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Failure Analysis of Bevel Gear System in Servo Geared Motor Drive System of Cnc Machine to Improve Strength Using Fem Approach

[1]Pratik M Hadani, [2] Vasim G. Machhar,

[1]Designation of 1st Author,[2]Perusing, M.E.(CAD/CAM),

[1] Mechanical Engineering Dept., Noble Group of Institutions, Junagadh,India

Abstract— **Gears are an integral and necessary component in our day to day lives. They are present in the satellites we communicate with, automobiles and bicycles we travel with. Gears have been around for hundreds of years and their shapes, sizes, and uses are limitless. For the vast majority of our history gears have been understood only functionally. That is to say, the way they transmit power and the size they need to be to transmit that power have been well known for many years. It was not until recently that humans began to use mathematics and engineering to more accurately and safely design these gears. Bevel gears are widely used because of their suitability towards transferring power between nonparallel shafts at almost any angle or speed. The American Gear Manufacturing Association (AGMA) has developed standards for the design, analysis, and manufacture of bevel gears. The bending stress equation for bevel gear teeth is obtained from the Lewis bending stress equation for a beam and bending stress value derive for the spiral bevel gear, straight teeth bevel gear and zero bevel gear. For above mentioned gear comparison between analytical value and value obtain by the ANSYS Workbench.**

***Index Terms*—Bevel Gears, Finite Element Analysis, Face Width, Strength**

I. Introduction

One of the best methods of transmitting power between the shafts is gears. Gears are mostly used to transmit torque and angular velocity. The rapid development of industries such as vehicle, shipbuilding and aircraft require advanced application of gear technology.

The bevel gears are used for transmitting power at a constant velocity ratio between two shafts whose axes intersect at a certain angle. The pitch surfaces for the bevel gear are frustums of cones.

"Mustang CNC Automation" are famous manufacturer of a broad range of Angular Cutter, Cutting Tool, Machine Tool, CNC Machine, SPM Machine and VMC Machine. They provide these products in diverse specifications to attain the complete satisfaction of the clients. This company is located at Rajkot (Gujarat, India) and constructed a wide and well functional infrastructural unit where these products are being manufactured as per the global set standards. They are maintaining a consistency in terms of quality for products and services.

II. Problem Definition

Mustang CNC machine, Rajkot manufacturing CNC machine since last 15 years facing problem in transferring power from servo motor drive system. Bevel gearing is used for power transmission at 90 ° Angle. Facing problem of Failure before premature life of gear and wanted to increase efficiency of transmission system. They wanted to get rid of problem considering bevel gearing system.

III. Data taken from the MUSTANG CNC

The SIMOTICS S-1FG1 servo geared motors are compact geared motors compared to standard geared motors with induction machines; they have smaller dimensions, weight less and have a higher dynamic response. The range of types covers helical, parallel shaft, bevel and helical worm geared motors in the usual frame sizes and speed/torque classes. The SIMOTICS S-1FG1 servo geared motors use the same operating heads as the SIMOGEAR geared motors.

Table 1 Motor Specification

MOTOR SPECIFICATION	
MAKE	SIEMENS STANDARD MOTORS LTD
MODEL NO	SIMOTICS S1 FL6
TYPE	SERVO MOTOR
M_N	2.39 N.M
M_O	3.5 N.M
U_N	230V
P_N	0.75 KW
I_N	2.1 A

I_O	2.9 A
n_N	3000 REVOLUTION / MINUTE
n_{MAX}	4000 REVOLUTION / MINUTE
MASS	6.2 KG
BRAKE	2 V DC
ALLOWABLE STRESSES ON BEVEL GEAR	
σ_g	95 MPA
σ_P	95 MPA

Based on above dimensions we calculated dimensions of gear and pinion.

Table 2 Dimensions of gear and pinion

	PINION	**GEAR**
BASE CIRCLE DIAMETER	24.24 mm	24.24 mm
OUTSIDE DIAMETER	32.24 mm	32.24 mm
INSIDE DIAMETER	22.95 mm	22.95 mm
WHOLE DEPTH	6.566 mm	6.566 mm
NO OF TOOTH	10 nos	10 nos
MODULE	3	3
PITCH ANGLE	45'	45'
FACE WIDTH	7 mm	7 mm
ADDENDUM	3 mm	3 mm
DEDENDUM	3.566 mm	3.566 mm
LENGTH OF PITCH CONE ELEMENT	20 mm	20 mm

IV. ANALYSIS OF EXISTING DESIGN

Here first of all we have made a model in modelling software Creo 3.0 and after that we are going to analyse it in Ansys 15.0 with respect to different loads to fatigue life.

