Study of the performance of multiple v-type fin arrangements as an enhancement of natural heat transfer from a vertical surface

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Abstract: Thermal influence on home appliances, devices and other engineering systems is so high that it causes a system failure. The dominant mode of heat transfer through these systems is the natural heat transfer. As the value of heat transfer coefficient is low, an extension in the heat transfer area becomes essential. The extension is merely to add fins with different shapes and configurations depends on the geometry of systems and space available. One of these configurations is the V type fin array. A testbed was built using two different configurations of fin array. The first array looks like the letter “V” with two discrete parts and the other array is the upset of the letter “V”. The array consists of two columns and five rows separated by a certain distance. The variable heat fluxes were achieved by changing the voltage of three heating elements fixed to the backside of the testbed. Seven thermocouples were fixed at different positions on the testbed and the experiment was run to calculate the natural heat convection through the testbed. The configuration with upset V shape fin shows better performance because it allows the air to flow faster through the middle passage separating the fin array.

Index Terms: V type fin, fin array, natural convection, optimizing airflow

1. Introduction

Extended surfaces (or so-called fins) are frequently used when the heat is to be dissipated from outside boundaries with a higher rate. It represents the most practical way since Newton’s law of cooling states that the heat transfer area is directly proportional to the rate of convective heat transfer. Subsequently, an increase in the surface area will result in an increase in the heat dissipation from the surface.

The application of extended surfaces can be seen everywhere there is a system of convective heat transfer, for instance, the condenser of air conditioners are made in a variety of shapes and size even though the principle is to merely increase the total heat transfer area using aluminium fins. When the system is getting smaller, the heat dissipation becomes more critical and rather complex because of space restrictions. However, the heat is still dissipated via extended surfaces and taken away through special narrow tunnels. Otherwise, there will be heat accumulation that might result in overheating the system.

As an improvement to the heat transfer systems forced moving media enhances the convective heat transfer rate as the heat transfer coefficient becomes higher in value, a hence higher rate of heat transfer from the extended surface resulted in rapid heat dissipation. In such configuration, the layout and shape have no much concern as long as the passages of moving media are kept away from the critical measure. Accordingly, the total heat transfer area and the numerical value of the convective heat transfer coefficient become the most prevailing parameters and nothing can be made with extra cost regarding the layout and shape of extended surfaces.

In situations where it is not possible to mount a source of media accelerating devices such as fans or stream turbulence inducers, the shape and layout of extended surfaces will have a significant role because they will play a role as self-turbulence inducers and impose such self-media accelerators. The configuration of extended surfaces will be so important that they should compensate for the reduction in the value of the convective heat transfer coefficient.
The term of natural heat convection refers to this situation where the buoyancy and gravity forces are responsible for media acceleration. In this context, the direction of movement is always upwards and downwards and thus the configuration of extended surfaces shall take the advantage of such direction so that the movement is accelerated in such a way an increase in the value of heat transfer coefficient takes place. Accordingly, not only the area of heat transfer governs the rate of heat transfer, but the layout of fins plays a significant role in enhancing the heat transfer coefficient and eventually increasing the rate of heat dissipation.

