (3 and 4) for different frequencies. Isofrax paper (1260°c k=0.073 W/m K) has been used to thermally insulate the bottom surface of the substrate followed by Insulfrax blanket (k = 0.07 W/m K) of 80 mm thickness. The experiments of the fluid flow and heat transfer were carried out for each tested model for seven values of Reynolds number based on hydraulic diameter of test section (1.0886*104 to 1.3*105). A hot-wire anemometer is used to measure air velocity and temperature at inlet of test section. Power supply is used to generate and amplify signals sent to the piezoelectric actuator to oscillate the plate fin. The range of input signal frequency to the piezoelectric actuator was (5HZ to 50HZ).

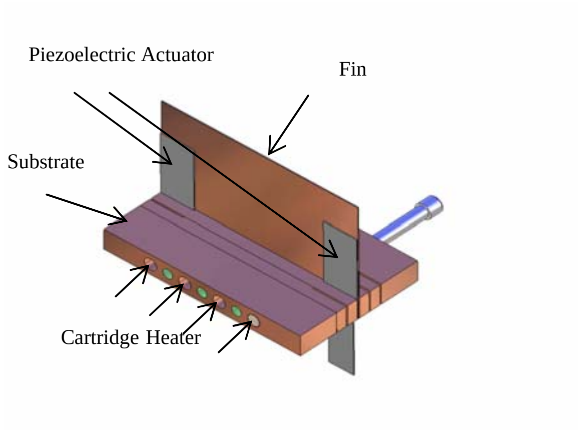


Fig. (1): The structure of oscillating fin.

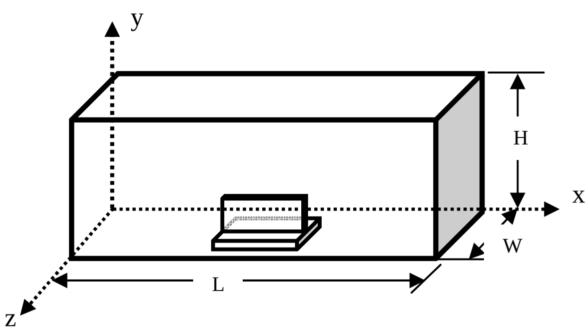


Fig. (2): Geometry of the Problem

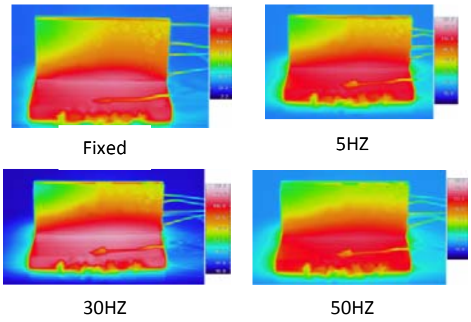


Fig. (3): Thermal Image , I.P=50W and u=0.5m/s (h=50mm and dis.=3mm)

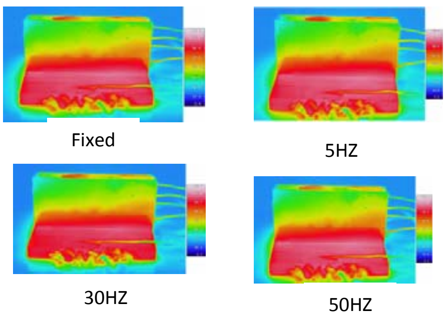


Fig. (4): Thermal Image , I.P=50W and u=0.5m/s (h=50mm and dis.=6mm)

DATA REDUCTION

In order to investigate the convection heat transfer in the test section, the heat transferred to the air is required to be determined. Considering the energy balance, the total heat flux q_{tot} generated by the heated plate is dissipated into the convective heat flux from the test section to the air q_{conv}, conductive heat loss through the insulation materials q_{cond} and radiant heat loss from the test section q_{rad}, as shown below (Saad 2002):

$$q_{tot}=q_{conv}+q_{cond}+q_{rad}, \qquad (1)$$

The essential quantities, which are obtained in the study of heat transfer are; the convective heat transfer (q_c), the local Nusselt number over the tested fins ($Nu_{x,f}$), the Average Nusselt number (Nu_f). The total heat generated from the heater (q_t) is distributed into the heat transferred by convection to the flowing air (q_c) and the heat losses through the insulation surrounds and under the tested model (q_L) as follows:

$$q_c = q_t - q_L \qquad (2)$$
$$q_L=q_s + q_p + q_b + q_{ft} + q_r \qquad (3)$$

Where q_s, heat lost by conduction in the stream-wise (X) direction (W); q_p, heat lost by conduction in the spanwise (Z) direction, (W); q_b, heat lost by conduction from the bottom surface of the base plate, (W); q_{ft}, heat lost from the fin tips surface (W); q_r, heat lost by radiation (W). Then, the value of losses is small compared with the heat input value and it can be neglected. Consequently the total heat transferred by convection to the flowing air (q_c) equals the heat generated by the heater (q_t):

$$q_c=q_t \qquad (4)$$

The heat transferred by convection to the air stream is divided into two parts:

The convection heat transfer from fins (q_{cf}) which can be calculated as

$$q_{cf} = q_c \ A_{ct,f}/A_{ct} \qquad (5)$$

where

$A_{ct,f}$, total cross-sectional area of fins at the base (sum of surface area of the fins tip) (m^2)

Act, is the total cross-sectional area of the base plate and fins (m^2), and the convection heat transfer through one fin (q_f) is given by

$$q_f = q_{cf}/N \qquad (6)$$

where N is Number of fins

The convection heat transfer from substrate area which exposed to flowing air (without fins) (q_{cb}) which can be calculated as

$$q_{cb} = q_c \, A_{cb,f}/A_{ct} \qquad (7)$$

Where

$A_{cb,f}$, total free surface areas of the base plate (without fins), (m^2).
The local heat transfer coefficient over the tested fins can be calculated by applying the Newton's law (F.B. Incropera 1985)

$$h_{x,f} = q_f / A_{sf} \, (T_f - T_i) \qquad (8)$$

where
T_f, Local fin temperature (^oC); T_i, air inlet temperature (^oC); q_f, convection heat transfer through one fin (W); A_{sf}, total surface area of the tested fin (m^2).

$$Nu_{x,f} = h_{x,f} \, .x/k_a \qquad (9)$$

where
$h_{xf,x}$, is local heat transfer coefficient over the tested fins (W/m^2. ^oC); ka, is air thermal conductivity at T_i (W/m. ^oC).
The fin-average Nusselt number can be defined as

$$Nu_f = h_f L / K_a \qquad (10)$$

where h_f is fin average heat transfer coefficient which is calculated as:

$$h_f = 1/A_{sf} \int_A h_{f,x} dA_{sf} \qquad (11)$$

Newton Raphson Method is used to calculate the average heat transfer coefficient over the tested fins such that:

$$h_f = \frac{h}{3} [h_{f,x0} + 4 \sum_{i=1,3,5}^{n-1} h_{f,xi} + 2 \sum_{i=2,4,6}^{n-2} h_{f,xi} + h_{f,xn} \qquad (12)$$

where h is division size of the fin length = L/n

RESULTS AND DISCUSSION

The presented temperature distribution has been extracted from thermal image taken for substrate integrated with fixed and oscillated fins. The following cases have been studied in this work:

