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Heat Transfer Analysis of Continuous and Staggered Fin Arrays

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Abstract **- There are many ways to transfer heat from one medium to another medium efficiently. Heat dissipation is an important issue to tackle due to compacting of equipment and miniaturization. Heat exchangers are not only chosen for their thermal better performance but also for their other design parameters such as weight, cost, ease of handling etc. This paper briefly focuses on the study of continuous and staggered fin arrays.**

Keywords**: Staggered fin arrays, Heat enhancement, Heat transfer coefficient, Nusselt number**

I. INTRODUCTION

Heat exchanger is a device which transfers heat energy from one medium to another medium. It is very essential to remove excessive heat from any system component e.g electronic devices, to avoid the serious damaging effects or overheating or burning. The rate of heat transfer from any surface may be increased by increasing heat transfer coefficient between that surface and surroundings or by increasing surface area. For the improvement in performance of any heat exchanger several heat transfer enhancement techniques have been successfully implanted. These techniques may be classified as-active, passive and combined. There is an important passive technique in which surface area of the system is increased which is commonly known as *extended surface* or *fin*. Extended surface or fins are commonly used to enhance heat transfer in many industries, power generators, automobiles etc. Now days, various types of fins like rectangular, circular pin fins, plate type fin, rectangular pin fins are widely used for both natural and forced convection. In 2006, Ugur and Kadir experimentally investigated the heat transfer characteristics in a horizontal rectangular channel having a rectangular fin. They studied and observed different flow pattern depending upon different flow rates. In 2009,Suryawanshi and Sane had modified geometry of fin arrays. They used fin arrays with inverted notch at the central bottom portion of the fin to modify fin geometry to enhance heat transfer characteristics. They experimentally found the average heat transfer coefficient for inverted notched fin arrays is nearly 30 to 40% greater than normal arrays. Many of the earlier investigators have studied problems concerned with different orientation of fin arrays, both theoretically and experimentally. It is proved that the use of special surface geometry or modification of surface may allow higher heat transfer coefficients than those given by common plain fin arrays. There are many different situations that involve combined conduction-convection effects, the most frequent application is one which the extended surface specifically used to enhance the heat transfer rate between solid and adjoin fluid.

II. FIN ARRAYS SYSTEM

Fins are the extended surfaces. The term 'extended surface' is commonly used to a solid where energy transfers by conduction as well as energy transfer by convection or radiation between boundary and surroundings of the system. Fins are classified as,

According to cross section
1. Uniform cross section
2. Non-uniform or Varying cross section

According to shape and geometry
1. Triangular fin
2. Rectangular fin
3. Trapezoidal fin
4. Parabolic fin
5. Annular fin
6. Pin fin or Cylindrical fin

The term fin arrays are referred to systematic arrangement of fins on a base plate. For better thermal performance, fins can be arranged systematically to enhance heat transfer characteristics from the system component.

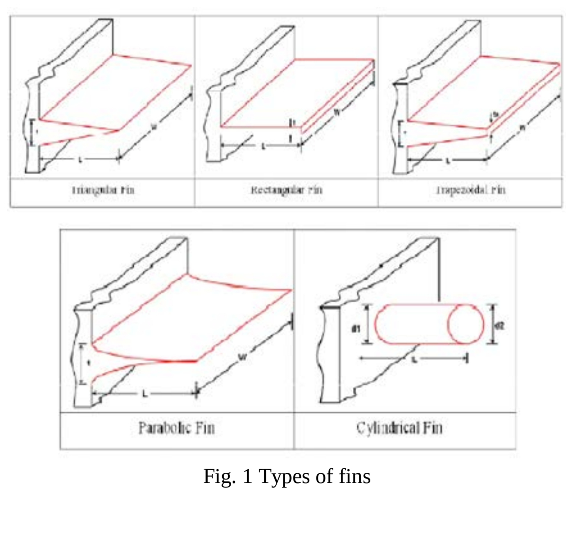

Fig. 1 Types of fins

II. NATURAL CONVECTION AND RELATED SIGNIFICANT PARAMETERS

In Natural or free convection, the type of process wherein fluid motion results in heat transfer. When any fluid is heated or cooled, the change in density and buoyant force produce a natural circulation in which the fluid moves of its own and it is replaced by similarly affected fluid by the energy transfer and process is continued.

