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OPTIMAL DESIGN OF PARAMETERS OF INVOLUTE SPUR GEAR USING FINITE ELEMENT ANALYSIS

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***Abstract*— Gear is one of the most critical components in mechanical power transmission system. The design of gear requires an iterative approach to optimize the design parameters. Failure of any one gear will affect the whole transmission system. To overcome this problem, it is necessary to find the root causes of gear failure and try to eliminate those causes. Bending and contact stresses were the major causes of spur gear failure. In this paper, an attempt has been made to reduce the stresses by modifying the face width, module and pressure angle of the spur gear. The stresses were analytically calculated using Hertz, AGMA and Lewis equations using MS-Excel. Finite Element Analysis is one of the best optimization tools which were used to optimize the design parameters of spur gear. It was observed that, the maximum bending and contact stresses were decreased while increasing face width, module and recommendable to use optimal values of tooth parameters during design work. This stress reduction improved their pitting resistance and prolonged the service life.**

Keywords—spur gear, parameters, bending stress, contact stress, FEA

I. INTRODUCTION

The history of gears is probably as old as civilization itself. Even today, the importance of gears in the manufacturing industry is undiminished and continues to grow. Gears are wheel-like machine elements that have teeth uniformly spaced around the outer surface of the blank. Gears deliver force (torque) and motion (rpm) from one part of a machine to another. Gear failure can occur in various modes. If care is taken during the design stage itself to prevent each of these failures a sound gear design can be evolved. It is clearly seen that pitting is the major reason for gear failure. 60% of gear failure is only due to pitting. So there is necessary to reduce the pitting in order to achieve maximum life. Exceeding limiting values of bending and contact stress will result into pitting and reduce the life. These stresses should be minimized in order to get maximum life.

II. LITERATURE REVIEW

Metin Zeyveli et al. applied Genetic Algorithm (GA) to the problem with the objective function of minimizing of volume of gear trains. The preliminary design parameters such as module, number of teeth, and width of teeth for pinion and gear pairs of the stages were optimized and gear ratios were determined in respect to the objective function and design constraints [1]. Alexander L. Kapelevich et al. minimized the bending stress of the spur by optimizing the fillet profile and provided size and weight reduction and lengthen the gear life[2]. R.C. Sanghvi et al. applied NSGA-II for minimizing the geometric volume of two stage helical gear train and concluded that the volume is reduced by 11.05%, while using the optimized results [3].

II. PROBLEM STATEMENT

From the literature survey, the problem statement of the current work is to carry out parametric analysis of the involute spur gear tooth and to predict the effect of these parameters on bending and contact stress.

For analysis purpose, spur gear which is used in drilling machine has been selected, whose specifications are shown in Table I.

TABLE I. SPECIFICATIONS OF PINION AND GEAR

Sl. No.	Parameters	Pinion	Gear
1	No. of teeth	27	30
2	Module	3 mm	3 mm
3	Pitch circle diameter	81 mm	90 mm
4	Addendum circle diameter	87 mm	96 mm
5	Dedendum circle diameter	73.5 mm	82.5 mm
6	Pressure angle	20	20
7	Tooth thickness	4.71 mm	4.71 mm
8	Face width	10 mm	10 mm