Case 1: fixed triple fins (height =50mm and fin spacing =3mm)Case 2: oscillated triple fins (height =50mm and fin spacing =3mm)

Case 3: fixed triple fins (height =35mm and fin spacing =3mm)

Case 4: oscillated triple fins (height =35mm and fin spacing =3mm)

Case 5: fixed triple fins (height =50mm and fin spacing =6mm)

Case 6: oscillated triple fins (height =50mm and fin spacing =6mm)

Case 7: fixed triple fins (height =35mm and fin spacing =6mm)

Case 8: oscillated triple fins (height =35mm and fin spacing =6mm)

Case 9: single fixed fin (height =50mm)

Case 10: single oscillated fin (height =50mm)

1- TEMPERATURE DISTRIBUTION

The front view of the fin is divided into four rows and five columns as shown in Fig.s (5and 6). The coordinate of each node has been transformed from the thermal image to digital value of temperature using thermal image camera software.

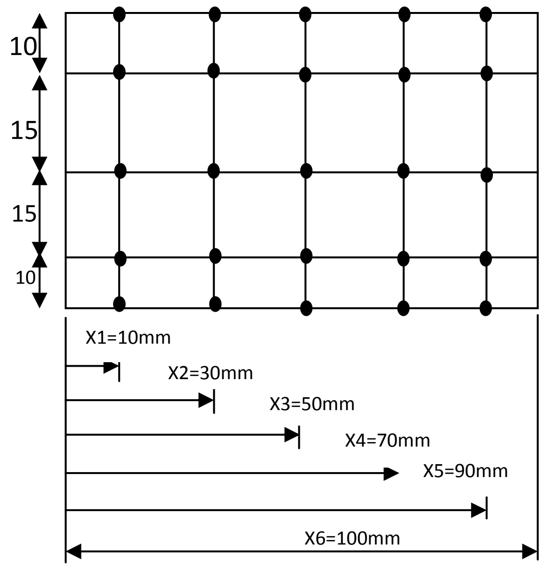


Fig. (5): Locations of temperature measured by thermal image (fin height =50 mm)

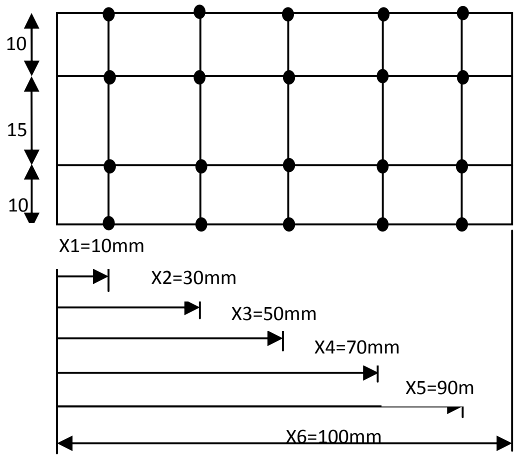


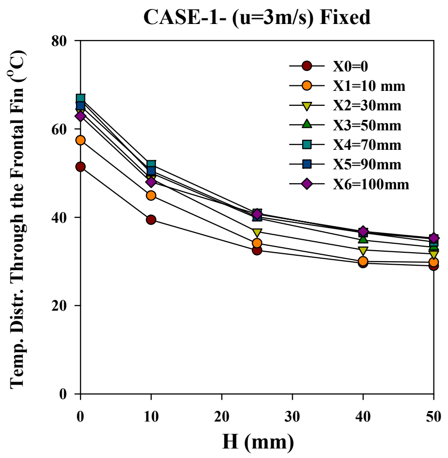
Fig.(6): Locations of temperature measured by thermal image (fin height =35 mm)

Figures (7, and 8) show the temperature distribution along the front fin height (when the substrate integrated with triple fins) for air free stream velocity of 3m/s and fins spacing =3mm. It is clear that the temperature decreases toward the tip of the fin. There is inconsiderable change in temperature variation for oscillated fin compared with that for fixed one except for H=35mm. Increasing the fin spacing from 3mm to 6mm has little Impact on the values of temperature distribution. The temperature distribution along the fins is shown in Fig.s (9 and 10). The fin leading edge (near the tip) was the coldest region while the fin root near the trailing edge was the hottest. Figure (11) shows the same trend of temperature distribution along the single fin height with approximately constant stream wise temperature variation. Quantitatively colder temperatures are indicated with increasing number of fins as shown in Figures (9 a, b) (10 a, b) and (11 c, d).

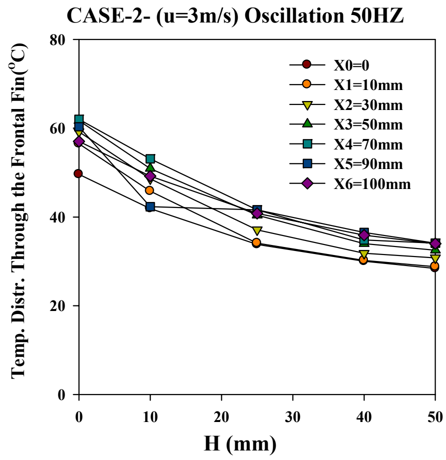
2) NUSSELT NUMBER

The calculated local heat transfer coefficient is increased toward the fin tip as shown in Fig. (12). Fin oscillation enhances the heat transfer as shown in this figure. Higher values of local Nusselt No. are found near the fin trailing edge as presented in Fig. (13). From this figure it is clear that stimulation the fins to oscillate rises the value of hf,x. Increasing the inter-fin spacing causes an increase of mass flow rate between fins and good washing for the fins wall takes place, hence an increase of local Nusselt No. was obtained consequently as indicated in Fig.s (12 and 14). Fig.(15) shows the same trend of local Nusselt No. on single fixed and oscillated fin.

