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The Effect of Circular Perforation on a V-Corrugated Fin Performance under Natural Convection

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ABSTRACT

An experimental investigation has been made to study the influence of using v-corrugated aluminum fin on heat transfer coefficient and heat dissipation in a heat sink. The geometry of fin is changed to investigate their performance. 27 circular perforations with 1 cm diameter were made. The holes designed into two ways, inline arrangement and staggered in the corrugated edges arrangement. The experiments were done in enclosure space under natural convection. Three different voltages supplied to the heat sink to study their effects on the fins performance. All the studied cases are compared with v-corrugated smooth solid fin. Each experiment was repeated two times to reduce the error and the data recorded after reaching the steady state conditions. The results showed that the v-corrugated fin dissipate heat twice and triple times than flat plate mentioned in past research with the same dimension. Also, the inline perforated fin gave higher enhancement percentage than solid one by 15, 32 and 36% for 110, 150 and 200 V voltages supplied. Finally, the staggered perforation arrangement gave the higher enhancement percentage with 22, 42 and 45% for the same voltages supply.

Keyword: Fin performance, heat sink, natural convection, perforated fin, corrugated fin

تأثير الثقوب الدائرية على اداء زعنفة مموجة تحت الحمل الحراري الطبيعي

مها علي حسين
مدرس مساعد
قسم هندسة تقنيات التبريد والتكييف-كلية دجلة الجامعة

الخلاصة

تم اجراء تجربة عملية لدراسة تأثير استخدام زعنفة مموجة مصنوعة من الالمنيوم على معامل التوصيل الحراري و التبدد الحراري لجهاز يستعمل لتقليل الحرارة الزائدة. اجريت تعديلات على تصميم الزعنفة لدراسة ادائها بعد التعديل. تم عمل 27 ثقب دائري بقطر 1 سم. ثم وزعت الثقوب بطريقتين: الاولى مرتبة في خط عمودي والثانية موزعة على حواف التموج للزعنفة. التجربة اجريت في حيز مغلق وتحت تأثير الحمل الحراري الطبيعي. استخدمت ثلاث قيم مختلفة للفولتية لدراسة تأثير اختلافها على اداء الزعانف. جميع الحالات تم مقارنتها مع الزعانف المموجة الصلبة. كل تجربة تم اعادتها مرتين لتقليل الخطأ و تم استحصال القراءات بعد الوصول الى مرحلة الاستقرار الحراري. اظهرت النتائج عند مقارنتها بنتائج سابقة ان الزعنفة المموجة تبدد حرارة ضعف الزعنفة المسطحة لنفس الحيز. ايضا الزعانف التي تحتوي على الثقوب المرتبة بشكل

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خطوط عمودية حسنت من النتائج التي استحصلت من الزعنفة الصلبة بمقدار 15، 32، 36% عند تجهيز ثلاث فولتيات مختلفة 110، 150، 200 فولت على التوالي . اخيرا الزعانف ذات الثقوب الموزعة على حواف الزعنفة اعطت اعلى قيم لنسبة التحسينات بنسبة 22، 42، 45% عند تجهيز ثلاث فولتيات مختلفة .
الكلمات الدالة: اداء الزعنفة، جهاز تبديد الحرارة، الحمل الطبيعي، زعانف مثقبة، زعانف مموجة

1. INTRODUCTION

In many industries, excess heat considered as a big problem and can cause damage in industrial parts. Therefore, heat transfer enhancements are essential to overcome this issue. Types of improvements can be classified as active, **Jadhav, et al., 2016,** and passive, **Sonawane, et al., 2016**. The active techniques involve external power input, but the passive techniques geometrical enhancements were introduced.

An extended surface like fin is one of the passive methods; it is used to dissipate heat from hot surfaces to the surrounding. New fins geometry are required to achieve maximum heat transfer rate with less cost. Many types of research have been made on improving the fins shapes, size and material, **Bahadure, and Gosavi, 2014, Prabisha, and Ramesh, 2015**. Experiments were done with various types of fins, such as continues, interrupted, wavy and perforated fins, **Hussain, et al., 2011, Mostafavi, 2012**.

