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THE EFFECT OF BLOWING DIRECTION ON HEAT SINK PERFORMANCE BY THERMAL IMAGING

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ABSTRACT

Heat sinks (HSs) are designed for the mechanical, electrical and electronic components that generate heat in considerable amount. For this purpose, an aluminum conical pin fin heat sink is designed. Aluminum conical pin-fins geometry has been experimentally investigated for the blowing direction (pushing or pulling) which is the energy efficient option for the heat sink. The heat sink was tested at the same fan power for pushing and pulling conditions for 25, 50, 75 and 100 W resistance heater power. Designed aluminum conical pin fin heat sink can be easily used in heat sweeping processes. It has found that pushing configuration of the fan is more efficient for this design.

Keywords: *Heat Sink, Performance, Blowing Direction, Conical Pin Fin*

INTRODUCTION

Mechanical, electrical and electronic components generate heat in considerable amount. Heat sink (HS) is a passive heat exchanger that transfers the heat generated by these components to a surrounding air, where it is dissipated away from the component, thus allowing regulation of the component's temperature at proper grades. Heat sinks (HSs) are used to cool heat sources such as different kind of processing units, high-power semiconductor devices such as power transistors and electronics such as light emitting diodes (LEDs), where the heat spreading ability of the component itself is inadequate to provide its operating temperature range. Expanded surfaces with fins and pin fins are often used in HSs for in order to increase the heat transfer between a heat source surface and the surrounding fluid. HSs such as pin fin, straight fin, cross cut, diamond cut, stacked fin, folded fin, skived fin, soldered fin, bonded fin, stamped and others have been a subject of extensive research. Aluminum or copper straight finned and cross cut HS (in low air velocity flows and in locations where their space is limited) are good known. Various forms of HS fins and pin fins have been used different geometries such as cylindrical, annular, rectangular, square, tapered shaped unit attached to a base plate, with the air passing in the parallel and cross flow over the HS. Ritzer and Lau [1] investigated the effect of two fan orientation (pushing and pulling) HS thermal performance for the aim of which orientation gives the best HS thermal performance and stated that the fan orientation on various types of HSs can affect thermal performance. The results showed that pulling air gives better performance in the high fin density besides pushing air gives better performance in the low fin density. Ahamed et al. [2] studied porous conical pin fin array under natural convection conditions and the experimental results showed that heat transfer coefficient, fin efficiency, and effectiveness were increased up to 5.16, 7.39 and 22.19% respectively than that of non-porous pin fin for the same power. Pirompugd and Wongwises [3] studied analytically the efficiencies for partially wetted fins for the uniform cross section spine, conical spine, concave parabolic spine, and convex parabolic spine. They stated that the fin efficiency is a function of the length of the dry portion. The results also indicate that a larger cross-sectional fin results in a higher conduction heat transfer rate. Contrariwise, the fin efficiency is lower. This is different from the longitudinal fin, for which the trend lines of heat transfer rate and fin efficiency are the same. This converse relationship is due to the effect of the ratio of the cross-sectional area to the surface area. Different applications of conical fins can also be seen in literature. Conic fins for absorber plate of solar air collector were applied by Abuşka et al. [4] and it was tested experimentally. Naphon and Sookkasem [5] worked the heat transfer characteristics of the in-line and staggered taper pin fin HS under constant heat flux conditions and stated that the fin arrangement has a significant effect on the flow characteristics of the air flowing through the HS and, increase the heat transfer rate. Also, first row's shadow affected the second row of pin fins but higher Reynolds number decreases the effect. Elshafei [6] studied geometry, heat flux, and orientation of circular pin fin

