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THE ANALYTICAL STUDY OF MESHING OF DOUBLE HELICAL GEAR

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ABSTRACT

Gears are the most important components of a Power Transmission System. The recent advancement and trend in technology requires gears operating at high load capacities and speeds with reduced space requirement. A gear normally fails when the tooth stress exceeds the allowable limit. So, it is essential to keep the maximum stress in a gear tooth under a specified limit for a given loading conditions. Analysis of gears is carried out to find out the possible laudation of failure and taking a corrective measure for the same. A gear failure is expensive not in terms of the cost of replacement but due to the downtime and loss of production associated with the failure. Gear design should consider all the possible loads that it can experience during its life. Surface fatigue and fracture is one of the major problem in gear and gearbox design. In order to find out and analyze the exact location of failure and the stress value at which gear fails, stress analysis of the double helical gear has been done in ANSYS. Load conditions to be experience by the gear has been applied along with other constraints.

1. Introduction

1.1 New Method of Gear Design

The advancement in Computer technology has made Finite Element Analysis a routine tool in design. This has lead Computer-Aided Design (CAD) using solid-body modelling. Some benefits of CAD are increased design productivity, shorter design lead times, more logical design process & analysis, fewer design errors, greater accuracy in calculations, standardization of design and improved procedures for reviews & revisions.

1.2 Finite Element Analysis (FEA)

It is widely used method of accessing a product performance. Using FEA, the requirement of prototype testing can be greatly reduced thereby shortening the development cycle time. This further facilitates quicker product launch. FEA consists of a computer model of the design on which a specified load is applied and analysed for the required results. FEA gives an insight into the design like probable failure areas or areas with excessive margin for optimization, which would otherwise will not be possible.

2. Literature Review

Ashish V Kadu and Sanjay S Deshmukh (2015)experimented that the transmission error in the actual gear system which arises because of a irregular tool geometry or imperfect geometry or imperfect mounting the characteristic of the involutes spur gear are analyzed by using finite element method. The contact stresses are examined by using 2D FEM Model. And the bending stresses in the tooth root are examined by using 3D FEM Model. The conventional method of calculating gear contact stress using Hertz's theory for verification by 2D FEM analyser using ANSYS, the stiffness relationship between two contact area is usually established using a spring place between source and target surfaces for the contact generation between two gears. The stresses are compared with theoretical result. The static transmission error and analysis of load sharing method using displacement vector and the effect of this error in the actual transmission power of mesh gear.

Deva Ganesh et al. (2015)studied that the meshing between two gears contact stresses are evolved, which are determined by using analyzing software called ANSYS. Finding stresses has become most popular in research on gears to minimize the vibrations, bending stresses and also reducing the mass percentage in gears. These stresses are used to find the optimum design in the gears which reduces the chances of failure. The model is generated by using Catia and ANSYS is used for numerical analysis. The analytical study is based on Hertz,s equation. Study is conducted by varying the geometrical profile of the teeth and to find the change in contact stresses between gears. It is therefore observed that more contact stresses are obtained in modified gears. Both the results calculated using ANSYS and compared according to the given moment of inertia.

Sarfraz Ali N. Quadri and Dhananjay R. Dolas (2015)experimented an attempt to summarize about stresses developed in a mating spur gear which has involute teeth. A pair of spur gears are taken from a lathe gear box and progressed onward to calculate stresses. Conventionally the analysis is carried out analytically using Lewis formulae and then Finite Element Analysis is used for the same. Some stress relieving features have been incorporated in the teeth to know their effect on the stress concentrations. A finite element model of teeth is considered for analysis and geometrical features of various sizes are introduced at various locations and their effect is analyzed.

Mohammad Jebran Khan et al. (2015)experimented that the gears or toothed wheels form a positive drive for power transmission system in precision machines wherein a definite velocity ratio is needed. Despite having high cost, complicated manufacturing, need of precise alignment of shafts and lubrication, the gear drives are preferred over other power transmission drives. One of the important reasons of preference being that of efficiency which is very high in gear drives, even up to 99 percent in case of spur gears. Spur gears are the simplest of the gear drives having teeth cut parallel to the axis of the shaft. The contact stress analysis of Stainless Steel spur gears by theoretical method using Hertz equations and by Finite Element Analysis using FEA software ANSYS workbench .

