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# Vibration Analysis of Misaligned Shafts

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***Abstract: Vibration is one of the common sources of machinery failure. Shaft misalignment is one of the sources of vibration. Correct shaft alignment gives surety of smooth, efficient transmission of power from the prime mover to the driven machine. Misalignment produces excessive vibration, noise, coupling, and bearing or shaft failure. Shaft misalignment can be divided into two components: offset misalignment, and angular misalignment. Offset (parallel) misalignment occurs when the centerlines of two shafts are parallel but do not meet at the power transfer point. Angular misalignment occurs when centerline of two shafts intersect at the power transfer point but are not parallel. Often misalignment in actual machinery exhibits a combination of both types of misalignment. In this work, vibration analysis of misaligned shafts is done by experimentation and finite element analysis. Vibration accelerations were measured using single channel vibration analyzer for baseline and the misalignment condition. The experimental results are in good agreement with the finite elements analysis results. The peak values are observed on the multiple of rotational frequency. This work will be helpful to condition monitoring of rotary machines which fails due to the misalignment between shafts. It will help for predictive maintenance and to optimize breakdown period.***
***Keywords: Condition monitoring, Flexible Coupling, Misalignment, Rotor Shaft, Vibration Analysis.***

## I. INTRODUCTION

José M. Bossio, Guillermo R. Bossio and Cristian H. De Angelo(2009) described the problem of angular shaft misalignment in motors. The load system coupled through flexible couplings is analyzed in this work. A model for the analysis and diagnosis of angular misalignment in induction motors is presented.

This work shows that the angular misalignment produces oscillations of torque and speed. A motor working under the misalignment of a shaft undergoes perturbation frequency that doubles that of rotation. Such perturbations due to misalignment are produced by the previously mentioned oscillation. In the similar way oscillation effects are also observed on the instantaneous active power consumed by the motor. For a three phase motor at constant load by the motor is constant but due to the misalignment angle, the instantaneous active power undergoes perturbations at the misalignment frequency. The effect of angular misalignment on the current spectrum shows to sidebands around the fundamental component [01].

Vaggeeram Hariharan and PSS Srinivasan (2011) have done experimental studies on a rotor dynamic test apparatus to predict the vibration spectrum for shaft misalignment. A self-designed simplified 3–pin type flexible coupling was used in the experiments. Vibration accelerations were measured using dual channel vibration analyser for baseline and the misalignment condition.

The rigid and pin type flexible coupling with shaft parallel misalignment is simulated and studied using the both experimental investigation and simulation. Finally the author other concluded that experimental and simulated frequency spectra are similar, the experimental predictions are in good agreement with the ANSYS results. Both the experiment and simulation results prove that misalignment can be characterized primarily by second harmonics (2X) of shaft running speed. He also found that by using new newly designed flexible coupling, the vibration amplitudes due to the shaft parallel misalignment are found to reduce by in percentage [02].

Piotrowski. J. (2006) described importance of misalignment phenomenon as Industry worldwide is losing billions of dollars a year due to misalignment of machinery.

The heart and soul of virtually every industrial operation pivots on keeping rotating machinery in good working order. Countless processes are dependent on the successful operation of rotating machines that produce electric power, fuels, paper, steel, glass, pharmaceuticals, the food we eat, the clothes we wear, the buildings we live and work in, and the vehicles that transport us across the surface of the Earth. Just about everything you see around has somehow been influenced by rotating machinery of some kind. The primary objective of accurate alignment is to increase the operating life span of rotating machinery. To achieve this goal, machinery components that are most likely to fail must operate well within their design limits. Despite popular belief, misalignment can disguise itself very well on industrial rotating machinery.

He observed that the secondary effects of misalignment as it slowly damage the machinery over long periods of time. Some of the common symptoms of misalignment are as follows:

1) Premature bearing, seal, shaft, or coupling failures.
2) Elevated temperatures at or near the bearings or high discharge oil temperatures.
3) Excessive amount of lubricant leakage at the bearing seals.
4) Certain types of flexible couplings will exhibit higher than normal temperatures when running or will be hot immediately after the unit is shut down. If the coupling is an elastomeric type, look for rubber powder inside the coupling shroud.
5) Similar pieces of equipment seem to have a longer operating life.
6) Unusually high number of coupling failures or wear quickly.
7) The shafts are breaking (or cracking) at or close to the inboard bearings or coupling hubs [03].

