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Thermal Analysis on a Finned Tube Heat Exchanger of a Two Stage Air Compressor

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Abstract: Compressors are intended to compress a substance in a gaseous state. Air is compressed in more than one stage when high pressure ratio is required in view of reducing the power input to the compressor.In almost all multi stage applications, air will be cooled between stages which is also known as intercooling.With intercooling, the compression more closely approximates an isothermal compression with resulting lower power requirement. Intercooler is basically a heat exchanger, which is used to cool the compressed air between stages. In the present work, thermal analysis has been done on finned tube heat exchanger (Intercooler of two stage air compressor). Experimental analysis was done on the intercooler (Finned tube heat exchanger) model for the particular pressure ratio. For the finned tube, inside and outside convection heat transfer coefficients are estimated based on the experimental data.Theoretical analysis was done to estimate the heat transfer rate at different sections of the bare tube as well as at the sections where the fins are present. Modified Bessel's functions of the first and second kind were calculated for the annular fin of rectangular profile model and temperature distribution over each fin was calculated. Finite Element Analysis was done using ANSYS software.

Temperature distribution at different sections and total heat flow is estimated for the finned tube with fins of rectangular profile and with fins of triangular profile. A comparison has been made among experimental data, theoretical data and finite element analysis.

Keywords: Extended surfaces, Heat Transfer, Bessel's function, Temperature distribution, Intercooler.

1. INTRODUCTION

Thermal Analysis is also often used as a term for the study of heat transfer through structures. Many of the basic engineering data for modeling such systems comes from measurements of heat capacity and thermal conductivity.

A reciprocating compressor or piston compressor is a positive-displacement compressor that uses pistons driven by a crankshaft to deliver gases at high pressure. The intake gas enters the suction manifold, then flows into the compression cylinder where it gets compressed by a piston driven in a reciprocating motion via a crankshaft, and is then discharged. Applications include oil refineries, gas pipelines, chemical plants, natural gas processing plants and refrigeration plants.

In the study of heat transfer, a fin is a surface that extends from an object to increase the rate of heat transfer to or from the environment by increasing convection. The amount of conduction, convection or radiation of an object determines the amount of heat it transfers. Increasing the temperature difference between the object and the environment, increasing the convection heat transfer coefficient or increasing the surface area of the object increases the heat transfer. Sometimes it is not economical or it is not feasible to change the first two options. Adding a fin to an object, however, increases the surface area and can sometimes be an economical solution to heat transfer problems. Finite element analysis (FEA) has become

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commonplace in recent years for the analysis of most of the systems which are subjected to different conditions.

2.SCOPE OF WORK

In the present problem, to analyze a finned tube heat exchanger, Intercooler of a Two stage reciprocating air compressor is considered. Intercooler consists of 20 fins of rectangular cross section, which are also known as annular fins of rectangular profile. An experiment was conducted on the compressor at different pressure ratio and temperature of air entering and leaving the intercooler was measured. The main objective of this work is to approximate the inside convective heat transfer coefficient and outside convective heat transfer coefficient. Temperatures at different sections of the finned tube is measured by using Infrared Thermometer. ANSYS is used to analyze the temperature distribution and heat flow for finned tube with fins of rectangular profile and fins of triangular profile.

3. EXPERIMENTAL DETAILS

3.1. Specifications of Finned tube Heat Exchanger(Intercooler):

Innerdiameter of Bare tube:34.65 × 10^{-3}m
Outer diameter of Bare tube:41.65 × 10^{-3} m
Tube material:Aluminium
Length of Intercooler:166.18 × 10^{-3} m
No. of fins:20
Fin material:Aluminium
Thickness of each fin:2.73 × 10^{-3} m
Type of fins:Circular fins of rectangular cross section
Height of fin:11.325 × 10^{-3} m

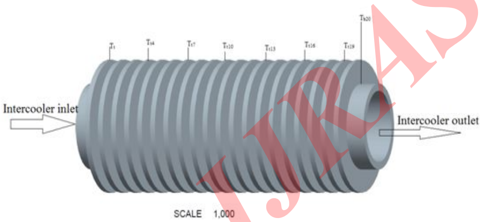

Fig.1.Extended surface Heat Exchanger

3.2. Experimental observations

Temperatures at different positions are measured for a particular pressure ratio by using thermocouples and Infrared thermometer.

