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Numerical and Experimental Evaluation of Multiple Interrupted V-Type Fins Array

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Abstract: Experiments and numerical simulations have been performed vertically rectangular with interrupted fins with free convective heat transfer. The performance of interrupted V-type fins was calculated to provide the best space fin performance (Sf). Three distinct fin array configurations that changed the space fin (Sf) 10, 8, and 5 mm were used to construct a plate with fins. The power supply of the heat source is varied from 125 to 1,320 W. The configuration with Sf of 8 mm shows better performance because it has a maximum heat transfer coefficient =76.8 W/m^2 K compared with other configurations. The attractiveness of the fin's existing performance and the reduction in mass result from adding disturbance and using a V-type to a vertical fin. The different temperature is high with vertical plate without fins. Nevertheless, the value is low for fins of Sf =10 mm as compared with other fins.

Keywords: fin space, free convection, interrupted V-type fins arrays, fin space.

多間斷 V 型鰭片陣列的數值和實驗評估

摘要: 實驗和數值模擬已在具有自由對流熱傳遞的間斷翅片的垂直矩形上進行。 計算間斷 V 型翅片的性能以提供最佳的空間翅片性能)。 三種不同的翅片陣列配置改變了空間翅片）10、8 和 5 毫米，用於構建帶翅片的板。 熱源的功率從 不等。的配置表現出更好的性能，因為與其他配置相比，其最大傳熱係數。 翅片現有性能的吸引力和由於增加擾動和在垂直翅片上使用 V 型而導致的質量減少。 無翅片立式板溫差大。 但與其他翅片相比，的翅片的值較低。

关键词： 鰭空間, 自由對流, 間斷的 V 型鰭片陣列, 鰭空間。

1. Introduction

The use of natural convection to improve heat transmission is an important topic in thermal engineering. Excessive heat must be removed from system components to avoid the harmful effects of burning or overheating. Increase the heat transfer coefficient between a surface and its surroundings, or heat transfer area, to improve heat transfer. In many applications, fins or expanded surfaces are employed to improve heat transfer. The idea of free convection is related to the passage of a fluid through a solid barrier due to the consequences of the energy exchange on density [1]. Furthermore, heat transfer applications with vertical or horizontal plates are used in solar, electronic, and electric applications [2]. A new augmenting performance was developed for free convection. The vertical plate has a V-shaped fin attached to it. The heat transmission coefficient of a V-type surface is 40% higher than that of a traditional finned surface [3]. The different configurations of fins are used numerically to evaluate the performance of the heat sink. It found that the 90^o V-partition plate enhanced the heat transfer compared to another plate [4]. The research developed the experimental work of different designs of V type. They concluded V-type fin array design performs better than the rectangular vertical fin array and V-fin array with bottom spacing design [5]. Under natural-convection conditions, the nanocoated heat sink was affected heat transfer performance. The effect of nanocoating in improving heat transfer is more obvious only when the temperature differential is large enough [6].

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The correlating equation for free convection from a vertical plate has been presented. The optimal channel width is independent of heat sink height but dependent on heat sink length, the temperature difference between a heat sink's base and the surrounding temperatures, and fluid quality [7]. The simulated and experimented V-shaped ribs attached with vertical base plate have been studied. It was discovered that 15 mm is the ideal row spacing. It was also discovered that as the number of fins increases, thermal resistance reduces [8]. Interrupted fins were studied to check the natural convection. A higher enhancement in the performance of fins has been determined [9].

To reduce the material used and create a heat sink with low cost and high efficiency. Five configurations were modeled to deliver heat transfers of natural convection efficiently. It concluded Nusselt number of a heat sink with 4-interrupted fins was higher [10]. The experimental work investigated V-type fins, parallel horizontal and split fin patterns. The V fin pattern has the greatest effectiveness value than other patterns [11]. A compact high-performance heat sink has been developed. The heat transfer coefficient of the vertical plate with V-shaped fins was 40 percent higher than that of conventional fins. The V-type partition plates act as a flow disturber as well as a stretched surface. [12]. The heat sink with upset V shape and V shape was investigated numerically and experimentally. The upset V shape enhanced the natural convection[13]. The effect of fin space on fin performance has been presented in this study. This impact has been identified through experimental and numerical studies. The three heat sink with different fin space was used. The high range of power supply 125-1320W will cover all applications of natural convection.