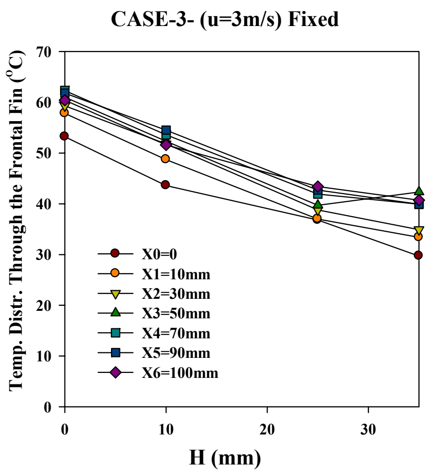
Figures (16 and 17) show a considerable increase in average Nusselt number (Nu) on the frontal fin is obtained with increasing inlet air velocity from 0.5m/s to 3m/s. Nusselts number increases at the tip of the fin much more than fins' root. Also an enhancement in heat transfer is obtained with increasing the frequency. For 0.5m/s, the best performance for oscillation was at frequency 50HZ, while for 3m/s it was at frequency 30HZ. The static fin develops convection driven thermal boundary layer on either side of the fin. It acts as a resistance to further heat dissipation. This layer is expected to rupture under the influence of local cross wise perturbation in the form of sinusoidal motion imposed on the fin by the piezoelectric actuator. As the fin starts to oscillate, one of the lateral surfaces of the fin behaves as the pressure side with the opposite surface being as the suction side. This gets reversed every half time period. As the fin is on its way to move, the fluid close to the fins' pressure surface is pushed by the fin and flows transversely to the stream wise direction. As a result, the heat transfer is enhanced. Conversely, the fluid near the opposite surface of the fin simultaneously replenishes the vacant space induced by the movement of the fin. Most of the fluid near the suction surface of the fin is difficult to catch up to this surface of the fin simultaneously, hence a small recirculation zone is developed, and a re attachment flow is presented. As a result a heat transfer enhancement is obtained. It is clear from Fig.s (16,and 17) that for cases 3, 4, 6 and 7, no significant effect of oscillation frequency on average Nusselt number is found. For cases 1, 2, 5 and 6, the results show obviously an increase in average Nusselt number exists with increasing the frequency, especially with 50HZ. For case 9 and case 10, the results show the increase of the average Nusselt number with increasing the frequency is very low for higher air velocity.



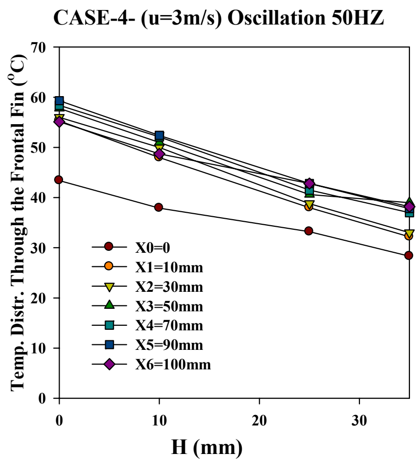
a) Height=50mm



(b) Height=50mm

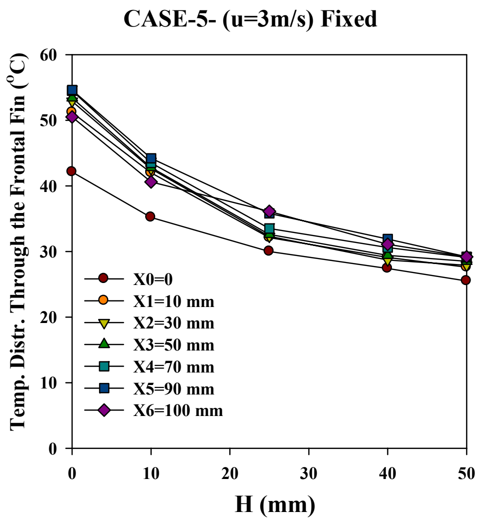


(c) Height=35mm

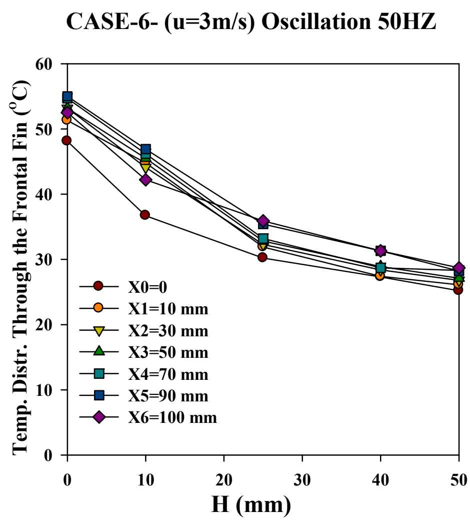


(d) Height=35mm

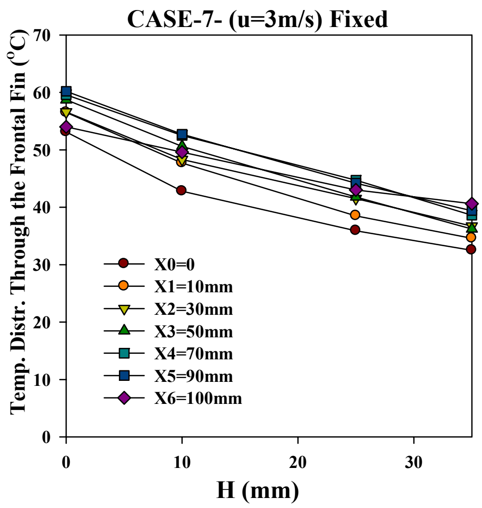
Fig. (7): Temperature distribution along the height of the frontal fin for (fins spacing=3mm).



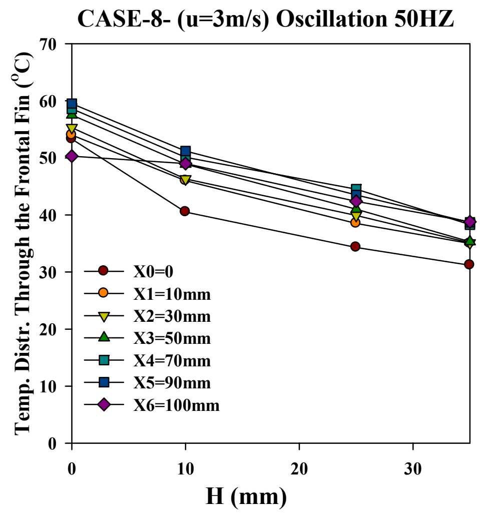
(a) Height=50mm



(b) Height=50mm



(c) Height=35mm



(d) Height=35mm

Fig.(8): Temperature distribution along the height of the frontal fin for (fins spacing=6mm).