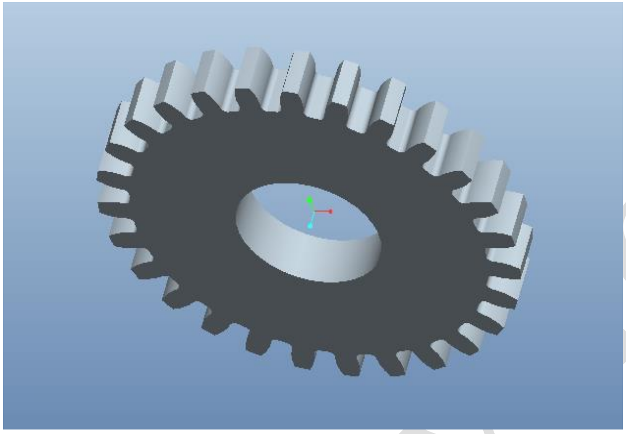

Fig. 1. Pro-E Model Pinion

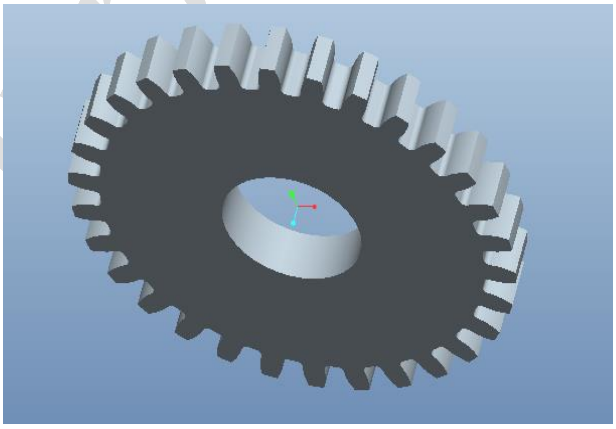

Fig. 2. Pro-E Model Gear

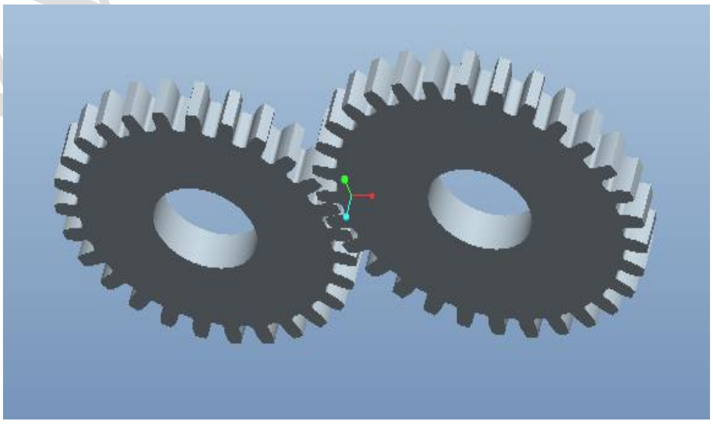

Fig. 3. Pro-E Model Mated Position

III. ANALYTICAL MODEL

A. BENDING STRESS

Stresses developed by Normal force in a photo-elastic model of gear tooth is called bending stress.

Lewis Equation:

$$\text{Bending stress} = \frac{F}{bmY} K_v$$

where,

F is Tangential Load
K_v is Velocity Factor
b is Face width
m is module
Y is Lewis Form Factor

Parameters:

The parameters selected for this study is,

- Face width
- Module

TABLE II. BENDING STRESS FOR FOUR DIFFERENT MODULE AND FACE WIDTH (LEWIS EQUATION)

Load, F	Face Width, B	Module, m	Velocity Factor, K_v	Lewis form factor, Y	Bending Stress (N/mm^2)
2000	10	3	1.7	0.343	330.41
2000	10	4	1.7	0.343	247.81
2000	10	5	1.7	0.343	198.25
2000	10	6	1.7	0.343	165.20
2000	20	3	1.7	0.343	165.20
2000	20	4	1.7	0.343	123.90
2000	20	5	1.7	0.343	99.12
2000	20	6	1.7	0.343	82.60
2000	30	3	1.7	0.343	110.13
2000	30	4	1.7	0.343	82.60
2000	30	5	1.7	0.343	66.08
2000	30	6	1.7	0.343	55.06
2000	40	3	1.7	0.343	82.60
2000	40	4	1.7	0.343	61.95
2000	40	5	1.7	0.343	49.56
2000	40	6	1.7	0.343	41.30

