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Research and Finite Element Analysis of Circular Braking Plate

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Abstract: Noise and vibration associated with the deceleration process in passenger buses has come an important profitable and technological problem in the sedulity.The knowledge of natural frequency of factors is of great interest in the study of the response of structures to various excitations.Hence, a copse slice plate with a central hole, fixed at the inner edge and free at the external edge, is chosen, and its dynamic response is excavated.The ideal of the current discussion work is to anatomize the vibration characteristics as the natural frequency and mode shapes, of copse discs with drilled holes of different compasses,worn discs & enlarged bolt hole fringe discs at the external end due to deceleration but with the same rates of inner to external compass for inner edge clamped and external edge free boundary conditions. Also, the FEM software package is used for vibration analysis of copse discs with the same boundary condition but having different compasses of cooling holes, wear viscosity & enlargedbolt hole compasses for determining different parameters like natural frequency and mode shapes. Thus, results attained are to be compared. This work deals with chancing the natural frequency and mode shapes of slice chaparrals.The slice copse is modelled using marketable computer- backed design(CAD) software, ANSYS.

I. INTRODUCTION

Slice boscage noise and vibration are known to involve structural coupling between similar factors as the rotor, pads, caliper, and knuckle.Depending on the frequency range of interest, the hydraulic system, body panels, steering column, and other vehicle factors can also come active.In an aggregate sense, the slice thickets of only a many percent of new vehicles parade sufficient noise and vibration to induce significant client complaints,but the volume and expenditure of remediation sweats, in addition to the perception of reduced product- line quality, place pressure on boscage noise and vibration.An acute problem is called "squeal" noise, which is generally defined as that being within the range 1.5 to 20 kHz at one or furtherof the rotor's natural frequencies and its harmonics.For voiced and solid core designs, rotors have the distinction of being structural rudiments, members of the slice-pad disunion brace, and effective radiators of sound because of their large face area.

The study of the dynamic gesteof boscage discs is important, as are several machine factors.It can be considered as annular plates with radial holes for the purpose of analysis.This study is abecedarian for high- threat shops.In each case, the rotor comprises the "slice" element which is in frictional contact with the pads during operation and the chapeau element which provides thegeometric neutralize necessary for mounting the rotor to the vehicle.The consistence, inner and external periphery of the slice, and the figures and distance of the cooling vanes and mounting super studs are some of the geometricparameters that set the rotor's natural frequencydiapason and vibration modes.

II. OBJECTIVE

1) To determine the natural frequencies of the system using FEM throughANSYS software.
2) To identify the corresponding mode shapes of the system using FEM in ANSYS.
3) To compare and evaluate the discrepancies in natural frequencies and mode shapes due to design optimization, and to conclude the impact of optimization on dynamic behavior.

III. PROBLEM STATEMENT

An acute problem is called "squeal" noise, which is generally defined as that being within the range 1.5 to 20 kHz at one or further ofthe rotor's natural frequency and its harmonics. For voiced and solid core designs, rotors have the distinction of being structural rudiments, members of the slice- pad disunion brace, and effective radiators of sound because of their large face area. The study of the dynamic geste of boscage discs is important, as are several machine factors.It can be considered as annular plates with radial holes for the purpose of analysis.

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IV. METHODOLOGY

Dissertation exploration papers are bandied dealing with the vibration analysis of the slice thickets, which includes the adding operation of vibration analysisgeneralities in design has urged experimenters to gain an understanding of the dynamic gesteof structures.Then dynamic parcels of boscage discs are delved using the FEM analysis software(ANSYS) and discussion of mode shapes by entrapping them.Conversations on FEM results will be done then to get the conclusions as per variables applied on boscage discs.

V. LITERATURE REVIEW

Unwanted noise and vibration associated with the braking system in passenger automobiles has become an important economic and technological problem in the industry.Improved understanding of disc vibration through research papers offers one opportunity for targeted improvements in brake rotor design**.** To help the dissertation, research papers dealing with the vibration analysis of the brake disc have been studied.