T_b-Temperature at fin base ,T_t-Temperature at fin tip

S.No.	Temp.	(°C)
1	T_{int}^{in}	78
2	T_{int}^{out}	63.7
3	T_{t1}	73.7
4	T_{b2}	76.3
5	T_{t4}	77.7
6	T_{b5}	76.1
7	T_{t7}	76.7
8	T_{b8}	77.5
9	T_{t10}	77.7
10	T_{b11}	77.2
11	T_{t13}	78.9
12	T_{b14}	76.8
13	T_{t16}	74.7
14	T_{b17}	73.6
15	T_{t19}	74.0

Table.1.Temperatures on the surface of the intercooler

4. THERMAL ANALYSIS

4.1. Estimation of inside convective heat Transfer coefficient

Mean temperature of air inside the intercooler = 80 °C

Discharge (Q)= $c_d a_o \sqrt{(2gH_a)}$

Where c_d= coefficient of discharge of orifice -meter= 0.62

$$a_o = \frac{\pi}{4} d_o^2$$

$$H_a = \frac{\rho_{water}}{\rho_{air}}$$

$$d_o = 20\text{mm}$$

$$a_o = 3.14 \times 10^{-4}\text{m}^2$$

$$Q = 0.62(3.14 \times 10^{-4})\sqrt{(2 \times 9.81 \times 0.025)}$$

$$Q = 3.79 \times 10^{-3}\ \text{m}^3/\text{s}$$

$$V_{air_{inside}} = \frac{Q}{a_i}$$

$$a_i = \frac{\pi}{4} d_i^2$$

$$d_i = 34.65 \times 10^{-3}\text{m}$$

$$V_{air_{inside}} = 4.21\text{m/s}$$

$$\text{Re} = \frac{V_i \times d_i}{\vartheta}$$

$$\text{Re} = \frac{4.21 \times (34.65 \times 10{-}3)}{21.09 \times 10{-}6}$$

$$\text{Re} = 6600.74$$

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**Dittus-Boeltor equation for fully developed flow is considered for evaluation of inside convective heat transfer coefficient*

$Nu=0.023\ Re^{0.8}Pr^{0.3}$

$Nu=23.39$

$Nu=\frac{h_i \times d_i}{K}$

$h_i=21.11\ W/m^2K$

4.2. Estimation of outside convective heat transfer coefficient

$Film\ temperature\ of\ air=51\ ^oC$

Properties of air are evaluated at 51^{o}C,

Density (ρ) =1.093 Kg/m^3

Kinematic viscosity (ϑ) = 17.95 × 10^{-6} m^2/s

Thermal conductivity(K)=0.03047 W/mK

Prandtl number (Pr)=0.692

**flow across cylinders is considered for evaluation of outside convective heat transfer coefficient*

$Nu=C\ Re^m Pr^{0.333}$

$Re=\frac{V_o \times D_e}{\vartheta}$

v_o =2m/s

d_o=41.65 × 10^{-3}m

ϑ =17.95 ×10^{-6} m^2/s

D_e =Equivalent diameter of the fin

$=\frac{2[A_f+A_{bare\ tube})}{\pi(projected\ perimeter)}$

Where,

A_f = fin area = $\frac{\pi(d_f-d_o)(d_f+d_o)}{4}$ = 1.8847 × 10^{-3}m^2

$A_{bare\ tube}$ = $\pi d_i L_i - [20(\pi \times 0.04165 \times .00273)]$ = 14.599 × 10^{-3}m^2

$projected\ perimeter=\pi d_f$=20.2 ×10^{-2}m

D_e =51.949 ×10^{-3}m

$Re=\frac{V_o \times D_e}{\vartheta}$

$=\frac{2\times(51.94 \times 10-3)}{17.95 \times 10-6}$

=5788.189

For the calculated Reynold's number, the value of C and m are referred from the heat transfer data book as,

C=0.193

m=0.618

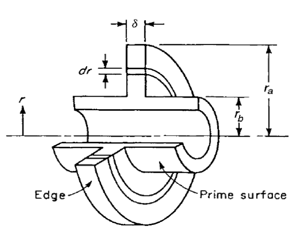
Nu=0.193(5788.189)$^{0.618}$(0.698)$^{0.333}$