2. The Experimental Rig

A testbed is constructed for this study. The authors used a heat sink with fins to determine free convection heat transfer. Three configurations of fin array with different fin spaces are selected to carry out the experimental work. All are made from aluminum. Figure 1 explains the photo of fins geometry.

Fig. 1 Interrupted V-type fins

The size of the baseplate is length = 300, width = 110 and thickness=10 mm. The fins are attached to a plate. Individual fins have a thickness of 5, a height of 50, and a width of 10 mm. The fins are arranged in two columns and five rows for a total of 30 fins. The gap between the rows is 55 mm, while the spacing between the two columns of fins is 10 mm. The heating system consists of four heaters 600 W for each one to supply the heat to fins. The back of the base plate is connected to the heating system, ensuring a steady heat flux. Seven thermocouples (type K) were used to evaluate the base plate's average temperature and monitor the ambient temperature. Experimental RIG with the measuring instruments sees Figure 2.

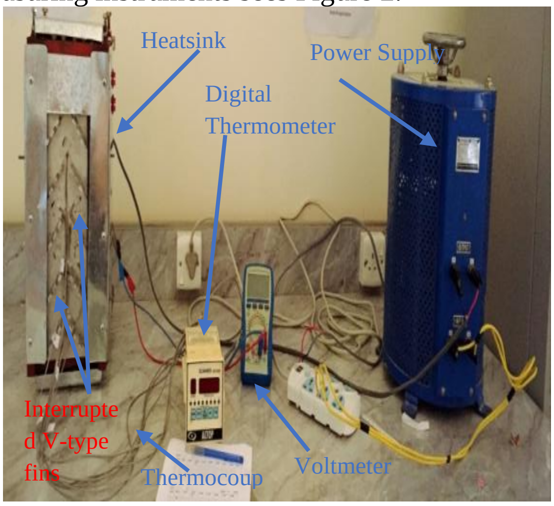


Fig. 2 Experimental RIG with the measuring instruments

3. Numerical Analysis

The Computational Fluid Dynamics (CFD) software has been implemented to simulated natural convection of interrupted V-type fins. The different mode of heat transfer takes place convection, conduction, and radiation. The governing equation is solved to get the numerical data.

3.1. Computation Domain

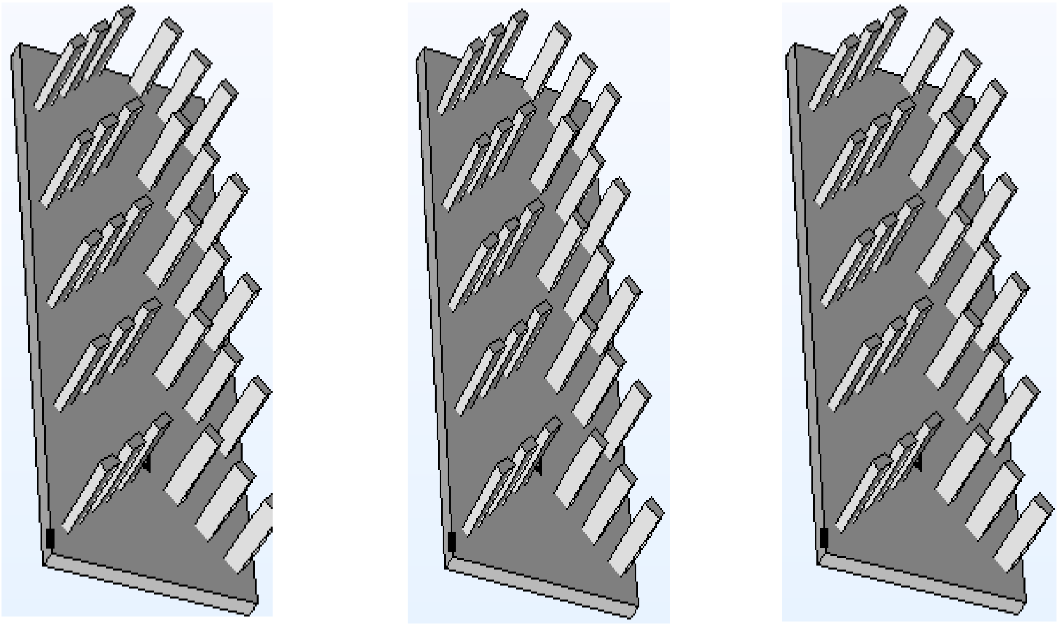
The three arrays of fins with different fin spaces 5,8 and 10 mm have been modeled as shown in Figure 3.