Tsuyoshi Inoue, Yukio Ishida [1] investigated forChaotic Vibration and Internal Resonance Phenomena in Rotor Systems,studied rotating machinery has effects of gyroscopic moments, but most of them are small. Then, many kinds of rotor systems satisfy the relation of 1 to (−1) type internal resonance approximately. In this paper, the dynamic characteristics of nonlinear phenomena, especially chaotic vibration, due to the 1 to (−1) type internal resonance at the major critical speed and twice the major critical speed are investigated.

Mehdi Ahmadian, Kristina M. Jeric, Daniel J. Inman, Nagoya, Aichi[2],“An Experimental Evaluation of Smart Damping Materials for Reducing Structural Noise and Vibrations”, investigated an experimental evaluation of the benefits of smart damping materials in reducing structural noise and vibration is presented. The construction of a special test rig for measuring both vibrations and structure-borne noise is discussed. Next, the application of smart damping materials, specifically piezoceramics with electrical shunts, in reducing the vibrations of a test plate is discussed. It is shown that the smart damping materials are able to effectively reduce the vibration peaks at multiple frequencies, with minimal amount of added weight to the structure, as compared to passive viscoelastic damping materials.

Albert C. J. Luo, Mote [3], “Nonlinear Vibration of Rotating Thin Disks”, investigated the response and natural frequencies for the linear and nonlinear vibrations of rotating disks are given analytically through the new plate theory proposed by Luo in 1999. The results for the nonlinear vibration can reduce to the ones for the linear vibration when then on linear effects vanish and for the von Karman model when the nonlinear effects are modified. They are applicable to disks experiencing large-amplitude displacement or initial flatness and waviness.

Albert C., J. Luo [4], “An approximate theory for geometrically nonlinear thin plates”, investigated an approximate theory of thin plates is developed that is based on an assumed displacement field, the strains described by a Taylor series in the normal distance from the middle surface, the exact strains of the middle surface, and the equations of equilibrium governing the exact configuration of the deformed middle surface. In this theory, the exact geometry of the deformed middle surface is used to derive the strains and equilibrium of plates.

Singh & Saxena [5]have been investigated that transverse vibrations of skew plates of variable thickness with different combinations of boundary conditions at the four edges by using Rayleigh Ritz method. The two dimensional thickness variations are taken as the Cartesian product of linear variations along the two concurrent edges of the plate.The study of transverse vibration of plates of various shapes under different boundary conditions is important owing to a wide variety of applications in engineering design.

Ouyan& Mottershead [6]have been investigated that the velocity-dependent friction law with the Stribeck effect in a moving load model for the vibration and squeal of a car disc brake. Simulated numerical results produce a bounded region of instability for the rotating speed of the disc which is compatible with observed squeal phenomenon.

Bambill [7] has been evaluated the analytical and experimental investigation on transverse vibrations of solid circular and annular plates carrying a concentrated mass at an arbitrary position with marine applications.

Ambili [8] t has been investigated the free vibrations of annular plates coupled with fluids. The natural frequencies of annular plates on an aperture of an infinite rigid wall and in contact with a fluid on one side are theoretically obtained by using added mass approach.

Lee & Singh [9] has been investigated that Annular disk idealization can be used to analyse many real-life mechanical components such as gears, brake rotors, clutches, flywheels, railway wheels, circular saws, and electric motor. In many cases, thickness is not negligible relative to other dimensions of the component.

C.S. Huang [10] have done this work applies the famous Ritz method to analyze the free vibrations of rectangular plates with internal cracks or slits. To retain the important and useful feature of the Ritz method providing the upper bounds on exact natural frequencies, the paper proposes a new set of admissible functions that are able to properly describe the stress singularity behaviors near the tips of the crack and meet the discontinuous behaviors of the exact solutions across the crack.

VI. DESIGN ANALYSIS

The natural frequencies and mode shapes are determined for same ratio of inner to outer radius of the disc. Discs are having radial holes but of variable diameter and variable disc wear for inner edge clamped and outer edge free boundary condition**.**

Assumptions:

1) Scratching of the middle surface of the plate is neglected in order to keep the equation of motion linear.
2) Influence of the shear force and rotary inertia is ignored. It is also assumed that the plane cross section before and after deformation remain plain.
3) The deflection considered is also small.
4) Plate is thin, i.e. thickness of the plate is small than its outer dimensions.

