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Design and Analysis of Flat Head Pressure Vessel in Petroleum Industry

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Abstract: Naphtha is a volatile commercial product which is obtained by the distillation process in petroleum refinery of coal tar. Petroleum naphtha is a name for the petroleum distillate which contains aliphatic hydrocarbons and boiling point higher than the gasoline and lower than kerosene. In the crude oil distillation the overhead liquid distillate is called virgin or straight naphtha. Heat is used to maintain viscosity of naphtha that is stored. As naphtha has a non-linear thermal property after distillation it is in the semi-fluid form. If the temperature drops the fluid changes from semi- liquid to semi-solid form and can cause considerable damage to the system and becomes difficult for further processing. Hence to avoid this heat pipes are inserted in pressure vessel from top to ensure efficient distribution of heat. In this project we will study the effect of thermal stress due the presence of embedded heat pipes and try to simulate the distribution of heat in the system using ANSYS.
Keywords: Flat head Pressure vessel, embedded heat pipes, FEA, ASME

I. INTRODUCTION

A. Overview of Pressure Vessel

Pressure Vessel is single most important aspect of mechanical engineering in the industrial field. A pressure vessel is defined as a container with a pressure differential between inside and outside. Pressure vessel often has a combination of high pressure together with high temperature and in some cases flammable fluids or highly radioactive material. Because of such hazards it is imperative that the design be such that no leakage can occur. In addition vessel has to be design carefully to cope with the operating temperature and pressure. Cylindrical pressure vessels are divided into two groups, thin and thick cylinders.

B. Embedded Heat Pipe

Heat pipe is the capillarity structure in which working fluid flows through the capillarity structure which is a vacuumed closed system. The heat pipes are used to transfer the heat from one medium to the medium which are in contact with minimum temperature difference. The fundamental construction of the traditional heat pipe is shown in figure 1.1

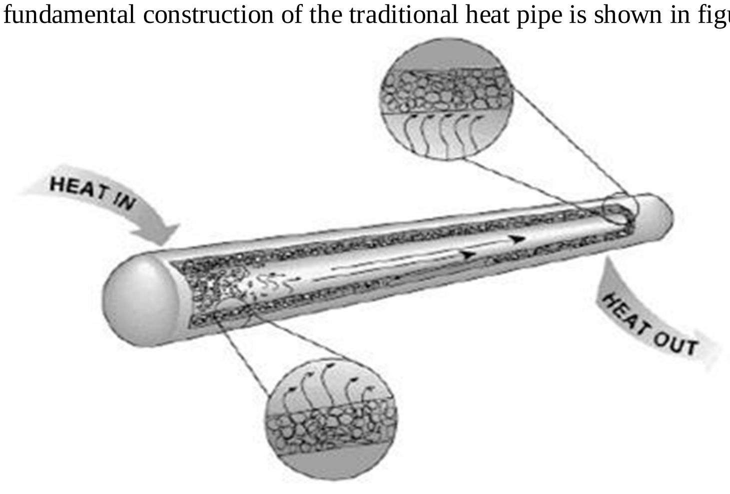

Fig. 1.1 Basic structure of conventional heat pipe

Although the heat pipe has good thermal performance for lowering the temperature of the heat source, its operating limitation is the key design issue called the critical heat flux or the heat capacity quantity

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C. Naphtha

Naphtha has the initial boiling point of about 35ºC and final boiling point of about 200ºC. Chemical basis, naphtha contains different amount of constituents (paraffins, naphthenes, aromatics, and olefins) in different proportions.