3.2. Governing Equation

The conduction heat transfer can be represented by eq. 1.

$$\rho\, C_p u.\nabla T + \nabla(-k\,\nabla T) = Q \qquad (1)$$

where T, t, ρ, Cp, k, and Q are temperature, time, density, specific heat, thermal conductivity, and heat source, respectively.



S_f = 10 mm S_f = 8 mm S_f = 5 mm

Fig. 3 Array fins model

The convection heat transfer from surfaces to the environment can be written as shown in eq. 2.

$$-n.(-k\,\nabla T) = h(T_a - T\ ) \qquad (2)$$

where n is normal to the surface component vector, h is the free convection coefficient [W m-2 K-1], and Ta is the ambient temperature [K].

The heat radiation from all outside faces to the surrounding environment is controlled by;

$$-n.(-k\,\nabla T) = \varepsilon\,\sigma\,(T_a{}^4 - T\ {}^4) \qquad (3)$$

where ε and σ are emissivities and Stefan-Boltzmann constant, respectively.

3.3. Boundary Conditions

The natural convection held to fins. The heat sink (baseplate) is represented at constant heat flux and transfer by conduction to the fins—the heat transfers between other surfaces by radiation and convection. The emissivity of aluminum is 0.04.

4. The Independent Mesh

The examination of independent mesh has been established. Sixty thousand five hundred eleven elements were used to create the array fins (mesh 1). The average temperatures are calculated to compare between the meshes. The findings of this mesh were compared to a mesh with 619,167 elements, which was substantially finer (mesh 2), as shown in Table 1. The small difference between the two meshes, the mesh with 60,511 elements, was considered sufficient for simulation array fins. Mesh 1 and mesh 2 are shown in Figure 4.

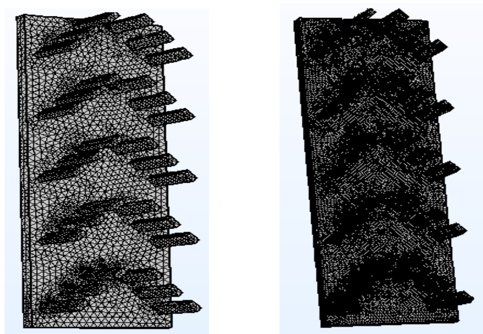


Fig. 4 Observation of array fins for (a) mesh 1. (b) mesh 2

Table 1 Independent of mesh

Mesh	**Tetrahedral element**	**Heat source W**	**Tabag K**
1	60511	560	406.6
2	619167	560	407.5

5. Cases Studies Authentication

Table 2 shows the results of the validation investigations, which compare the experimentally recorded heat sink baseplate temperature to the calculated temperature using CFD software. The deviation between them according to errors in experimental work, where Tbag represents average baseplate temperature.

Table 2 Comparison between numerical and experimental

Heat sink	**Power 125 W**	**Power 315 W**	**Power 560 W**	**Power 925 W**	**Power 1320 W**
Sf = 5 mm	Ta 297 K	Ta 297 K	Ta 297.5 K	Ta 297.5 K	Ta 298 K

	Num. Tbag 333	Exp. Tbag 333.4	Num. Tbag 371.9	Exp. Tbag 369.6	Num. Tbag 407.7	Exp. Tbag 406	Num. Tbag 456.8	Exp. Tbag 455.2	Num. Tbag 522.2	Exp. Tbag 518
Sf = 8 mm	Ta 297 K		Ta 297.5 K		Ta 299 K		Ta 297.5 K		Ta 298 K	
	Num. Tbag 332.4	Exp. Tbag 333.0	Num. Tbag 372.5	Exp. Tbag 365.7	Num. Tbag 409.2	Exp. Tbag 403	Num. Tbag 456.9	Exp. Tbag 453	Num. Tbag 521.4	Exp. Tbag 514
Sf = 10 mm	Ta 296 K		Ta 296 K		Ta 296.5 K		Ta 296.5 K		Ta 297 K	
	Num. Tbag 331.4	Exp. Tbag 327	Num. Tbag 370.9	Exp. Tbag 363	Num. Tbag 406.6	Exp. Tbag 400	Num. Tbag 455.8	Exp. Tbag 448	Num. Tbag 521.1	Exp. Tbag 510