The attempts for altering the configuration of fins have been made by many researchers in order to obtain sensible enhancement of total heat dissipation. For instance, Churchill and Chu [1] found out an empirical relationship through which Nusselt number for natural convection under the assumption of steady-state can be predicted. Their experiment was run along a vertical heated baseplate placed in a flow. Vermeulen and Baudoin [2] also achieved experimental correlations for both inclined and vertical flat plate after running sequences of experiments. The results show an increase in the convective heat transfer coefficient of about 10%. Ra and others [3] examined numerically the combined effect of natural convection and radiation through a vertical finned plate. The principal equations were solved applying Alternate Direct Implicit (ADI) method. The conclusion obtained was the free convection rate increases when fin separating distance decreases which are accompanied by an increase in fin length. Through the work of Abid [4] the effect of fin shape on laminar natural convection was calculated experimentally. Correlations for the pin and vertical fins array under laminar flow were developed empirically. An enhancement for natural convection from a heated vertical plate with several V-type fin array was studied by Sable and others [5]. Their results showed that the V-type fin array always performs better heat transfer performance compared to both vertical fins and V-type fins. Naidu and others [6] made an experimental and numerical study for the problem of natural convection from fin arrays with different inclination angles using Alternate Direct Implicit (ADI) method. The fins have two geometric fin array orientations, horizontal and vertical. The results obtained showed that the rate of convection rate through the vertical fin array is higher than the one obtained from the horizontal fin array for the same inclination angles. Fahiminia and others [7] run the computer-based analysis using the finite volume method to estimate the natural convection heat from extended vertical surfaces. The conclusion which was approached showed that the relationship between heat convection rates and fin space is still proportional until an optimum spacing is reached. More and others [8] made a review study of free convection from a heated plate with different fin array configurations and inclinations. The results show that all configurations of fin have a positive effect on thermal design and results in an increase in the rate of heat dissipation from the heated surfaces at different percentages. Hireholi and others [9] examined the natural heat convection from a heat sink of electronic chips. The optimum fin spacing was found and a comparison was made between the experimentally measured and theoretically predicted temperatures of heat sink applying two-dimensional model which shows a very good agreement. Tiwari and Malhotra [10] studied the laminar natural convection on a flat plate with enclosures in order to investigate the effect of certain parameters among others are; ambient temperature, surface roughness, surface inclinations and flow velocity on the convective heat transfer coefficient at varying input heat flux. It was noted that an increase in input heat flux would increase heat transfer rates. Contrarily, an increase in surface inclination would result in a decrease in heat transfer coefficient. [11] run an experimental study on three geometric orientations these are; (a) vertical fin array (b) horizontal fin array and (c) inclined fin array. The conclusion was the maximum average heat transfer coefficient occurs at 60° V-fin arrays. This result has a good agreement with the one obtained from Computational Fluid Dynamics (CFD) applied. The convective heat transfer coefficient increases with increasing the angle of inclination until it reaches a maximum value of 60°, afterwards the heat transfer coefficient decreases. Moreover, they concluded that the vertical plate with V-fin array of 60 ° included angles having the same surface area as that for the horizontal and inclined plates always results in a greater heat transfer coefficient. [12] explored empirically and numerically the free convection heat transfer coefficient from V-type fin-arrays on a vertical base plate. FLUENT software was used to develop CFD simulations to the arrangement. The case studies under consideration involved different shapes of a 90-degree V-type fin which were
separated into discrete pieces. The number and shape of these pieces and the gap between them were investigated in different cases. The results showed that the maximum natural convection coefficient and heat transfer rate occurs when the fin has the tiniest effect on the airflow within the vicinity of the baseplate and also when the thickness of thermal boundary layer on the fin is thin. Having found the optimum shape, the row spacing was also investigated. The results showed that an increase in the number of rows (i.e. a narrower row spacing), the natural convective heat transfer coefficient increases significantly reaching a maximum value at which the optimum row spacing is determined. After that, the natural convection heat transfer coefficient decreases steeply because of the interference of boundary layers through the rows.

In [13] an experiment was run for comparing the value of heat transfer coefficient from three models; plain vertical surface, plain vertical surface with horizontal fins and a plain vertical surface with V type fin apex facing downwards. The heat flux at the base surface was varied using electrical heaters. They concluded that the surface with V type fins apex facing downwards was able to dissipate much heat. i.e. the highest convective heat transfer coefficient is obtained. In addition, when the heat flux is varied from lowest to the highest values, the temperature difference between the fin surface and ambient was lowest in the case of V type fins apex facing downwards hence better performance. As a result, the Nusselt number for the vertical surface with V-type fins apex facing downwards was the highest.

In this research, an experiment was run to determine the performance of a V-type fin array using some testing instruments and models manufactured by a CNC machine. A comparison is made between V shape fin consists of two discrete parts and its counterpart. i.e. the upset configuration of the V shape fin. The total heat dissipation and convective heat transfer coefficient were calculated for both cases in order to be compared to each other and find out the arrangement that is able to dissipate much heat from the base plate.