Putti Srinivasa Rao et al. (2015)studied that the contact stress in the mating gears is the key parameter in gear design. Deformation of the gear is also another key parameter which is to be considered. Gears generally fail when the working stress exceeds the maximum stress. The complex design problem of spur gear which requires fine software skill for modeling and also for analyzing. The project aims at the minimization of both contact stress as well as deformation to arrive at the best possible combination of driver and driven gear. In this process of spur gears mating, 3 different materials were selected and the software programme was performed for 9 different combinations to get the best result possible. The results of the two dimensional FEM analysis from ANSYS are presented.

Sabah Khan(2015)studied that the gears are one of the most critical components in mechanical power transmission systems. Transmission error is considered to be one of the main contributors to noise and vibration in a gearset. Transmission error measurement has become popular as an area of research [14]. To estimate transmission error in a gear system, the characteristics of in volute gears were analyzed using ANSYS. The contact stresses were examined using 2-D FEM models. The bending stresses in the tooth root were examined using a 3-D FEM model.

3. Problem Definition

3.1 Definition of the Problem

Two stages of the Gear Box were analyzed for the given load conditions. The maximum stress generated under the load needs to be evaluated. Further, geometry of the gear was modified to reduce the stress concentration values for the same loading conditions.

3.2 Methodology Used

The Gear profile calculations of Double Helical gear were performed using reference from handbook data. The gear and the assembly of the Gears was then generated in Solid Works [5]. The 3D model generated in Solid Works was then transferred to Ansys Workbench. Material definition was given using Engineering Data. Mesh was generated using Ansys Mesher. Boundary conditions were supplied and stress analysis was performed.

3.3 Constraints

The following inputs were used:

1. **Material: -** The gears were made of forged steel having specification EN24 as it is most economical.
2. **Type of Gear: -** Type of gear is double helical gear.
3. **Helix Angle and Pressure Angle: -** The helix angle and pressure angle are 15 and 20 respectively.
4. **Pitch Circle Diameter: -** The pitch circle diameters of the gears mounted in the gearbox were fixed.
5. **Number of Gears: -** The number of gears mounted in the gearbox was fixed.
6. **Gear Module and PCD: -**Dimensions of the double helical gear to be used are as given in the table below :

Diameter (mm)	Number of teeth	Module	Centre Distance (mm)	Torque to be Transmitted (Nm)
204	18	10	420.0	156,000
636	60	10		
234	17	12	454.5	50,900
675	52	12		

Table 1.1: Dimensions of the Double Helical Gear

Shaft diameter is 75mm. Gears of same module will mesh together to form a stage of reduction. Tooth width was taken as 100mm and total width of the Gear was taken as 229mm. Input Power and rpm at Ist stage input is 800kW and 500 rpm respectively.All the above inputs were kept fixed and cannot be changed.

3.4 Basis of Modifications of the gearbox

Design modifications in the gears can be done considering factors such as strength and stiffness. But the gear module, width, material and loading conditions cannot be changed [3].

3.5 Procedure for the Analysis

The following steps were used for solving of the problem:-

Step I: Modelling of the gears as per the dimensions and making assembly of two gears to make one reduction stage using Solid Works.

Step II: The 3D model was imported in Ansys Workbench.

Step III: Material Property of EN24 was defined in Engineering Data in Ansys Workbench.

Step IV: The material EN24 was assigned to the Gears in Mechanical interface.

Step V: Frictional contact was specified between the faces of the gears mating each other.

Step VI: Frictional contact was specified between the faces of the gears mating each other.

Step VII: Mesh was generated for the gears using an element size of 4mm at the contact face.

Step VIII: The shaft bore of small gear was fixed and moment was applied at shaft bore of large gear. A moment of 50,900 Nm &156,000 Nm was applied at shaft bore of Gear Dia 636 &Dia 675 respectively.

Step IX: Analysis solution was performed and stress values were checked for the gears.

Step IX: Analysis solution was performed and stress values were checked for the gears. Von Misses stress was used to compare the results.

Step X: The gear geometry was modified in Solid Works and all the steps from Step I to Step IX were performed again to get the results for the revised geometry. The results of original gear and with modified geometry were compared.