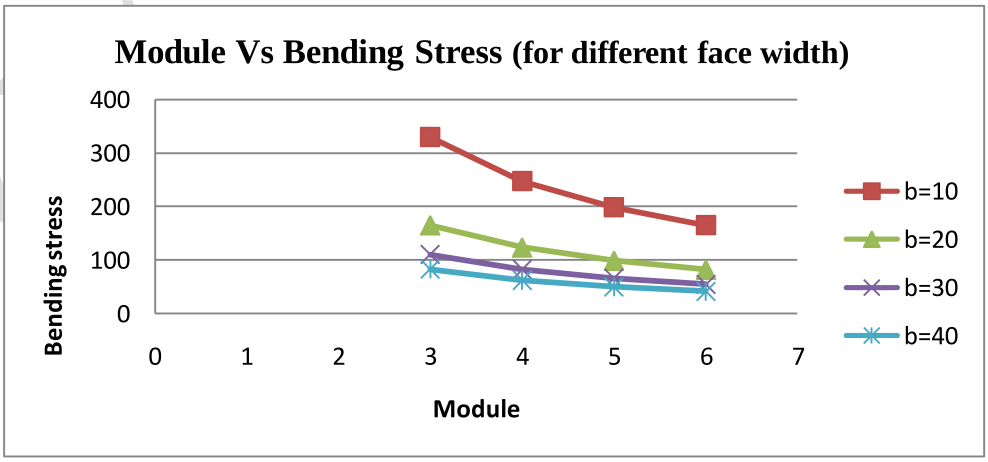

Fig. 4 Combined effect of face width and module on bending stress

Bending stresses have been analyzed for different face width and modules. The graph shows that, the bending stress is reduced while increasing the face width and module.

AGMA Equation:

Lewis equation considers only static loading and doesn't take the dynamics of meshing teeth into account. Below is the modified Lewis equation recommended by AGMA for practical gear design to account for variety of conditions.

$$\textbf{Bending stress} = \frac{F}{bmJ} K_v.K_o.K_m$$

where,

F is Tangential Load
K_v is Velocity factor
K_o is Overload factor
K_m is Load distribution factor
b is Face width
m is Module
J is Spur gear geometry factor

TABLE III. BENDING STRESS FOR FOUR DIFFERENT MODULE AND FACE WIDTH(AGMA EQUATION)

Load, F	Face Width, B	Module, m	Spur gear Geometry Factor, J	Velocity Factor, K_v	Overload Factor, K_o	Load Distribution Factor, K_m	Bending Stress (N/mm^2)
2000	10	3	0.37878	1.7	1	1.6	478.72
2000	10	4	0.37878	1.7	1	1.6	359.047
2000	10	5	0.37878	1.7	1	1.6	287.23
2000	10	6	0.37878	1.7	1	1.6	239.36
2000	20	3	0.37878	1.7	1	1.6	239.36
2000	20	4	0.37878	1.7	1	1.6	179.52
2000	20	5	0.37878	1.7	1	1.6	143.61
2000	20	6	0.37878	1.7	1	1.6	119.68
2000	30	3	0.37878	1.7	1	1.6	159.57
2000	30	4	0.37878	1.7	1	1.6	119.68
2000	30	5	0.37878	1.7	1	1.6	95.74
2000	30	6	0.37878	1.7	1	1.6	79.78
2000	40	3	0.37878	1.7	1	1.6	119.68
2000	40	4	0.37878	1.7	1	1.6	89.76
2000	40	5	0.37878	1.7	1	1.6	71.80
2000	40	6	0.37878	1.7	1	1.6	59.84