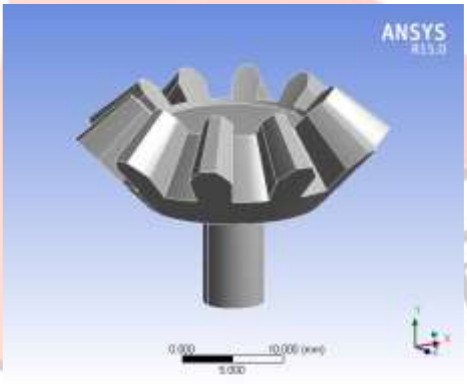

FIGURE 1: IGES file imported in Ansys 15.0

After getting model in Ansys we are going to analyse it with respect to various design considerations of bevel gear.

Table 3 Units

Unit System	**Metric (mm, kg, N, s, mV, mA) Degrees rad/s Celsius**
Angle	Degrees
Rotational Velocity	rad/s
Temperature	Celsius

Table 4 Material Data

Type	**Structural steel**
Density	7.85e-006 kg mm^-3
Compressive ultimate strength	0MPA
Compressive yield strength	250MPA
Tensile yield strength	250MPA
Tensile ultimate strength	460MPA

Vonmises stress, maximum principal stress, total deformation, safety factor are being calculated in ansys workbench 15.0 and plotted as shown below. Factor of safety is less then one in existing design which shows that current design is not safe.