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HSs under natural convection conditions and stated that perforated pin fin HS and sideward facing orientation was better than that of solid pins and upward facing orientation. Sahin and Demir [7] investigated a flat surface equipped with circular perforated pin fin HS in terms of heat transfer enhancement and pressure drop. The results indicated that circular pin fins increase the heat transfer between 1.4 and 2.6 depending on clearance ratio and fin spacing ratio. Deshmukh and Warkhedkar [8] investigated the comparative thermal performances of elliptical pin-finned and circular pin fin HSs. They found that the elliptical inline is 24% more effective (eff.) than circular inline, elliptical staggered is 50% more eff. than circular staggered and the elliptical staggered is 63% more eff. than elliptical inline. Yüncü and Anbar evaluated the plate-fin HSs and they obtained the optimal fin spacing decreases as fin height increases [9]. The failure rate of electronic components decreases with the decrease in temperature of the component. Therefore, a conic pin-finned HS was designed to reduce the temperature of a component connected and the effect of fan orientation on the performance of a conical pin-finned HS was investigated experimentally. Zhou and Rau [10] studied a heat sink to come up with a framework to correlate hand calculations and numerical simulations with experimentally obtained results. They investigated many parameters such as power levels, orientation (flat-vertical-horizontal), gravity, and altitude under natural convection conditions. Belhadj et al. [11] studied numerically the forced convective flow in micro channels with periodic expansion-constriction cross-section under laminar conditions. They focused the effect of cross-sections on the thermal behavior of water flow used for cooling systems. They stated that the Nu number is increasing (36% of maximum) with the rise of Reynolds number for all micro channels modifications and convective heat transfer rate was improved significantly via the periodic expansion-cross section. They concluded that micro channels with triangular cavities are more efficient than cylindrical grooves. Zunaid et al. [12] studied numerically the conjugate heat transfer in a heatsink with semi cylindrical geometry under the single phase flow condition. Reynolds number from 200 to 1000 with constant heat flux of 106 W/m^2 selected for the heat sink in the simulation. The heat transfer in microchannel heat sink with semi cylindrical geometry was better than that of heat sink with rectangular one.

MATERIALS AND METHODS

Aluminum conical pin fin geometry was selected for heat transfer enhancement also relatively low-pressure drop. In this study, a new design conical pin fin HS designed and manufactured via CNC machining. The HS made of aluminum. The dimension of HS base plate is 80x80x5 mm. The base plate has 39 conical pin fins. Each conical fin has 6 mm base, 1 mm tip diameter, and 10 mm fin height.

The experimental setup consisted of the air duct, fan, heat sink, plate type resistance heater, insulation, fire brick and test instruments. The air duct is made of transparent Plexiglas and measured temperature in the air duct by PT1000 type sensor (Comet SN234 with the precision of ±0.15°C). The system can be operated in pushing and pulling modes. The air flow direction is changed by reversing the fan. The test setup is from bottom to top; fire brick, insulation, resistance heater, heat sink, fan and air duct. The fire brick is opened to a bed for resistance heater and thermally insulated in accordance with the heat sink scale. The lower horizontal and side surfaces of the heater are insulated by a ceramic wool blanket in 5 mm thickness. The base of the HS was heated by an electrical resistance plate heater with 190 W (DC 24 V). The heater is 80x80x3 mm and mounted on the HS with M4 screw, and powered by a DC power supply (Sayntech 23003 power supply). A usual 12 VDC fan is used in 80x80x25 mm for providing air flow and worked by a power supply (TT Technic RXN-303D power supply). In order to measure the air outlet temperature in the push mode and the air inlet temperature in the pull mode, T type thermocouple sensor is mounted on each side between the heat sink and the fan also a thermocouple temperature sensor (Teflon coated, Elimko T type with precision of ±0.5°C) embedded within the center of the base plate for to measure temperature of the heat source. Hot wire type (Delta ohm HD403TS1 with precision of ±0.2 m/s + 3% f.s.) and hand type anemometer (Kestrel 3000 multifunction anemometer with a resolution of 0.1 m/s) were used to measure the air velocity in the air duct. A thermal camera (Flir SC325 with the precision of ±2%) was used to measure the surface temperature of the heat sink and analyzed by Researcher 2.10 software. Monitoring and recording of test data were done with Comet MS6D universal data logger. The test recording time interval was set to five seconds. Two electrical multimeters (Brymen BM807 multimeter) were used to measure the electrical values of the heater and the fan.

The HS geometry sketch is shown in Fig. 1 and photographic representation of the heat sink and, test bed in Fig. 2. The thermo-physical properties aluminum and air are given in Table 1. The experimental set up was performed. The test apparatus was placed in the flow channel.