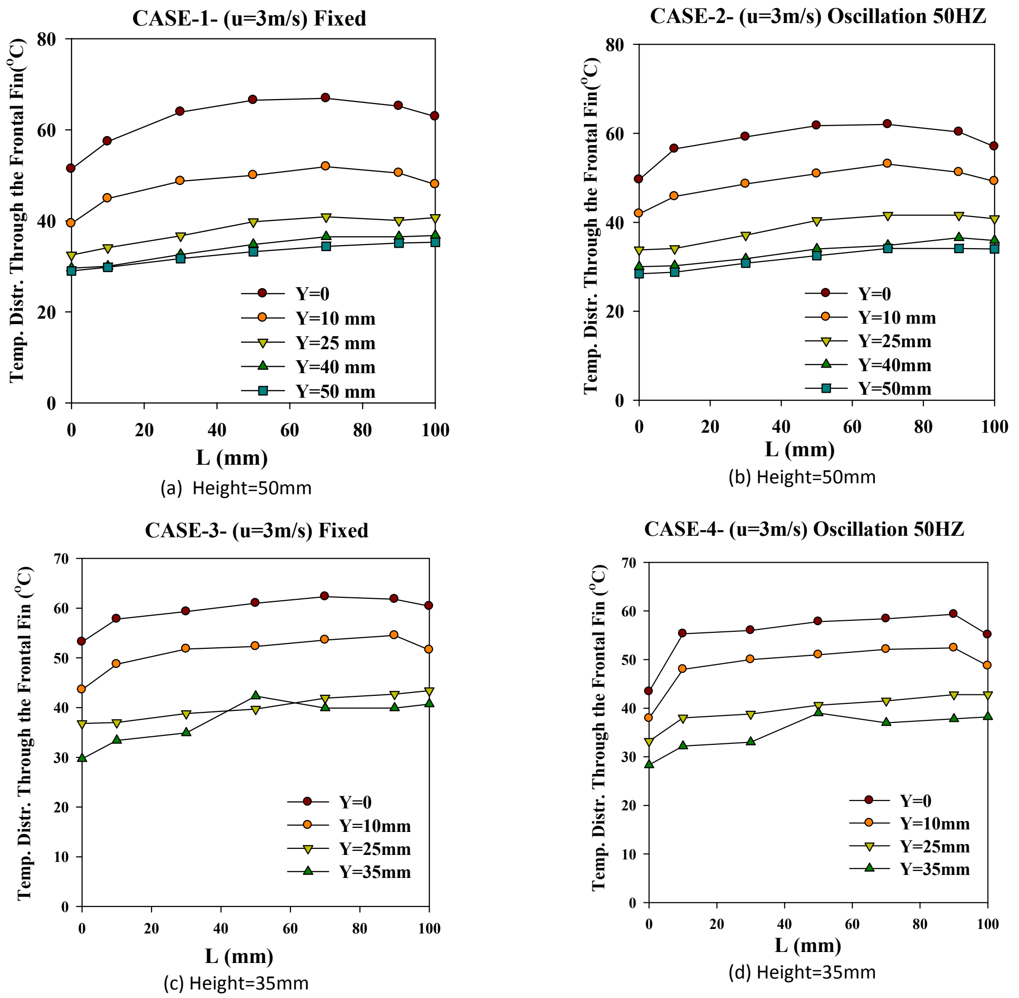
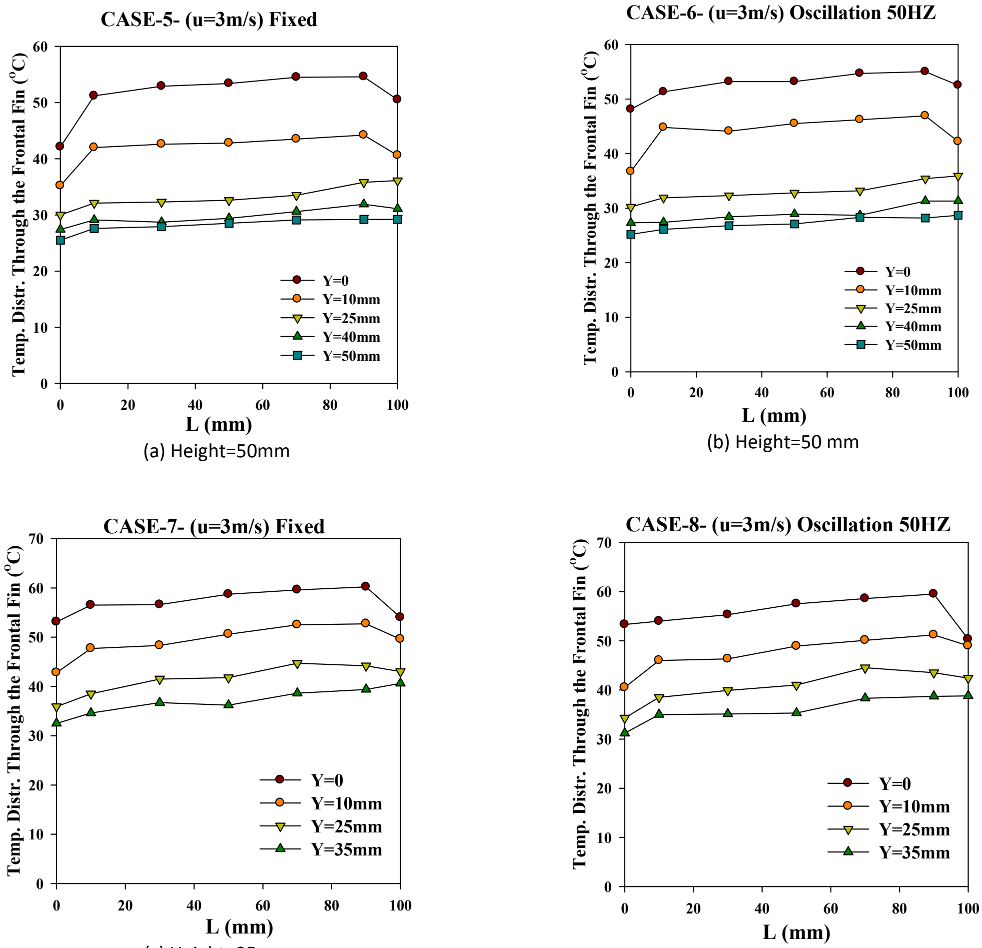


Fig. (9): Temperature distribution along the frontal fin for (fins spacing=3mm).