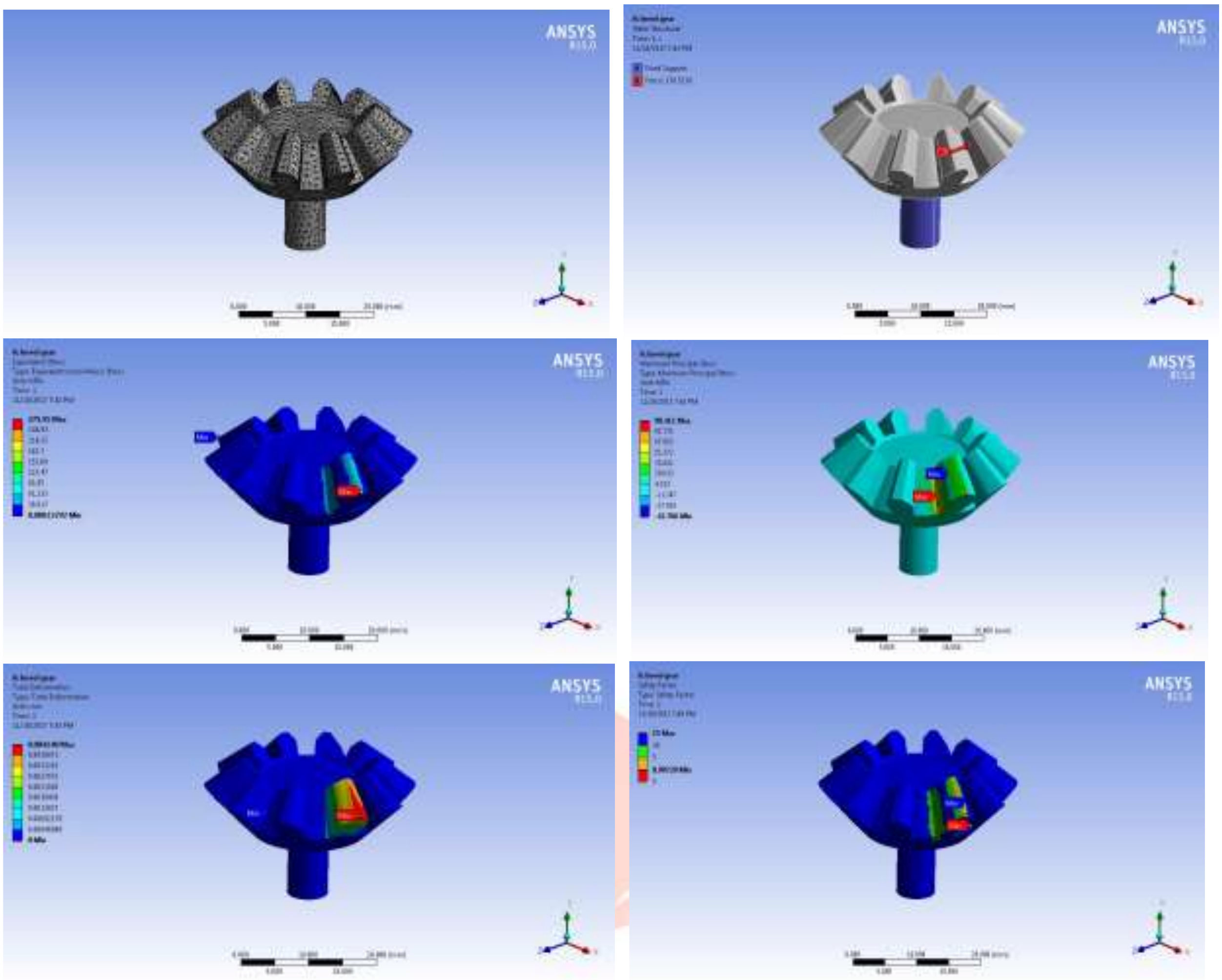

FIGURE 2 FEM ANALYSIS OF EXISTING DESIGN

V. ANALYSIS OF MODIFIED DESIGN

We are changing face width of bevel gear and calculating vonmises stress, maximum principal stress, total deformation, factor of safety for different face widths. (b=8,9,10,11,12mm)