Al-Doori, 2011, investigated the rate of heat transfer to a perforated rectangular fin plate in natural convection condition. The research focused on the number of perforation holes and their effect on the temperature distribution along the fin. The fins were divided according to holes number; each hole was 12 mm in diameter. The results showed that fins with a larger number of perforations have heat transfer rate more than a small number of holes. **AlEssa, 2012,** examined the effect of holes enhancement on heat transfer rate for a rectangular fin with square perforation and those for non-perforated fin under natural convection. The results indicated that the holes dimension and lateral spacing have a strong influence on heat dissipation rate. **Jassem, 2013,** investigated the temperature distribution for five aluminum fins under natural convection. The first fin is solid rectangular plate while the other perforated by different shapes (circular, square, triangular and hexagonal). All perforation area is 113 m^2. The results indicated that the triangular perforation shape gave the best heat transfer coefficient than the other shapes. **Ibrahim, et al., 2016,** carried out an experimental investigation to study the heat dissipation from a heat sink by using 15 rectangular aluminum fins with circular perforation. The number of holes in each fin is constant while the diameter varies from one fin to another. The results showed that as the diameter of the holes increased, the heat transfer coefficient also increased.

The present study involves geometrical modifications to fin by making circular perforation through corrugated fin and investigates experimentally. The effect of diameter and place of the perforation on the heat transfer rate along the wavy fins under natural convection was investigated. It was stated that the free convection coefficient for a non-flat plate is higher by 50% to100% than flat plate **Shaikh, et al., 2015**. The heat dissipation along two perforated fins under natural convection is compared to the corresponding solid fin.

2. EXPERIMENTAL ANALYSIS

The experimental work can be divided into two main parts which are:

2.1 The Rig Description

The schematic and photograph of the experimental rig were shown in **Fig. 1** and **Fig. 2** respectively. It mainly consists of a heat sink with one heating element (800)Watt concentrated on it. The heat sink used was an aluminum cylinder with 40 cm long and 6 cm in diameter. Six aluminum V-corrugated fins were fixed vertically around the heat sink with equal spacing. The fins dimensions were 25 × 25 cm and 2mm thickness. **Table 1** shows the specifications and geometry of the fins. Power supplied regulator was used to regulating the voltage of the heating element to change the source temperature. 36 thermocouples of type K were placed on the fins surfaces with equal space, while one thermocouple fixed on the heat sink surface and environment temperature is also measured. The whole thermocouples connected to an electrical display screen.

2.2 The Experimental Procedure

The following procedures were done during the experiment:

1- Thermocouples are connected to the data display

2- Switching the voltage regulator on.

3- Setting the voltage to 110 V, when the heat sink becomes hot, the heat transfers to the fins equally and dissipate to the environment. After approximately 1 hour, the steady state was achieved where no change in temperature along the fins, at that time the temperature was recorded.

4- The same previous procedure was repeated with two different voltages 150 V and 200 V to change the source temperature.

5- The data collected from above steps were used to calculate the temperature drop, heat transfer coefficient and heat dissipation for the fins. **Fig. 3** illustrates the experimental parameters measured in the experiments.

3. DATA REDUCTION

The heat transfer analysis of both wavy solid and wavy perforated fins is determined by using analytical solution which is based on the following assumptions:

1- The fin doesn’t generate heat.

2- Constant conductivity fin value.

3- Uniform base and ambient temperature.

4- Uniform heat transfer coefficient along the fin.

3.1 Solid and Perforated Fin Analysis

The temperature gradients of the fin in the x-direction and according to the above assumption can be explained as**, Holman, 2010 and Lienhard, 2008**:

$$-\mathrm{kA}\left.\frac{\mathrm{dT}}{\mathrm{dx}}\right|_{\mathrm{x+dx}} + \mathrm{kA}\left.\frac{\mathrm{dT}}{\mathrm{dx}}\right|_{\mathrm{x}} + \mathrm{h(pdx)(T - T_\infty)_x} = 0 \quad (1)$$

$$\frac{d^2T}{dx^2} - \frac{hp}{kA}(T - T_\infty) = 0 \quad (2)$$

$$\theta = T - T_\infty$$

$$\frac{d^2\theta}{dx^2} - \frac{hp}{kA}\theta = 0 \quad (3)$$

The fin is of finite length and loses heat by convection from its end,

Let $m = \sqrt{\frac{hp}{kA}} \approx \sqrt{\frac{2h}{kt}}$

The general solution for Eq. (3) may be written as:

$$\theta = C_1 e^{-mx} + C_2 e^{mx} \quad (4)$$

$$\frac{T - T_\infty}{T_o - T_\infty} = \frac{\cosh m(L-x) + \left(\frac{h}{mk}\right)\sinh m(L-x)}{\cosh mL + \left(\frac{h}{mk}\right)\sinh mL} \quad (5)$$

