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Heat Transfer Enhancement of Perforated Square Pin-Fin Array in a Rectangular Duct

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Abstract*: This investigation reports the analysis of heat transfer of a rectangular plain surface which equipped with square perforated pin fins in staggered arrangement in a rectangular duct. Dimension of the base plate in the rectangular duct 145mm x 220mm. Rectangular plain surface (Base plate) which is attached with a heater. The range of Reynolds number is fixed & about 2000– 12,000, the clearance ratio (C/H) as 0.0, the inter-fin spacing ratio (Sy /D) are 2.4, 3.6 and 4.8. Stream wise distance Sy is kept at constant and span wise distance Sx varies. The friction factor, enhancement efficiency and heat transfer correlate in this equation with each other. Here we are comparing square perforated pin fin with different number of perforations (n) and different dia. of perforation (Dp) of pin fins with each other. Staggered arrangement and optimum perforation modification will enhance the heat transfer rate and they relatively influence on the flow characteristics. Friction factor & Nusselt number are the key parameters which relates with efficiency enhancement and heat transfer rate.
Keywords*: Flow characteristics, Heat Transfer Enhancement, Perforated square, Pin fins, Performance Analysis, Staggered arrangement.

I. INTRODUCTION

For cooling of electronic based components , pin fin heat exchanger is one of the most efficient Passive cooling techniques. Pin fin arrays are often used in many industrial applications in order to augment convective heat transfer. The rate of heat transfer at a solid–fluid interface can be attained more by extending the surface area in the form of fins. Various types of fins, ranging from relatively simple shapes, such as rectangular, square, cylindrical, annular, tapered, and pin fins, to a combination of different geometries, have been used in different heat transfer applications. These are the most popular fin types because of their low cost of production and high thermal effectiveness. Study on the influence of geometric parameters viz. fin height, Fin length, fin spacing over heat dissipation is found (Sara [1], Chyu et al. [2] and Tahat et al. [3]).Heat transfer characteristics of pin fin have undergone great impact of the research in most heat exchanger applications. To increase heat transfer and reduce fluid flow loss due to the pin-fin arrays, it is need to understand the physical mechanisms that govern the heat transfer and pressure drop parameters. Earlier investigation on pin-fin arrays done by Sparrow et al. [4] and Tahat [5]. They experimentally investigated in-line and staggered pin-fin arrays fixed in an internal cooling channel. Metzger et al. [6] investigated the effects of span wise and stream wise distances between pin-fins. They found that the area averaged heat transfer coefficient on the heated surface depends on the distances between pin-fins for both staggered and in-line arrays. Chyu et al.[7] evaluated the heat transfer performance of pin-fin arrays in a cooling channel. The heat transfer rate was enhanced by at least a factor of two while introducing one of the pin fin arrays in the smooth channel, and the staggered pin fin arrays resulted higher thermal performance than the in-line arrays. Deqing et al. [8] experimentally investigated for the thermal hydro dynamic characteristics on micro-pin fin arrays. Isak kotcioglu et al. [9] studied on the cross flow heat exchanger using rectangular channel to estimate. The average heat transfer rate varies with Reynolds number, constant span wise ratio and change in stream wise ratio. Shaeri et al. [10] investigated experimentally heat transfer characteristics from heat sink with using perforated fins. The pin fin surface technology has broad applications that try to find out new design ideas, including fins made of perforated and interrupted plates. Due to the more requirements of compact, lightweight, and cost effective fins, the optimization of fin geometry is of importance. Therefore, fins must be designed to attain higher heat removal with lower material expenditure. Sahin et al. [11] used perforated square fins are commonly used different heat exchangers, solar collector and film cooling, applications as a result of their high heat transfer capacity at a relatively low material usage. Rasim Karabacak , Gülay Yakar [12] experimentally investigated the effect of holes placed on perforated finned heat exchangers in order to increase convective heat transfer. The holes created turbulence in a region near the heating tube surface on the base of the fin. They performed to analyze the effect of this turbulence on heat transfer with pressure drop. Kavita H. Dhanawade et al. [13] studied the heat transfer Enhancement over horizontal flat surface with rectangular fin arrays with lateral square and circular perforation by