Nu=36.21

$Nu=\frac{h_o \times D_e}{K_{air}}$

h_o =21.08 W/m^{2}K

5. CIRCULAR FIN OF RECTANGULAR PROFILE



For the circular fin of rectangular profile, the profile function is

$f_2(r)=\frac{\delta}{2}$

$\frac{df_2(r)}{dr}=0$

The Modified Bessel's equation is of the form,

$r^2\frac{d^2\theta}{dr^2} + r\frac{d\theta}{dr} - m^2r^2\,\theta$

Where $m=\sqrt{\frac{2\,h_o}{Kt}}$

The solution is in the form of Modified Bessel's functions,

$$\theta(r) = C_1 I_o(mr) + C_2 K_o(mr)$$

The arbitrary constants are evaluated using the boundary conditions, $\theta(r = r_b) = \theta_b$

When these boundary conditions are used, in $C1$ and $C2$ result

$$\theta(r) = C_1 I_o(mr_b) + C_2 K_o(mr_b)$$

$$\theta(r) = C_1 I_1(mr_a) + C_2 K_1(mr_a)$$

The equation for the temperature excess becomes,

$$\theta(r) = \frac{\theta_b\,[K_1(mr_a)I_o(mr) + I_1(mr_a)K_o(mr)]}{K_1(mr_a)I_o(mr_b) + I_1(mr_a)K_o(mr_b)}$$

Where,K and I are the modified Bessel's functions of zero and first order.

5.1. Temperature distribution for fin

**Temperature distribution for circular fins of rectangular profile is estimated by using*

$$\frac{T_r - T_\infty}{T_{fin\ base} - T_\infty} = \frac{K_1(mr_f)I_o(mr) + I_1(mr_f)K_o(mr)}{K_1(mr_f)I_o(mr_o) + I_1(mr_f)K_o(mr_o)}$$

Where,$m=\sqrt{\frac{2\,h_o}{Kt}}$

h_o =21.04 W/m^{2}K

K= 169.5 W/mK

t= 2.73 × 10^{-3}m

m=9.54

r_f =32.15 × 10^{-3}m

r_o =20.82 × 10^{-3}m

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Fin	Temperature at fin base	Temperature at centre of fin	Temperature at fin tip
1st fin	77.3 °C	76.933 °C	76.089 °C
2nd fin	76.0 °C	75.6434 °C	74.823 °C
3rd fin	76.6 °C	76.2387 °C	75.408 °C
4th fin	75.1 °C	74.7504 °C	73.946 °C
5th fin	75.6 °C	75.2465 °C	74.433 °C
6th fin	76.6 °C	76.2387 °C	75.407 °C
7th fin	76.7 °C	76.3380 °C	75.505 °C
8th fin	76.8 °C	76.4372 °C	75.602 °C
9th fin	78.2 °C	77.8263 °C	76.966 °C
10th fin	77.4 °C	77.0325 °C	76.187 °C
11th fin	76.5 °C	76.1395 °C	75.310 °C
12th fin	77.9 °C	77.5286 °C	76.674 °C
13th fin	76.9 °C	76.5364 °C	75.7 °C
14th fin	75.9 °C	75.5442 °C	74.725 °C
15th fin	77.7 °C	77.3302 °C	76.47 °C
16th fin	75.5 °C	75.1472 °C	74.335 °C
17th fin	73.2 °C	72.8651 °C	72.094 °C
18th fin	74.0 °C	73.6589 °C	72.874 °C
19th fin	73.6 °C	73.2620 °C	72.484 °C
20th fin	70.0 °C	69.6900 °C	68.976 °C

"Table 2. Comparison of Base, Centre line and Tip temperatures of the fins"

5.2. Calculation of Fin efficiency

$L_c = L + \frac{t}{2}$ =11.325mm

K =169.5W/mK

$A_p = L_c$ t=3.091725 × 10^{-5}m^2

By referring [8], the fin efficiency(η) is calculated as 94 % .

5.3. Estimation of Heat transfer through each fin

Q_{fin}= η $A_{fin} h_o (T_{surface} - T_\infty)$

Where, η=0.94, A_{fin} =1.88 × 10^{-3} m^2, h_o=21.04 W/m^{2}K

Fin	Base temperature	Q_{fin}
1st fin	77.3 °C	4.04209 W
2nd fin	76.0 °C	3.93099 W
3rd fin	76.6 °C	3.98227 W
4th fin	75.1 °C	3.85408 W
5th fin	75.6 °C	3.89681 W
6th fin	76.6 °C	3.98227 W
7th fin	76.7 °C	3.99082 W
8th fin	76.8 °C	3.99936 W
9th fin	78.2 °C	4.11900 W
10th fin	77.4 °C	4.05064 W
11th fin	76.5 °C	3.97372 W
12th fin	77.9 °C	4.09336 W
13th fin	76.9 °C	4.00791 W
14th fin	75.9 °C	3.92245 W
15th fin	77.7 °C	4.07627 W
16th fin	75.5 °C	3.88827 W
17th fin	73.2 °C	3.69171 W
18th fin	74.0 °C	3.76008 W
19th fin	73.6 °C	3.72590 W
20th fin	70.0 °C	3.41826 W