The heat conducted from the base at x=0 was lost by the convection from the fin, the equation for the temperature distribution is:

$$q = -kA\left.\frac{dT}{dx}\right|_{x=0} \quad (6)$$

The amount of heat flow in the fin can be calculated from:

$$q = \sqrt{hpkA}(T_o - T_\infty)\frac{\sinh mL + \left(\frac{h}{mk}\right)\cosh mL}{\cosh mL + \left(\frac{h}{mk}\right)\sinh mL} \quad (7)$$

The surface area of the vertical corrugated fin can be calculated as:

$$A_{sf} = 2(L.W) + 2(L.t) + W.t \quad (8)$$

In this case, a corrected length is used:

$$L_c = L + \frac{t}{2} \quad (9)$$

While the surface area of the corrugated fin with perforation can be calculated as:

$$A_{pf} = A_{sf} + A_{ic} - 2.N_x.N_{y.}A_c$$

$$A_{pf} = A_{sf} + \pi dt - 2.N_x.N_{y.}(\frac{\pi}{4}d^2) \quad (10)$$

The length and width of the perforation fin can be express as:

$$L = N_x.d + (1 + N_x)S_x \quad (11)$$

$$W = N_y.d + \left(1 + N_y\right)S_y \quad (12)$$

Where S_x and S_y are the space between the circular perforation in X and Y direction respectively.

3.2 Heat Transfer Coefficients Analysis

In this study the heat dissipation from the vertical corrugated fin by free convection. Many empirical correlations were made to estimate the heat transfer coefficient for both solid and perforated fin as follows:

For free convection from a solid vertical fin, Churchill and Chu**, Holman, 2010,** provided a complex relation.

$$Nu = \left[0.825 + \frac{0.387\, Ra^{\frac{1}{6}}}{\left[1+(\frac{0.492}{Pr})^{\frac{9}{16}}\right]^{\frac{8}{27}}}\right]^2 \quad (13)$$

$$Ra = \frac{g.\beta.(T_s - T_\infty)L^3}{v^2}.\mathrm{Pr} \quad , \quad \beta = 1/T_f \quad (14)$$

In case of the perforated fin, there are two heat transfer coefficients. The first one for the solid space of the fin which can express in eq. (15), **AlEssa, 2012, Jassem, 2013, Ibrahim, et al., 2016.**

$$(h_s)_{pf} = (1 + 0.75\, ROA).h \quad (15)$$

Where ROA is the ratio of open area. It can be calculated by dividing the actual open area on the maximum open area of the perforation as follows:

$$ROA = \frac{N_x.N_{y.}A_c}{(N_x N_y)_{max}.A_c}$$

Where $(N_x N_y)_{max}$ can be achieved by making the space between holes equal zero.

The second heat transfer coefficient within the perforation **AlEssa, 2012,** can be calculated from eq.(16).

$$Nu_c = \left[(\frac{Ra_c}{16})^{-1.03} + (0.62.Ra_c{}^{0.25})^{-1.03} \right]^{\frac{1}{1.03}} \qquad (16)$$

4. RESULTS AND DISCUSSION

The convection heat transfer from extended surfaces like fins under natural convection was studied by many researchers. The investigation in this field aimed to increase the heat transfer rate with lower cost, space, and weight.

In this work, a cylindrical heat sink with V-corrugated fins is used in to increase the surface area with less space. A geometry modification is identified for these fins by making a circular perforation through them. The perforations are arranged into two types, inline and staggered in the corrugated edges to study the influence of their location on the heat transfer distribution. The experiments were done with three different power sources which achieved by changing the power supply voltage. The experimental run repeated two times for each voltage to increase the run accuracy. The results can be summarized as follows:

4.1 Model Validation

The experimental heat transfer coefficients were calculated and compared with empirical heat transfer coefficient values from previous literature.

