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NATURAL CONVECTION HEAT TRANSFER FROM VERTICAL TRIANGULAR FIN ARRAYS

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Abstract Experimental investigations of natural convection heat transfer from vertical triangular fin arrays, namely an array of four fins, an array of seven fins and an array of thirteen fins were carried out. All fins were prepared by sand casting and were of same dimension. From the experimental results two correlations were developed in terms of dimensionless Nusselt number and Rayleigh number, which satisfied all of the experimental data with reasonable accuracy.

Keywords: Triangular fin array, Fin spacing, Heat transfer enhancement.

INTRODUCTION

A growing number of engineering disciplines are concerned with energy transitions requiring the rapid movement of heat. They produce an expanding demand for high-performance heat-transfer components with progressively smaller weights, volumes, costs or accommodating shapes. Triangular or any type of fins are extended surfaces. Extended surface heat transfer is the study of high-performance heat-transfer components with respect to these parameters and of their behavior in a variety of thermal environments. When a prime surface is extended by appendages intimately connected with it, such as the metal tapes and spines on the tubes, the additional surface is known as extended surface. Fins provide compact heat transfer surfaces. Compactness refers to the ratio of heat transfer surface per unit of exchanger volume. More recently the demands of aircraft, aerospace, gas-turbine, air-conditioning and cryogenic auxiliaries have places particular emphasis on the compactness of heat transfer surface and particularly on those surface elements which induce small pressure gradients in the fluids circulated through them. Natural convection cooling using fins is of interest in the thermal management of different electronic components.

So far, a lot of investigations have been conducted with fin or fin arrays of different profiles. But triangular fins are something, which could draw a very little attention specially for arrays of triangular fins. Considering these facts the present work has been selected for further study. The main objectives of the present work were to establish the effect of number of fins in an array on heat transfer enhancement and to develop correlation for estimation of natural convection heat transfer from fin arrays. For this purpose, an experimental setup was designed or studying the natural convection heat transfer from vertical triangular fin arrays. The present work also involved study of effect of temperature on emissivity for different fin arrays and the effect of radiation heat loss on convection heat loss for different fin arrays.

To design a fin properly one needs to know the convective heat transfer coefficient to the surrounding fluid. Interestingly, it appears that very few measurements have been reported in the literature of the natural convective heat transfer coefficients from triangular fins mounted in a vertical surface, which is the most common orientation. It seems likely that designers (Karagiozis[1], Elenbaas[4]) of triangular fins have used recommended equations to calculate heat transfer from the triangular fins assuming, for example, that the triangular fin will convect the same as a rectangular fin of the same perimeter facing a passage of the same cross-sectional area.

In the present work cast iron was used as base plate as because of it's availability and low cost, and the fins were the integrated part of the base plate to minimize the surface contact resistance. A triangular fin would have less boundary layer interference near its base. Also, heat transfer from the fin ends is going to be different for above mentioned two cases. Even if the rectangular fins and triangular fins did dissipate heat at the same rate, there would still be a problem. For vertical rectangular fins there are substantial differences between the recommended equations of different researchers is most pronounced at low Rayleigh numbers. The reason is that the corrections for the radiant losses and for the back losses become relatively

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large at low Rayleigh number, and there is usually a large uncertainty in both of these corrections. In the present measurements, every effort was made to minimize the radiant and back losses.

EXPERIMENTAL SETUP AND PROCEDURE

The setup consists of test bench, test section and measuring instruments. The experimental setup is shown in the Fig. 1.

➢ The Test Bench

The test bench was fabricated from mild steel angle frame. It has two parts. In one part the test section might be installed. The other part is moveable. The test bench was mounted on a trolley.

➢ The Test Section

The test section consists of specimen (fin array) and heater. All the specimens were prepared by sand casting. The specimens were painted black-mat and were attached with a mild steel plate installed in the test bench. A sketch of the specimens is shown in Fig. 2. The dimensions of the specimen are listed in Table 1. A 100 watt electric heater made of 28 BWG nicrome wire was mounted on the back portion of cast iron base plate of fin array. Back portion of the test section was insulated by asbestos sheet to minimize the heat losses.

➢ Measuring Instruments

The setup was instrumented by a stabilizer, a voltage regulator, an ammeter, a voltmeter, an infrared thermometer and a digital thermometer. The digital thermometer was used to measure the temperature of the fin surface and that of the ambient air. Five thermocouples were embedded in 8mm deep holes in the base plate, another three were attached on the fin surface and one thermocouple was used to measure the temperature of the surrounding air. The iron constantan thermocouples (J-type) were used in this experiment. The contribution of radiation heat transfer in natural convection is quite significant. So in this experiment a non-contact type infrared thermometer was used to estimate the emissivity of the in surface. After knowing the emissivity the heat loss from the specimen by radiant mode was calculated.