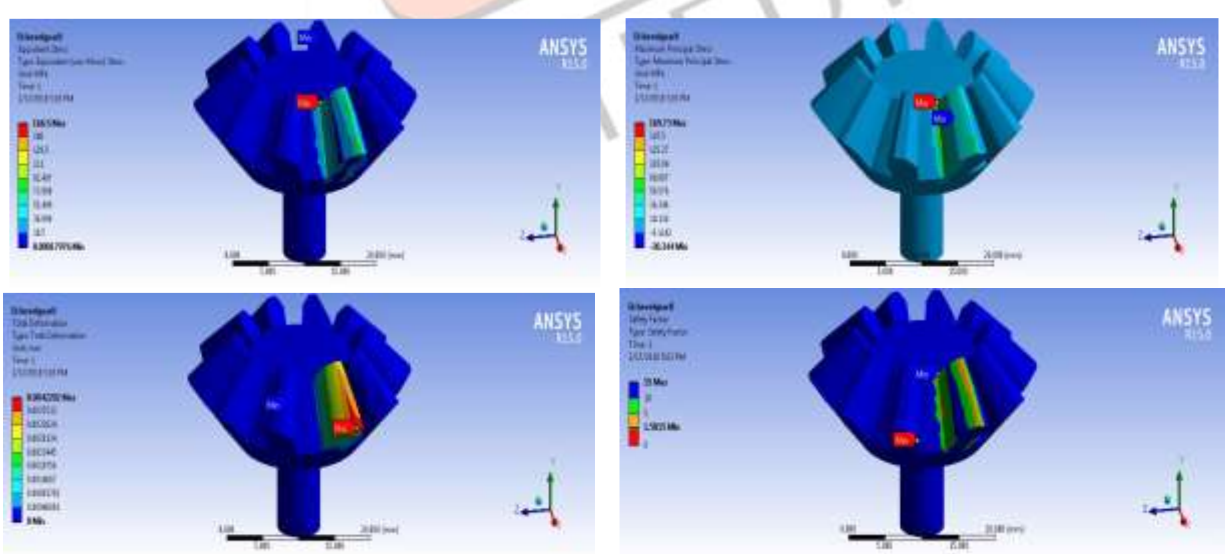

Figure 3 NEW DESIGN OF FACE WIDTH B=8mm

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