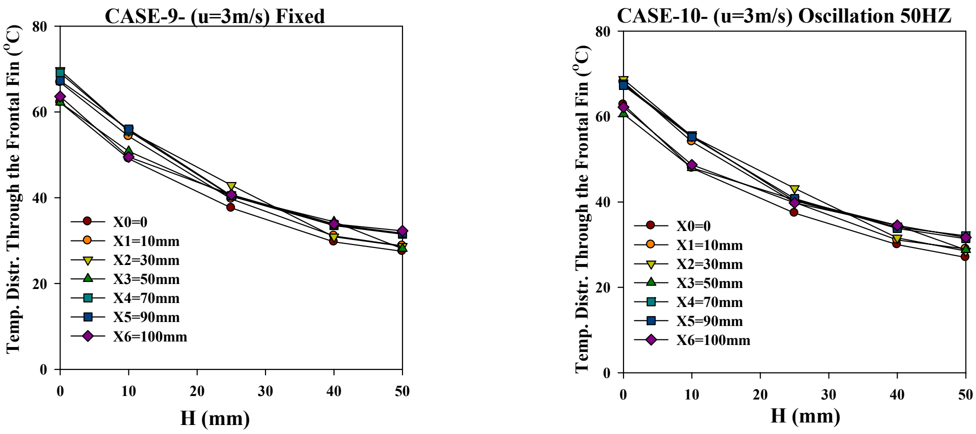
(a) Height=50mm

(b) Height=50 mm

(c) Height=35mm

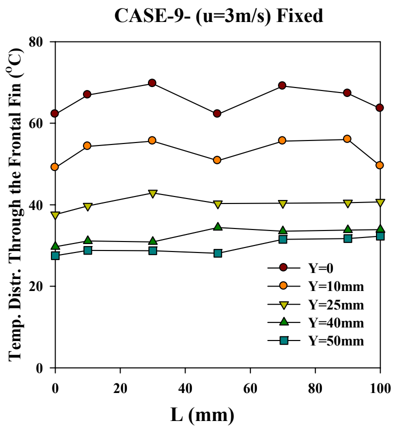
(d) Height=35mm

Fig. (10): Temperature distribution along the frontal fin for (fins spacing=6mm).

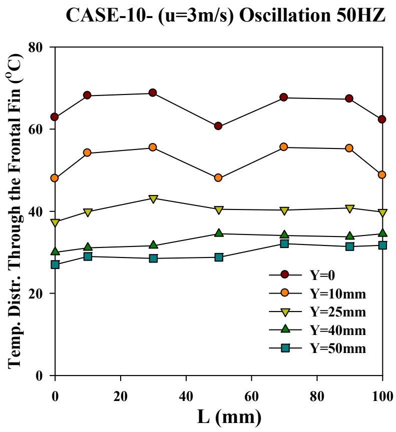


(a)Single fixed fin

(b) Single oscillated fin

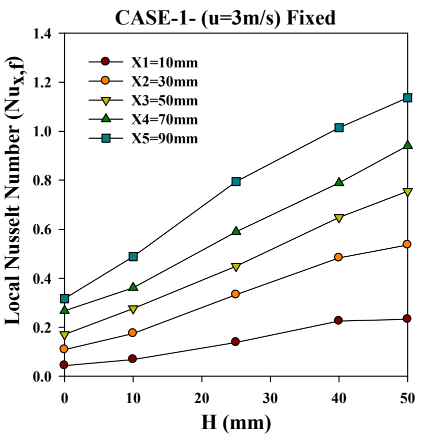


(c) Single fixed fin

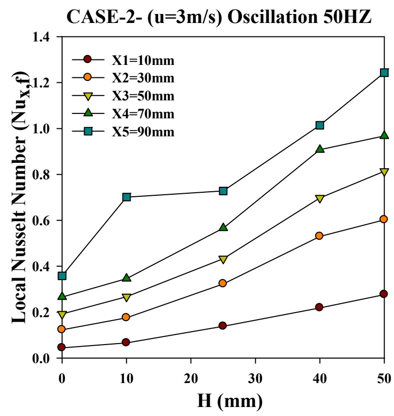


(d) Single oscillated fin

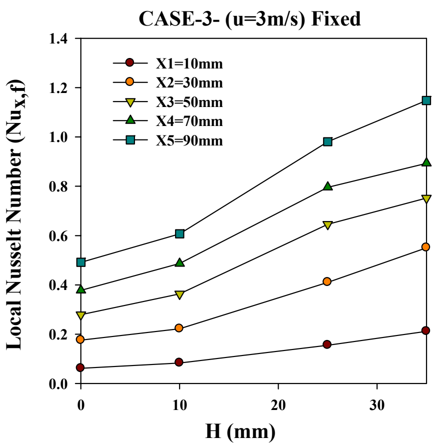
Fig.(11): Temperature distribution along the fin; a,b) height; c,d) length



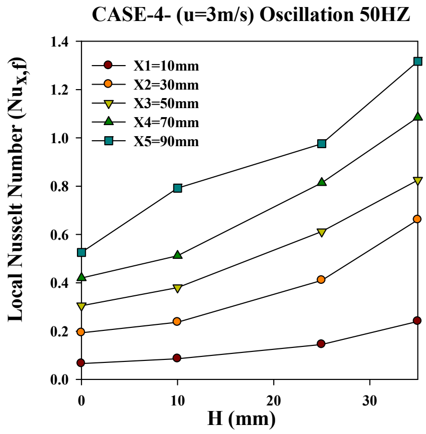
(a) Height=50mm



(b) Height=50mm

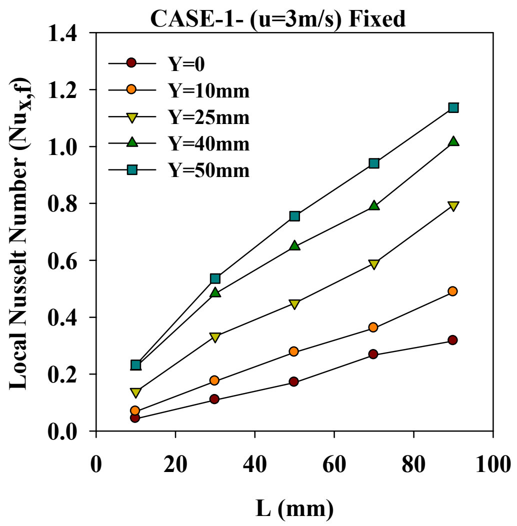


(c) Height=35mm

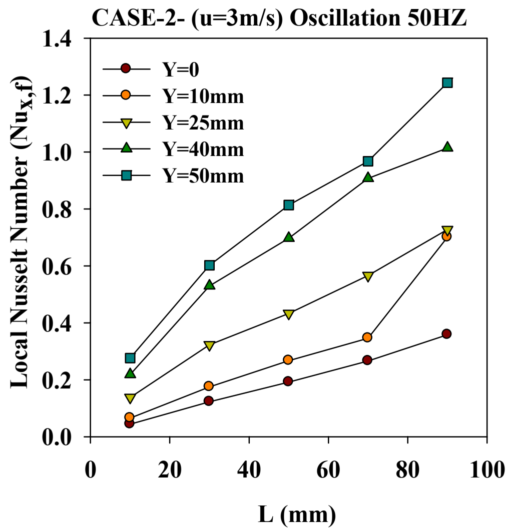


(d) Height=35mm

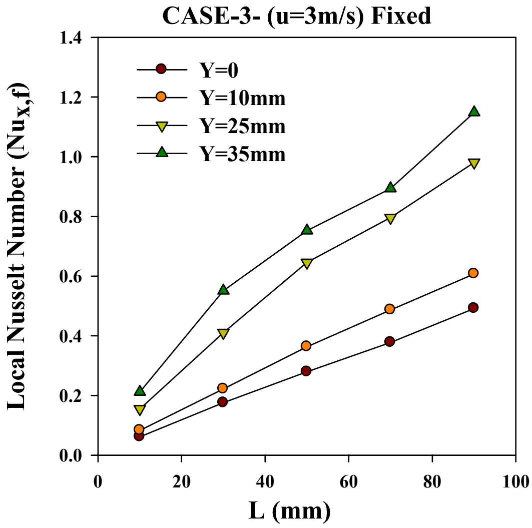
Fig. (12): Variation of local Nusselt number along the height of the frontal fin (fins spacing=3mm).