3.6 Stress Analysis of the Gears

3D Model of the Gears used for Analysis

The 3D model and the drawing of the Gears and their assembly is in Fig. 3.1 to Fig. 3.4

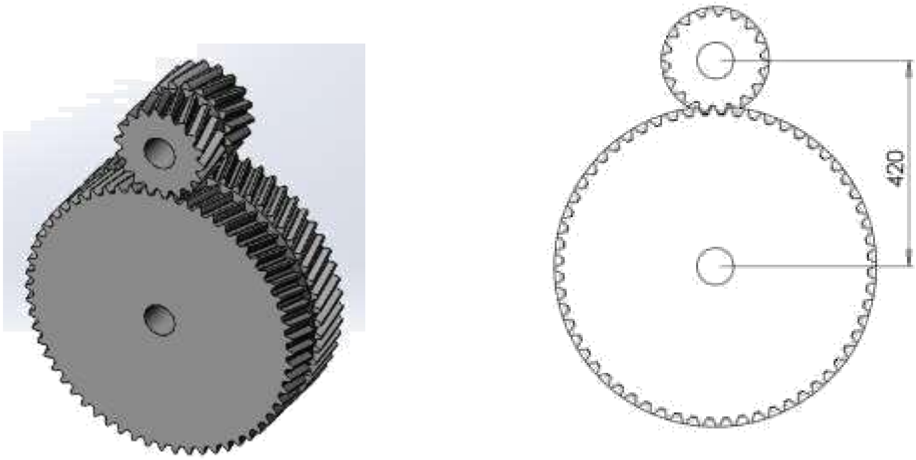

Fig.3.1: 3D Model & 2D Drawing of Gear Assembly of Module 10 Gear Dia 204 &Dia 636