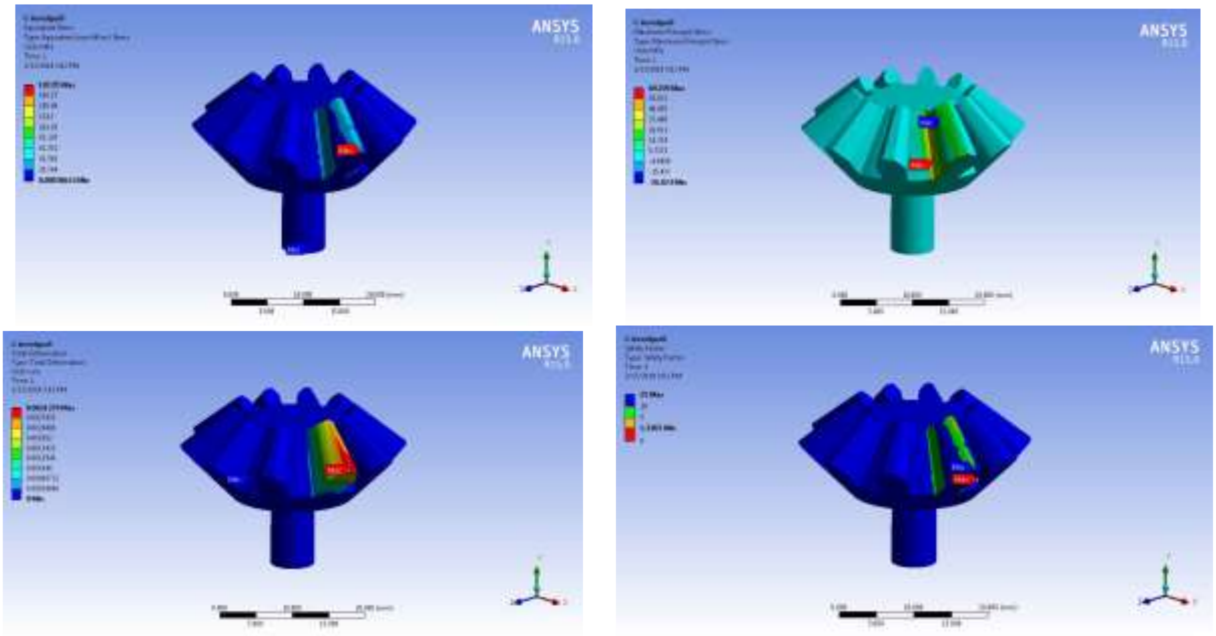


Figure 4 NEW DESIGN OF FACE WIDTH B=9MM

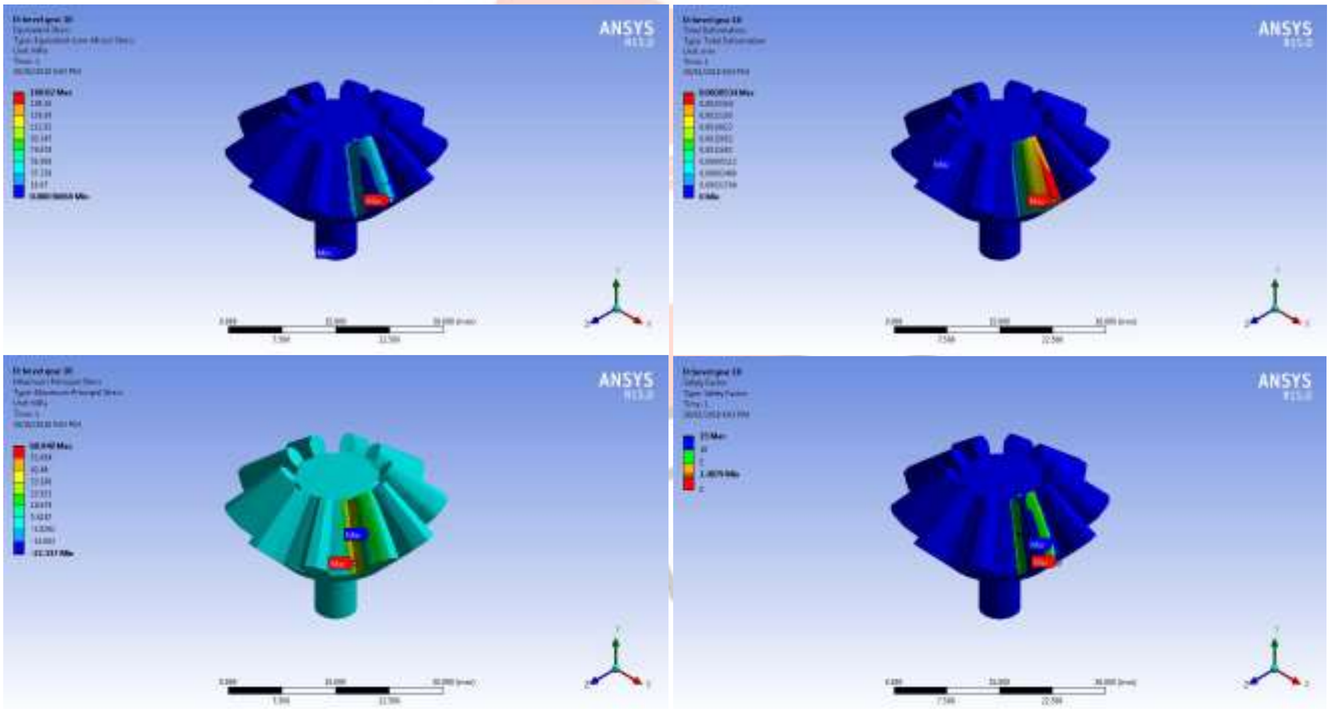
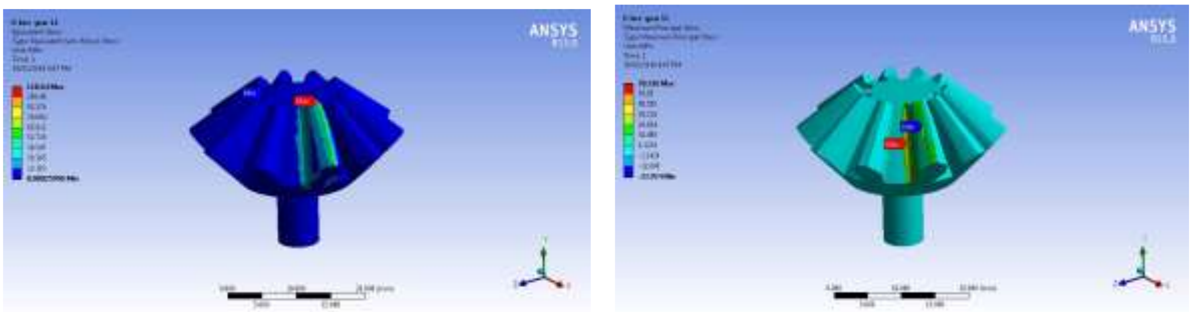