1. Nusselt Number (Nu) - It is ratio of conductive heat transfer to convective heat transfer. The dimensionless expression is,

$$Nu = \frac{hL}{k}$$

Where h=convective heat transfer coefficient
L=Characteristics length (In present study, fin height H)
k=Thermal conductivity of fluid

2. Prandtl Number (Pr) - It is the ratio of molecular diffusivity of momentum to molecular diffusivity of heat. The expression is given as,

$$Pr = \frac{\mu\, Cp}{k}$$

Where µ=absolute or dynamic viscosity
C_p=Specific heat capacity
k=Thermal conductivity of fluid

3. Grashoff Number (Gr) - In natural convection, the flow is produced by buoyant effects resulting from temperature difference. These effects are included in the Grashoff number. The Grashoff number is the ratio of buoyancy force to viscous force of a fluid. A critical value of the Grashoff number is used to indicate transition from laminar to turbulent flow in free convection.

III. EXPERIMENTAL SETUP

The experimental set up is as shown in fig.2. Thermal analysis is done on continuous and staggered fin arrays. All fin arrays are equi-spaced. The fin dimensions are 200mm X 180mm. The thickness of fin is 3mm. The material of fin used is Aluminium-Silicon alloy.

Fig. 2 Experimental set-up

The experimental setup consists of test section, dimmerstat, power supply etc. The test section includes fin arrays, having base plate connected to an electric heater. The fins are attached to the base plate with aluminum tape to get minimum contact resistance and heat loss. Total 18 K-thermocouples are used to measure temperatures at different location of the setup.

A. Arrangement of Fin Arrays

Set-I is of staggered fin arrays having 40% staggering lengthwise. Set-II is of continuous fin arrays of same length as that of first one. The fin/assembly is made of aluminum alloy because of its high thermal conductivity, low emissivity and easy machinability. The wooden insulation is provided for heat losses from side of experimental setup. The test section is as shown in fig.3. selection of appropriate materials to conduct and transfer heat at better rate and efficiently is very important criteria in heat exchanger design. In this experiment, the Copper material is selected for inner tube as it is thermally efficient and durable and outer tube is made of M.S. material.

Fig. 3 Test section

Specifications of test loop:

Length of test section = 200 mm
Thickness of fin = 3 mm
Type of thermocouples = K-type (16 in Number)
Heater = Mica Paper Heater (2 in Number)
Insulator = Bakelite and Wood

Data reduction

The convective heat transfer rate (Q_N) from electrically heated surface is calculated by using relation,

$$Q_N = Q_{electrical} - Q_{conduction} - Q_{radiation} \quad (1)$$

The electrical heat input is calculated from electrical potential and current supplied to the surface.

$$Q_{electrical} = V \times I \quad (2)$$

The conduction heat transfer is calculated by using the relation,

$$Q_{conduction} = -k \times A \times \frac{dT}{dx} \qquad (3)$$

The radiation heat losses are neglected during test.
The heat transfer rate from any surface by convection is given by the relation,

$$Q_N = h \times A \times \Delta T \qquad (4)$$

where 'A' is the area of base area of fin and ΔT can be calculated as,

$$\Delta T = T_S - [\frac{T_i + T_o}{2}] \qquad (5)$$

From equation (4), heat transfer coefficient (h) is calculated and dimensionless numbers like Nusselt number is calculated as,

$$N_u = \frac{h \times L}{k} \qquad (6)$$

Also, the product of Grashoff number and Prandtl number is calculated.