Procedure

The investigations were carried out with four different test specimens having varied number of fins in the array. The specimens were: (a) a flat plate with single triangular fin, (b) an array of four triangular fins, (c) an array of seven triangular fins and (d) an array of thirteen triangular fins. First the specimen was firmly mounted in the test bench. The heater was switched on to heat the specimen. After attaining the steady state (generally it took 2 to 3 hours to attain the steady-state) all the readings (base plate temperature, ambient temperature, voltmeter, ammeter, etc.) were noted. The emissivity of the specimen was measured by a non-contact type infrared thermometer.

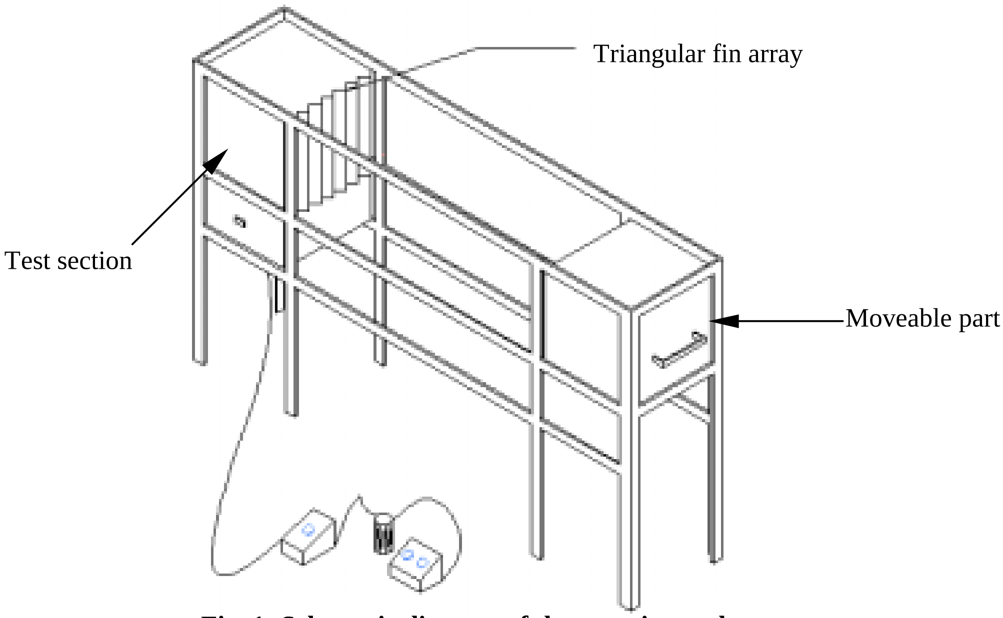

Fig. 1: Schematic diagram of the experimental setup

DESCRIPTION OF PROBLEM

The objective was to measure the convective heat transfer, Q_{conv}. If Q_{conv} is the convective heat transfer to the ambient fluid from the surface area As, where As is the entire surface area of the array except the back and side (Fig. 2), the average heat transfer coefficient is embodied in the Nusselt number as follows:

$$Nu_b = \frac{\bar{h}b}{k} = \frac{Q_{conv}b}{A_s \Delta TK} \quad \text{--------------------------------(1)}$$

Nu numbers should be applicable to all geometrically similar arrays, for the same Ra and Pr. To measure the total heat loss from fin arrays, a steady state technique was used. Heat loss consists of three components and heat balance from te specimen's surface becomes:

$Q_{total} = Q_{conv} + Q_{rad} + Q_l$ ------------------------------(2)

where Q_{rad} is the radiation heat loss and Q_l is the loss from the back of fin base.

Q_{conv} can be found by subtracting Q_{rad} and Q_l. Raithby and Holland's method was used to measure the Q_{rad}. The equation is:

$$Q_{rad} = \sigma A_s F_{1-2}\left(\overline{T}_{b'}^4 - T_a^4\right) \quad \text{-------------------------------(3)}$$

where the radiant exchange factor F_{1-2} (Krieth, 1968) accounts for both geometric and surface emissivity effects governing radiant exchange between the fins and the surroundings. The value of F_{1-2} equals to ε as the area A_s is very small with respect to room.