Fig. 4 illustrate the comparison values of the heat transfer coefficients for the V-corrugated plate (solid and perforated one) at different voltages with those calculated from empirical correlations. The results show good agreement between the empirical and experimental data as follows: (a) for the solid plate with less than 8 % average error, (b) for an inline perforated plate with 8 % average error and (c) for the staggered perforation plate with 2 % average error. The error equation used is as following:

$$\text{Error \%} = \left| \frac{h_{theoritical} - h_{experimental}}{h_{theoritical}} \right| \times 100$$

4.2 Effect of Enhanced Surface Geometries on Thermal Boundary Layer

Specially configured fin should be used to increase the heat transfer rate. Such enhanced fin will provide an efficient heat sink with size reduction. In this study two main concepts of enhancement mentioned in **Kraus, et al., 2001,** were employed. The V-corrugated fin considers as a particular shape with boundary layer separation which provides good thermal mixing. Also, the circular perforations through the fin help to form a wake region to dissipate the thermal boundary layer. These geometry modification fins gave best results as compared with corresponding flat and solid one as discussed briefly later.

4.3 Temperature Distribution along the Fins

A comparison between the temperature distributions along fin length for different fins is the best method to represent the fin performance. **Fig. 5** illustrate these comparisons for three supplied voltages. It can be seen that the temperature along the perforated fins (inline and staggered) decreases with distance from the base to the tip. Moreover, the perforated fin with staggered arrangement gives higher temperature difference about 10 % more than the inline arrangement, because the staggered holes allow the air to pass through the whole hot surface of the fin which increases the heat transfer coefficient values.

4.4 Heat Transfer Coefficient (h)

The heat transfer coefficient considers as a significant value for the fins effectiveness. As the convective heat transfer coefficient increase, the fin performance will increase. One of the methods used to raise the (h) value of the fin is to modify its geometry.

In this study, the highest heat transfer coefficients value for the V-corrugated solid plate is 6.7 $W/m^2.°C$ and is achieved by supplying 200 V to the heat sink. This value is considered good and satisfactory as compared with the heat transfer coefficient value for flat solid fin obtained in previous literature. The highest value of (h) for a flat fin calculated by **Jassem, 2013,** is 3 $W/m^2.°C$**.** While, a heat transfer coefficient value obtained by **Shitole, and Arkirimath, 2016, and Muthuraja, et al., 2015,** is 5 $W/m^2.°C$. As a result, the present work gave about 25 % enhancement in (h) value only by using V-corrugated fin instead of a flat fin.

Fig. 6 shows the heat transfer coefficients for perforated and non-perforated fins with the different voltage supply. As can be seen from the figure, the (h) value is greater for perforated fin than the solid one by 20 % and 27 % for inline and staggered holes respectively. The reason for that, the perforation fins have two independent heat transfer coefficient. The first one associated with the perforation and the second within the solid portion of the perforation. Also, the (h) value is slightly increased about 8 % when the voltage supply increases because of the excess heat which transferred to the fin from the heat sink.

4.5 Heat Loss (Q)

Fig. 7 illustrate the amount of heat loss by the fins for different voltages supplied. The figure shows that the heat dissipated from the V-corrugated solid fin is 75 % more than from flat plate fin as calculated in **Prasad, et al., 2016**. Furthermore, heat dissipation increases by 15, 32 and 36 % by using the perforated fin with the inline arrangement and 22, 42 and 45 % higher heat dissipation achieved in the perforated fin with the staggered arrangement. Finally, as the voltage supply increase more heat would conduct to the fins, therefore more heat would convict the surrounding from that fins.

A geometry modification of the fins enhanced their performance. **Fig. 8** shows the enhancements percentage in heat dissipation of the perforated fins in three different voltages supplied. These percentages obtained by comparing the results of the perforated fins with the solid one. Clearly, there is a difference between the enhancement percentage of the perforated fin with the staggered arrangement and inline arrangement about 6, 8 and 9% for 110, 150 and 200 V respectively.

5. CONCLUSIONS

An experimental study of a v-corrugated perforation and non-perforation fins was made with three different voltage supplies 110,150 and 200 V. The conclusions can be summarized as:

1. The heat transfer coefficient and the heat dissipation for V-corrugated solid fin are greater than the flat as compared with previous researches.

2. The temperature drop along the fin is higher in the perforated fin than in solid one. In addition, the fin with staggered holes arrangement has the greatest temperature drop values.
3. A comparison between the fins, the heat transfer coefficient of the perforated fin is enhanced by 20 % and 27 % more than solid fin for the inline and staggered arrangement respectively. Also, the value of (h) increased with the increase of voltage supplied.
4. The heat dissipation from heated fin is strongly affected by the perforation arrangement. The enhancement percentages for the heat loss in the inline arrangement is about 15, 32 and 36 % more than solid fin and about 22, 42 and 45 % for the staggered arrangement.