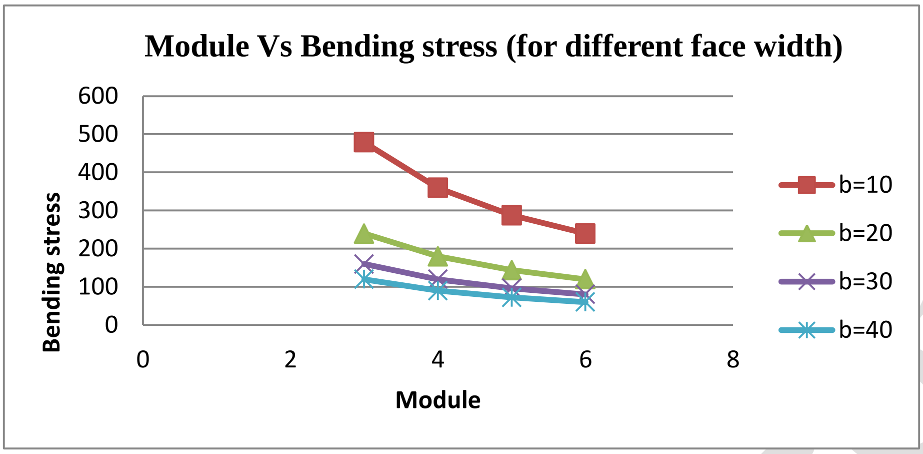

Fig. 5 Combined effect of face width and module on bending stress

According to AGMA equation also, it is cleared that while increasing the face width and module, the bending stress is reduced.

B. CONTACT STRESS

When two solid deformable bodies are comes in contact with each other, lot of stresses will be produced in the contact region. This stress is called contact stress.

Hertz equation:

$$\text{Contact stress} = Y_m Y_p \sqrt{\frac{F}{bd_1}\frac{u+1}{u}}$$

Where,

Y_m is material coefficient
Y_p is pitch point coefficient
F is tangential load
u is gear ratio
b is face width
d_1 is pitch circle diameter of the pinion

TABLE IV. CONTACT STRESS FOR FOUR DIFFERENT MODULE AND FACE WIDTH (HERTZ EQUATION)

Load, F	Face Width, b	Module,m	No. of teeth	Diameter of the pinion, d_1	gear ratio, u	material coefficient, Y_m	Pitch point coeffcient, Y_p	Contact stress (N/mm^2)
2000	10	3	27	81	1.11	264.5	1.76	1008.53
2000	10	4	27	108	1.11	264.5	1.76	873.41
2000	10	5	27	135	1.11	264.5	1.76	781.20
2000	10	6	27	162	1.11	264.5	1.76	713.14
2000	20	3	27	81	1.11	264.5	1.76	713.14
2000	20	4	27	108	1.11	264.5	1.76	617.59
2000	20	5	27	135	1.11	264.5	1.76	552.39
2000	20	6	27	162	1.11	264.5	1.76	504.26
2000	30	3	27	81	1.11	264.5	1.76	582.27
2000	30	4	27	108	1.11	264.5	1.76	504.26
2000	30	5	27	135	1.11	264.5	1.76	451.03
2000	30	6	27	162	1.11	264.5	1.76	411.73
2000	40	3	27	81	1.11	264.5	1.76	504.26
2000	40	4	27	108	1.11	264.5	1.76	436.70