Figure 5 NEW DESIGN OF FACE WIDTH B=10MM



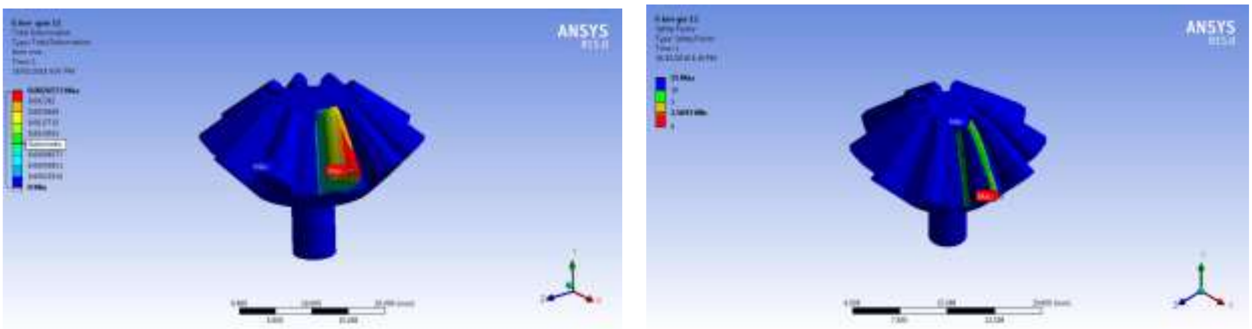


Figure 6 NEW DESIGN OF FACE WIDTH B=11MM

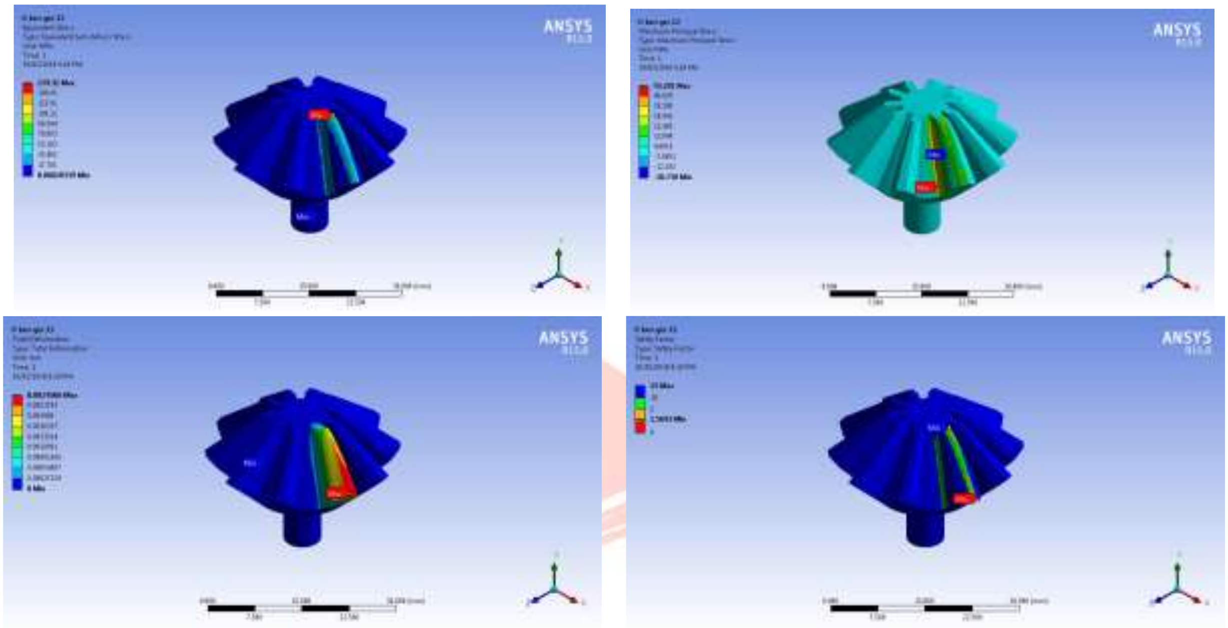


Figure 7 NEW DESIGN OF FACE WIDTH B=12MM

For making analysis in ansys workbench 15.0 we are input different parameters (like volume, nodes, elements, element size, smoothing, environment temperature, load) in preprocessing as shown in table5

Table 5 Parameter table

	7MM	8MM	9MM	10MM	11MM	12MM
VOLUME	3823.7 mm³	4132.4 mm³	4386.2 mm³	4478.2 mm³	4648.3 mm³	4775.9 mm³
MASS	3.0016 e-002 kg	3.2439e-002 kg	3.4432e-002 kg	3.5154e-002 kg	3.6489e-002 kg	3.7491e-002 kg
NODES	27356	29972	20578	21035	38296	40579
ELEMENTS	15160	16840	11414	11594	21604	23063
ELEMENT SIZE	0.560 mm	0.560 mm	0.70 mm	0.70 mm	0.50 mm	0.50 mm
SMOOTHING	Medium	Medium	Medium	Medium	Medium	Medium
ENVIRONMENT TEMPERATURE	22 C	22 C	22 C	22 C	22 C	22 C
LOAD	170.52N	170.52N	170.52N	170.52N	170.52N	170.52N

After solving for different face widths we are getting the results from ansys workbench15.0 as shown in table15.0

Table 6 Result From AnsyS

	7MM	8MM	9MM	10MM	11MM	12MM
VONMISES STRESS	275.55MPA	166.5MPA	187.05MPA	168.02MPA	118.64MPA	159.31MPA
MAXIMUM PRINCIPAL STRESS	98.411MPA	169.73MPA	69.259MPA	60.948MPA	70.316MPA	55.281MPA
TOTAL DEFORMATION	0.004MM	0.004MM	0.003MM	.002MM	.002MM	.002MM
SAFETY FACTOR	0.907	1.5015	1.3365	1.4879	1.5510	1.5693
LIFE	16191	46806	67790	1.006*10^5	2.155*10^5	86039

VI. LEWIS BENDING STRESS EQUATION

We are calculating maximum principal stress by lewis bending stress equation for validation. The calculations for different face widths are as follows.

FOR FACE WIDTH B= 7MM

$$\Sigma_B = \frac{F_T}{C_V * B * \Pi * M * Y'}$$
$$\Sigma_B = \frac{170.52}{0.577*7*\Pi*3*0.0724}.$$
$$\Sigma_B = 81.44MPA$$

FOR FACE WIDTH B= 8MM

$$\Sigma_B = \frac{F_T}{C_V * B * \Pi * M * Y'}$$
$$\Sigma_B = \frac{170.52}{0.577 * 8 * \Pi * 3 * 0.0724}$$
$$\Sigma_B = 71.26MPA$$

FOR FACE WIDTH B= 9MM

$$\Sigma_B = \frac{F_T}{C_V * B * \Pi * M * Y'}$$
$$\Sigma_B = \frac{170.52}{0.577 * 9 * \Pi * 3 * 0.0724}$$
$$\Sigma_B = 63.34MPA$$

FOR FACE WIDTH B= 10MM

$$\Sigma_B = \frac{F_T}{C_V * B * \Pi * M * Y'}$$
$$\Sigma_B = \frac{170.52}{0.577 * 10 * \Pi * 3 * 0.0724}$$
$$\Sigma_B = 57.01MPA$$

FOR FACE WIDTH B= 11MM

$$\Sigma_B = \frac{F_T}{C_V * B * \Pi * M * Y'}$$
$$\Sigma_B = \frac{170.52}{0.577 * 11 * \Pi * 3 * 0.0724}$$
$$\Sigma_B = 51.83MPA$$

FOR FACE WIDTH B= 12MM

$$\Sigma_B = \frac{F_T}{C_V * B * \Pi * M * Y'}$$
$$\Sigma_B = \frac{170.52}{0.577 * 12 * \Pi * 3 * 0.0724}$$
$$\Sigma_B = 47.51MPA$$

VII. AGMA STRESS EQUATION

We are calculating maximum principal stress by AGMA stress equation for validation. The calculations for different face widths are as follows.