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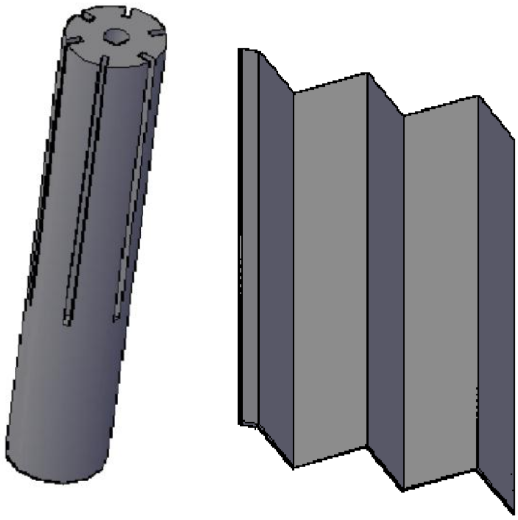
Nomenclature

A = cross-sectional area of the fin, m^2.
A_{sf} = area of the solid fin, m^2.
A_{pf} = area of the perforated fin, m^2.
A_c = the surface area of cutting perforation, m^2.
A_{ic} = inner surface area of cutting perforation, m^2.
d = diameter, m.
g = gravity acceleration, m/s^2
h = heat transfer coefficient, $W/m^2.°C$
k = thermal conductivity, W/m.K
L = length of the fin, m
N_x = number of perforation in x-direction.
N_y = a number of perforation in the y-direction.
Nu = Nussults Number, dimensionless.
p = the perimeter of the fin, m
Pr = Prandtl number, dimensionless.
q = energy, W.

Ra = Rayleigh number, dimensionless.
Ra_c = Rayleigh number of the perforation, dimensionless.
t = the thickness of the fin, m.
T_s = temperature of fin surface, °C.
T_∞ = ambient Temperature, °C.
T_f = film temperature, °C.
W = the width of the fin, m.
ν = kinematic viscosity, m^2/s.
θ = temperature difference, °C.
β = inverse film temperature, $°C^{-1}$.

Table 1. The specification and geometries of fins.

Type of fin plat used	No. of perforation	Diameter of perforation	Location of perforation	Photo of fin plate
Plate (1)	Non	-	-	
Plate (2)	27	10 mm	In line	
Plate (3)	27	10 mm	Staggered in corrugated edges	



(a) (b)

Figure 1. Schematic 3D apparatus (a) cylindrical heat sink (b) plane corrugated fin.

Figure 2. Photograph experimental apparatus parts.