M. LI, and L. YU (2001) found that misalignment of a gear coupling in a multi rotor system is an important problem; it can cause various faults. In this work the non-linear coupled lateral torsion vibration model of rotor-bearing-gear coupling system is developed based on the engagement conditions of gear couplings. From the theoretical analysis author concluded that the forces and moments acting on gear couplings due to the initial misalignment are from the inertia forces of the sleeve and the internal damping between the meshing teeth, and depend on the misalignment, internal damping, the rotating speed, and the structural parameters of the gear coupling [04].

## II. METHODOLOGY

### *A. Simulation (Finite Element Analysis)*

Rotary shaft, coupling and disk are modelled using CATIA V5 R19 with the same dimensions which are used in the experimental setup. The figure 1 shows the assembly. The following table I shows the dimensions used for making a component models.

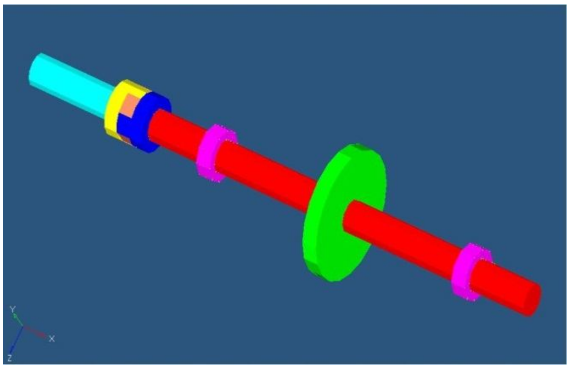

Fig. 1: Assembly of rotor shaft and coupling

TABLE I
Dimensions Of Shaft And Coupling Assembly

| Sr. No. | Description | Value |
|---|---|---|
| 1 | Input Shaft Diameter | 19mm |
| 2 | Output Shaft Diameter | 19 mm |
| 3 | Disk Diameter | 75 mm |
| 4 | Disk thickness | 10 mm |
| 5 | Coupling Inner Diameter | 19 mm |
| 6 | Keyway depth | |
| | In shaft | 4 mm |
| | In hub | 3 mm |
| | Keyway cross section | |
| | Height | 6 mm |
| | Width | 6 mm |
| 7 | Bolt diameter | 6 mm |

Then this assembly is imported in hypermesh software and meshing is carried out.

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While doing meshing or even developing the model, it is important to decide which type of meshing is more suitable – a free mesh or a mapped mesh for the analysis. A free mesh has no restrictions in terms of element shapes, and has no specified pattern. A mapped mesh on the contrary, is restricted in terms of the element shape it contains and the pattern of the mesh. A mapped area mesh contains either quadrilateral or triangular elements, while the mapped volume mesh contains hexahedron elements. In addition, a mapped mesh typically has a regular pattern, with obvious rows of elements. In this type of mesh, first it is necessary to build the geometry as a series of fairly regular volumes and/or areas and the mapped mesh. In the present model, free mesh has been used with the element type of PSOLID. The meshed model is shown in Figure 2.

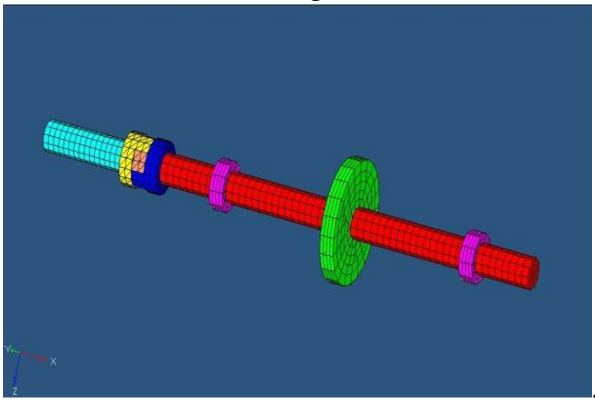

Fig. 2: Meshed Model

The table II shows the properties of material used for analysis.
The rotating shaft is supported between two identical ball bearings of 190 mm span on non-drive end and one bearing on the drive end. The bearing P 204 type is represented by COMBIN 40 element and the stiffness of the bearing is 1.5 x 10 4 N/mm. The rotor shaft model rotates with respect to global Cartesian X-axis. The angular velocity is applied with respect to X-axis. The degree of freedoms along UX, UZ, ROTY, ROTZ are used at bearing ends. Different angular velocities are given as input and corresponding accelerations are measured at the non-drive end because at this location the maximum vibrations are generated.