Now, we can derive the final form of the governing equation, which is as follows:

$$Q_{conv} = Q_{Total} - Q_{rad} - Q_l$$

$$\Rightarrow Q_{conv} = Q_{Total} - \sigma A_s \varepsilon\left(\overline{T}_{b'}^4 - T_a^4\right) \quad \text{------------------(4)}$$

[As Q_l is negligible]

Corrected equation for convective heat transfer

G.M. Dusinberre [3] has introduced an equation for triangular fin efficiency, $\theta = 1/(1 + hH^2/kt)$.

Again,

$$h = \left[A_{b'}\left(\overline{T}_{b'} - T_a\right) + A_f\theta\left(\overline{T}_{b'} - T_a\right)\right]$$

$$or, h = \overline{h}A_s\left(\overline{T}_{b'} - T_a\right)$$

$$\therefore \quad \overline{h} = h = [x + \theta(1-x)]$$

$$where, x = \frac{A_{b'}}{A_s} \quad and \quad A_{b'} + A_f = A_s$$

Therefore, the corrected Q_{conv} is

$$Q_{conv} = \overline{h}A_s\Delta T$$

Corrected Equation for Film Temperature

The physical properties used in the calculation of Nusselt and Rayleigh numbers were evaluated at a film temperature of

$$T_F = T_a + 0.62\left(\overline{T}_{b'} - T_a\right) \quad \text{--------------------------------(5)}$$

as recommended by Sparrow and Gregg [2].

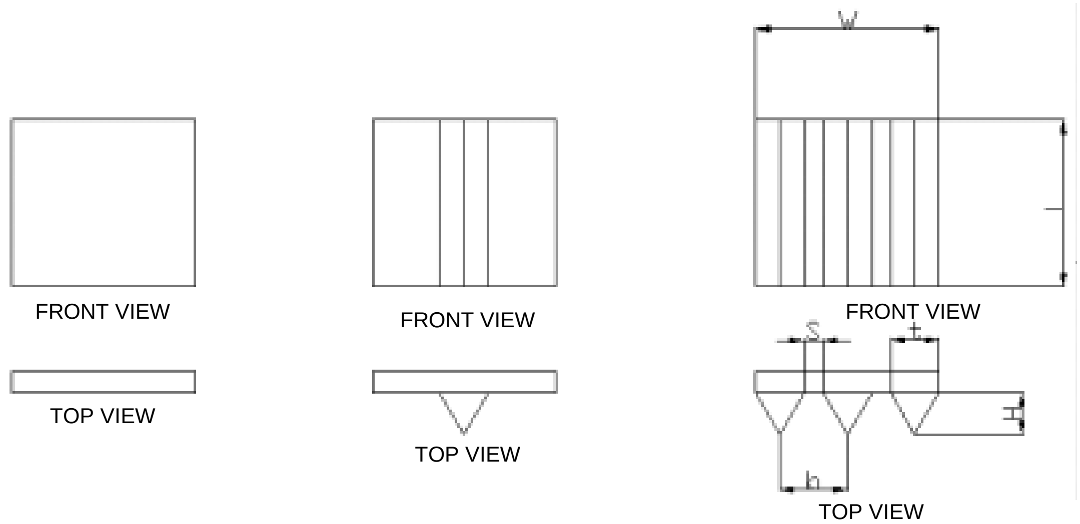

Fig. 2: Different specimens that were used in the experiment

Table 1: Dimension of test specimen

	L (m)	W (m)	H (m)	S (m)	b (m)	A_s (m)
Flat plate	0.215	0.215	--	--	--	0.0462
Plate with single fins	0.215	0.215	0.0385	--	--	0.0602
4 –fin arrays	0.215	0.215	0.04	0.45	0.0525	0.1033
7 –fin arrays	0.213	0.216	0.04	0.01566	0.02266	0.1492
13 –fin arrays	0.215	0.215	0.04	--	0.0075	0.2225
		t_{base} = 0.010 m			t = 0.015 m	

Table 2: Effect of Temperature on Emissivity for different fin arrays

Sl. No.	Single Fin with Flat Plate		Array of 4 fins		Array of 7 fins		Array of 13 fins	
	T_b`	ε	T_b`	ε	T_b`	ε	T_b`	ε
1	31.13	0.57	32.30	0.59	22.80	0.59	32.88	0.68
2	31.19	0.57	33.20	0.594	33.74	0.595	35.05	0.689
3	31.44	0.58	35.47	0.598	38.45	0.60	34.51	0.69
4	33.31	0.59	37.38	0.60	42.48	0.605	38.79	0.696
5	35.68	0.59	45.68	0.61	43.85	0.61	47.78	0.71