2000	40	5	27	135	1.11	264.5	1.76	390.60
2000	40	6	27	162	1.11	264.5	1.76	356.57

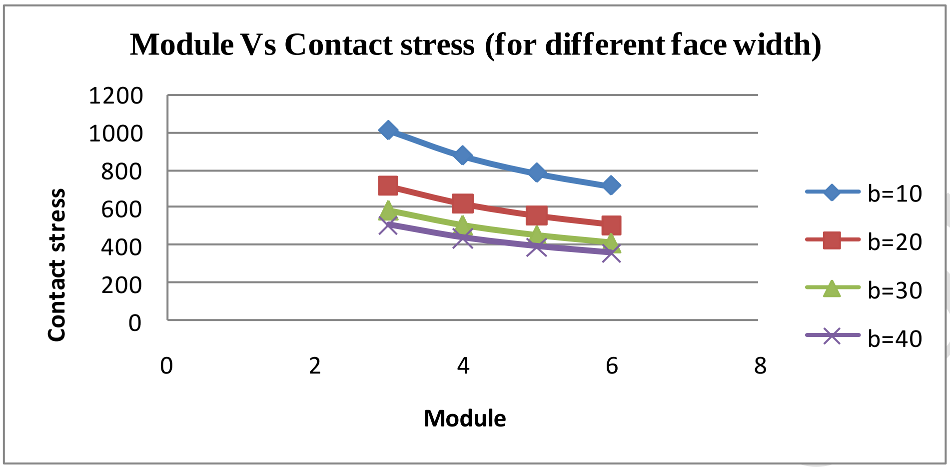

Fig. 6 Combined effect of module and face width on contact stress

Contact stresses have been analyzed for different face width and modules. The graph shows that, the contact stress is reduced while increasing the face width and module.

AGMA Equation:

$$\textbf{Contact stress} \quad \sigma_H = C_p \sqrt{\frac{F}{b d_1 I}}\, K_v.K_o.K_m$$

where,

C_p is Elastic coefficient
F is Tangential load
K_v is Velocity factor
K_o is Overload factor
K_m is Load distribution factor
b is face width
d_1 is pitch circle diameter of the pinion
I is spur gear geometry factor

TABLE V. CONTACT STRESS FOR FOUR DIFFERENT MODULE AND FACE WIDTH (AGMA EQUATION)

Material coefficient C_p	Load, F	Face Width, B	Diameter of the pinion d_1	Geometry factor, I	Velocity Factor, K_v	Overload Factor, K_o	Load Distribution Factor, K_m	Contact stress (N/mm²)
191	2000	10	81	0.085	1.7	1	1.6	1697.77
191	2000	10	108	0.085	1.7	1	1.6	1470.31
191	2000	10	135	0.085	1.7	1	1.6	1315.09
191	2000	10	162	0.085	1.7	1	1.6	1200.51
191	2000	20	81	0.085	1.7	1	1.6	1200.51
191	2000	20	108	0.085	1.7	1	1.6	1039.67
191	2000	20	135	0.085	1.7	1	1.6	929.91
191	2000	20	162	0.085	1.7	1	1.6	848.88
191	2000	30	81	0.085	1.7	1	1.6	980.21
191	2000	30	108	0.085	1.7	1	1.6	848.88

191	2000	30	135	0.085	1.7	1	1.6	759.26
191	2000	30	162	0.085	1.7	1	1.6	693.11
191	2000	40	81	0.085	1.7	1	1.6	848.88
191	2000	40	108	0.085	1.7	1	1.6	735.15
191	2000	40	135	0.085	1.7	1	1.6	657.54
191	2000	40	162	0.085	1.7	1	1.6	600.25