2. Experimental Model

In this research, a testbed is fabricated for measuring natural convection heat transfer from a heat sink with fins. The frame of testbed was made of low carbon steel square pipe with a height of 400 mm and width of 200mm. The depth of the frame is shaped in V-type and fixed to four corners. The frame is covered with an aluminium sheet of 0.9 mm thick which is insulated with a layer of thermal woof having a thickness of 15 mm. A polished plate with a thickness of 0.5mm is used inside the testbed as a reflector of heat. Four 600W heaters are used in the test. The heaters are distributed in equal distances along the front side of the testbed.

The fins are attached to a baseplate through grooves which were milled on the front face of the baseplate. The dimension of the baseplate is L × W × H, where L = 300 mm, W = 110 mm and H=30mm. The thickness of each fin is 10 mm, the height is 50mm and the width is 60mm. The total number of fins is 10 arranged in an array of two columns and 5 rows. The distance between the two columns of fins is 15mm and the distance between the rows is 55mm. The fin and baseplate are made of aluminium because of its high thermal conductivity and low emissivity.

The backside of the base plate is fixed to the front face of testbed so that it receives a constant heat flux while the front finned surface of the baseplate is exposed to the environment. Nevertheless, it is assumed that the heat sink is fully shrouded and the air temperature $T_i$ is the bulk-mean air temperature, see Figure 1.

The measuring instruments used in the experiment are listed below

2.1. The Electrical Circuit
The electrical circuit consists of heating elements (heaters) with a total power of (2400 watt).

2.2. Power Supply
A Single-phase power supply SPS-110 Amp Is used to control the electrical power supplied to the electrical heaters with a range of (0-220V).
2.3. Voltage and Current Digital Measurement Device
HEME ANALYST 2050 is used as a measuring instrument for the supplied electrical power to the heater.

2.4. Thermocouples
Thermocouples type K (chromel – alumel) are used to measure the baseplate and fin surface temperatures. Figure 2 shows the experimental RIG along with the measuring instruments.

![Figure 1](image1.png) the test bed used in the experiment  
![Figure 2](image2.png) The experimental RIG with the measuring instruments.

3. Theoretical Analysis
The analysis of natural convection heat transfer from V-type fins with the distance between the fins is discussed in this section. A focus will be shed on the dimensionless groups, namely Nusselt Number, Grashof Number and Rayleigh Number. These numbers represent practical indicators for the amount of heat dissipation from the fin array.

Let us start with heat generated within the heating elements and being converted into heat flux. The length of the finned surface, which is subjected to constant and steady heat flux is \( L \) and the total heat generated in the heater \( Q_{\text{gen}} \) is changed into heat.

This heat is transferred across the fin by conduction \( Q_{\text{cond}} \), and then to the surroundings by natural convection \( Q_{\text{conv}} \) and by radiation \( Q_{\text{rad}} \), i.e.

\[
Q_{\text{gen}} = Q_{\text{cond}} + Q_{\text{conv}} + Q_{\text{rad}}.
\] (1)

The total amount of heat generated is calculated as follows:

\[
Q_{\text{gen}} = V \times I
\] (2)

Where \( V \) and \( I \) are voltage and electrical current respectively. The heat transferred by radiation is calculated from Stefan – Boltzmann law:

\[
Q_{\text{rad}} = \sigma \times \varepsilon \times A_t \left( T^4_{\text{av}} - T^4_{\text{air}} \right)
\] (3)
Where:

\[ \varepsilon = \text{emissivity surface factor and equal to } 0.04 \]

\[ \sigma = \text{Stefan-Boltzmann constant} = 5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4 \]

\[ A_t = \text{total exposed area} \]

The heat transferred by convection is calculated as follows:

\[ Q_{\text{conv.}} = Q_{\text{gen.}} - Q_{\text{rad.}} \quad (4) \]

The free convective heat transfer coefficient \((h)\) is estimated from Newton’s cooling law which states that:

\[ h = \frac{Q_{\text{conv.}}}{A_t \Delta T} \quad (5) \]