Table 3. Rate of Heat Transfer through each fin

$\sum Q_{fin}$ = 78.41 W

5.4. Estimation of Heat transfer from the Bare surface (Un-finned surface)

$Q_{unfinned} = A_{unfinned} h_o (T_{surface} - T_\infty)$

Where

$A_{unfinned} = h_o(\pi d_o dx)$(1.88 × 10^{-3})

dx = Distance between two fins= 4.82 × 10^{-3}m

h_o = 21.04 W/m^{2}K

ISSN: 2321-9653

S.No.	Base temperature	$Q_{unfinned}$
1	77.3 °C	0.6077 W
2	76.0 °C	0.621
3	76.6 °C	0.617 W
4	75.1 °C	0.6157 W
5	75.6 °C	0.6117 W
6	76.6 °C	0.6024 W
7	76.7 °C	0.6183 W
8	76.8 °C	0.6303 W
9	78.2 °C	0.6356 W
10	77.4 °C	0.6303 W
11	76.5 °C	0.6263 W
12	77.9 °C	0.6303 W
13	76.9 °C	0.6263 W
14	75.9 °C	0.621 W
15	77.7 °C	0.6276 W
16	75.5 °C	0.597 W
17	73.2 °C	0.5785 W
18	74.0 °C	0.5307 W

"Table 4.Rate of Heat Transfer through the tube"

$\sum Q_{unfinned}$=11.03 W

Total heat transfer rate,
$\sum Q_{fin} + \sum Q_{fin}$= 89.44 W

5.5. Estimation of inside surface temperature at different lengths of intercooler

Temperature of inside air is estimated at different locations by using the following relation assuming the steady state heat transfer.

$$\sum Q_{conduction} = \sum Q_{convection}$$

$$\frac{T_{inside_{surface}} - T_{outside_{surface}}}{\left(\frac{\ln\left(\frac{r_o}{r_i}\right)}{2\pi K dx}\right)} = h_o(\pi d_o dx)(T_{outside_{surface}} - T_\infty)$$

Fin	Distance from intercooler inlet	Inside surface temperature of intercooler
$T_{bare\ inside\ 2}$	18.915 mm	76.0218 °C
$T_{bare\ inside\ 3}$	26.465 mm	76.622 °C
$T_{bare\ inside\ 5}$	41.565 mm	75.622 °C
$T_{bare\ inside\ 6}$	49.115 mm	76.622 °C
$T_{bare\ inside\ 8}$	64.215 mm	76.822 °C
$T_{bare\ inside\ 9}$	71.765 mm	78.2229 °C
$T_{bare\ inside\ 11}$	86.865 mm	76.522 °C
$T_{bare\ inside\ 12}$	94.415 mm	77.9227 °C
$T_{bare\ inside\ 14}$	109.515 mm	75.92 °C
$T_{bare\ inside\ 15}$	117.065 mm	77.722 °C
$T_{bare\ inside\ 17}$	132.165 mm	73.22 °C
$T_{bare\ inside\ 18}$	139.715 mm	74.021 °C

6. Discussions of Results:

6.1. Variation of inside and outside temperature along the length of intercooler :

ISSN: 2321-9653

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The following figure shows the variation of outside and inside surface temperature along the length of intercooler. The difference between temperature at each section of intercooler is small as the thickness of tube material is only 3.5 mm and the thermal conductivity of tube material is considerably high.

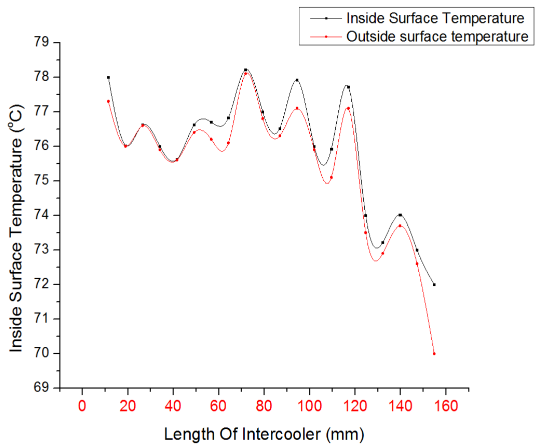
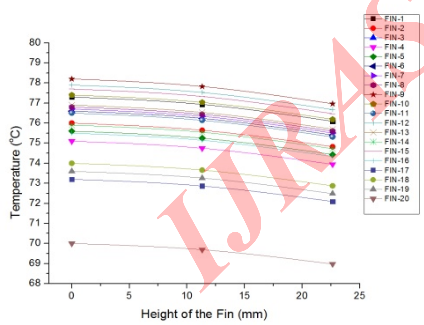