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forced convection . They varied sizes of perforation and found that average of percentage improvement of square perforated fin arrays is more than circular perforated fin of same size. Friction factor slightly increases with increase in the size of perforation. A.B.Ganorkar, V.M.Kriplani [14] studied overall performance of suitably designed perforated fins in a rectangular channel. Different types of perforated fins are used to analyze the effect of perforated fins in a rectangular duct which is observed for different Reynolds numbers. Reynolds number range taken 2500-10000, diameter range of perforated holes 6-10 mm.They reported As Reynolds number increases the ratio of Nusselt numbert of perforated fin to Nusselt number of solid fin ($N_u$perforated/$N_u$solid) increases. Increase in no. of holes, the enhancement ratio (Nu_p/Nu_s) increases. Increase in diameter of perforation the enhancement ratio increases. The review of the literature showed a variety of modifications and the alterations of the fins by introducing the holes, slits and struts, which augment the heat transfer; however, the usage of ribs or struts are not literally recommended. This aspect of choice increases the mass or weight of the existing fin and in such cases the usage of grooved fin is not advisable due to the reduction of material. However, it is noticed that the effectiveness of heat transfer in the perforated type of fins are larger for the same conditions of the temperature difference, which is basically a driving potential. The literature survey also reveals very few investigations in the literature showed the effect of geometrical alteration in modifying the characteristics of the heat transfer; tested on the perforated type of fin.

II. DATA REDUCTION

The data generated during the various trials are used to evaluate the heat transfer and friction characteristics. The steady state heat transfer from the finned surface is,

$$Q_{tot} = Q_{conv} + Q_{rad} + Q_{loss} \quad (1)$$

In the present study the data reduction is similar to that followed by Tahat et al. (2000) and Sara (2003). They conducted experiments and fin arrays are comparable and reported that the total heat loss from the assembly ensue less than 5%. Under the present operating conditions together with the fact that the test section is well insulated and assuming the loss is very minimum, eqn. (1) is rewritten as,

$$Q_{conv} = mc_p(t_{out} - t_{in}) \quad (2)$$

The heat transfer by convection from finsurface including base plate is given by,

$$Q_{conv} = h A_s \left[t_b - \left(\frac{t_{in} + t_{out}}{2} \right) \right] \quad (3)$$

where, t_{in} and t_{out} are the temperatures ofair flow, t_b is the average temperature at certain designated locations on the base assembly and A_s is the total surface area of base assembly and fins, which is given as,

$$A_s = WL + \pi d H N_{xy} - \frac{\pi d^2 N_{xy}}{4} \quad (4)$$

The average heat transfer coefficient for the heated pin-fin assembly can be calculated by combining eqns. (2) and (3) under the present operating conditions together with the fact that the test section was well insulated.

$$h = \frac{m c_p (t_{out} - t_{in})}{A_s \left[t_b - \left(\frac{t_{in} + t_{out}}{2} \right) \right]} \quad (5)$$

The free flow area A_{ff} is calculated as,

$$A_{ff} = W(H+C) - N_x H d \quad (6)$$

The Reynolds number (Re) is defined in the conventional way as,

$$Re = \frac{G d}{\mu} \quad (7)$$

where, $G = m/A_{ff}$ is the mass flux.

$$N_u = h_{av} D_h / k_{air}$$

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III.EXPERIEMENTAL SETUP

The various components involved in the set-up are given below.