FOR FACE WIDTH B= 7MM

$$\sigma_b = \frac{F_t K_v K_0 K_m}{BmJ}$$
$$\sigma_b = \frac{170.52 * 1.6 * 1 * 1.25}{8 * 3 * 0.2}$$

$$\Sigma_B = 81.2MPA$$

FOR FACE WIDTH B= 8MM

$$\Sigma_B = \frac{F_T K_V K_0 K_M}{BMJ}$$
$$\Sigma_B = \frac{170.52 * 1.6 * 1 * 1.25}{8 * 3 * 0.2}$$
$$\Sigma_B = 71.05MPA$$

FOR FACE WIDTH B= 9MM

$$\Sigma_B = \frac{F_T K_V K_0 K_M}{BMJ}$$
$$\Sigma_B = \frac{170.52 * 1.6 * 1 * 1.25}{9 * 3 * 0.2}$$
$$\Sigma_B = 63.16MPA$$

FOR FACE WIDTH B= 10MM

$$\Sigma_B = \frac{F_T K_V K_0 K_M}{BMJ}$$
$$\Sigma_B = \frac{170.52 * 1.6 * 1 * 1.25}{10 * 3 * 0.2}$$
$$\Sigma_B = 56.84MPA$$

FOR FACE WIDTH B= 11MM

$$\Sigma_B = \frac{F_T K_V K_0 K_M}{BMJ}$$
$$\Sigma_B = \frac{170.52 * 1.6 * 1 * 1.25}{11 * 3 * 0.2}$$
$$\Sigma_B = 51.67MPA$$

FOR FACE WIDTH B= 12MM

$$\Sigma_B = \frac{F_T K_V K_0 K_M}{BMJ}$$
$$\Sigma_B = \frac{170.52 * 1.6 * 1 * 1.25}{12 * 3 * 0.2}$$
$$\Sigma_B = 47.37MPA$$

VIII. Comparison table between analytical value and lewis bending stress equation

Table 7 Comparison Between Lewis And Ansys Stress Value.

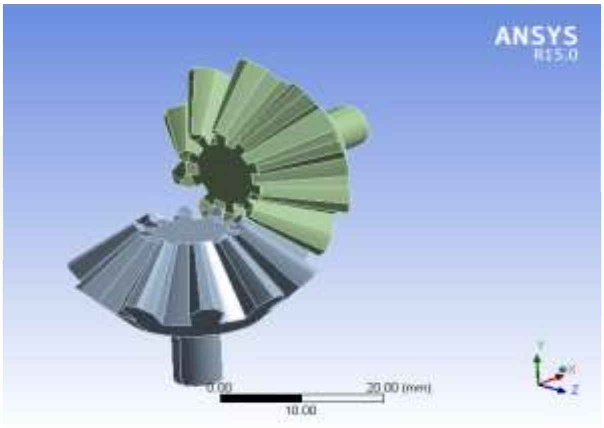
FACE WIDTH IN MM	ANSYS VALUE (N/ mm^2)	LEWIS VALUE (N/ mm^2)	DIFFERENCE %
7	98.41	81.44	17.24
8	169.73	71.26	58.01
9	69.26	63.34	8.54
10	60.95	57.01	6.46
11	70.31	51.83	26.28
12	55.28	47.51	14.05

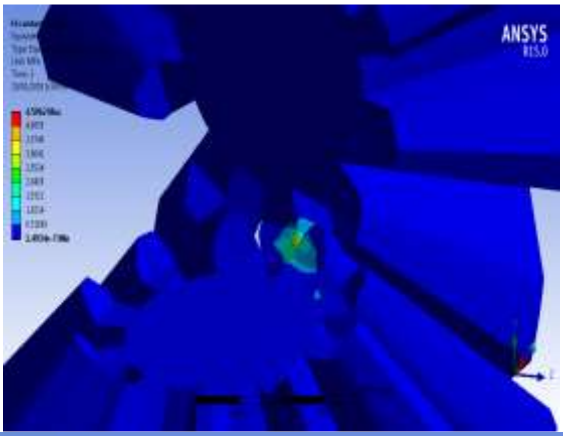
Table 8 Comparison Between Agma And Ansys Stress Value