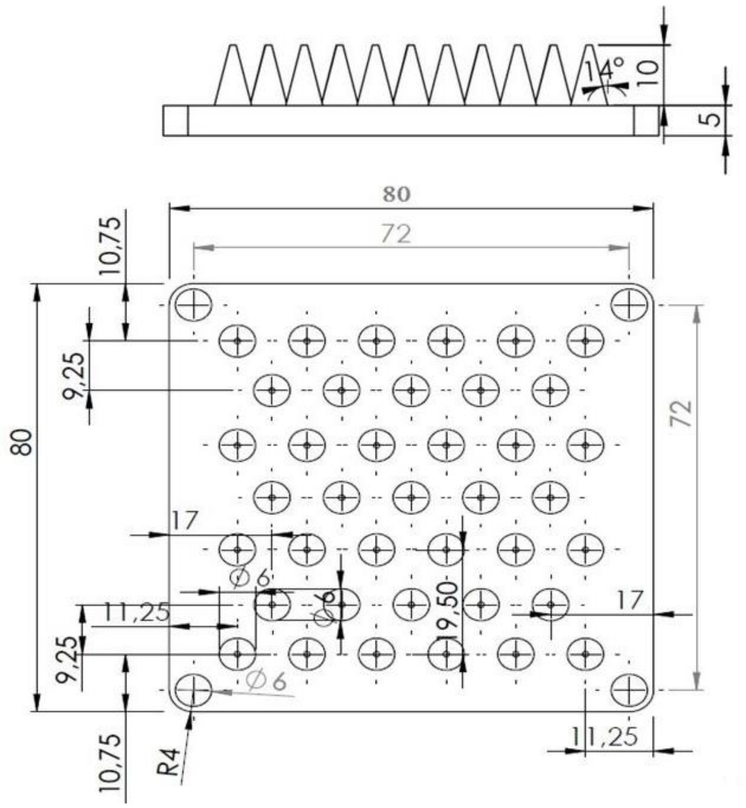

Figure 1. The HS geometry sketch

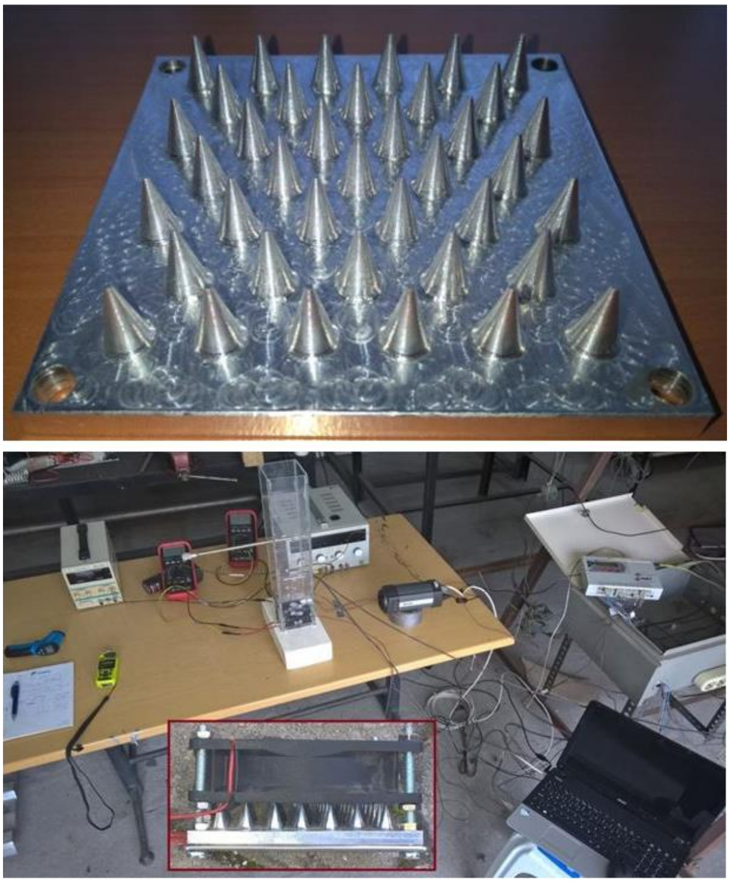

Figure 2. The heat sink and test bed

Table 1. Thermo-physical properties of air and aluminum at 293 K

Thermal conductivity (aluminum, $k_{w,}$ W/mK)	237
Thermal conductivity (air, $k_{a,}$ W/m K)	0.025
Specific heat capacity (air, $c_{p,a,}$ J/kg K)	1006
Specific heat capacity (aluminum, $c_{p,al,}$ J/kg K)	871
Dynamic viscosity (air, μ_a kg/sm)	$1.76x10^{-5}$
Bulk thermal conductivity (aluminum, W/m K)	180
Specific thermal conductivity (aluminum, W/m K)	63
Density (aluminum, kg/m^3)	2719
Density (air, kg/m^3) $\rho_{air} = 1.2x10^{-5}T^2 - 0.01134T + 3.498$	