TABLE II
Material Properties

| Properties | Brass | Mild | Rubber |
|---|---|---|---|
| Young's Modulus(MPA) | $1.05\times10^5$ | $2.1\times10^5$ | 30 |
| Poisson's Ratio | 0.33 | 0.290 | 0.49 |
| Density($kg/m^3$) | 8500 | 7800 | 1100 |

The solver used for the analysis is Nastran. Hypergraph is used for post processing and results are obtained.

*B. Experimentation*

Figure 3 shows the experimental set up used to study effects of shaft misalignment. It consists of a 3 phase AC induction motor of 0.75 kW.

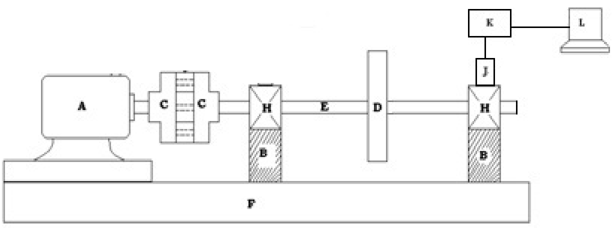


Fig. 3: Experimental setup

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A: AC Motor, B: Bearing Support, C: Coupling, D: Disk, E: Shaft, F: Base Plate, H: Ball Bearing, J: Accelerometer,
J: Accelerometer, K: FFT analyser, L: Computer.

The electric motor shaft is connected to driven shaft through flexible jaw coupling. The driven shaft has length 280 mm and diameter 19 mm. The driven shaft is supported between two identical ball bearings having span of 190mm. These two ball bearings are mounted on two supports. A circular disk of diameter 75 mm and thickness 10 mm is mounted on the driven shaft.

The motor end is adjustable in horizontal direction so that different misalignments between the shafts are created. Three phase A.C. auto transformer is used for voltage controller adjust the power supply so that motor speed can be changed.

Practical methods are developed for monitoring alignment of systems especially at static condition. Due care should be taken during the installation of all systems and their reconditioning, to ensure precise alignment between relative position of assembled system.

Major methods to diagnose alignment condition are as follows.

1) Reverse indicator method
2) Face & rim indicator method
3) Laser method
4) Double radial method
5) Shaft to coupling spool method
6) Face-face method.

Out of all above methods, face and rim method is used to correct the shaft alignment.

### *C. Face and Rim Method*

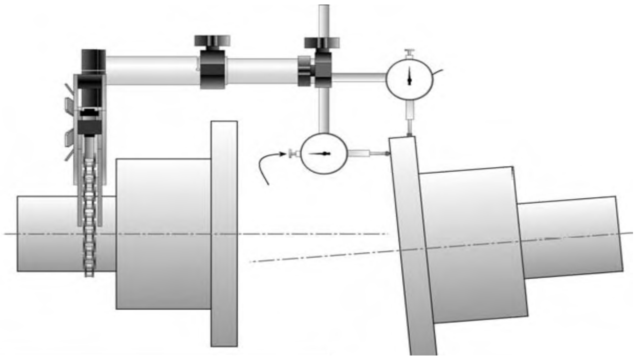

Fig. 4: Face and rim method set up

The basic setup is as shown in Figure 4. A clamp is fixtured to the rotating shaft or a part rigid with it, like the coupling hub. An extension bar is spanned to the other coupling hub where two dial indicators are attached one to take rim reading and another to take a face reading. The second shaft can be rotating or stationary better if it rotates with first shaft so that coupling imperfections are not added into the measurement. The coupling can be disassembled or not better if it is not.

The indicator is always attached or fixture onto the rotating shaft and reads on a rotating or stationary shaft. If second shaft rotates than dial indicator registers only misalignment data. If the second part is stationary, then the dial indicator measures the sum effect of misalignment plus run out of surface being read. Therefore, it is important that the surface, i.e. the coupling half, be round and concentric with the shaft axis if the second shaft is stationary. A stationary indicator reading on a rotating surface will only show run out. For this reason, the indicator fixture is never clamped to stationary shaft unless the purpose is a run out measurement.