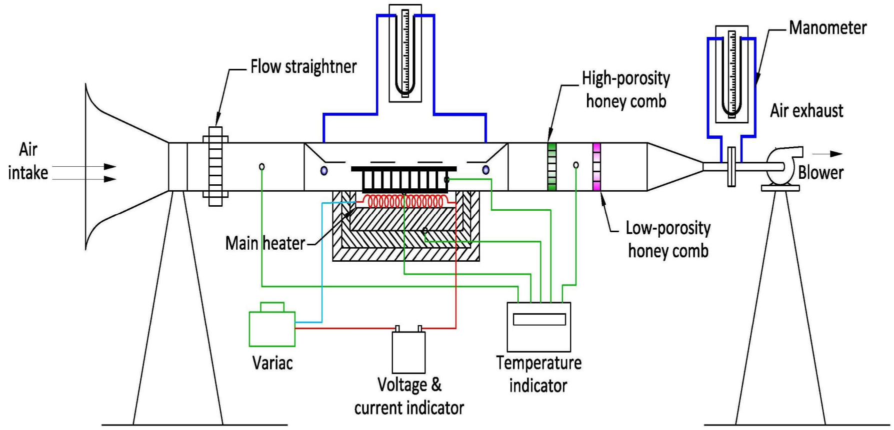

Fig-1: schematic diagram of experimental setup

A. Pin-fin assembly

The pin-fin arrays considered for this experimental study are having cylinders (fin-size 90 mm of height and 10mm of diameter) are of protruding vertically upwards from a 250 mm x 145 mm horizontal rectangular base having thickness 25.4 mm as shown in Fig.1. The minimum to maximum numbers of pin-fins used in this investigation are 36 to 60 respectively. Spacing is varied from 12 to 60 mm in the span-wise direction and from 12 to 60 mm in the stream-wise direction. The rectangular base as well as the pin-fins was manufactured from a light aluminum material (i.e.; duralumin).

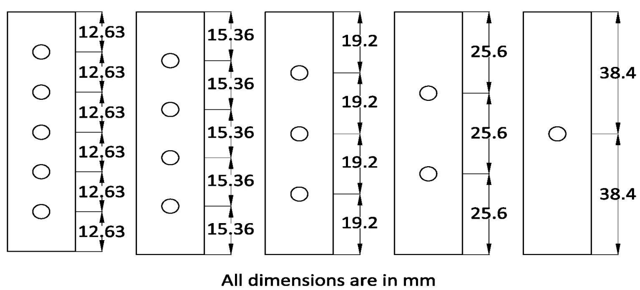

Fig-2: Perforated pin fin model

For each test, the pin-fin height was kept constant with clearance ratio(C/H) as 0.0.This (C/H) is the ratio between the tips of the pin-fins and to the shroud (adjustable roof). The pin fins were perforated with variable diameters (2 mm,4mm,6mm) and with the variation of 1 to 5 numbers also at different locations.