W. T. Norris and J. E. T. Penny have given values of $(\beta a)^2$ for various ratios of outer radius to inner radius and different mode characteristics. First six harmonics are given in order for aspect ratio (b/a =0.1).

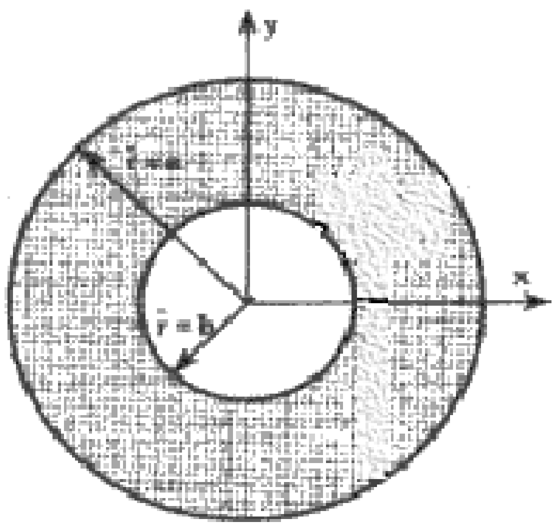

Fig. 1 Brake disc

$(\beta a)^2$ isnon-dimensional frequency parameter used to calculate natural frequencies of annular disc.

$$(\beta a)^2 = \omega a^2 \sqrt{\frac{3\rho(1-\upsilon^2)}{Eh^2}}$$

Where,

h = half thickness in mm

E = Young's modulus N/m^2

ρ = Density of the disc material in kg/m^3

ω = frequency in rad/s.

υ = Poisson's ratio taken as 0.3

A = Outer radius of disc in mm

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Calculations to find natural frequencies:
Dimensions of annular plate test specimen are as below:
Natural frequency for disc with aspect ratio= 0.1
Inner radius of the plate b=10 mm
Outer radius of the plate a=100 mm
Thickness of the plate 2*h*= 1.5mm

We know $(\beta a)^2 = \omega\, \mathrm{a}^2 \sqrt{\dfrac{3\rho(1-\upsilon^2)}{Eh^2}}$

For first mode i.e. (0, 1)

$(\beta a)^2 = 3.478$

$$3.478 = \omega\,(100 \text{ x } 10^{-3})^2 \sqrt{\frac{3x7850(1-0.3^2)}{2.1x10^{11}x(0.75x10^{-3})^2}}$$

$\therefore\ \omega =$ 1088 rad/s

$\therefore\ \omega =$173.27 Hz

For Second mode i.e. (0, 0)

$(\beta a)^2 = 4.237$

$$4.237 = \omega\,(100 \text{ x } 10^{-3})^2 \sqrt{\frac{3x7850(1-0.3^2)}{2.1x10^{11}x(.75x10^{-3})^2}}$$

$\therefore\ \omega =$1326.34 rad/s

$\therefore\ \omega =$211.00 Hz

For Third mode i.e. (0, 2)

$(\beta a)^2 = 5.623$

$$5.263 = \omega\,(100 \text{ x } 10^{-3})^2 \sqrt{\frac{3x7850(1-0.3^2)}{2.1x10^{11}x(o.751x10^{-3})^2}}$$

$\therefore\ \omega =$1760.21 rad/s

$\therefore\ \omega =$280.00 Hz

For Forth mode i.e. (0, 3)

$(\beta a)^2 =12.45$

$$12.45 = \omega\,(100 \text{ x } 10^{-3})^2 \sqrt{\frac{3x7850(1-0.3^2)}{2.1x10^{11}x(0.75x10^{-3})^2}}$$

$\therefore\ \omega =$3897.32rad/s

$\therefore\ \omega =$620.00 Hz

For Fifth mode i.e. (0, 4)