The ideal practice is to rotate both shafts together such that the indicator tips, always read on the same spot. This way the condition of the coupling is not relevant, and in fact, it is possible to accurately alignment the shaft even if the coupling is not true or is mounted crooked. The face and rim setup takes two independent measurements. The rim reading measures parallel offset while the face reading measures angularity. Because the two conditions are measured separately, the ability exists to easily separate the two effects in the field with simple observations of the readings. This is the beauty, or simplicity of the face and rim method.

By using this method alignment is done at stationary. After this a different misalignments between the shafts are created and measured using the dial gauge indicator. The misalignments are given to the motor end at the speed range of 720 rpm to 300 rpm. Signals are acquired using accelerometer at the second bearing end at which maximum vibration energy is transferred. The instruments used in the experiment include dual channel vibration analyser (FFT Analyser) and accelerometer.

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## III.RESULTS AND DISCUSSIONS

Table III, IV and V shows the simulated vibration amplitudes in $m/s^2$ of non-drive end at different speeds. The numerical frequency spectra of non-drive end for parallel alignment is shown in the figure 5, for angular misalignment is shown in the figure 6 and for combined misalignment is shown in the figure 7. From this figures it is observed that peak amplitude values are multiple of rotational frequency(X). It also shows that as misalignment increases the vibration amplitudes are also increases.

TABLE III OVL For Parallel Misalignment

| Sr.No. | Misalignment 'mm' | Speed(RPM) OVL (m/s2) | | | |
|---|---|---|---|---|---|
| | | 720 | 900 | 1440 | 2880 |
| 1 | 0 | 0.25 | 0.275 | 0.39 | 0.48 |
| 2 | 0.05 | 0.31 | 0.325 | 0.414 | 0.477 |
| 3 | 0.1 | 0.405 | 0.456 | 0.533 | 0.654 |
| 4 | 0.2 | 0.413 | 0.478 | 0.62 | 0.786 |
| 5 | 0.4 | 0.434 | 0.45 | 0.685 | 0.89 |

TABLE IV OVL For Angular Misalignment

| Sr.No. | Misalignment 'Degree' | Speed(RPM) OVL (m/s2) | | | |
|---|---|---|---|---|---|
| | | 720 | 900 | 1440 | 2880 |
| 1 | 0 | 0.25 | 0.275 | 0.39 | 0.48 |
| 2 | 0.029 | 0.284 | 0.296 | 0.453 | 0.55 |
| 3 | 0.058 | 0.312 | 0.285 | 0.426 | 0.65 |
| 4 | 0.087 | 0.35 | 0.33 | 0.489 | 0.766 |
| 5 | 0.116 | 0.43 | 0.444 | 0.51 | 0.844 |

TABLE V OVL for combined Misalignment

| Sr.No. | Misalignment 'Combined' | Speed(RPM) OVL (m/s2) | | | |
|---|---|---|---|---|---|
| | | 720 | 900 | 1440 | 2880 |
| 1 | 0 | 0.25 | 0.275 | 0.39 | 0.48 |
| 2 | 0.05 and 0.029 | 0.356 | 0.456 | 0.879 | 0.945 |
| 3 | 0.05 and 0.058 | 0.469 | 0.546 | 0.89 | 0.876 |
| 4 | 0.1 and 0.029 | 0.657 | 0.776 | 0.91 | 0.934 |
| 5 | 0.2 and 0.029 | 0.884 | 0.493 | 0.786 | 0.76 |