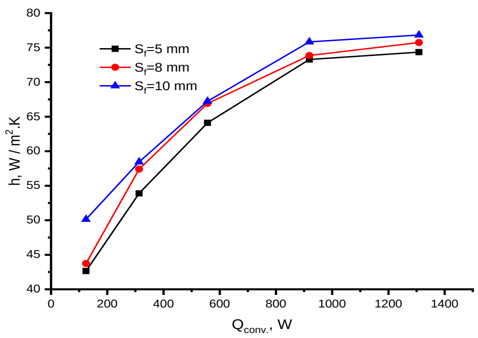


Fig. 5 The coefficient of heat transfer (h) and the rate of heat transfer (Q_{conv})

6. Results and Discussion

The free convection of interrupted V-type fins has been studied in order to determine the best fin performance. The electricity source is varied 125 to 1,320 W. The three different configurations were used to estimate the V-type fin's performance. The distance between the fins was changing between 5,8 and 10 mm. The angle between the rows was 90º. Figure 5 shows the heat transfer coefficient (h) with heat transfer rate fin. The distance between fins had changed 5,8 and 10 mm. The maximum heat transfer coefficients for all fins were 74.3, 75.7, and 76.8 W/m^2. K respectively. Thus fins with Sf =10 mm are more than other fins.

This is due to the increase in heat transfer—the temperature distribution for fins of Sf= 10 mm, as shown in Figure 6.

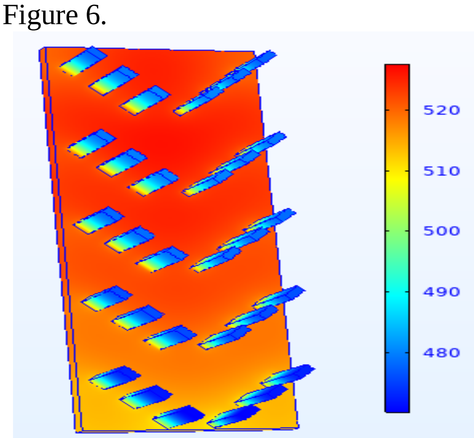


Fig. 6 The distribution temperature of fins of Sf = 10 mm

The coefficient of heat transfer of base (hb) and temperature differential (Tbag.- Ta) is shown in Figure

7. We notice the high-value base heat transfer hb=186 W/m^2. K for the distance between fins Sf=10 mm. The streamline of fins of Sf= 10 mm is shown in Figure 8.

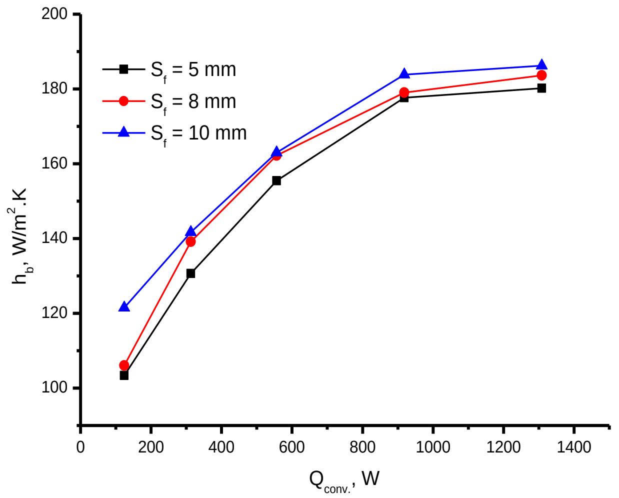


Fig. 7 The coefficient of heat transfer of base (hb) with (Tav-Tair)