Table 3: Array of 4 fins

Sl. No	Q_{Tot} = VI	$\overline{T}_b$`	T_a	T_F	ΔT	Ra_b	Nu_b (expt)	Nu_{rad} (expt)
1	0.179	31.326	30.9	31.164	0.426	3.4×10^2	1.932	7.59
2	0.37889	31.872	31.0	31.541	0.872	1.91×10^3	2.625	7.662
3	0.6972	32.3	31.3	31.882	1.0	3.991×10^3	3.32	7.63
4	1.695	33.2	30.7	32.25	2.5	7.62×10^3	4.25	7.65
5	2.4637	35.47	31.3	33.886	4.17	1.2379×10^4	5.715	7.66
6	4.8465	37.3857	30.9	34.92	6.4857	1.89×10^4	6.447	7.69
7	12.34249	45.6857	31.0	40.105	14.6857	3.9×10^4	7.64	7.9

Table 4: Heat transfer enhancement due to increase of Ra for different fin arrays

Sl No.	Single fin with flat plate			Array of 4 fins			Array of 7 fins			Array of 13 fins		
	Q_{fin}	$Q_{eq.unfin}$	Q_{tot}	Q_{fin}	$Q_{eq.unfin}$	Q_{tot}	Q_{fin}	$Q_{eq.unfin}$	Q_{tot}	Q_{fin}	$Q_{eq.unfin}$	Q_{tot}
1	.0051	.005	.039	.0430	.030	.1790	.205	.09	.700	.183	.100	.093
2	.0110	.010	.5790	.1198	.068	.3788	.405	.151	1.144	.427	.120	1.212
3	0.020	.018	.9210	.1740	.090	.6970	.632	.214	1.812	1.070	.300	3.010
4	.2520	.225	.9314	.5560	.248	1.695	1.33	.420	3.210	2.770	.830	7.148
5	--	--	--	1.247	.502	2.463	6.67	2.03	14.08	20.38	6.00	40.37

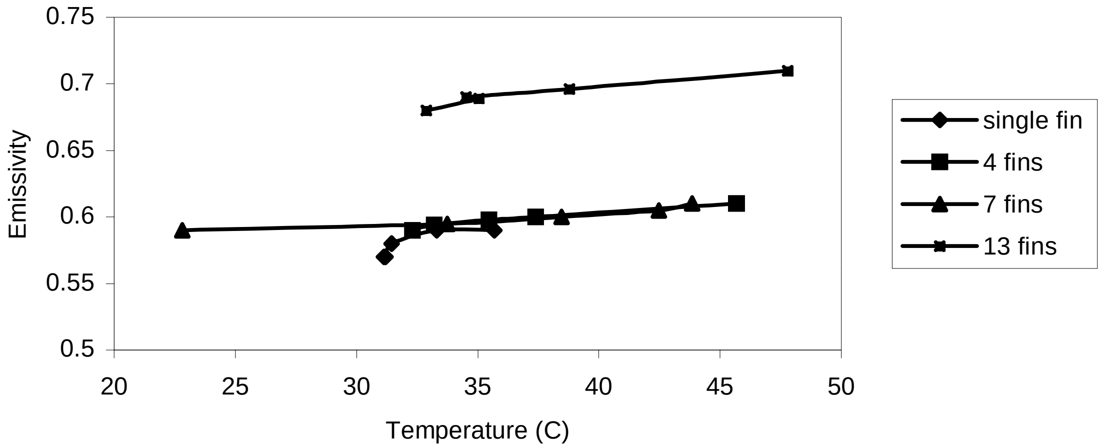

Fig. 3: Effect of temperature on emissivity for different fin arrays

RESULTS AND DISCUSSIONS

Investigations were carried out for natural convection heat transfer from different triangular fin arrays. Some of the experimental data (heat input, prime surface and fin surface temperature, ambient temperature, emissivity, etc.) are given in table 1-3. The outcome of the experiments were expressed by dimensionless group, Nusselt & Rayleigh number and was incorporated in the above mentioned tables.

Fig. 3 represents the experimental measurement of emissivity for emissivity for different fin arrays. The emissivity of the fin arrays increases with the increase of fin surface temperature. At a definite temperature emissivity of the fin arrays increases with the increase of number of fin in the array. The results of the emissivity shows that there is no variation of emissivity in between 4 fin and 7 fin array at a definite temperature.

Fig. 4 shows the comparison of radiation and convection heat transfer from an array of four fins. Similar graphs can be plotted for different number of fins per array but are not incorporated in this paper. The radiation heat transfer, Q_{rad} was calculated by using equation 3. Q_{rad} was used to calculate "radiation Nusselt number" Nu_{rad}, where Q_{conv} in equation 1, is replaced by Q_{rad}. It is clear from this figure that failure to account the radiation will result a severe under-estimation of the heat loss from the fin surface.