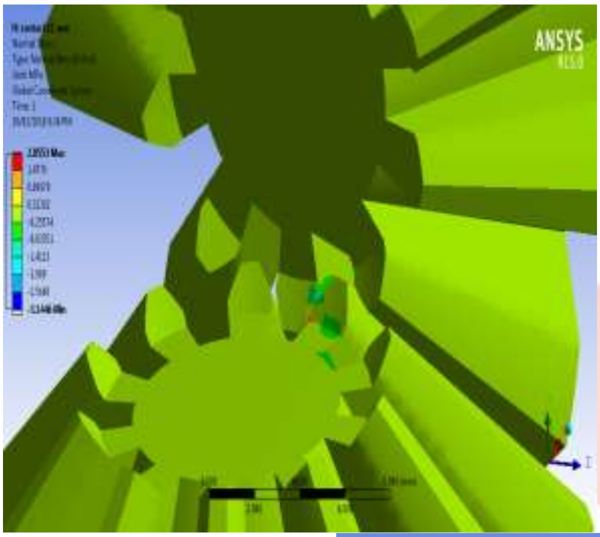
FACE WIDTH IN MM	ANSYS VALUE (N/ mm^2)	AGMA VALUE (N/ mm^2)	DIFFERENCE %
7	98.41	81.2	17.48
8	169.73	71.05	58.13
9	69.26	63.16	8.8
10	60.95	56.84	6.74
11	70.31	51.67	26.51
12	55.28	47.37	14.3

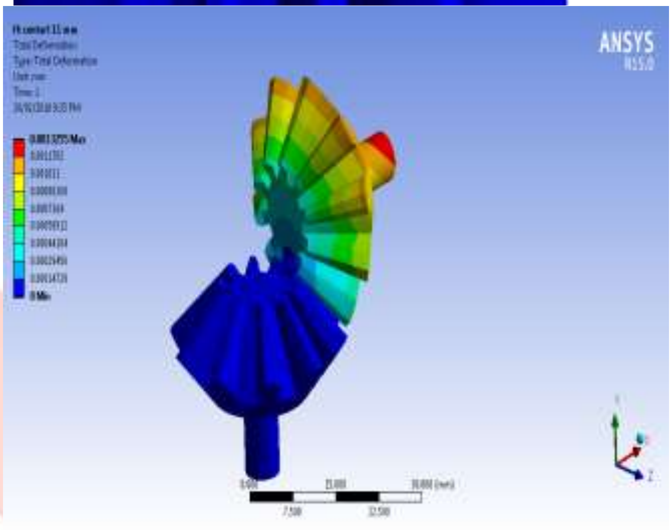
IX. Check For Contact Stress For Face Width B=12mm

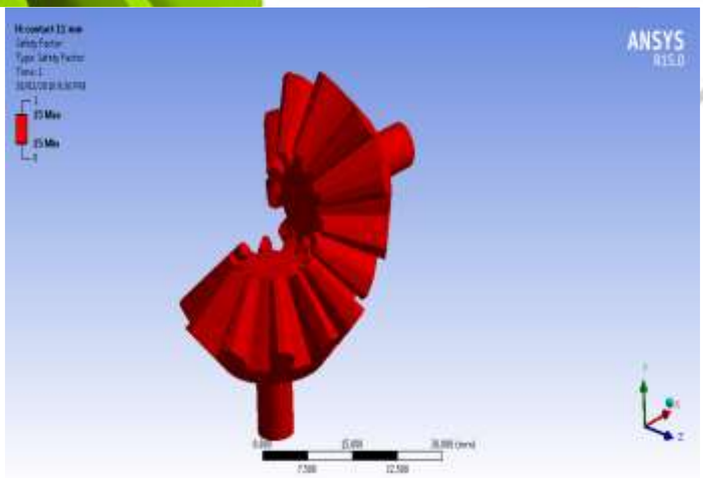
Considering above results and table maximum principal stress which is responsible for bending Failure of tooth is lower in face width of 12mm compare to face width of 7 ,8, 9,10 and 11mm. So the contact streeses are calculated and plotted for face width b= 12mm in workbench 15.0 and are shown below. Stress distribution, total deformation, factor of safety shows that bevel gear will not fail due to contact stresses. So the design is safe for face width b=12mm.











X. CONCLUSION

Considering above results and table maximum principal stress which is responsible for bending Failure of tooth is lower in face width of 12mm compare to face width of 7 ,8, 9,10 and 11mm. Factor of safety is also satisfactory and compatible with achieve results. Design. Contact stresses are also being checked for face width b=12mm.and it is safe. So face width change of 12 mm is suitable for safe drive and long life of gear compare to existing

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