THEORETICAL MODEL

The heat transfer modeling of a heat sink considered in this study is schematically showed in Fig. 3. Assuming that the air flow and ambient temperature is steady state also the bottom and the side surface of the heater is perfectly insulated.

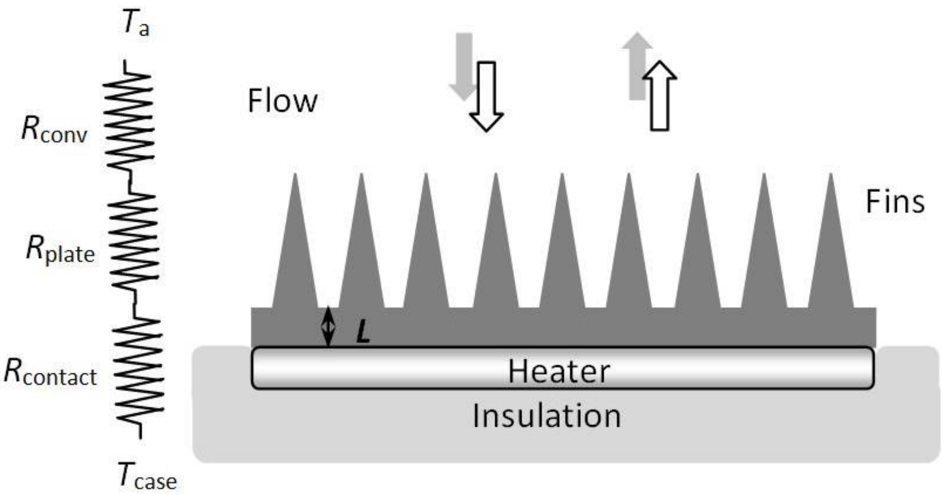

Figure 3. Heat transfer modeling of the heat sink

Total surface area ($A_{s,fin}$) of the cone = Area of the base + area of the curved surface

$$A_{s,fin} = \frac{\pi D}{2}\sqrt{L^2 + \left(\frac{D}{2}\right)^2} \tag{1}$$

The multiplication of total surface area of a cone and fin number equals summed surface area of n fins. Volume of cone is;

$$v = \frac{\pi D^2 L}{12} \tag{2}$$

where, (L) length of the fin and (D) diameter of the fin at the base. Total amount of heat supplied by heater is calculated as

$$Q_{heater} = VI\cos\theta \tag{3}$$

where, *(Cosθ)* is power factor 1 (no inductive and capacitive load), (*V)* voltage supplied and (*I)* current. The rate of heat transfer from the heater is determined to be;

$$\dot{Q} = \frac{\Delta T}{R_{th}} \tag{4}$$

Thermal resistance is

$$R_{th} = \frac{\Delta T}{\dot{Q}} \tag{5}$$

The thermal contact resistance can be defined as

$$R_{th,cont} = \frac{1}{h_c A_c} \tag{6}$$

where, (A_c) the contact area and (h_c) the contact conductance. The thermal resistance for conduction can be expressed as:

$$R_{th,\,plate} = \frac{T_1 - T_2}{q} = \frac{L}{kA} \tag{7}$$

The thermal resistance for convection can be expressed as:

$$R_{conv} = \frac{1}{h_o A} \tag{8}$$

$$R_{total} = R_{th,cont} + R_{th,\,plate} + R_{th,conv} \tag{9}$$

h_o : the convection conductance. The heat balance is;

$$\Sigma Q = Q_{conv} + Q_{rad} + Q_{loss} \tag{10}$$

Due to well-insulated surfaces and irradiative heat and losses is very small, they could be neglected. Thus, equation is written as