D. Working Of Process Reactionary Vessel

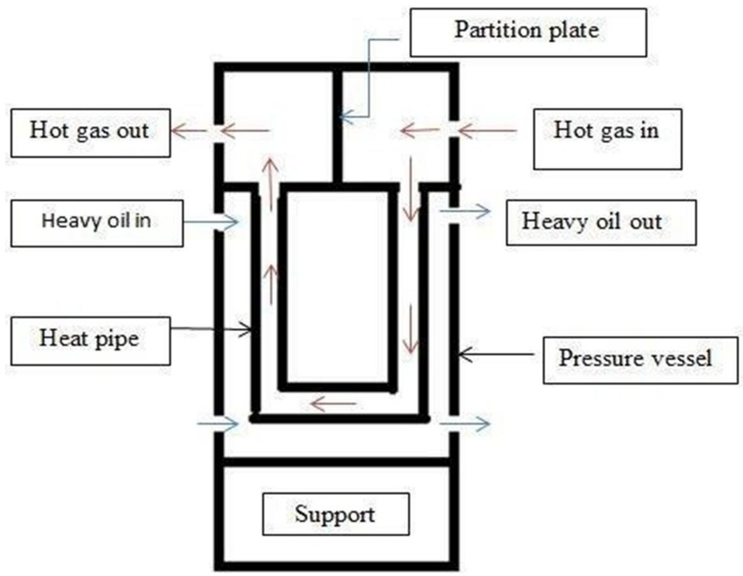

Fig. 1.2 Basic layout of Process reactionary vessel with heat pipes

Figure1.2 illustrates the working of process reactionary vessel with embedded heat pipe. Heavy oil is stored in the vessel called as process volume (Vs). This oil is stored for further process at a certain temperature to avoid the phase change. For that heat pipe is inserted in the vessel and hot gas is allowed to pass through it by maintaining pressure difference. Due to pressure and temperature difference the uneven thermal stresses are developed in the system which will compromise the safety of the structure. So to avoid this optimum embedded pipe is needed to stabilize the design.

II. PROBLEM DEFINITION

From the literature survey it was noticed that not much work is carried out on heat treatment embedded pipes inside pressure vessel using finite element method. Also there is no such design procedure stated which can help to determine the optimum pipe thickness for efficient heat transfer so Finite element analysis and optimization of embedded heat pipe inside the pressure vessel

A. Objective of Study

1) The objective of the present work is an attempt to study of different parameters affecting the heat flow inside pipes.
2) Select the material to ensure proper heat distribution and minimal thermal stresses.
3) Design and do finite element analysis of heat pipe and storage vessel.

III. METHODOLOGY

A. CAD Model generation of a pressure vessel.
B. Design the pressure vessel using ASME codes.
C. Modeling and Drafting of pressure vessel design in ANSYS workbench 14.5
D. Mesh generation of ANSYS model in hypermesh 12.0
E. Structural and thermal simulation of pressure vessel and embedded pipes using ANSYS workbench.
F. Static structural and transient thermal nonlinear analysis is done using numerical method.

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IV. CASE STUDY

A. The Thick Wall Cylinder

Thick-wall cylinders are used widely in industry as pressure vessels, pipes, gun tubes, etc. in many applications the cylinder wall thickness is constant and the cylinder is subjected to a uniform internal pressure p1, a uniform external pressure p2, an axial load p, and a temperature change ΔT. Often the temperature change ΔT is a function of the radial coordinate r only. Under such conditions, the deformations of the cylinder are symmetrical with respect to the axis of the cylinder.

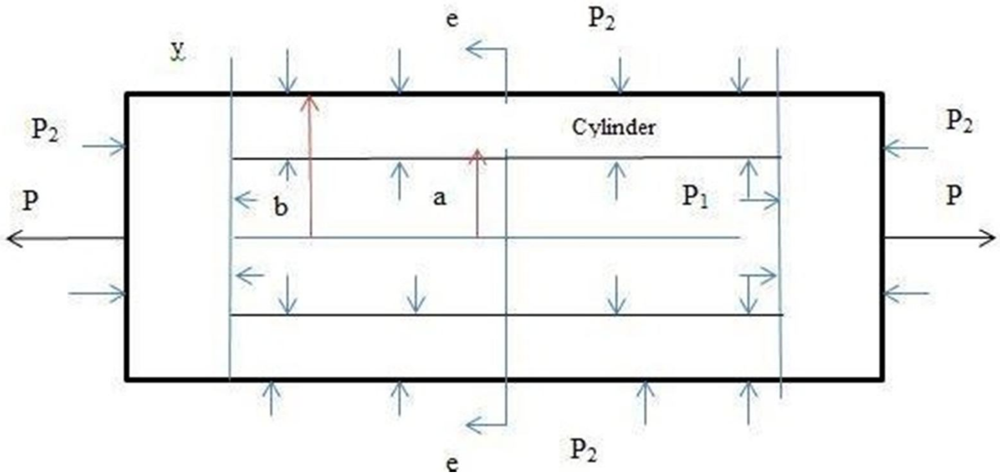

Fig. 3.1 Closed cylinder with internal pressure, external pressure, and axial loads.