$(\beta a)^2 = 21.84$

$$21.84 = \omega\,(100 \text{ x } 10^{-3})^2 \sqrt{\frac{3x7850(1-0.3^2)}{2.1x10^{11}x(0.75x10^{-3})^2}}$$

$\therefore\ \omega =$ 6836.75 rad/s

$\therefore\ \omega = 1088.10$ Hz

For Sixth mode i.e. (1, 0)

$$(\beta a)^2 = 25.26$$

$$25.26 = \omega\,(100 \text{ x } 10^{-3})^2 \sqrt{\frac{3x7850(1-0.3^2)}{2.1x10^{11}\,x(0.75x10^{-3})^2}}$$

$\therefore\ \omega = 7907.34$rad/s

$\therefore\ \omega = 1258.00$ Hz

Theoretical natural frequencies for particular mode shape of simple annular plates without cracks with aspect ratio 0.1 are tabulated as below in Table no. 1.

Table1 :VariousModes & Natural Frequencies

Sr. No.	Mode (Nodal circle, Nodal diameter)	Theoretical natural frequency in Hz
1	(0, 1)	173
2	(0, 0)	211
3	(0, 2)	280
4	(0, 3)	620
5	(0, 4)	1088
6	(1, 0)	1258

VII. ANALYSIS OF OWNER CONTRIBUTED FACTORS

To carry out FEM analysis of any component, the solid model of the same is essential. It is also called body in white. So the solid model of brake disc is requiredand this can be done in special CAD package like CATIA.

Solid Models:

Following are the ways in which the solid model of fender is created in CATIA V5.

- CSG: It is the constructive solid geometry process. In this case final model consists of union of various primitive objects such as brick, cube, sphere, cone, cylinder, etc. Depending upon the shape of final model Boolean operations are performed over these shapes to obtain the required geometry of solid model. Thusit builds a solid model from fundamental shapes.
- B-REP: It is the boundary representation of solid objects. In this case final model is only represented by its boundary surfaces. The relations between surfaces are maintained so as to give required solid model. It is a quick process. This is used to get surface for auto mesh for the outer surface of fender
- FBM: It is a feature based modeling. The features such as extrude, protrude, cut, revolve, copy, etc. are used to build a solid model. Many CAD packages use FBM method. It is easy and gives 'model tree' for completed part, so that modification at any point at any branch can be passed through whole model. Thus FBM is suited for parameterization of model. It will be helpful to generate similar models from existing one just by changing the parameter values.

Table 2: Comparison of theoretical and FEM natural frequency results for annular disk without cracks.

Sr.No.	Mode (c,d)	Theoretical Natural frequency in Hz	FEM(ANSYS) Natural frequency in Hz	Percentage Error%
1	(0,1)	173	161	7
2	(0,0)	211	212	0.5
3	(0,2)	280	292	4
4	(0,3)	620	610	1.6
5	(0,4)	1088	1058	2.8

VIII. FEM ANALYSIS OF BRAKE DISK

In short, clean geometry can be defined as a solid CAD model that maximizes the possibility for a' mesh which in turn captures the features required for correct results. Two key points are made in this statement. First, the geometric features must not prevent the mesh from being created and must also contain surfaces of consistent size and shape ratios to prevent forcing high, aspect ratio elements and/or transitions between element edges that may compromise accuracy. Second, simplification or manipulation of features in an attempt to clean up the geometry should not reduce thestructural integrity of the part. The best mesh is the smallest model that yields correct data. Consequently, manipulation of the model, either by adjusting dimensions or suppressing features far from any area of interest, is acceptable, as effects local to the simplification will most likely not affect the global behavior of the system. However, care should be taken when adjusting a model near an area of concern.

IX. REVIEW OF OBTAINED RESULTS

Results are reviewed in general post-processor and time history in post-processor. The general postprocessor is used to review the results during a particular time step for the whole or part of body. Different results like deformations, stresses, and tabular listings of displacements and animations of vibration can be obtained by using general post-processor. Time-history postprocessor is used to find out results of a particular node through all the time-steps. By using time-history postprocessor, graphs of data vs. time can he obtained. The arithmetic operations can also be performed on graphs.Sample FEM results are shown below in table 3 for annular disc with three cracks of 45mm length.