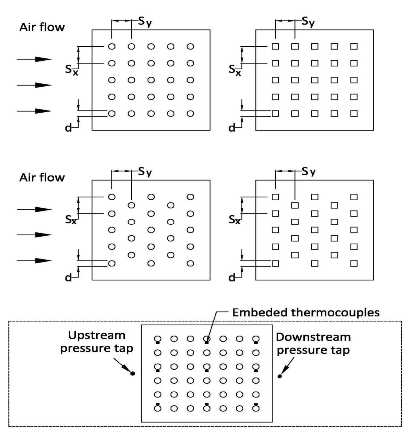

Fig-3: Pin fin arrangements

B. Wind-tunnel

A rectangular shaped wind-tunnel duct was manufactured by 19 mm thick plywood and about 2 m long with a non variable internal width of 150 mm. Fig.1 shows the schematic of the experimental set-up. A bell-mouth section was fitted at the entry of the wind-tunnel duct followed by a low porosity, cardboard honeycomb flow-straighter. The heated air from the pin-fin assembly was passed through an insulated chamber, where mixing was accomplished by two cardboard honeycombs fitted perpendicular to the flow-stream, one being of relatively low porosity and the other of higher porosity. The latter was situated upstream of the air flow. At the exhaust end of the duct, a gradual area-contraction section attached is used to connect a single-speed, single-stage blower (via G.I. pipe). Blower has the capacity of providing a maximum flow rate of 0.242 kg/s, and is preceded by a butterfly throttle control valve. A differential manometer was employed to observe the pressure drop across calibrated orifice plate to indicate mass-flow rate of the air. The wind tunnel was operated in the suction mode, i.e. the blower induces atmospheric air via the bell-mouthed entrance section that flow through pin-fin assembly in the test section. This avoids the air-stream being heated by the blower during compression/friction prior to its travel to the heat-exchanger assembly and enhanced the cooling capability of the air. A plate electric heater having nichrome wire wound on a mica sheet and covered on either side with another mica sheet and electric connections are also provided appropriately to supply power for heating. It was capable to deliver 1500 W. The heat exchanger base was heated uniformly to maintain constant temperature. The whole heat-exchanger base, the main heater with associated thermal insulations, was located and protected in a well-fitted open-top wooden box. The upper edges of this box and the top surfaces of the laterally-placed thermal insulant are levelled in order to get smooth flow with the upper surface of the rectangular base plate where the fins protrude upwards.

C. Heating system

The steady-state temperatures at the base of the pin-fin array are measured by a nine set of copper-constantan (T-type) thermocouples embedded and appropriately distributed throughout the rectangular base. The base plate was thick enough to maintain uniform temperature of 50^{o}C rather than uniform heat flux achieved normally employing a thin plate heater. The inlet and the outlet air-stream temperatures in the wind-tunnel duct were measured by employing eight thermocouples and 6 RTDs. Experiment is continued for half an hour after steady-state conditions were attained.

IV.RESULTS AND DISCUSSION

The experiment is carried out on the test rig with and using perforated fins (i.e. passive heat transfer enhancement methods). Heat transfer coefficient and Nusselt Number are calculated for all the observations. From the outcome it is found that heat transfer rate is more in the event of perforated fins with the different perforation diameter and number of perforations as compare to solid fins. This is because degree of turbulence increases by inserting the fins inside the duct. Heat transfer rate is further increased by using perforated fins. Thermal and flow characteristics for a particular design of perforated fins are better for all ranges of Reynolds Number which is discussed below .

A. Effect of location of perforation

The location of perforation on the pin fin is varied to study the influence on thermal performance. Chart-3 shows the location effect of perforation for staggered cylinder .

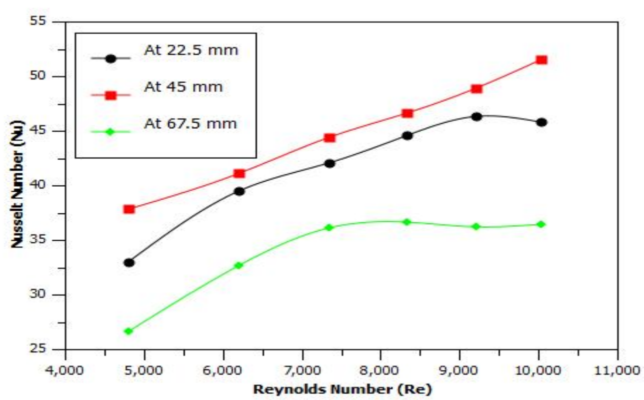

Chart- 1 Effect of Nusselt number Vs Reynolds number in staggered arrangement

Three configurations are considered for this study i.e., 22.5 mm, 45 mm and 67.5 mm from the base. Among these arrays the pin-fiarray with perforation at 45 mm from the base indicated a higher rate of heat transfer than that of other location 22.5mm and 67.5mm perforation. This indicate or the reason is that perforations at 45mm in the fins introduce convection rates higher in addition to, that perforation at middle position affecting the wake region flow and thereby main flow strikes on the downstream pin-fin. In other words the flow through the perforations at that point not only increases the turbulence but also control the flow separation in comparison to that of others.