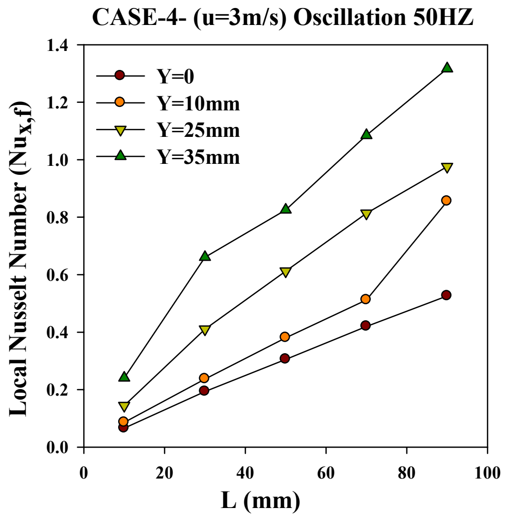
(a) Height=50mm



(b) Height=50mm

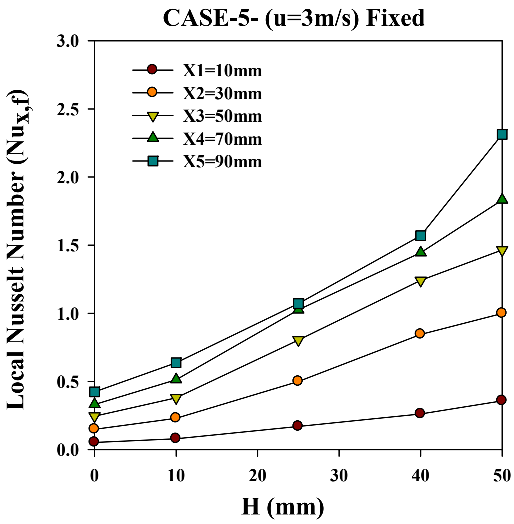


(c) Height=35mm

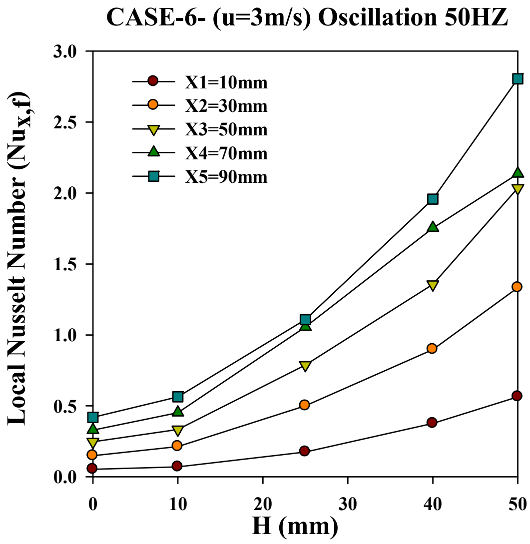


(d) Height=35mm

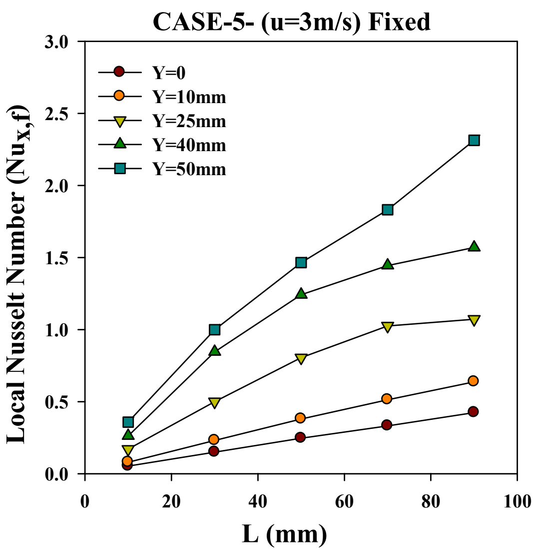
Fig.(13): Variation of local Nusselt number along the frontal fin (triple fins with fins spacing=3mm).



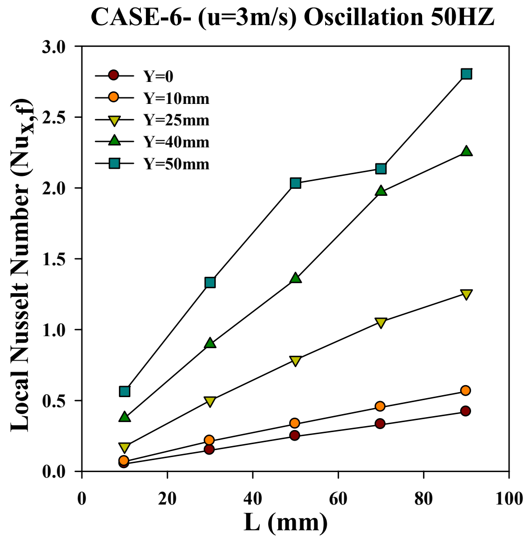
(a)Height=50mm



(b) Height=50mm



(c) Height=35mm



(d) Height=35mm

Fig.(14): Variation of local Nusselt number along the frontal fin (fins spacing=6mm).