Postprocessor: Results: Model > Modal > Solution

Table 3:FEM results for brake disc specimen No.01

Mode	Frequency [Hz]
1.	1003
2.	1092
3.	1095
4.	1102
5.	1102

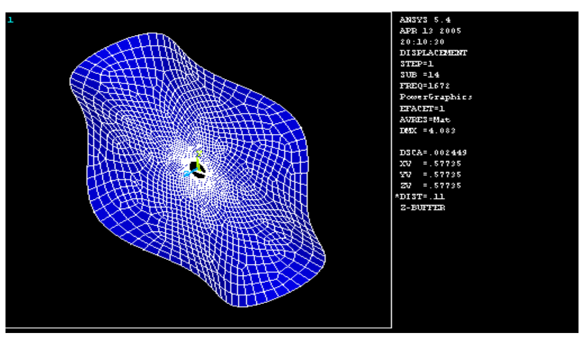


Fig.2: Diametrical mode shape in meshed model (0, 3)

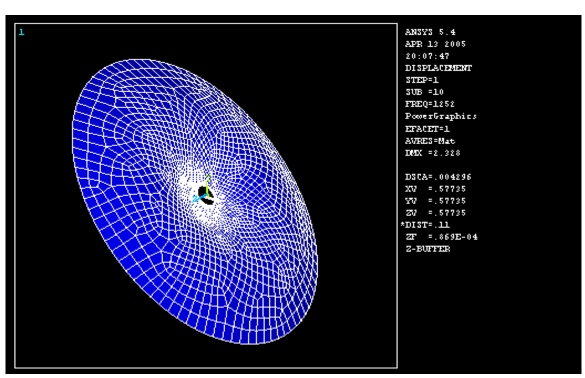


Fig.3: Circular mode shape meshed model (1, 0)

X. PERFORMANCE ANALYSIS

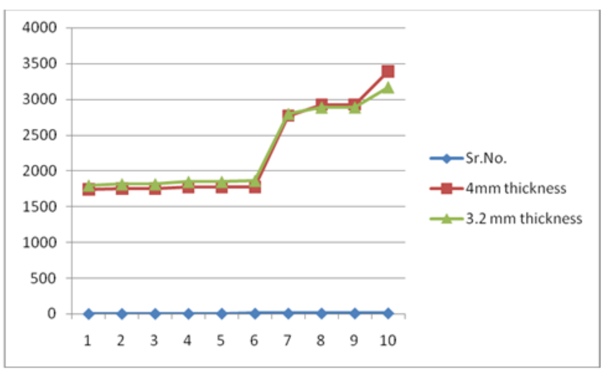
Performance analysis is carried out here to reach the conclusion. Differences in results are compared for same ratio of the inner to outer diameters, but variable thickness & hole sizes of brake discs for inner edge clamped and outer edge free boundary condition. To find natural frequency & modes by FEM (Ansys software):Remarkable change in natural frequency received as diameter of air holes or holder holes or disk brake thickness will be changed in FEM results. For most engineering circumstances the aspect ratio gets no smaller than 0·05.We also made a finite element model of a disc with using ANSYS to get natural frequencies and mode shapes.

XI. RESULTS BY FEM ANSYS SOFTWARE

We shall now attempt to compare the predicted and measured results for individual modes or pairs of modes. As these are the modes with a low number of diameter, which are most readily identified to concentrate our interest on those with 2, 3 or 4 nodal diameters. Comparisons on the results, concern the natural frequencies and the mode shapes is done here.

Table 4: Comparison of FEM Natural frequencies of disk brakes

	[Bajaj-Pulser Frequency Hz]			
Sr.No.	4mm thickness (sample 1)	3.2 mm thickness (sample 2)	Disc Holder hole dia.13mm (sample 3)	Air ventilation hole dia.9mm (sample 4)
1	1741	1794	1741	1288
2	1752	1812	1752	1297
3	1752	1812	1753	1297
4	1770	1845	1761	1309
5	1771	1845	1771	1309
6	1778	1860	1778	1313
7	2772	2795	2771	2102
8	2929	2885	2929	2219
9	2930	2886	2930	2219
10	3394	3168	3394	2572



Graph 1: Effect of wear of brake disc on FEM natural frequency.