B. Effect of number of perforations

The number of perforations on the pin fin is varied to study the heat transfer characteristics. The results are shown in chart-4 .

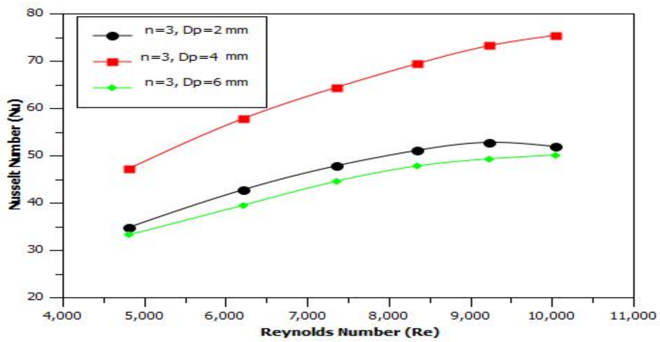

Chart- 2 Effect of Nusselt number Vs Reynolds number in staggered arrangement

It is evident from the graph that the Nusselt number increases with increasing number of perforations. This is due to the excessive air exposed area and surface contact area made by increasing number of perforation. Further increasing the no. of perforation which affects conduction heat due to the loss of material density .So it has resulted less heat transfer enhancement. In other case lower number of perforation results less heat transfer enhancement because of having reduced surface contact area.In this case, The 3 number of perforation with perforation diameter of 4 mm results better in thermal performance. In this study, it is the optimal number and diameter of the perforation on the pin fin.

C. Effect of diameter of perforation

The diameter of the perforation on the pin fin is varied in this study.

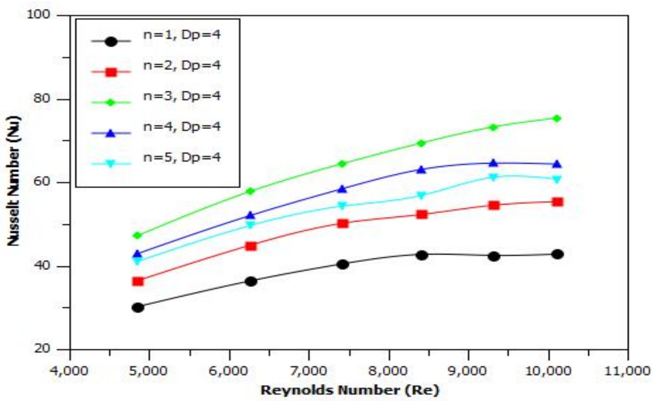

Chart- 3 Effect of Nusselt number Vs Reynolds number in staggered arrangement

IV. CONCLUSIONS

The various heat transfer parametric results are studied for the different cases viz. cylindrical perforated pin fins with different perforation diameter and different number of perforations including locations. Experimental investigations have been carried out in the rectangular duct to study the effect of heat transfer enhancement and flow properties. These characteristics are compared to each other modification and the conclusions are made as follows .

A. For certain packing density of the pin-fin (or N_{xy}) there exist maximum values of Nusselt number.
B. The average Nusselt number increases monotonically with increasing Reynolds number.
C. Perforation at certain height from the base shows better performance .
D. Number of Perforation affects the thermal performance. In optimum number of perforation shows better thermal performance.
E. Diameter of perforation influences on thermal performance. At optimum diameter of perforation gives better thermal performance.