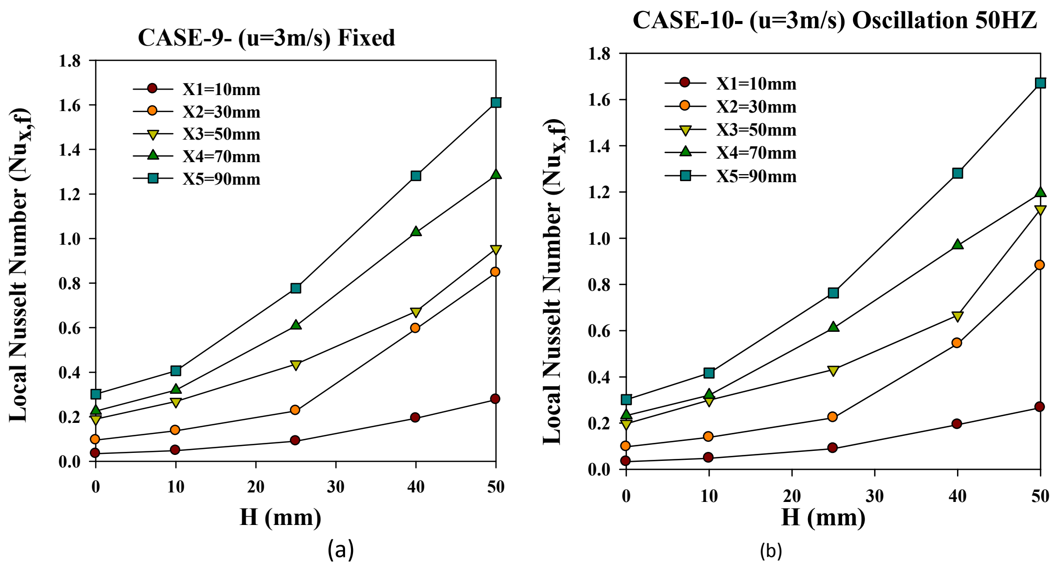
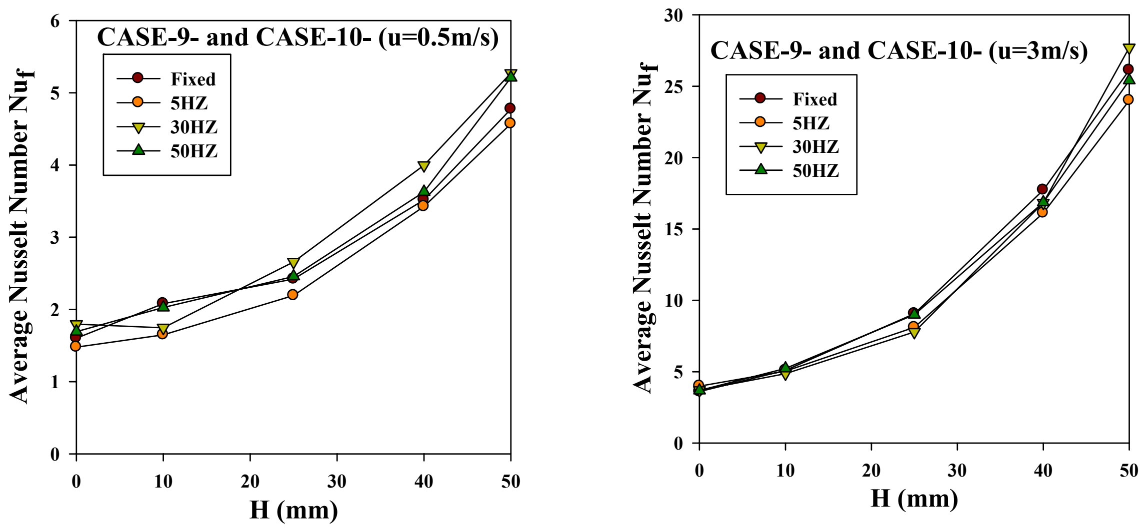
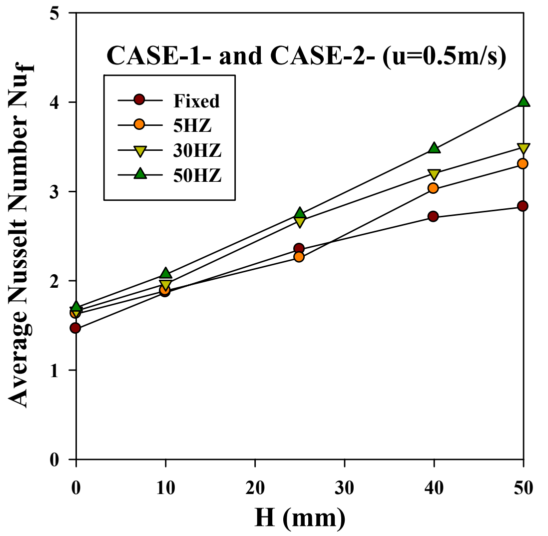


Fig. (15): Variation of local Nusselt number along the height of a) single fixed fin; b) single oscillated fin.