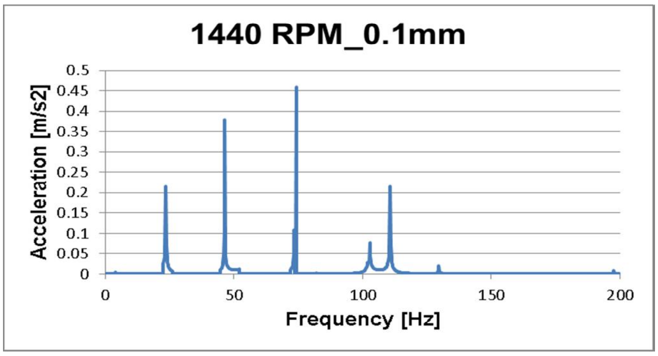


Fig. 5: Vibration Spectra for 0.1 mm misalignment at 1440 RPM

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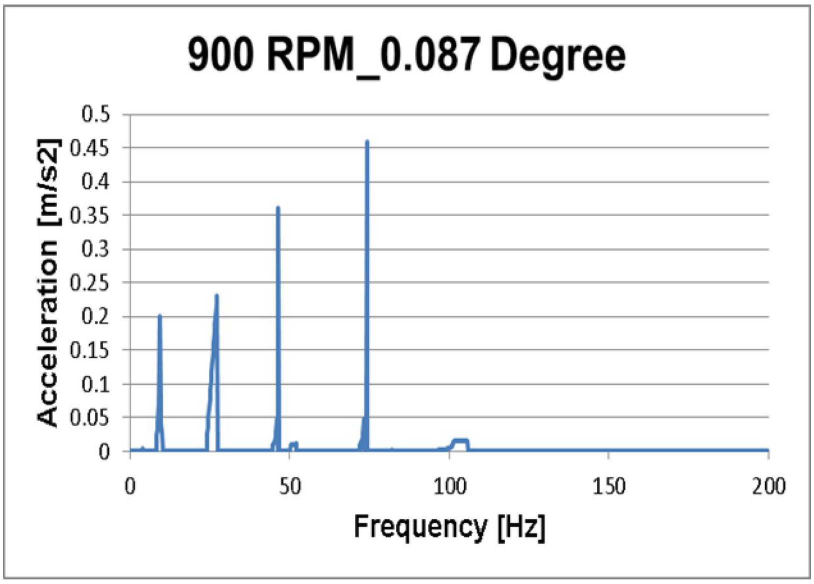


Fig. 6: Vibration Spectra for 0.07 degree misalignment at 1440 RPM

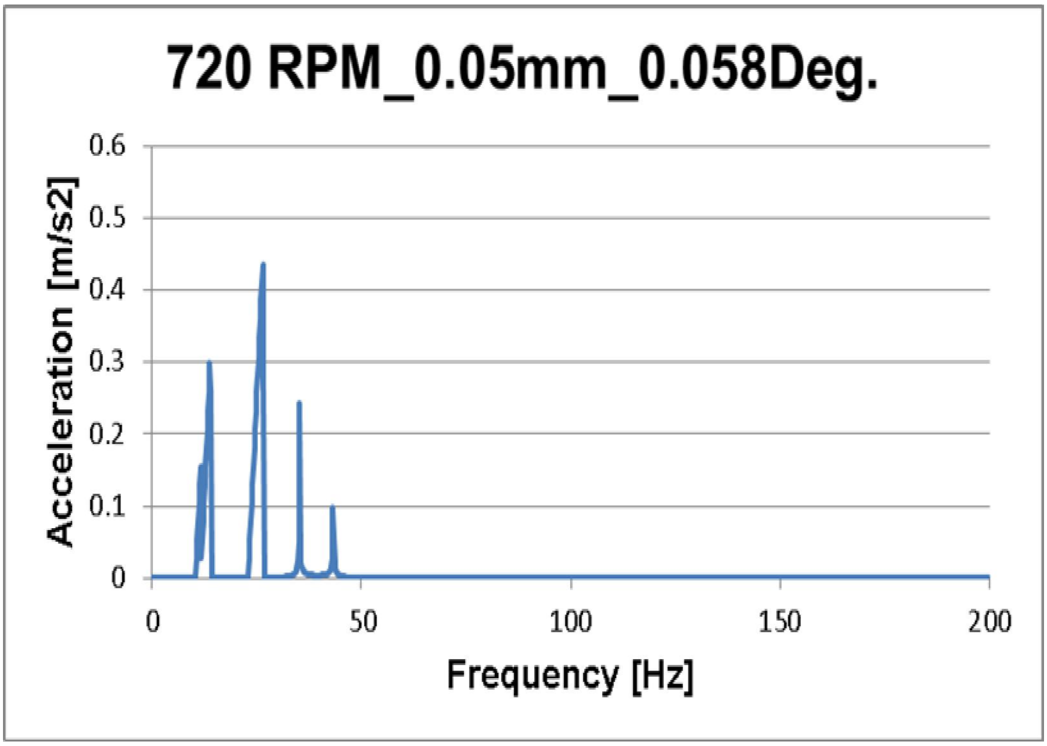


Fig. 7: Vibration Spectra for 0.05 mm and 0.058 degree misalignment at 720 RPM

## IV.CONCLUSIONS

In these work the effect of different misalignments was studied for different speeds by finite element method using software. It is found that the vibration amplitudes the 2X and 3X are gradually increases with increase in misalignment and rotational speed of shaft. The vibration graphs shows that the peak values are multiple of the rotational frequency(X). As vibration analysis is one of the most parameter in condition monitoring, this work will helpful for condition monitoring of rotary machines. By using one of the methods of alignment, the future failure of the rotary machine can be avoided. So this work will be helpful for predictive maintenance.