XII. SELECTION OF DISK BRAKES

Selection of Disc Brakes:

Total 4 brake discs specimens of Bajaj Pulser of alloy steel are chosen with same b/a ratio i.e. aspect ratio (Inner to outer radius ratio).

Following are the material properties for the specimen plates.

Young's modulus (E) $= 2.1 \text{ X } 10^{11}$ N/m^2

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Poisson's ratio (γ) = 0.3
Density of material (ρ) = 5833 N/m^3

Specimen dimensions tabulated in table no. 5 shows the range of same ratio of inner to outer radius with variables & inner edge clamped and outer edge free boundary condition.

Table 5: Test specimen dimensions

Specimen	Aspect ratio b/a	Inner diameter in mm	Outer diameter in mm	Disc thickness at friction area	Dia. of hole of holding wheel hub	Dia. of air cooling holes
1st Disk	0.5416	130	240	4	10.5	8
2nd Disk	0.5416	130	240	3.20	10.5	8
3rdDisk	0.5416	130	240	3.20	13	8
4thDisc	0.5416	130	240	3.20	13	9

Inner diameter of the disc is 130mm while outer diameter is 240 mm; thickness of the brake disc is kept 4 to 3.2 mm. These specimen sizes are chosen to facilitate the measurements by using the same fixture for all the specimen plates. As boundary conditions for plate specimen are inner edge fixed and outer edge free, with these boundary conditions same ratios of inner to outer radius but variable hole sizes and different wear at disc end. Specimen dimensions tabulated in table no. 5 shows the range test specimens for inner edge clamped and outer edge free boundary condition.

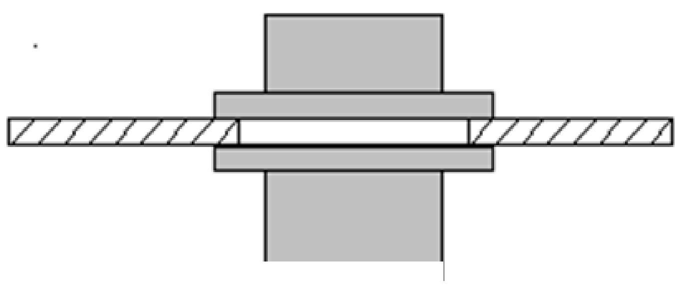

Fig.4: First clamping details of test specimen

XIII. DISCUSSION OF RESULTS

Although the end of this discussion is fulfill by chancing natural frequency, mode shapes of annular boscage slice with different holes & patterns with same rates of inner to external compass for inner edge clamped in shaft and external edge kept free still farther disquisition can be carried out on dynamic geste of slice boscage similar as rotor, caliper.

XIV. CONCLUSION

Understanding and mitigating brake "squeal" noise is crucial due to its high frequency and its occurrence at the rotor's natural frequencies and harmonics. Given that rotors in both ventilated and solid core designs serve as critical structural and acoustic elements, their dynamic behavior significantly impacts noise generation. Analyzing rotors as annular plates with radial holes allows for a deeper insight into their vibrational characteristics.

Key geometric parameters such as disc thickness, diameters, vane arrangement, and mounting configuration directly influence the rotor's natural frequencies and consequently its susceptibility to squeal. Therefore, a comprehensive study of these factors is essential, particularly in high-risk applications, to improve noise control and ensure mechanical reliability.Reckoned results have been attained for the annular slice boscage clamped at inner edge and free at the external edge system.

- Natural frequentness of slice boscage of bike increases as the slice consistence decreases till first six natural frequentness but rear effect after 7th natural freq.but seventh frequency is changed veritably less.
- Natural frequentness of slice boscage of bike decreases as the boscage slice holder hole periphery increases till first 7 natural frequentness but rear effect after 7^{th} natural frequency.
- Natural frequentness of slice boscage of bike decreases as the air ventilation hole periphery increases.