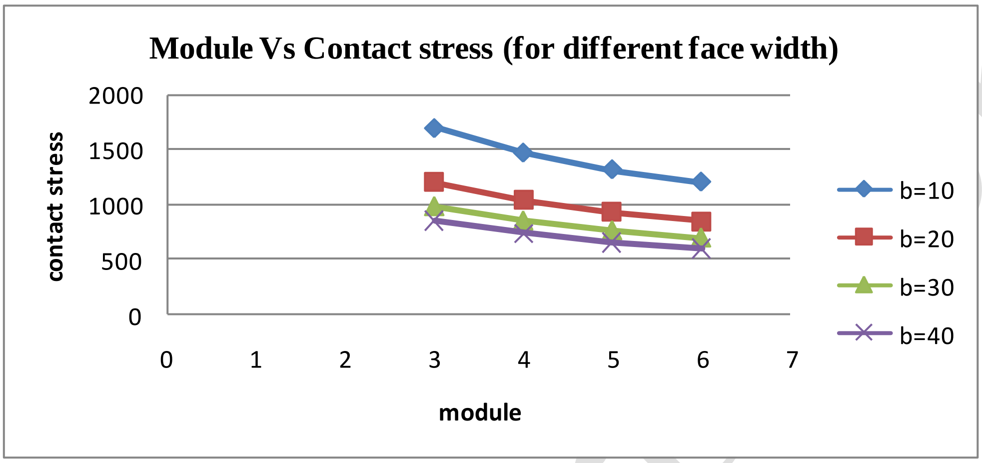

Fig. 7 Combined effect of module and face width on contact stress

According to AGMA equation also, it is cleared that while increasing the face width and module, the contact stress is reduced.

V. NUMERICAL MODEL

Modeling of the spur gears is based on the design parameters. The material properties and gear parts assembly are also assigned in this stage. This is followed by the FEA pre-processing stage where the mesh element, the load, and constraint are set on the involute spur gear model. In order to determine the bending stress and contact stress accurately, different mesh element settings and FEA solver are required in separate setup. In the FEA post-processing stage, the FEA solution for the gear is completed. The maximum von Mises stress is obtained from the contour plot of the gear. Thus, the bending and contact stress analysis can be performed by correlating the change in applied load and gear parameters.

Among the various FEM packages, in this work ANSYS is used to perform the analysis. The following steps are used in the solution procedure using ANSYS:

- The geometry of the gear to be analyzed is imported
- The element type and materials properties such as Young's modulus and Poisson's ratio are specified.
- Meshing the three-dimensional gear model.
- The boundary conditions and external loads are applied.
- The solution is generated based on the previous input parameters.
- Finally, the solution is displayed.