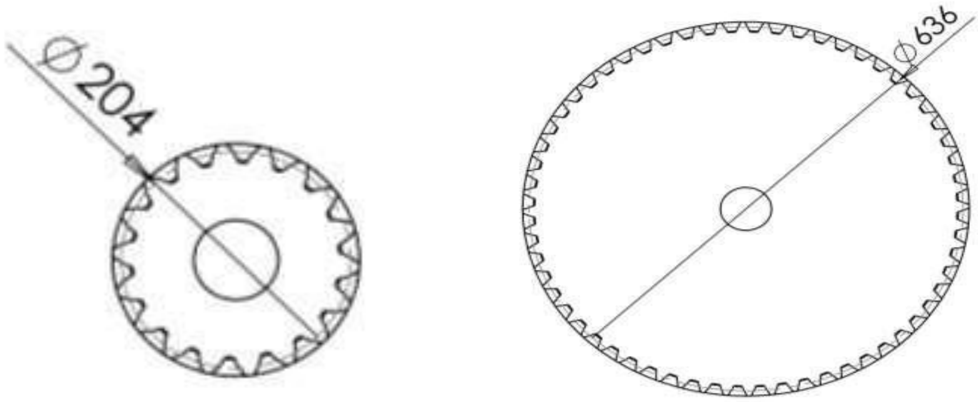

Fig. 3.2: Module 10 Gear Dia 204 &Dia 636

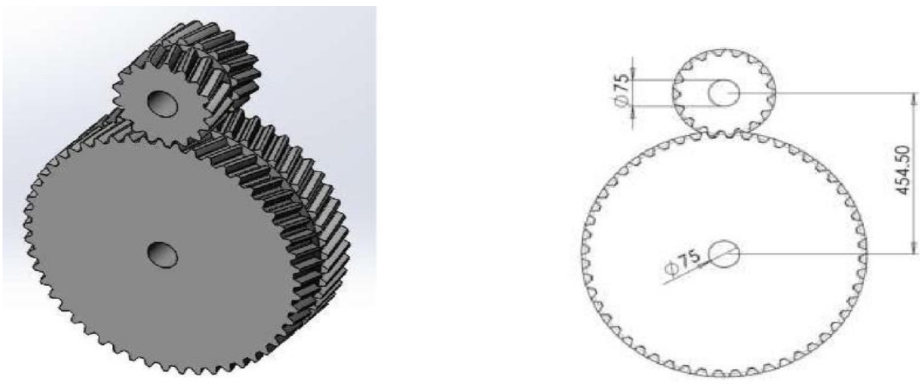

Fig. 3.3: Model & 2D Drawing of Gear Assembly of Module 12 Gear Dia 234 &Dia 675

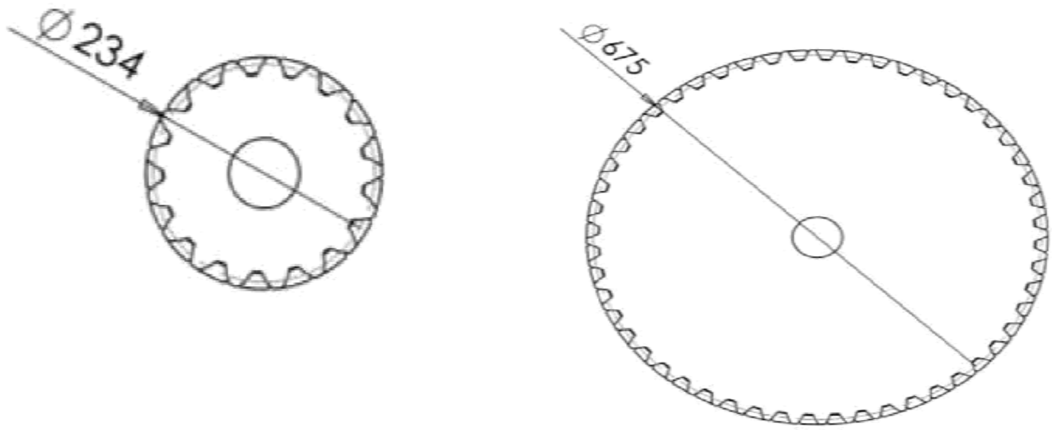

Fig. 3.4: Module 12 Gear Dia 234 &Dia 675

The following dimensions were maintained for the Gear Model:

Diameter (mm)	Tip Diameter (mm)	Root Diameter (mm)	Tooth Depth (mm)
204	218.8	183.8	17.5
636	635.6	618.6	
234	249.6	212.0	18.8
675	693.8	656.3	

Table 1.2: Dimensions of the Gear Model

3.7 Detailed Steps for performing the Analysis using ANSYS Workbench

ANSYS is a computer aided engineering tool that allows us to simulate the physical behavior of a part or assembly, to understand and improve mechanical performance of a design. It enables us to analyze and optimize the designs for structural, thermal and dynamic requirements. A Static Structural Analysis was performed for the gears. Pre-processing- It involves all the steps related to setting up of the problem [2]. The following steps were performed for the said analysis:
a) Material Property of EN24 to be used for the gears was defined in Engineering Data.

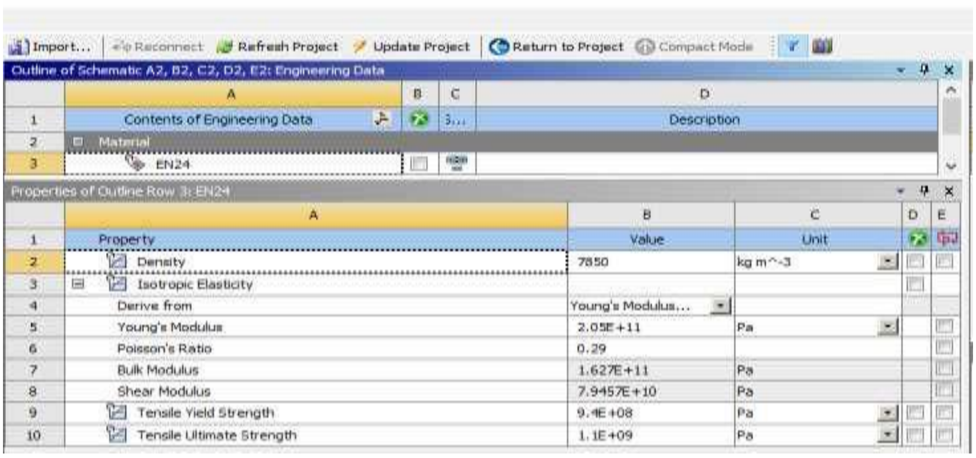

Fig. 3.5: Material Property in Engineering Data

a. The gear assembly was imported in Design Modeller.
b. Material EN24 was assigned to the Gears in Mechanical interface.
c. Frictional contact was defined for the touching faces of the gears
d. Contact size element was specified as 4mm. And the mesh was generated giving the total no. of elements as 50767 for M12 Gears & 81289 for M10 Gears.
e. Boundary conditions were then applied to the gears. Cylindrical support was specified for the hole of gears. A moment of 156,000 Nm & 50,900 was applied at the hole of Big gear Dia 675mm and Dia 636 mm respectively