V.NOMENCLATURE

A area, m^2
C clearance between fin tip and the roof,mm
C_p specific heat of air, J/kg K
d diameter of the pin-fin, mm
D_p diameter of the perforation on pin fin,mm
f friction factor
G mass flux, kg/m^2 s
H height of the pin-fin, mm
K thermal conductivity, W/m K
L length of the base plate, mm
M mass flow rate of air, kg/s
N number of pin-fin
n-number of perforation
Nu Nusselt number
Q heat transfer rate, W
Re Reynolds number
S spacing temperature (^{0}C)., mm
T_b- Base plate

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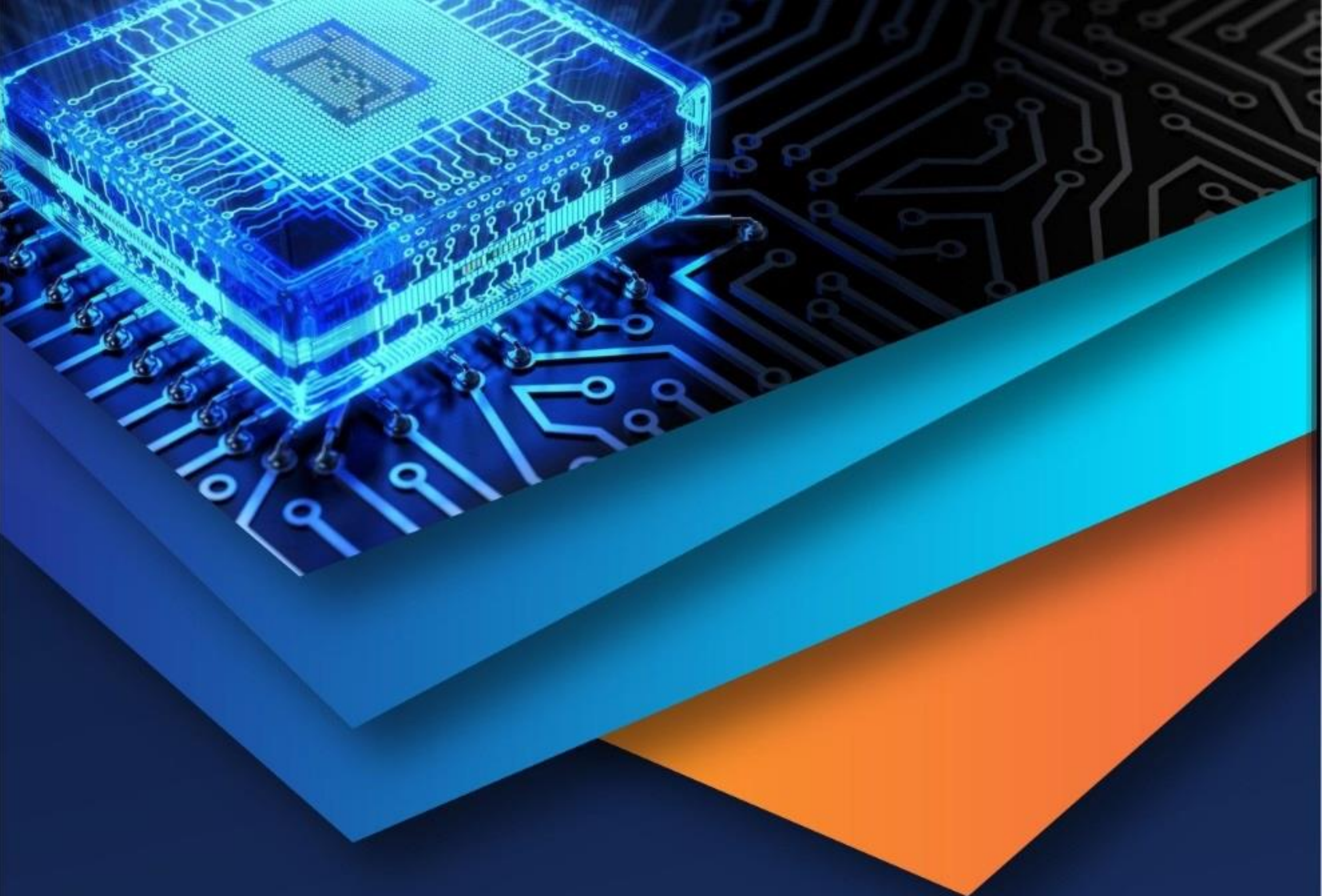

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