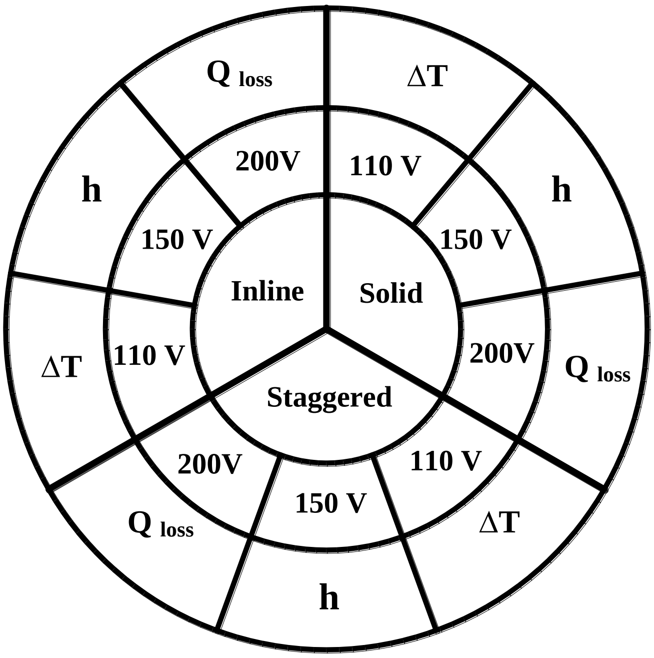
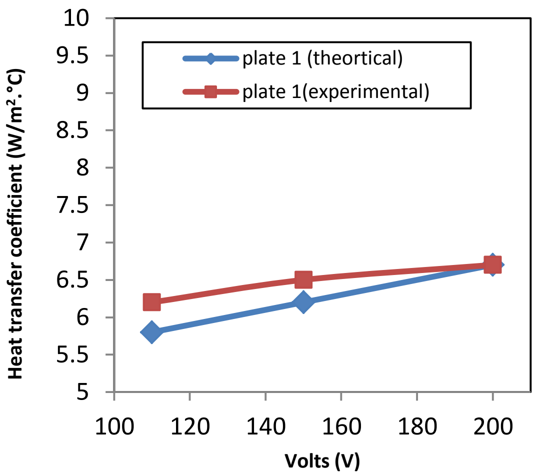
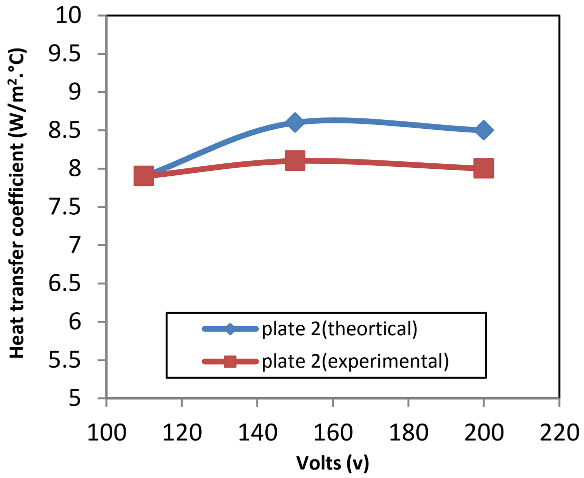


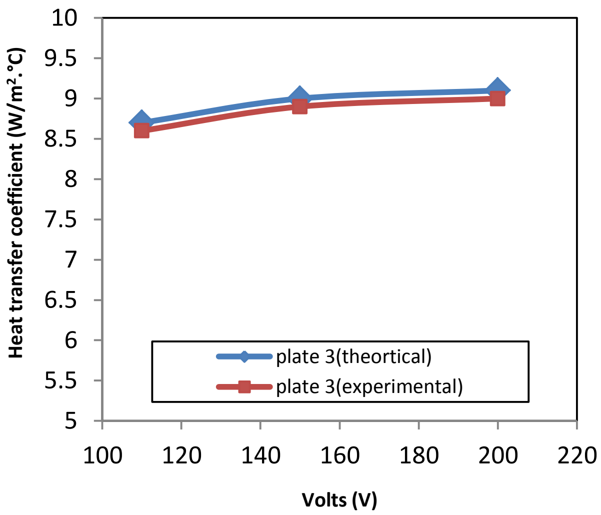
Figure 3. Experimental parameters.