The subscripts in the following formulas refer to baseplate \((b)\) and fin \((f)\) where \(L\) is length, \(W\) is width, \(H\) is height and \(t\) is thickness. (here the circumferential area of the holes is considered equal to the area of an upper and lower circular hole, hence it is neglected), as shown in Figure 3.

\[ A_b = L_b \times W_b - n(L_f \times t_f) \quad (6) \]

Exposed fin area \((A_f)\) is

\[ A_f = (L_f \times H_f \times 2) + (t_f \times H_f \times 2) + (L_f \times t_f) \quad (7) \]

So, the total exposed area \((A_t)\) is

\[ A_t = n \times A_f + A_b \quad (8) \]

The total exposed surface area for our sample was calculated using the above equations and it is \((0.081 \text{ m}^2)\).

\((\Delta T)\) in (5) represents the difference between the average temperature for the combination of V-Fin array & baseplate and the air temperature. Then the average temperature for the combination of V-fins and baseplate \((T_{\text{sav}})\) is calculated as follows:

\[ T_{\text{sav}} = \frac{(T_1 + T_2 + \cdots + T_n)}{n} \quad (9) \]

Where \((T_1 + T_2 + \cdots + T_n)\) are the numerical values of temperature which are measured using thermo-couples. The average film temperature \((T_f)\) is calculated as follows:
\[ T_{\text{film}} = \frac{T_{\text{avg}} + T_{\text{air}}}{2} \]  \hspace{1cm} (10)

The physical properties of the working fluid (air) is taken at the film temperature.

Nusselt Number \((Nu)\) is the ratio of convection to the conduction heat flux. It is expressed as follows:

\[ Nu = \frac{hL}{k} \]  \hspace{1cm} (11)

Prandtl number \((Pr)\) is the ratio of momentum diffusivity (kinematic viscosity) to thermal diffusivity and it is written as:

\[ Pr = \frac{\mu C_p}{k} \]  \hspace{1cm} (12)

Grashoff Number is the ratio of the buoyancy force to the viscous force acting on the fluid.

\[ Gr = \frac{L^3 g \beta \Delta T}{\nu^2} \]  \hspace{1cm} (13)

Grashoff Number defines the relationship between buoyancy and viscosity within a fluid and Prandtl number defines the relationship between momentum diffusivity and thermal diffusivity. Accordingly, Rayleigh number \((Ra)\), which the product of \(Gr\) and \(Pr\), may also be considered as the ratio of buoyancy and viscosity forces times the ratio of momentum and thermal diffusivities, i.e

\[ Ra = \frac{L^3 g \beta (T_{\text{avg}} - T_{\text{air}})}{\nu^2} Pr \]  \hspace{1cm} (14)

4. Experimentation Procedure

The experiment was run with various heat fluxes through the control of the heater voltage. The voltages supplied were \((50, 75, 100, 125 \text{ and } 150 \text{ V})\) and the current was measured at each voltage to find the heat generated expressed in equation (2). The indoor lab temperature was measured to calculate film temperature expressed in equation (10) and heat radiation expressed in equation (3). Seven temperature values were measured using the thermocouples. These thermocouples were distributed evenly, i.e. three at each column and one at the mid of baseplates. These temperatures are to calculate the \(T_{\text{avg}}\) appeared in equation (9).

The testbed was first attached with fins form the letter “V” and the experiment was run under different heat fluxes. At the second stage, the testbed was reattached with the fins shape the upside-down of the letter “V” and the experiment was run under different heat fluxes, see Figure 4 for these two arrangements.

The results from each stage are used to calculate the Grashof and Rayleigh Numbers in order to find the heat transfer coefficient for each arrangement.