$$\Sigma Q = Q_{conv} \tag{11}$$

Dimensionless surface temperature of the heater (θ_s) is

$$\theta_s = \frac{T_s - T_i}{\frac{qH}{k}} \tag{12}$$

where, *(q)* the heat flux, (*h)* the local convective heat transfer coefficient, (k) the thermal conductivity of air, (T_i) the inlet temperature of the air and *(H)* the channel height. Fan power can be expressed as:

$$P_{fan} = uA\Delta p \tag{13}$$

$$\Delta p = p_{in} - p_{out} \tag{14}$$

RESULTS AND DISCUSSION

The stages of the experiments consisted of the first five minutes of non-heating operation, the heating of 25, 50, 75 and 100 W each for ten minutes and finally the observing of the cooling of the HS for 15 minutes after shutting the heater.

The variation of the heating resistance plate surface temperature with time according to the pushing and pulling of the fan conditions is given in Fig. 4. The resistance heater surface temperatures (T_{res}) under the pushing and pulling fan states are very similar to each other and an average temperature difference of 3.5°C is measured.

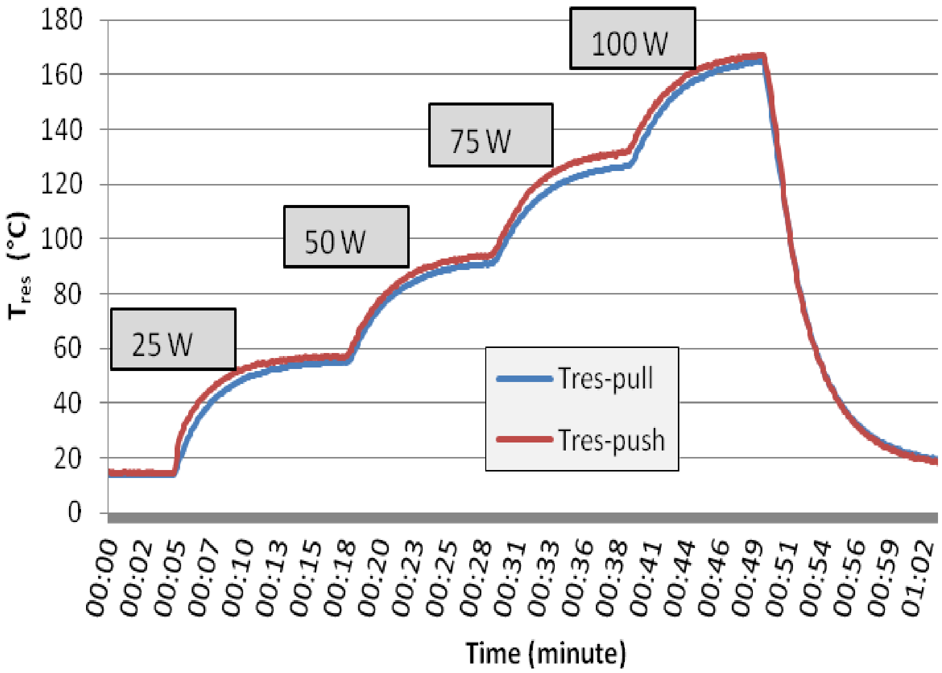

Figure 4. Change of heater surface temperature with time

The surface temperature of the pin fins of the HS was measured by the thermal camera and the averages were taken. Table 2 shows the surface temperature average values for the push and pulling mode. Change of surface temperature versus heater power is given in Fig. 5. As the heater power increases, the fin temperatures increase during pulling and pushing. The transferred heat energy was higher because the air flow rate in the pushing mode was about twice as high. As a result, the fin surface temperature was lower than expected. The average surface temperatures of the HS in the pushing mode were lower by 2.0-3.9-5.1-6.9°C, respectively for 25-50-75-100 W resistance heat power.