Furthermore, the deformations at a cross section sufficiently far removed from the junction of the cylinder and its end caps are practically independent of the axial coordinate z. In particular, if the cylinder is open and unconstrained, it undergoes axisymmetric deformations owing to pressure p1 and p2 and temperature changes ΔT= ΔT(r), which are independent of z. If the cylinder's deformation is constrained by support or end caps, then in the vicinity of the supports or junction between the cylinder and end caps, the deformation and stresses will depend on the axial coordinate z.

B. Stress Components and Radial Displacement for Constant Temperature

1) Stress Components

As there is no temperature change, ΔT=0

The expressions for the stress components in a closed cylinder (cylinder with end caps)

As there is no temperature change, ΔT=0

The expressions for the stress components in a closed cylinder (cylinder with end caps)

$$\sigma_{rr} = \frac{(P_1 a^2 - P_2 b^2)}{(b^2 - a^2)} - \frac{a^2 b^2}{r^2(b^2 - a^2)}(P_1 - P_2) \qquad \ldots (1)$$

$$\sigma_{\theta\theta} = \frac{(P_1 a^2 - P_2 b^2)}{(b^2 - a^2)} + \frac{a^2 b^2}{r^2(b^2 - a^2)}(P_1 - P_2) \qquad \ldots (2)$$

$$\sigma_{zz} = \frac{(P_1 a^2 - P_2 b^2)}{(b^2 - a^2)} - \frac{P}{(b^2 - a^2)} = constant \qquad \ldots (3)$$

C. Radial Displacement for a Closed Cylinder

For no temperature change, ΔT=0. Then the radial displacement *u* for a point in a thick-wall closed cylinder (cylinder with end caps).

The resulting expression for *u* is

$$U_{(closed\ end)} = \frac{r}{(b^2-a^2)}[(1-2V)(P_1a^2-P_2b^2)+\frac{(1+V)a^2b^2}{r^2}(P_1-P_2)-V\frac{P}{\pi}] \ldots (5)$$

D. Stresses and Deformations in a Hollow Cylinder

A Thick-wall closed-end cylinder is made of an aluminum alloy (E=72 GPa and *v*=0.33), has an inside diameter of 200 mm, and has an outside diameter of 800mm. the cylinder is subjected to an internal pressure of 150 MPa. Determine the principle stresses, maximum shear stress at the inner radius(r=a=100 mm), and the increase in the inside diameter caused by the internal pressure.

1) Analytical Solution

The principal stresses are given by Equations (1) & (3).

$$\sigma_{rr} = \frac{(P_1a^2-P_2b^2)}{(b^2-a^2)} - \frac{a^2b^2}{r^2(b^2-a^2)}(P_1-P_2)$$

$$\sigma_{zz} = \frac{(P_1a^2-P_2b^2)}{(b^2-a^2)} - \frac{P}{(b^2-a^2)} = constant$$

For the conditions P_2 that =0 and r=a, these equations give

$$\sigma_{rr} = P_1\frac{(a^2-b^2)}{(b^2-a^2)} = -P_1 = -150\ MPa$$

$$\sigma_{\theta\theta} = P_1\frac{(a^2+b^2)}{(b^2-a^2)} = 150\frac{100^2+400^2}{400^2-100^2} = 170\ MPa$$

$$\sigma_{zz} = P_1\frac{a^2}{(b^2-a^2)} = 150\frac{100^2}{(400^2-100^2)} = 10\ MPa$$

The maximum shear stress, given by

$$\tau_{max} = \frac{\sigma_{max}-\sigma_{min}}{2} = \frac{170-(-150)}{2} = 160\ MPa$$

The increase in the inside diameter caused by the internal pressure is equal to twice the radial displacement given by equation(5).

$$U_{(closed\ end)} = \frac{r}{(b^2-a^2)}[(1-2V)(P_1a^2-P_2b^2)+\frac{(1+V)a^2b^2}{r^2}(P_1-P_2)-V\frac{P}{\pi}$$

For the conditions =p = 0 and r = a. Thus,

$$U_{(r=a)} = \frac{P_1a}{(b^2-a^2)}[(1-2V)^2_a+(1+V)b^2$$

$$= \frac{150(100)}{7200(400^2-100^2)}[(1-0.66)100^2+(1+0.33)400^2$$

$$= 0.3003\ mm$$

The increase in the inside diameter caused by the internal pressure is 0.6006 mm.