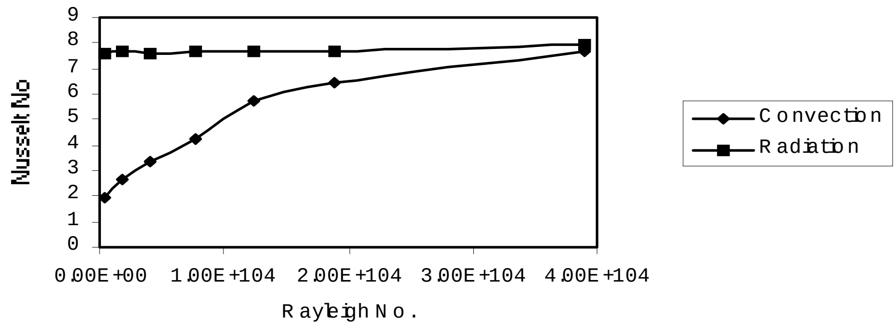

Fig. 4: Effect of radiation heat loss on convection heat loss from an array of 4 fins

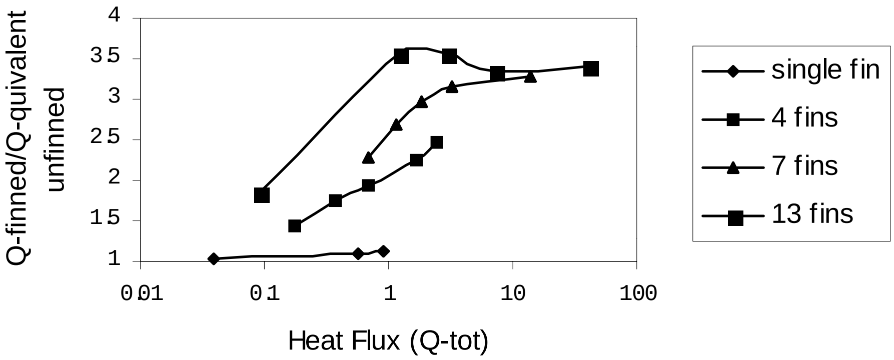

Fig. 5: Heat transfer enhancement due to increase of Ra for different fin arrays

The heat transfer augmentation with the increase of total heat flux for different fin array is depicted in Fig. 5. The Fig. 5 shows that natural convection heat transfer increases with the increase of number of fin in the array.

Experimental result depicts the results of natural convection heat transfer from different fin arrays. The results obtained from the generated data were also compared with that of a single fin and flat plate. From experimental data, it is evident that for an array consists of 13 fins, Nu number increases approximately at a constant rate with the increase of Rayleigh number from 0.439 to 9.714. For an array consists of 7 fins, Nu number increases approximately at a constant rate with the increase of Rayleigh number from 47.8 to 344.7. And for an array consists of 4 fins, Nu number increases approximately at a constant rate with the increase of Rayleigh number from 340 to 3991.

From the experimental data of this work the following correlation may be obtained for Rayleigh number ≥ 10^4:

$$Nu_L - 3.524 = 0.515Ra^{1/4}; \qquad Ra \geq 10^4$$

The measured Nu values are best fit by the above equation. The data that were plotted does not deviate more than 3.7 percent.

Analysing generated experimental data, again the following correlation may be recommende:

$$Nu_L - 0.779 = 0.515Ra^{1/4}\left(1+\left(\frac{3.26}{Ra^{0.21}}\right)^3\right)^{-1/3}$$

$$\text{for } 10^{-1}\langle Ra \langle 10^4$$

The measured Nu values are best fit by the above equation. The data that were plotted deviates not more than 1.6 percent.

CONCLUSION

The present work reports the measurements of natural convective heat transfer from triangular fin arrays to ambient air. The important conclusions as a consequence of the present investigation are enumerated below:

- The investigations revealed that the use of fin arrays provide better results only up to Rayleigh number less than or equals to 10^4, beyond that fin arrays does not enhance the heat transfer.
- For Rayleigh number greater than or equals to 10^4, the following correlation may be used in estimating

the natural convection heat transfer from triangular fin arrays.

$$Nu_L - 3.524 = 0.515Ra^{1/4}; \quad Ra \geq 10^4$$

- For Rayleigh number from 0.1 to 10^4, the following correlation may be used:

$$Nu_L - 0.779 = 0.515Ra^{1/4}\left(1+\left(\frac{3.26}{Ra^{0.21}}\right)^3\right)^{-1/3}$$

$$\text{for} \quad 10^{-1} \langle Ra \langle 10^4$$

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