(a) 110 V

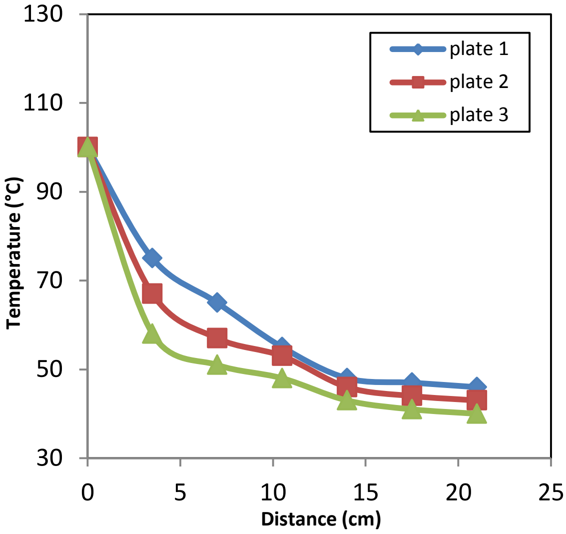


(b) 150 V

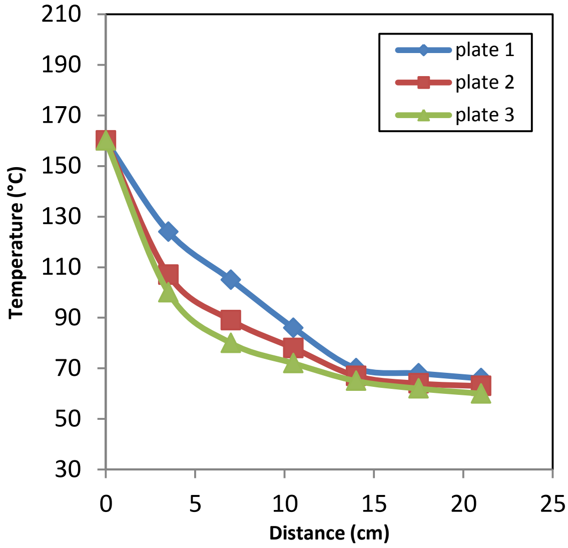


(c) 200 V

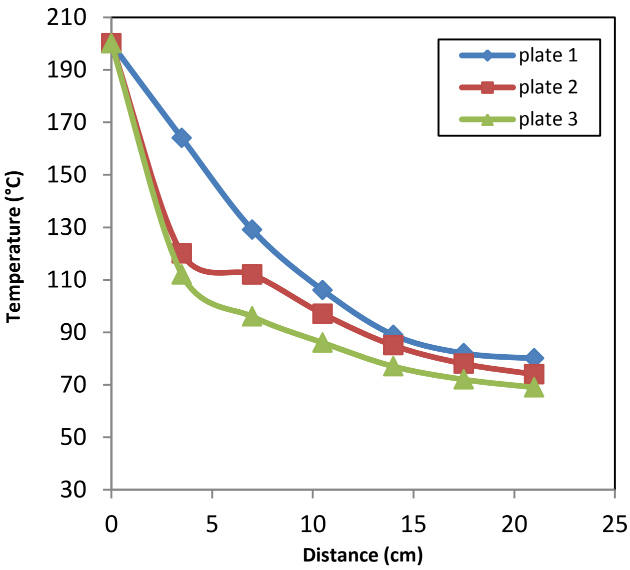
Figure 4. Comparisons of experimental and theoretical heat transfer coefficient vs. volts.



(a) 110 V



(b) 150 V



(c) 200 V

Figure 5. Temperature distribution comparison along different fins for three different supplied voltages.

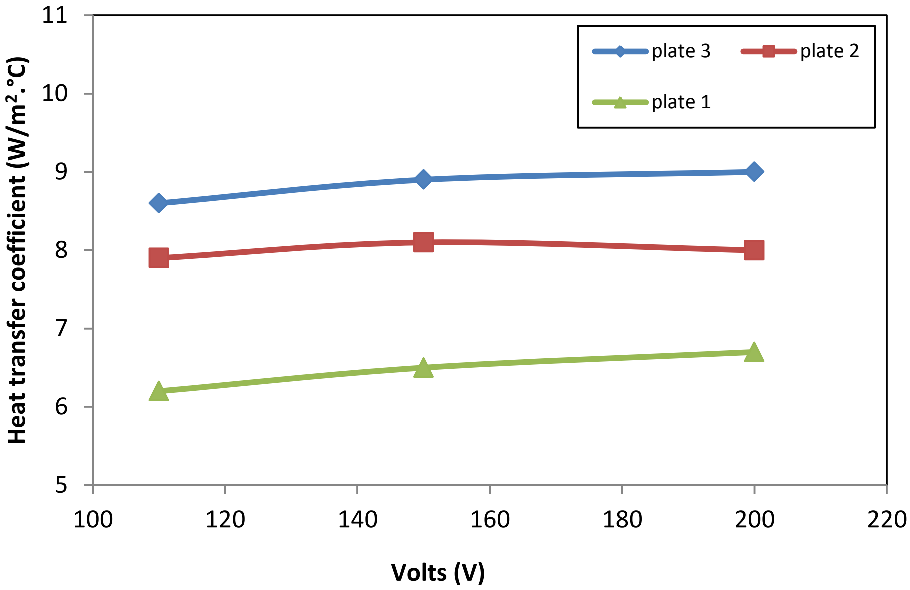


Figure 6. Heat transfer coefficients vs. voltage supplied for different fins.