Table 2. Average fin temperatures of the HS

Heater power (W)	Pulling (°C)	Pushing (°C)
25	21.4	19.4
50	28.1	24.2
75	34.9	29.8
100	42.4	35.5

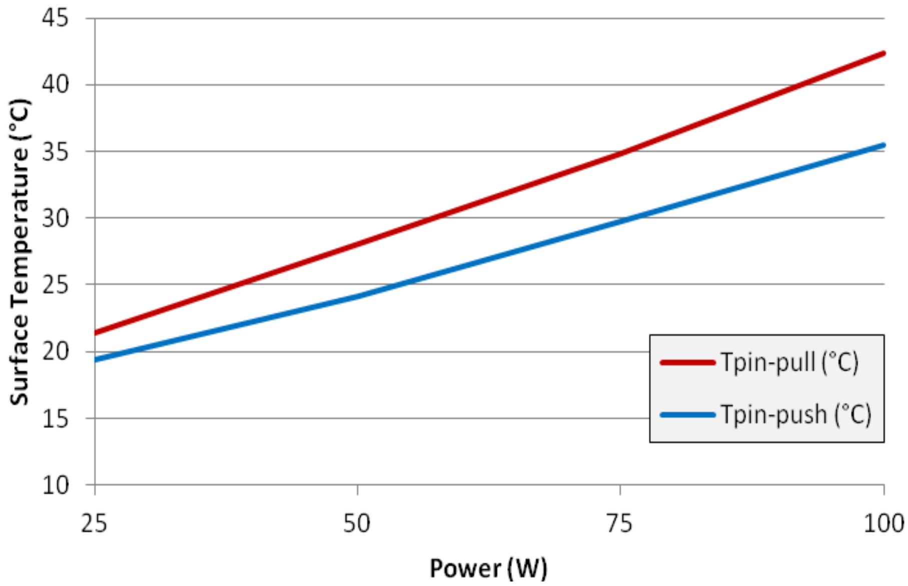

Figure 5. Change of surface temperature versus heater power

The blowing direction of the air in the test bed is changed by reversing the fan. Air velocity is measured with a hotwire type anemometer placed in the duct. The air velocity was also measured with a vane type anemometer every five minutes to compare. The air velocity in the pulling and pushing situations is given in Fig. 6. On average, the air velocity was approximately 1 m/s in the pulling mode and 2 m/s in the pushing mode for same power source. The air velocity was twice as high in the pushing mode comparing to the pulling mode. This can be explained by the fin density of the HS. The air velocity in the push mode is very slightly fluctuating. The air flow is perpendicular to the fins in the pushing mode as it flows parallel to the fins in the pulling mode. The fluctuation is thought to be due to turbulence, which is caused by the air flowing perpendicular to the fins.