Fig.2.Inside,Outside Surface Temperature of Intercooler vs Length of Intercooler

6.2 Temperature distribution over the fins

The following figure shows the variation of temperature over each fin with respect to its base temperature.
The temperature distribution is calculated by using the modified Bessel's function.



6.3 Variation of heat transfer rate along the length of intercooler :

The following figure shows the variation of heat transfer rate along the intercooler. The heat transfer rate is more at the section where the fins are present. Heat transfer rate is also depends on the difference between the local surface temperature and the ambient temperature.

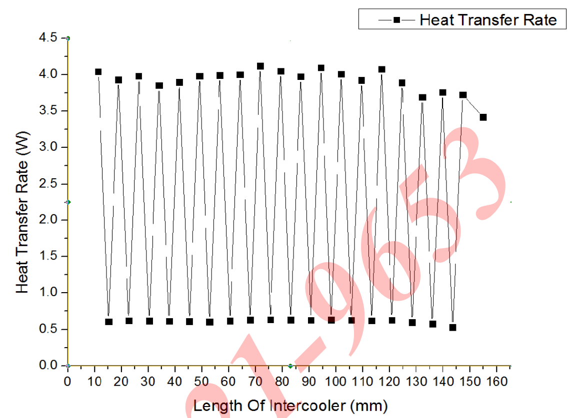


Fig 3. Rate of Heat Transfer vs Length of Intercooler

6.4 Modelling of 2-D geometry of Intercooler Using ANSYS

1. Fin type:

Circular fins of Rectangular cross section

The following figures shows the direction of heat flow and the temperature distribution on the fins

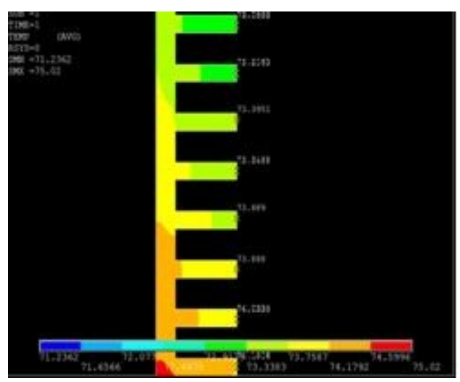


Fig. 4. Temperature distribution on the intercooler (rectangular profile)

Heat transfer rate is calculated from the ANSYS is 89.773 W

2. Fin type :

Circular fins of Triangular cross section

The following figures show the temperature distribution for circular fins of triangular profile. Triangular fins are generally adopted as the heat transfer is more when compared to the rectangular fins. To know the heat transfer rate with triangular fins, The intercooler is modeled and analyzed in ANSYS with

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the same conditions as that of the circular fins of rectangular profile.

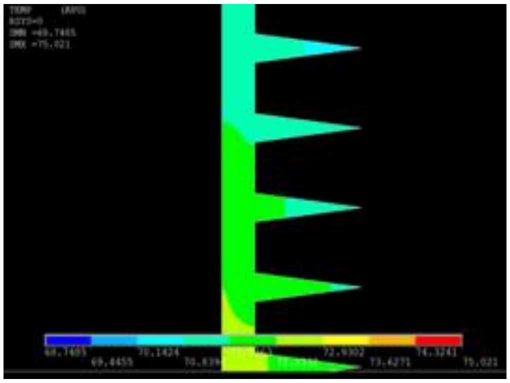

Fig.5.Temperature distribution on the intercooler (triangular profile)

The total heat transfer rate calculated from the ANSYS for this case is 92.82 W.

7. CONCLUSIONS:

The compression is approximated to be isentropic which is evident from the temperature of exit air (from the first stage of compression)

Heat flow for annular fins of triangular profile from the analysis using ANSYS is obtained as 92.82 W which is more than the heat flow for annular fins of rectangular profile which is obtained as 89.773 W.

Fins of triangular profile are suggested because for equivalent heat transfer it requires much less volume (fin material) than rectangular profile.