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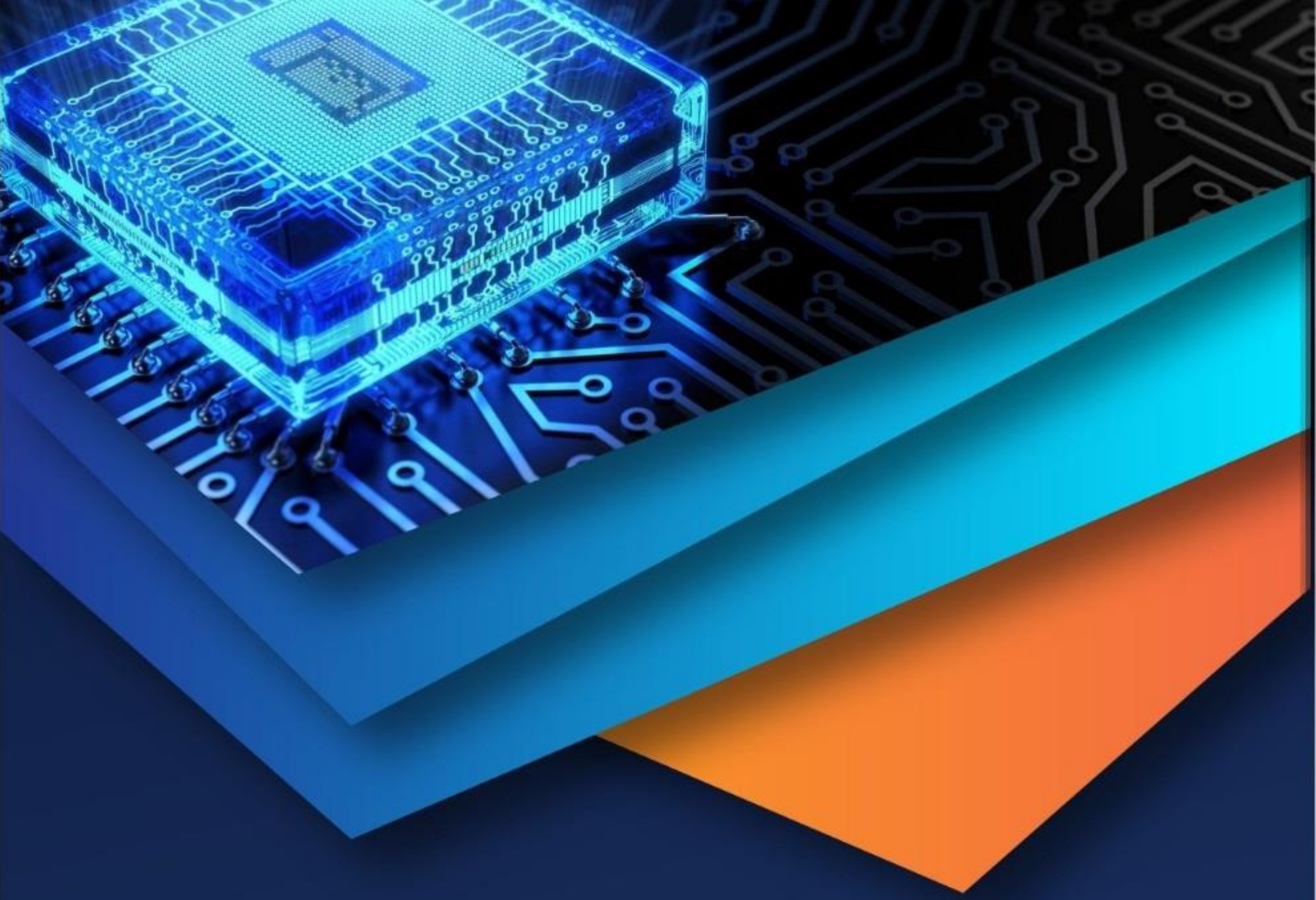

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