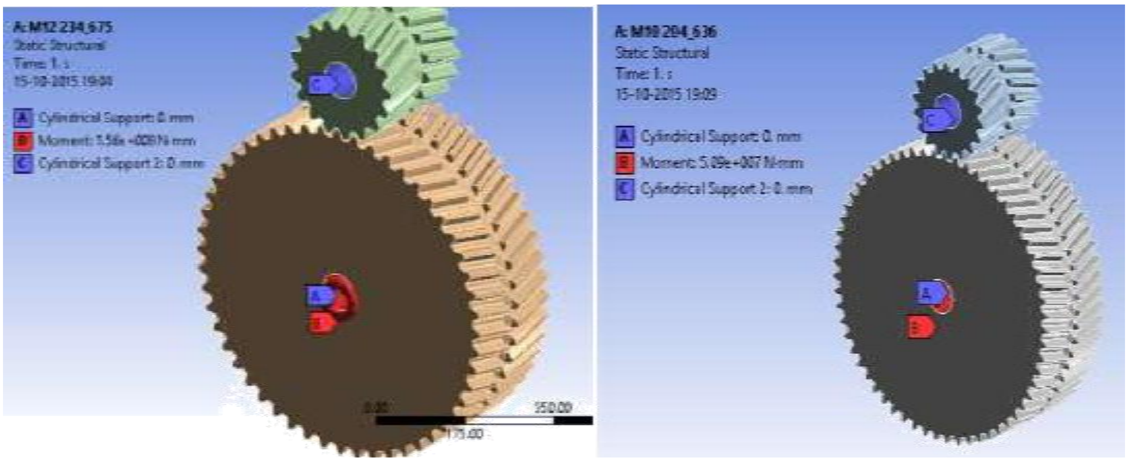

Fig. 3.6: Boundary Conditions for M10 and M12 Gears

f. Solution was then run and the results were plotted for both the Gears.

4. Results of Original Gear

4.1 Results for M12 Gear Assembly

The stress contours for M12 gears has been shown in the figures below. Equivalent von-Mises stress was plotted. The maximum stress value is 1168.2 MPa which occurs on Gear Dia 675. The stress value on Gear Dia 234 comes to be 1011.0 MPa.