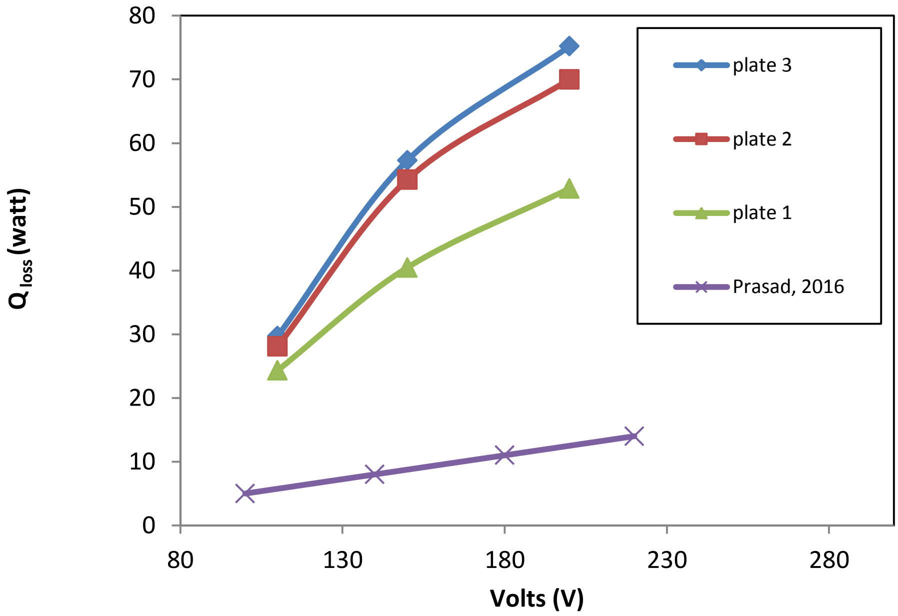
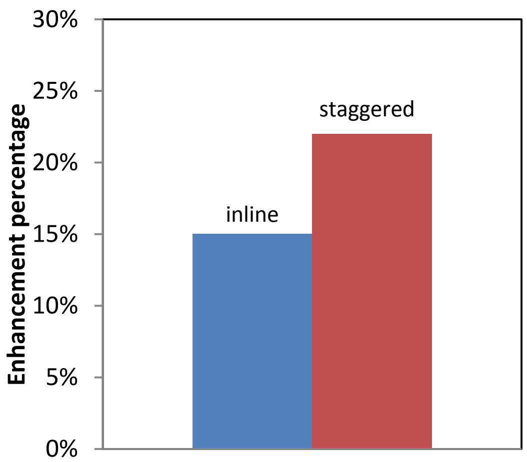
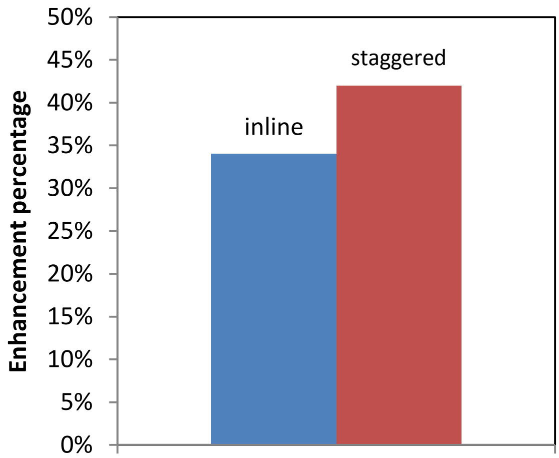


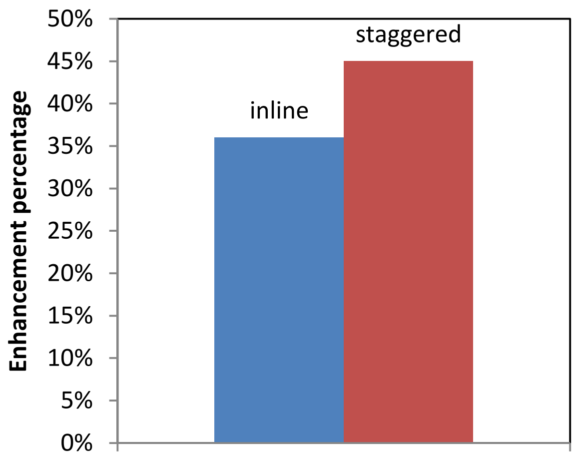
Figure 7. Heat loss from different fins vs. voltage supplied.



(a) 110 V



(b) 150 V



(c) 200 V

Figure 8. Enhancement percentage in heat loss for inline and staggered perforation arrangement for three different supplied voltages.