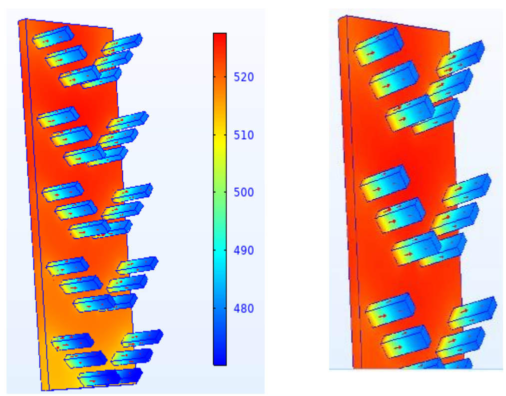


Fig. 8 The direction of conduction heat transfer of fins

Figure 9 shows the heat generated with different temperatures (Tav-Tair) for vertical plate and three configurations of fins with 5, 8, and 10 mm. it is clear the high ΔT with a vertical plate but the low-value fins of sf = 10 mm. This is the best performance with these fins.

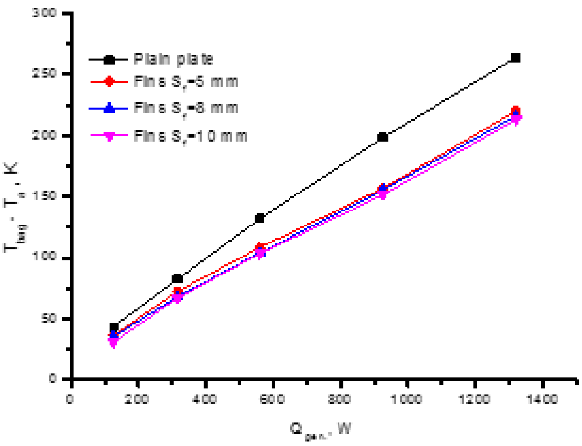


Fig. 9 The power supply with a variation of temperature for plain plate and fins

In order to make the experimental data more understandable, a methodology was developed. The Fluke Ti32 Thermal Imagers ("the Imager") displayed the fin surface's temperature distribution from various angles. The temperature gradient of the fin area inside the Imager's field of view is displayed as shown in Figure 10. It can show from these images the temperature gradient through the fins.

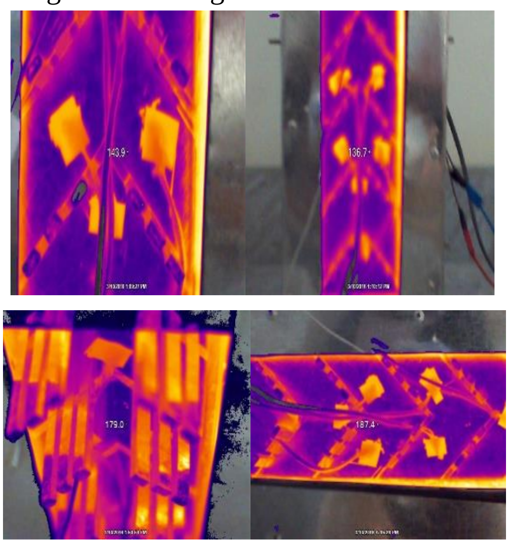


Fig. 10 Thermal imagers of fins

7. Conclusion

The Fins with a V shape have been the subject of experimentation. The effect of distance between fins investigated to learn about the fins' performance from this study can be concluded:

• With Qconv, the heat transfer coefficient rises. Compared to other fins, the best spacing between fins is 10 mm (Sf =10), which has the highest heat transfer coefficient =76.8 W/m^2 K. With a vertical plate without fins, the temperature difference (T) is significant. However, when compared to other fins, the value for Sf =10 mm fins is low.

• When numerical data is compared to experimental data, there is little variation.

Nomenclature

Cp	**specific heat, [J kg K–1]**	**Subscripts**	
h	Heat transfer coefficient for convection from the outer surface, [Wm–2K–1]	a	Ambient
		b	Base
		bag	Base temperature average
k	Thermal conductivity, [Wm–1K–1]	conv	Convection
Q	Heat source, [W]	f	Fin
S	Space		
t	Time, [s]		
T	Temperature, [K]		
	Greek symbols		
ρ	Density, [kgm-3]		
ε	Emissivity coefficient of surface		
σ	Stephan Boltzmann's coefficient, (= 5.67·10-8 Wm-2 K-4)		

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