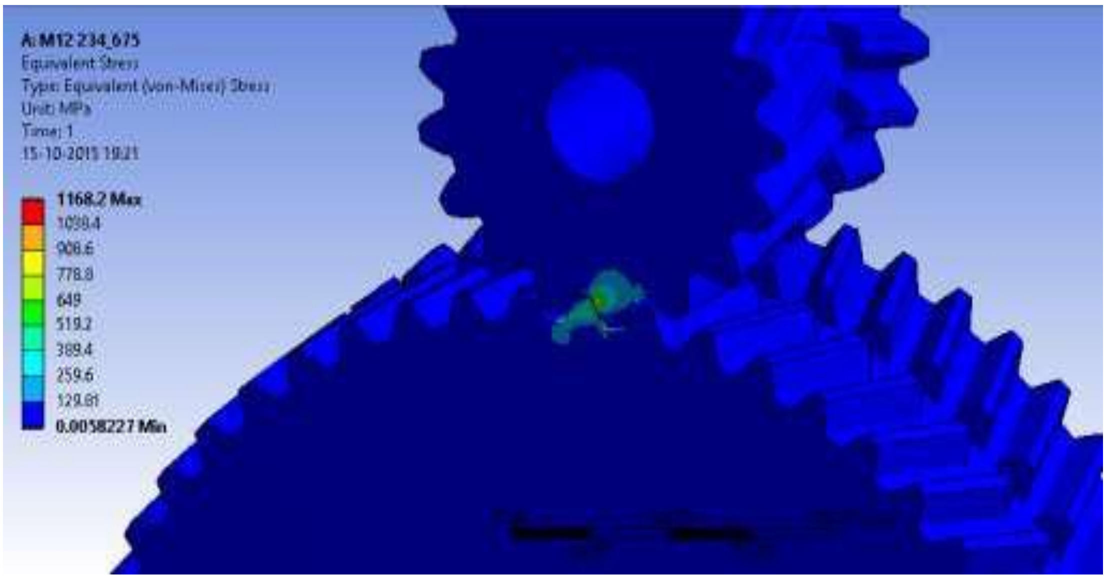

Fig. 4.1: Equivalent Stress Contours for M12 Gear Assembly

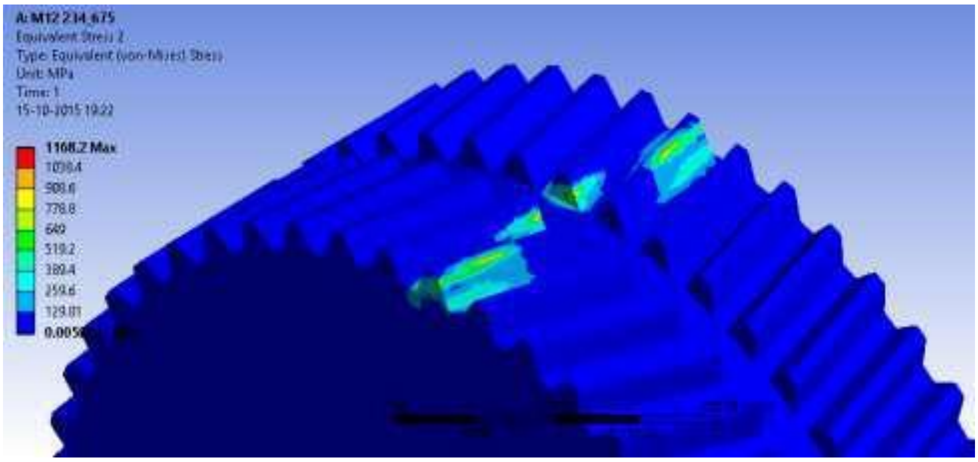

Fig. 4.2: Equivalent Stress Contours for M12 Gear Dia 675

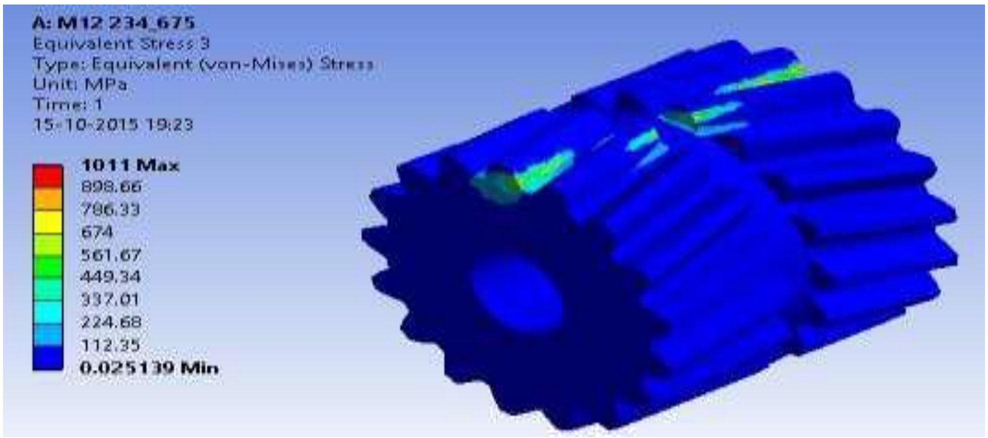

Fig. 4.3: Equivalent Stress Contours for M12 Gear Dia 234

Clearly the stress is above the allowable Yield value of 940 MPa for EN24 material. So, the geometry of the gears weremodified and the analysis was again performed to get the stress values.

4.2. Results for M10 Gear Assembly

The stress contours for M10 gears has been shown in the figures below. Equivalent von-Mises stress was plotted.The maximum stress value is 421.05 MPa which occurs on Gear Dia 636. The stress value on Gear Dia 234 comes to be 363.13MPa.