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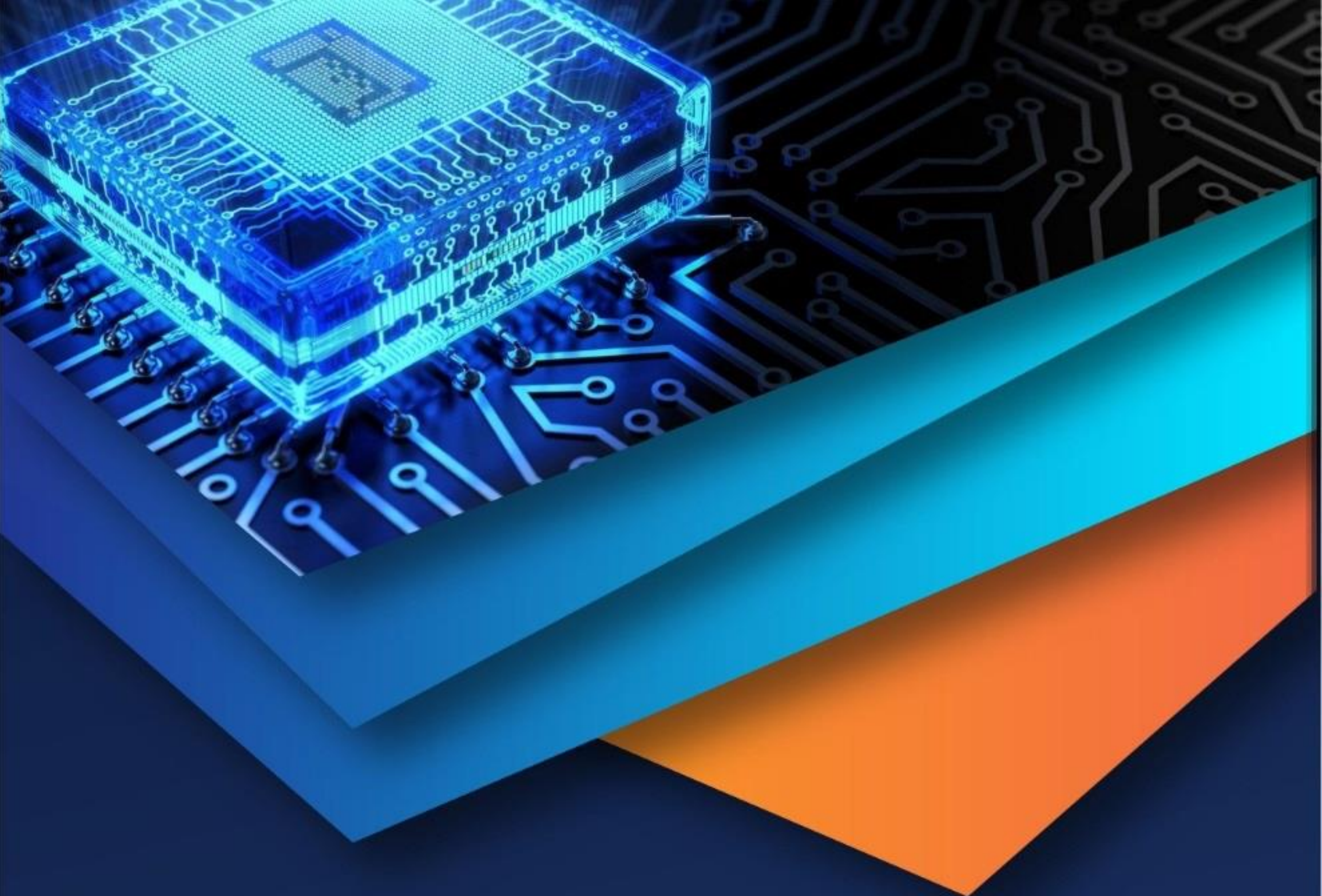

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