IV. RESULTS AND DISCUSSION

A. Effect on Average Heat Transfer Coefficient (h)

TABLE I CONTINUOUS FIN ARRAYS AND STAGGERED FIN ARRAYS

Heat input	49.5		66.5		88		110.25	
Parameter	$h (W/m^2k)$	% rise in h_a		% rise in h_a		% rise in h_a		% rise in h_a
Set 1	13.98	-	14.8	-	15.0	-	15.3	-
Set 2	18.01	28.6	18.1	22.6	18.3	22.5	20.2	34.6

Above tables shows the result continuous fin arrays (Set-1) and staggered fin arrays (Set-2). By using above results, a graph of heat input (Q) versus heat transfer coefficient (h) can be plotted as,

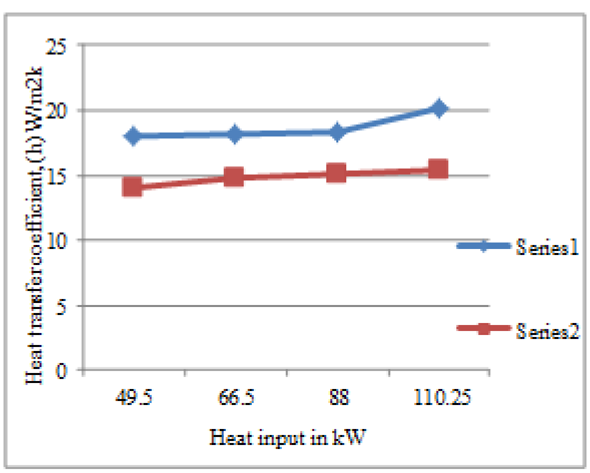

Fig. 4 Graph of Heat transfer rate (Q) v/s Heat transfer Coefficient (h)

Above graph shows the relation between heat input and heat transfer coefficient as the average heat transfer coefficient increases 20 to 30% in case of staggered fin arrays than continuous fin arrays. This is because as the amount of heat supplied is higher the average fin array temperature increases. As surrounding temperature is near to constant temperature difference which is the cause of convection increases and it causes the higher heat transfer rates.

B. Effect on Average Nusselt Number (Nu)

The increase of heat transfer through a fluid layer due to convection when compared to conduction across the same fluid layer is calculated by the Nusselt number. Larger the Nusselt number, the more effective the convection. Fig.5 shows that average Nusselt number (Nu) increases in heat input for each set. This is because fin array temperature is increased due to higher heater input and temperature is near to constant. It is found that Nusselt number increases 20 to 30% in case of staggered fin arrays than that of continuous fin arrays.

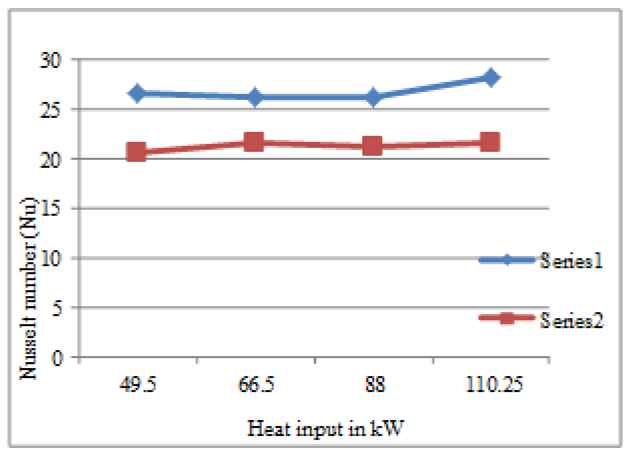

Fig. 5 Graph of Heat transfer (Q) v/s Nusselt number (Nu)

C. Temperature Difference versus Average Heat Transfer Coefficient

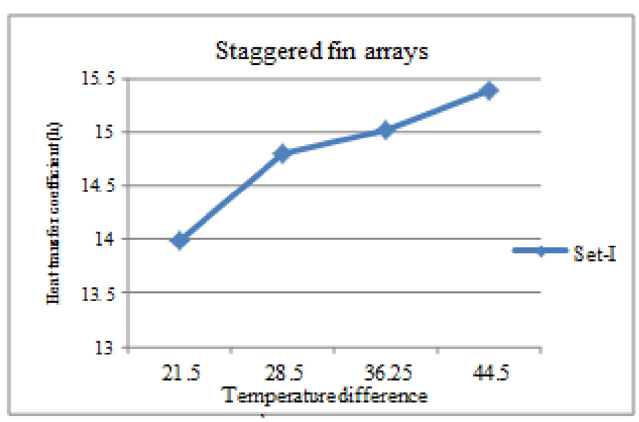

Fig. 6 Graph of Temperature difference (ΔT) v/s Heat transfer coefficient (h) for staggered fin arrays

Heat transfer rate by convection is given by the basic equation Q = h.A.ΔT. As heat input for each set is fixed, increase in heat transfer coefficient. Fig .6 and Fig.7 shows graph of temperature difference versus heat transfer coefficient for staggered fin arrays and continuous fin arrays respectively. It is found that better results were obtained in case of staggered fin arrays than that of continuous fin arrays. The main reason behind that more heat is transferred to atmosphere from staggered fin arrays.