E. Numerical Solution Model

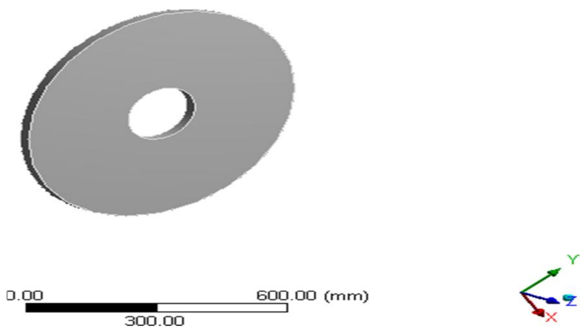

Fig. 3.2 Model

Model was created having inside diameter of 200mm, and outside diameter of 800mm it is extruded to 30 mm keeping symmetry in both the direction. Then defined material properties are assigned to this geometry.

1) Material and Its Properties

Material Selected: Aluminum alloy Properties Aluminum

Sr no.	Properties	Value
1	Young's Modulus	72 GPa
2	Poisson's Ratio	0.33
3	Bulk Modulus	69 GPa
4	Shear Modulus	26 GPa

Table 3.1 Properties of Aluminum

2) Meshing

METHOD: Hex Dominant method ELEMENT MIDSIDE NODES: Kept FREE FACE MESH TYPE: Quad/Tri NUMBER OF NODES: 29195 NUMBER OF ELEMENTS: 5696
ELEMENT TYPE: SOLID 186

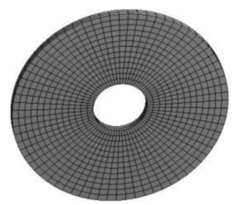

Fig. 3.4 Meshing

Mapped meshing is done on both faces of the model to remove the irregularities in the meshing and making the mesh more refine.

F. Boundary Condition

PRESSURE=150MPa

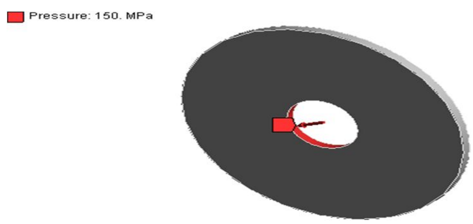

Fig. 3.5 Boundary Condition

Figure 3.4 shows the model with boundary conditions. Pressure of the magnitude 150 MPa is applied at the inner surface of the hollow cylinder. For cylinder no fix support is required since its structure itself acts as fix boundary conditions.

G. Solution

1) Maximum Principal Stress

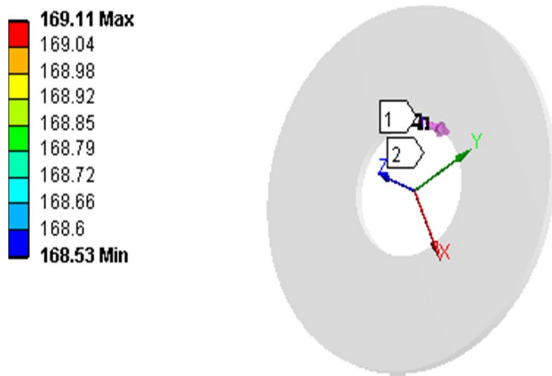

Fig. 3.6 Maximum Principal Stress

After applying the boundary conditions, simulations was run and above results are obtained. It can be seen from the results that maximum stress coming on the surface is 169.11 MPa and minimum stress is 168.53MPa. The value which was found during the simulation is less than analytical solution and is within the permissible limit.