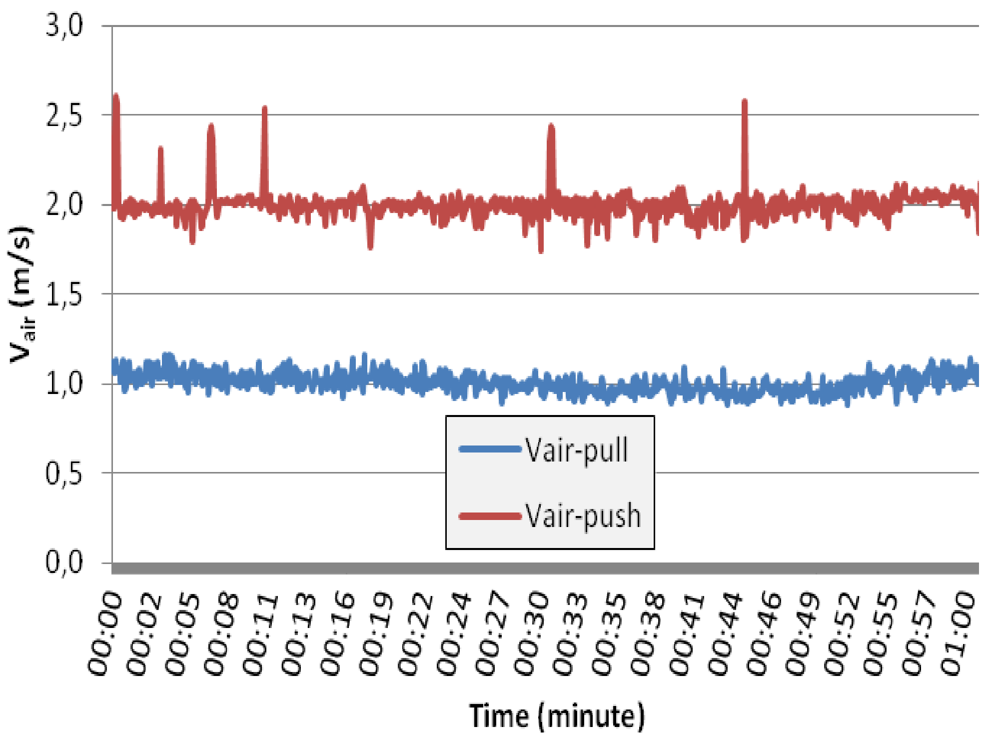

Figure 6. Change of air velocity with time

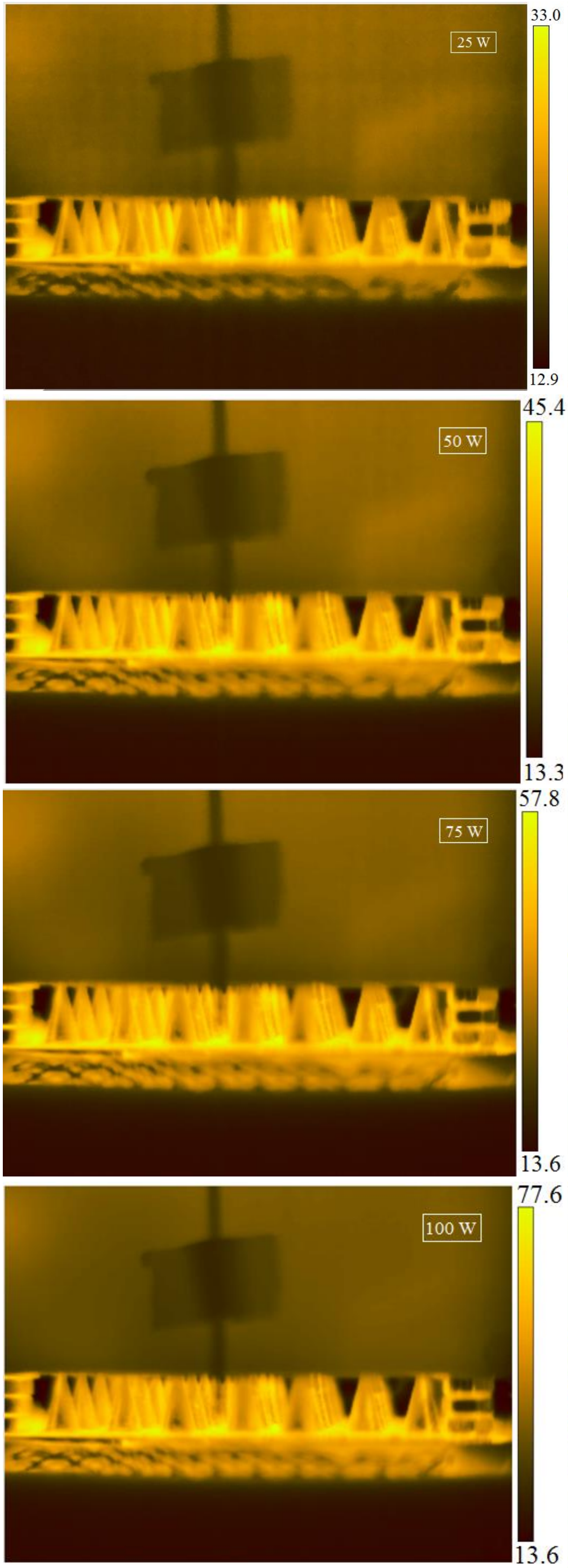

Figure 7. Thermal images of the HS in pulling mode

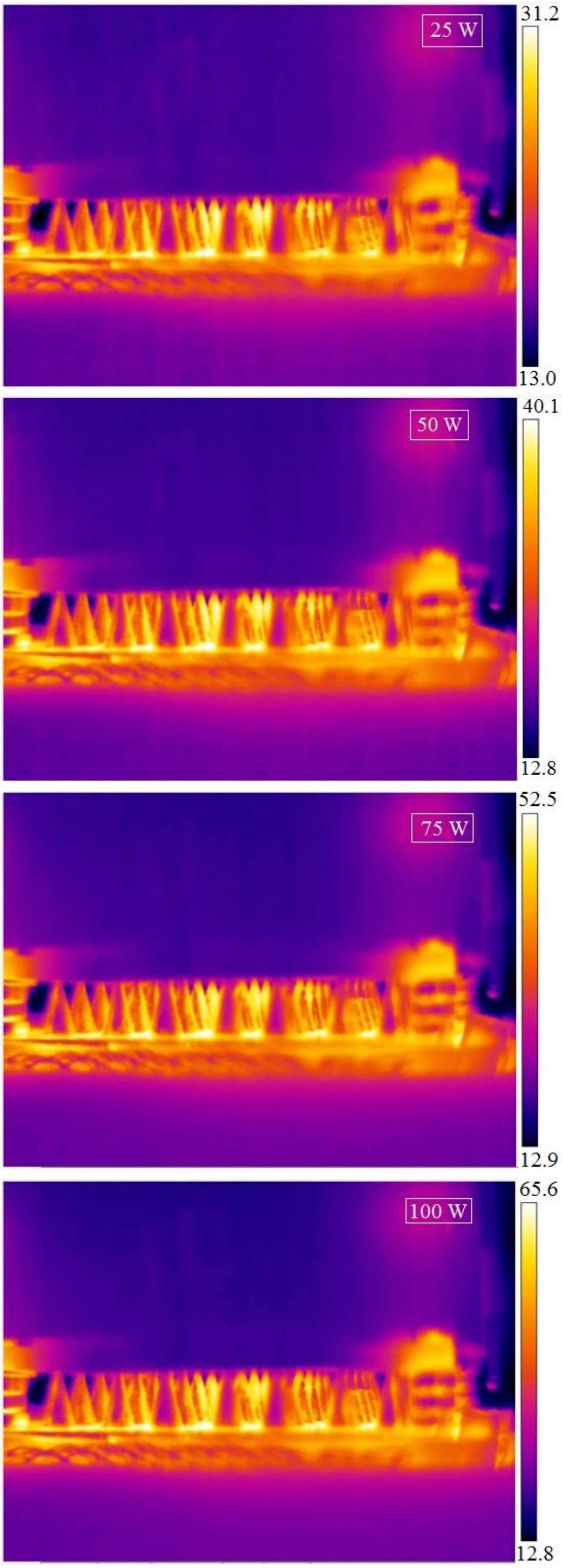

Figure 8. Thermal images of the HS in pushing mode

The surface temperature of the HS was about 5 °C lower in overall average in pushing mode and it can be said that this is caused by the difference of the air flow rate. Considering the present HS design, it has been concluded that the operation of the fan in the pushing mode is more appropriate in terms of thermal efficiency.

The thermal images in the pulling mode are shown in Fig. 7 and the pushing mode in Fig. 8 for four heater power.

CONCLUDING REMARKS

In this study, HS with the conical surface was designed, fabricated and the effect on the thermal performance of the fan orientation was investigated experimentally. The surface temperatures of the HS for thermal performance are measured and taken with the thermal camera. Tests were conducted at 25, 50, 75 and 100 W heating powers. The fan was run at the same electrical values and the fan orientation was reversed for to change the blowing direction. The air velocity in the pushing mode was twice as high, which can be explained by the fin density of the HS. The air velocity in the push mode is very slightly fluctuating. The air flow is perpendicular to the fins in the pushing mode as it flows parallel to the fins in the pulling mode. The fluctuation is thought to be due to turbulence, which is caused by the air flowing perpendicular to the fins. As a result of the experiments in the same conditions, the surface temperature in the pushing mode as average; 2°C at 25 W, 3.9°C at 50 W, 5.1°C at 75 W and 6.9°C at 100 W was lower than pulling. The surface temperature of the HS was about 5°C lower in overall average in pushing mode and it can be said that this is caused by the difference of the air flow rate. As a result, it has been found that the use of fan in pushing mode for the conical surface HS gives better results in terms of thermal performance.

NOMENCLATURE

A_c	the contact area (m^2)
$A_{s,total}$	total surface area (m^2)
$Cos\theta$	power factor
D	diameter (m)
H	channel height (m)
h_c	contact conductance (W/m^2 °C)
I	current (A)
k	thermal conductivity
L	length (m)
P	power (W)
Q	amount of heat (W)
q	heat flux (W/m^2)
R_{th}	thermal resistance (°C/W)
T	temperature (□)
u	velocity (m/s)
ν	volume (m^3)
V	voltage (V)
θ_s	dimensionless surface temperature

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