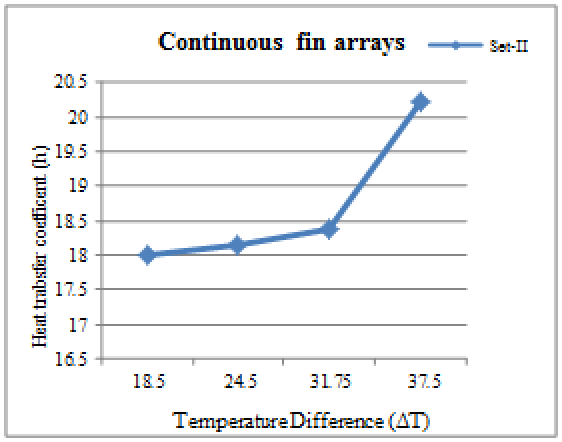

Fig.7 Graph of Temperature difference (ΔT) v/s Heat transfer coefficient (h) for continuous fin arrays

D. Heat Input versus Temperature Difference (ΔT)

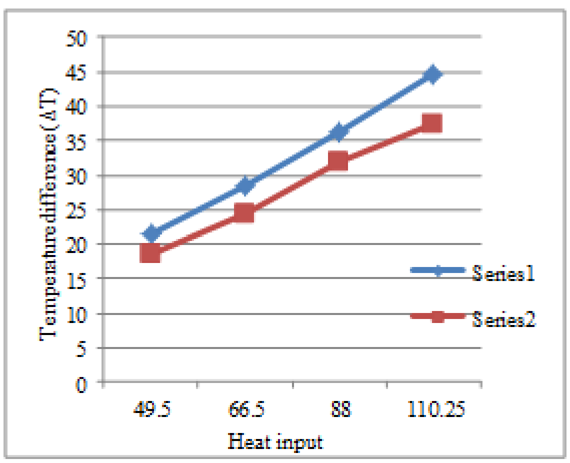

Fig. 8 Graph of Heat input v/s Temperature Difference (ΔT)

Fig. 8 shows the comparison graph of heat input versus temperature difference temperature clearly showing more temperature difference for the same heat input.

E. Percentage increase in Nusselt number

Fig.9 shows the percentage increase in Nusselt number for staggered fin arrays. It is experimentally found that Nusselt number 20 to 30% increases in case of staggered fin arrays as compared to continuous fin arrays.

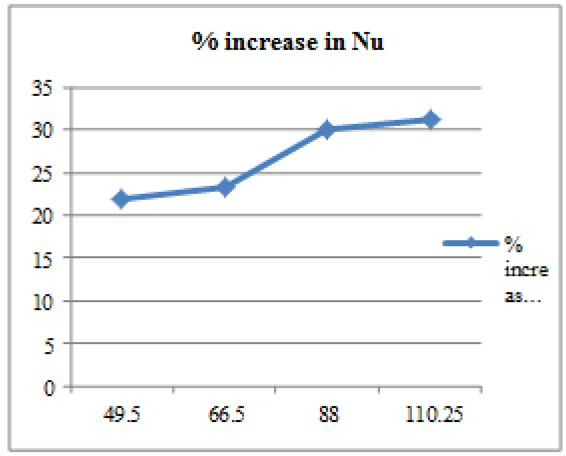

Fig.9 Percentage increase in Nu

V. CONCLUSION

Experimental study of staggered fin arrays and continuous fin array is performed at various heat input. Staggered arrays under study are tested under natural convection and compared with continuous fin array. The heat transfer rate increases in case of staggering because of enhancement in flow pattern leading to increased wetted surface. In case continuous fin array some air passes ineffectively (i.e. It doesn't make contact with fins) between two successive fins but in case of staggered fin array due to staggering of fins air makes full contact with fin surface.

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