2) Minimum Principal Stress

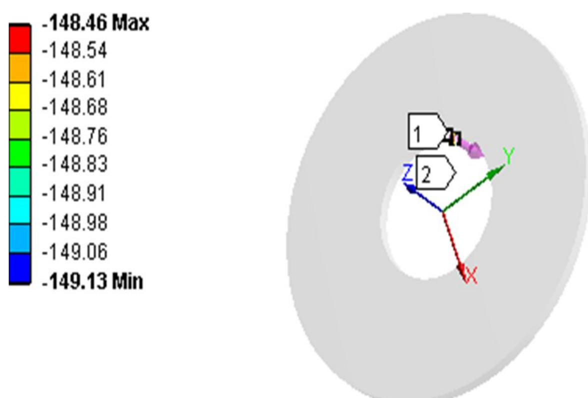

Fig. 3.7 Minimum Principal Stress

The result shows that maximum stress coming on the surface is -148.46 MPa and minimum stress is -149.13 MPa. The value which was found during the simulation is less than analytical solution and is within the permissible limit.

3) Maximum Shear Stress

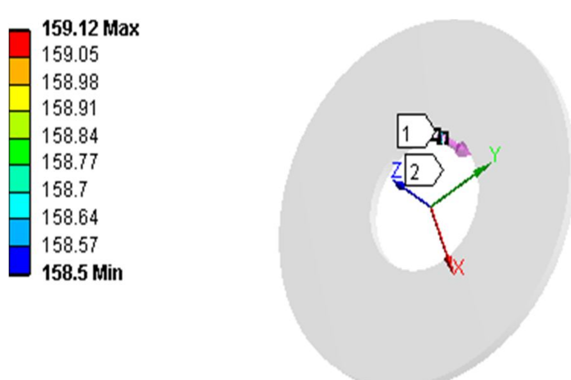

Fig. 3.8 Maximum Shear Stress

4) Total Deformation

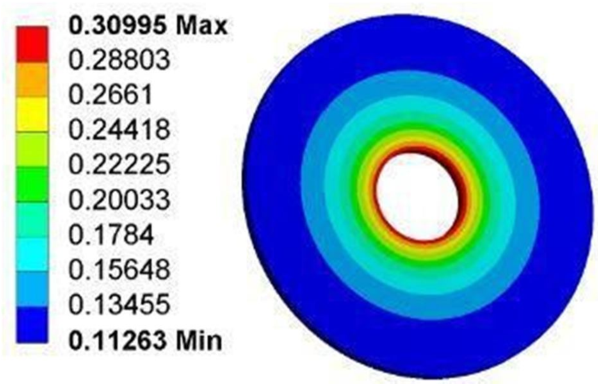

Fig. 3.9 Total deformation

The increase in the inside diameter caused by the internal pressure is equal to the twice the radial displacement. The maximum deformation is 0.30995mm

5) Percentage Error in Directional Deformation: The results obtained analytically and by simulation are compared and percentage error is found out which came 2.94%. This value of percentage error is within the allowable limit of 10%.

V. DESIGN OF PRESSURE VESSEL

A. Design Data

Design a pressure vessel for the following specifications

Sr. No.	Parameter Description	Parameter Code	Value
1	Internal Pressure	P	0.05 MPa
2	External Pressure	Po	Atm.
3	Process Volume	Vp	205 m3
4	Expected Stagnant Volume	VS	57 m3
5	Buffer Volume Requirement	Vb	50 m3
6	Vessel radius	R	2.5 m
7	Tube porosity volume	Tp	25
8	Radius of tube sheet	R	2.5 m
9	Tube diameter	Td	200 mm
10	Skirt Support height	H	3 m

Table 4.1 Design specification for pressure vessel

B. Rules for Characterization of Vessel

1) If $Vp = 0$, then the vessel will be characterized as a storage vessel.
2) If $Vs < 0.1Vp$, then the vessel will be characterized as a full process flow vessel.
3) If $Vs > 0.1Vp$, then the vessel will be characterized as process reactionary Vessel.

$$0.1 \times Vp \text{ ……(1)}$$

$$= 0.1 * 205$$

$$= Vs(57 \text{ m3}) > 20.5 \text{ m3}$$

Hence the vessel will be characterized as Process reactionary vessel.

VI. CONCLUSION

The aim of this research work is to design and optimized the process reactionary vessel and the heat treatment pipe. The structural and thermal analysis of embedded pipes inside pressure vessel is carried out using numerical simulation.