5. Results and Discussion

The natural convection of V - type fin array has been investigated to determine the best performance of fins. The power supply was set to different values between \((125 \text{ to } 1320\text{W})\) so that the heat supplied to the testbed and the heat losses are affected accordingly. Figure 5 shows the relationship between the average convective heat transfer coefficient \((h_a)\) and the heat dissipation from both arrangements. It is easily seen that both configurations give comparable behaviors with a minor advantage in favor of upset shape of V fin.
6. Results and Discussion

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Figure 6 shows the relation between the average heat transfer coefficient \( (h_a) \) and temperature differences \( (T_{av} - T_{air}) \). As it is seen, \( h_a \) is axiomatically increased when the temperature difference increased yet there is a minor advantage in favor of upset V shape fin. Figure 7 shows the relation between the convective heat transfer coefficient \( (h) \) and the convective heat transfer through the finned surface \( (Q) \). It is obvious the upset V shape configuration dissipate much amount of heat than the V shape configuration.

![Figure 4](image1.png) the two fin arrangements used in the experiment

![Figure 5](image2.png) the relation between average convective heat transfer coefficient and heat dissipation from the testbed

![Figure 6](image3.png) the relation between average convective heat transfer coefficient and temperature difference
Figure 8 shows the relation between the baseplate heat transfer coefficient ($h_b$) and temperature differences ($T_{av} - T_{air}$). Again, $h_b$ is axiomatically increased when the difference in temperature increased, yet there is a minor advantage in favor of the upset V shape fin.

![Figure 7](image1.png) **Figure 7** the relation between average convective heat transfer coefficient baseplate and transfer through the finned surface ($Q$)

![Figure 8](image2.png) **Figure 8** the relation between convective heat transfer coefficient ($h$) and temperature differences ($T_{av} - T_{air}$)

Figure 9 shows the relationship between $Nu$ and $Ra$ for both arrangements. The value for $Nu$ is higher for the upset V shape array compared with V shape array. This is expected from the previous shapes where the convective heat transfer for the upset V shape fin array exceeded the value for the one with V shape array.

![Figure 9](image3.png) **Figure 9** a comparison between Nusselt Number ($Nu$) and Rayleigh Number ($Ra$) for both arrangements

![Figure 10](image4.png) **Figure 10** A schematic diagram shows the direction of airflow through each configuration; left: Upset V shape; right: V shape

Having a glance at the figures above, it can be easily concluded that the upset V shape has an advantage over the V shape fin array. This advantage is expected to become more remarkable when the difference in temperature increases. This was seen in fig. 5&6 where the V shape shows an equal or higher value of the convective heat transfer coefficient when the temperature difference is relatively low. Thereafter, the upset V shape performs better performance with a higher value of heat transfer coefficient.

In order to explain the reason behind this advantage, figure 10 represents a schematic diagram of the active airflow (or passages) through the finned surfaces. As it was mentioned early, because of the prevailing forces, namely gravity and buoyancy force, the direction of airflow through the finned surface
is always upwards. With this context, the upset V shape will guide the flow to the middle distance between fins where the pressure undergoes a drop because of the change in air velocity and where the temperature is at its highest value. On the contrary, the V shape fin array tends to guide the air to the outer boundaries of the testbed where it loses acceleration because of the exterior insulation which results in a considerable reduction of heat dissipation. The vicinity of outer boundaries is the region of no flow or stagnation so that the hot air will try to escape through a longer passage which results in a lower rate of heat transfer.

Figures 11 and 12 represent heat transfer simulations obtained using SOLIDWORK features for each configuration with different heat fluxes. The heat flux, heat transfer coefficient and material of the baseplate and fins represent the input to the software. These values of heat transfer coefficient are taken from Fig. 6 and the input flux is the heat supplied by the heaters. These figures help to mark the areas where the heat is maximum and where it is minimum as an indication of the active airflow through the fins.

Figure 11 SOLIDWORK heat transfer simulation for upset V shape fin array with different heat fluxes

Figure 12 SOLIDWORK heat transfer simulation for V shape fin array with different heat fluxes

7. Conclusions

Comparing the results obtained from the present work, it can be concluded that a vertical surface with V-type fins having an arrangement that looks like upset V letter compared to the V shape enhances the natural heat transfer. i.e. higher heat transfer coefficient, higher heat dissipation and higher Nusselt number are obtained.

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