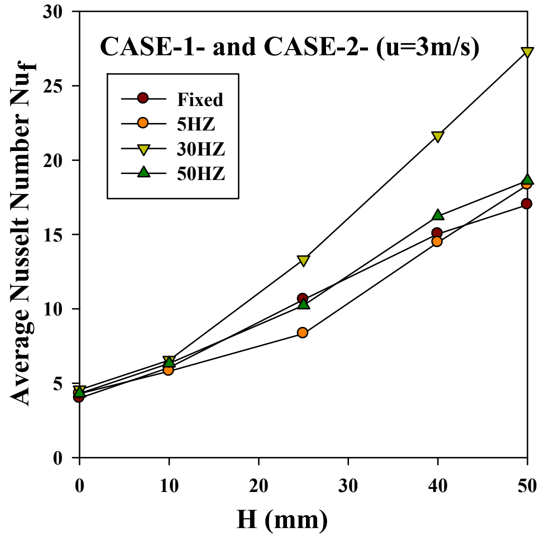


(a) Single fin, u=0.5 m/s

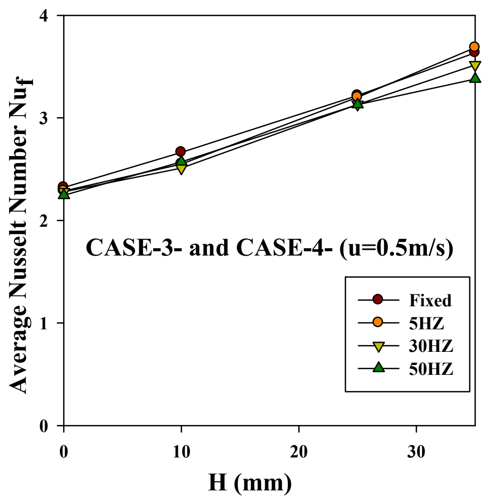
(b) Single fin, u=3 m/s



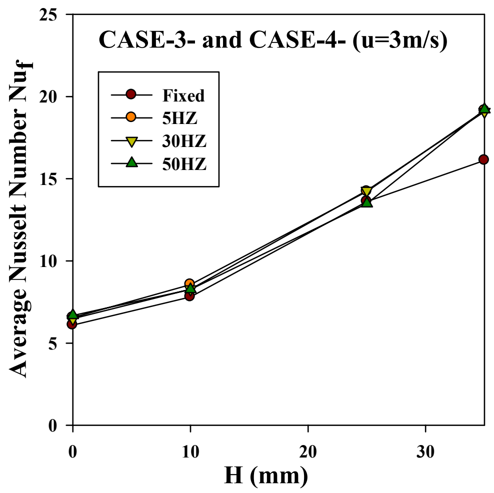
(c) Triple fins, u=0.5 m/s, spacing=3mm



(d) Triple fins, u=3 m/s , spacing=3mm

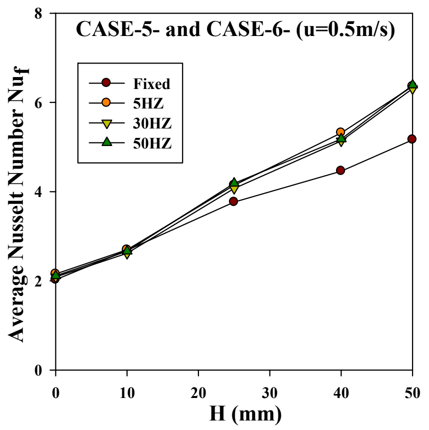


(e) Triple fins, u=0.5 m/s , spacing=6mm

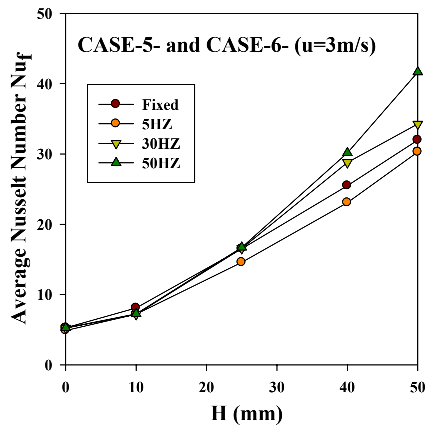


(f) Triple fins, u=3 m/s , spacing=6mm

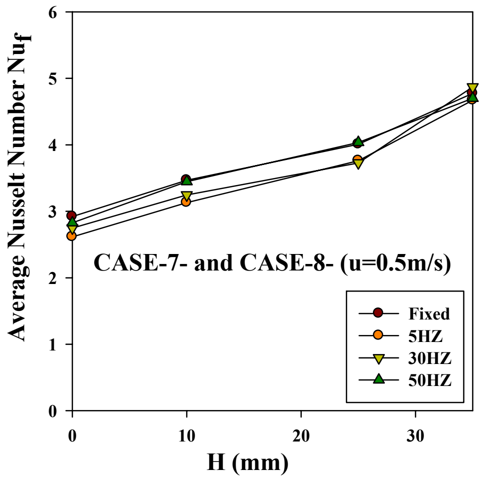
Figure (16): Variation of average Nusselt number along fins' height for different frequencies



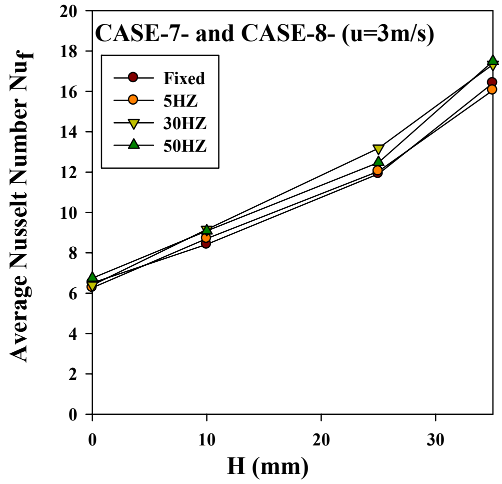
(a) Triple fin, u=0.5 m/s, spacing=6mm,H=50mm



(c) Triple fin, u=3 m/s, spacing=6mm, H=50mm



(c) Triple fins, u=0.5 m/s, , spacing=6mm,H=35mm



(d) Triple fins, u=3 m/s , spacing=6mm,H=35mm

Fig. (17): Variation of average Nusselt number along fins' height for different frequencies

CONCLUSION

According to the previous discussion the following conclusions can be extracted:

1. There is an effect for the height, Reynolds number, frequency and distance between the fin on heat transfer.
2. Triple fin (case 1) is the best case for enhancing heat transfer and the triple fin (case 2) gives minimum heat transfer with respect to other height.
3. Heat transfer increases with increasing frequency and stream wise air velocity.
4. Maximum heat transfer occurs at fins leading edge near it tip i.e X/L=0, Y/H=1 for all investigated heights.
5. Average heat transfer through the middle fin is enhanced by (4-25) % by oscillating the fin with 50HZ.
6. More effective enhancement of heat transfer utilizing fin oscillation is obtained for low stream wise air velocity (0.5m/s to 2m/s).
7. Increasing stream wise air velocity attenuates the enhancement in heat transfer due to oscillation because the air works as a damper to the oscillation.
8. The enhancement of heat transfer increases significantly with increasing oscillation frequency from 5HZ to 50HZ for all cases, except for height 35mm, because this fin height is not effective with oscillation, while, this height is found an optimal height with fixed fin only.
9. The effect of the inter-fin space obviously appears in the thermal image, especially with a distance of 6mm, the heat transfer through the surface of the frontal fin is improved due to the increase of flow rate between fins.

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Nomenclature

$A_{ct.f}$	Total Cross-Sectional Area of Fins at the Base (m^2)
Act	Surface Area of the Base Plate (m^2)
$A_{cb.f}$	Total Free Surface Areas of the Base Plate (m^2)
A_{sf}	Surface Area of the Tested Fin (m^2)
dis.	Distance Between the Fin (mm)
Distr.	Distribution
$h_{x.f}$	The Local Heat Transfer Coefficient ($W/m^2.^oC$)
$h_{xf.x}$	Local Heat Transfer Coefficient over the Tested Fins ($W/m^2.^oC$)
h_f	the Average Heat Transfer Coefficient over the Fin ($W/m^2.^oC$)
H	Height of the Fin (mm)
Ka	Thermal Conducyivity of the Air (W/m.k)
$Nu_{x.f}$	Local Nusselt Number over the Tested Fins
Nu_f	The Fin-average Nusselt number
N	Number of fins
q_{tot}	the Total Heat Flux (W)
q_{conv}	Convective Heat Flux (W)
q_{cond}	Conductive Heat Loss (W)
q_{rad}	Radiant Heat Loss (W)
q_c	the Convective Heat Transfer (W)
q_t	The Total Heat Generated from the Heater (W)
q_L	the Heat Losses (W)
q_s	Heat Lost by Conduction in the Stream-Wise (X) Direction (W)
q_p	Heat Lost by Conduction in the Spanwise (Z) Direction (W)
q_b	Heat Lost by Conduction From the Bottom Surface of the Base Plate (W)
q_{ft}	Heat Lost From the Fin Tips Surface (W)
q_r	Heat Lost by Radiation (W)
q_{cf}	the Convection Heat Transfer Through the Fins (W)
q_f	the Convection Heat Transfer Through one Fin (W)
q_{cb}	the Convection Heat Transfer Through the Free Area (W)
t	Thickness of the Fin (mm)
T_f	Temperature of the Fin at (x) (^oC)