A. After the design calculation of pressure vessel by ASME codes the pressure vessel should be under permissible failure limit according to the ASME codes guidelines.
B. The thermal distribution in the pressure vessel should be in range of 33 to 38 degree centigrade maintaining the low thermal expansion of the embedded heat pipe.

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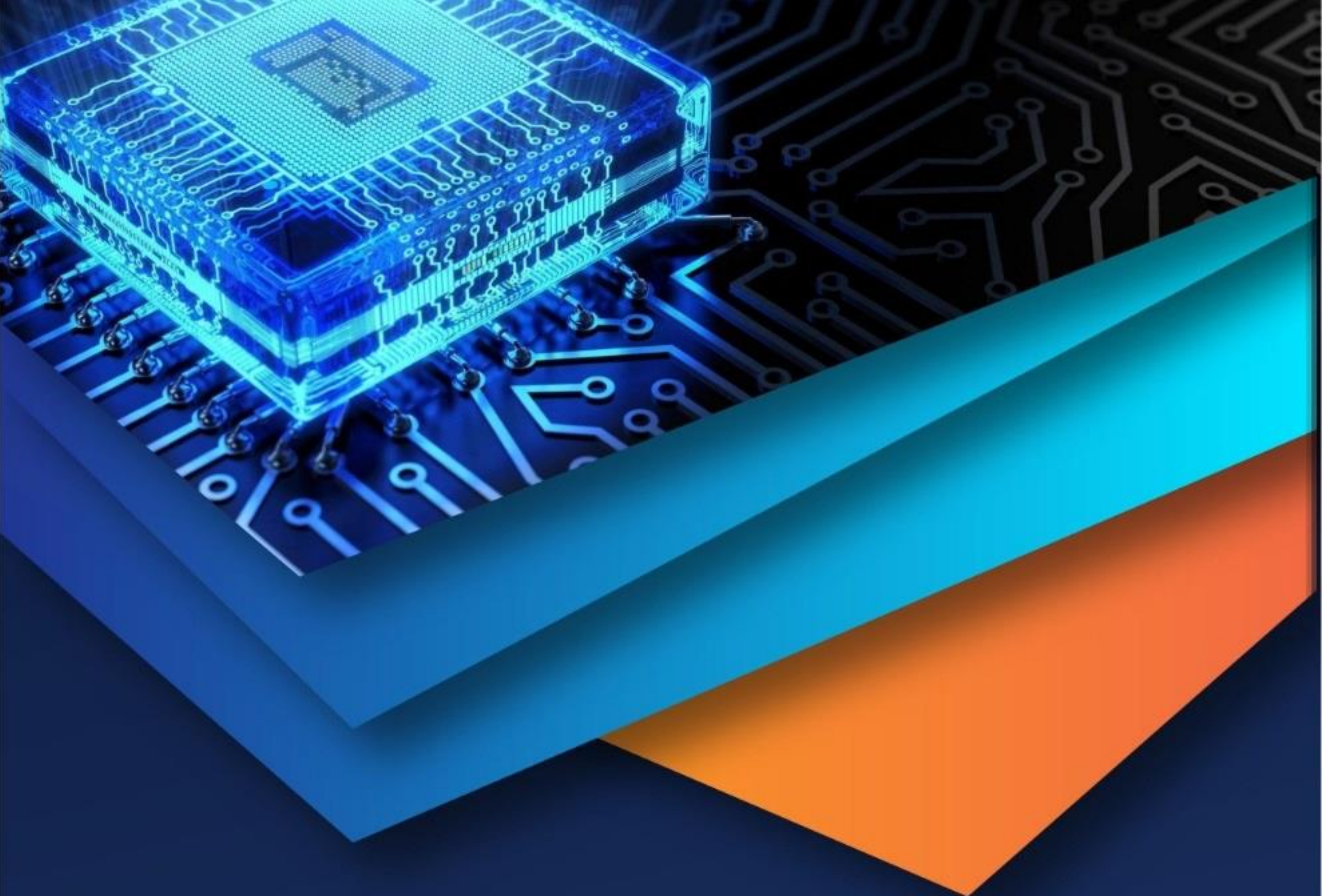

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