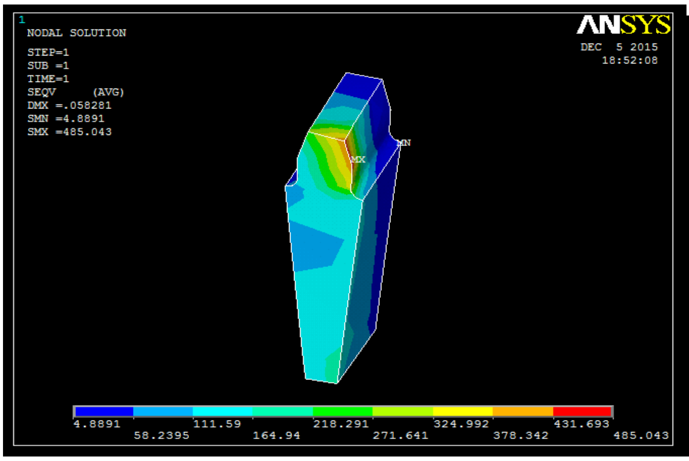

Fig. 8 Bending stress when face width is 10mm

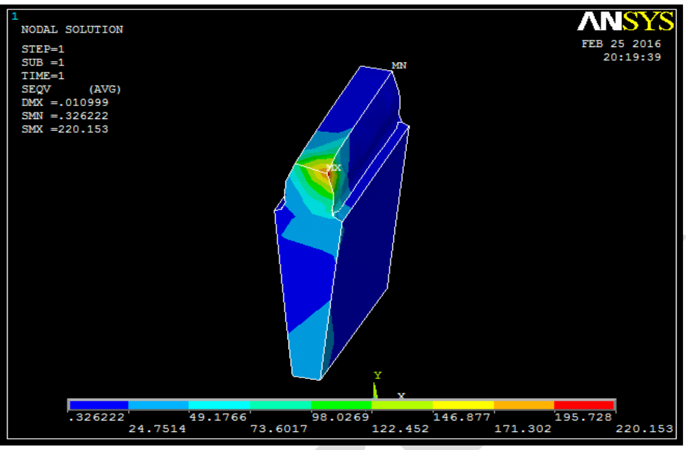

Fig. 9 Bending stress when face width is 20mm

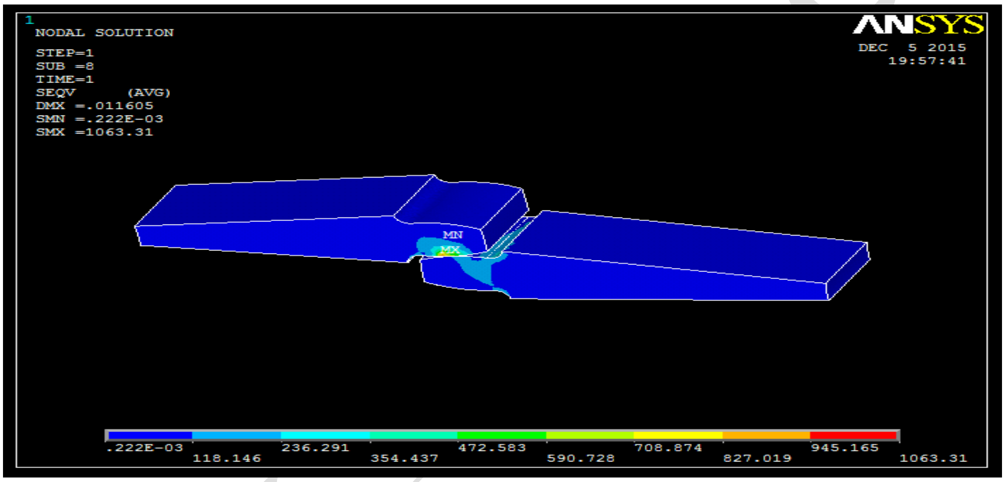

Fig. 10 Contact stress when face width is 10mm

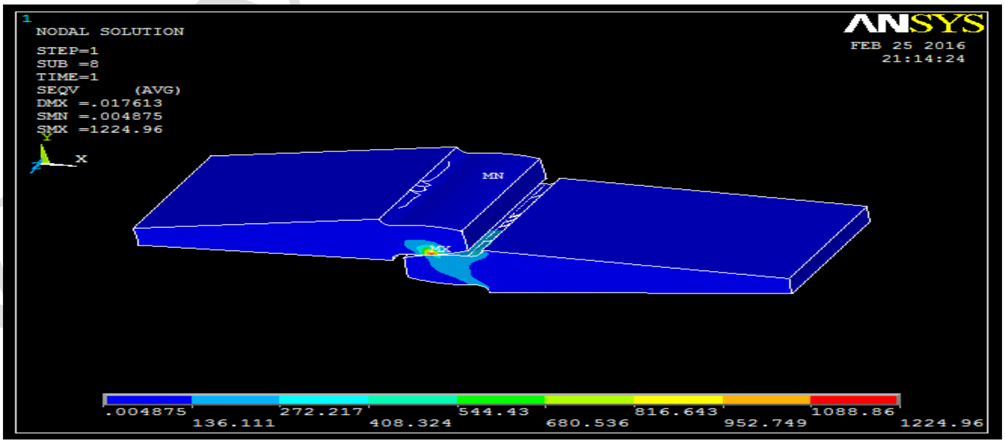

Fig. 11. Contact stress when face width is 20mm

VI. RESULTS AND DISCUSSIONS

Gear failure is the major concern for power transmission. Exceeding the limiting values bending and contact stresses will result into pitting which leads to gear failure. Lot of works has been undertaken by many researchers to avoid these failures. In the present work, an attempt has been made to optimize the bending and contact stresses by modifying the parameters of the spur gear tooth. The effects of changing these parameters are discussed here.

Both the analytical and numerical results reveals that, the gear parameters having significant effect on the bending and contact stress. In this work, the effect of module and face width is considered for analysis. From the analysis, it is clearly seen that, when the module and face width of the gear tooth are increased, the bending and contact stress will be reduced by a certain amount. This stress reduction will lengthen the gear life.

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