But, in view of their larger manufacturing costs, annular fin of rectangular profile is commonly used , which is justified as the heat flow is only 3.39% more in case of triangular fins.

Temperature distribution for fins which is in the form of modified Bessel's equation is obtained for all the fins and calculated tip temperatures are approaching the experimentally measured tip temperatures and tip temperatures which are obtained from ANSYS.

Total heat flow obtained from the theoretical analysis is 89.44 W, while that from the analysis using ANSYS is 89.77W.The percentage deviation between theoretical analysis and using ANSYS is 0.003 %.

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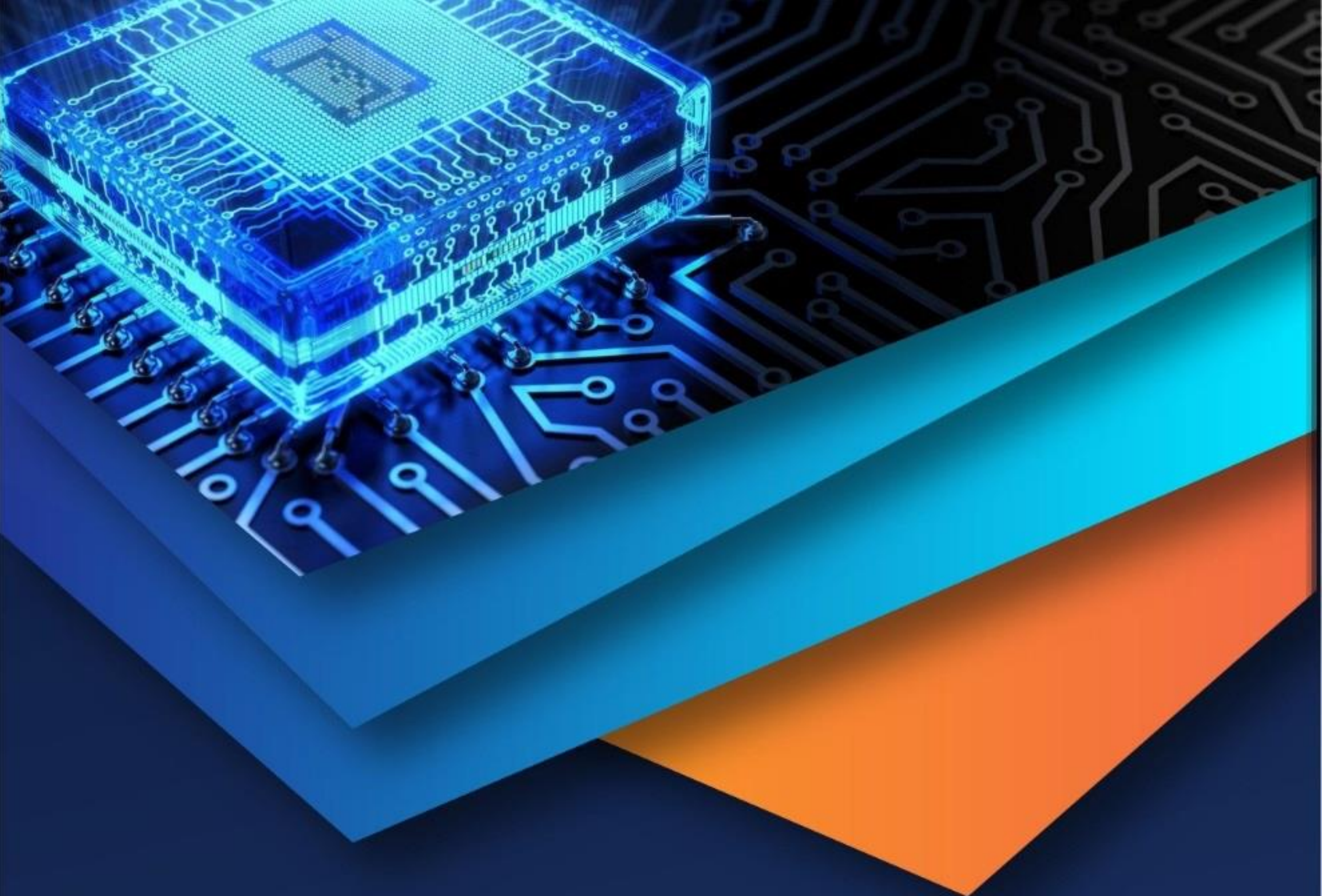

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