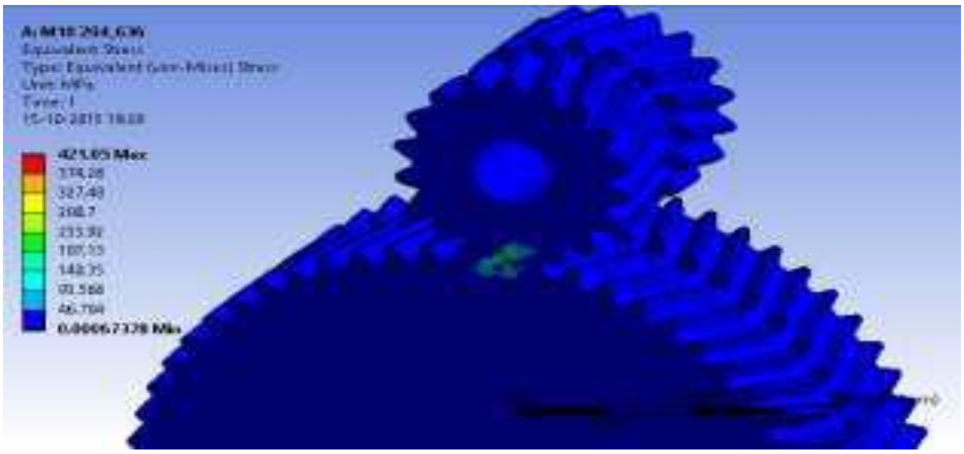

Fig. 4.4: Equivalent Stress Contours for M10 Gear Assembly

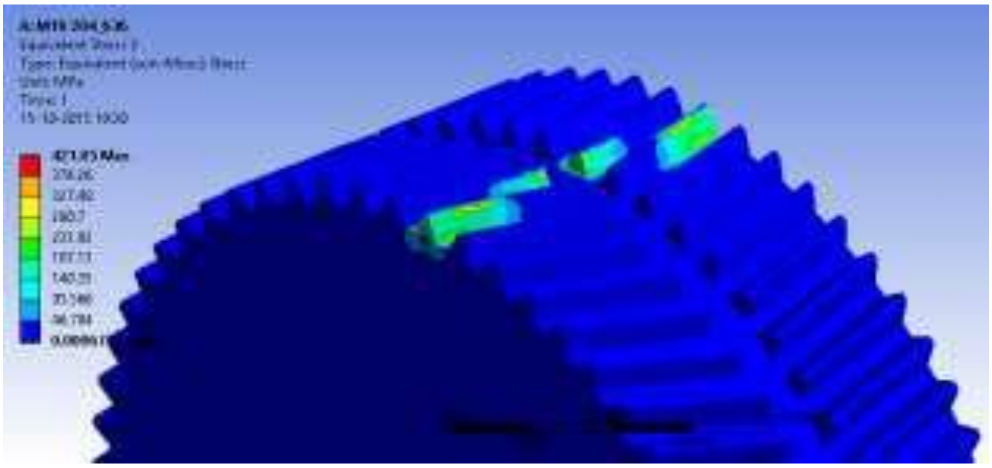

Fig. 4.5: Equivalent Stress Contours for M10 Gear Dia 636

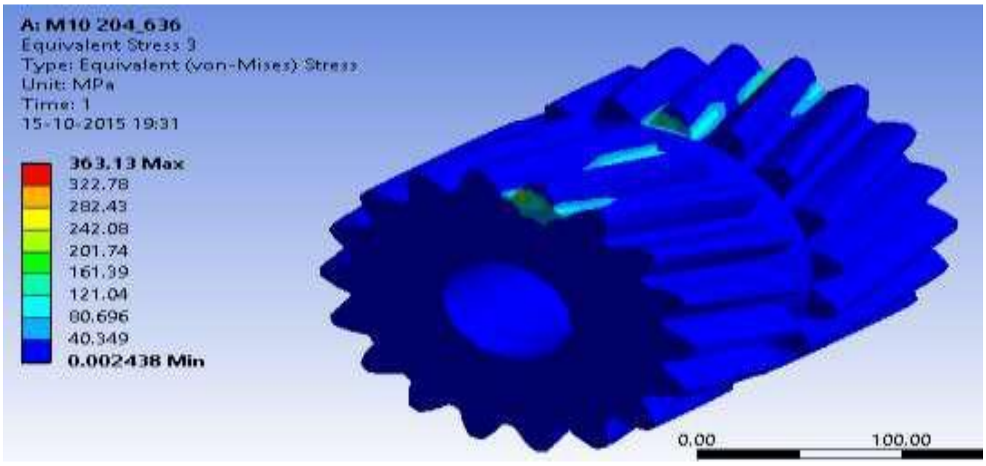

Fig. 4.6: Equivalent Stress Contours for M10 Gear Dia 204

The stresses in M10 gear assembly was on safe side as expected as the Torque to be transmitted is low. But, the same modifications were applied to M10 gears as was applied to M12 gears to have a clear understanding and comparison of the effect of modifications on the stress in the gears.

5. Modification in the Gear Geometry

From the analysis results, it is clear that stress value in M12 gear assembly exceeds the Yield Strength value of EN24 material. So, the following modifications in the gear geometry was done and analysis was performed for each of the modified geometry:

5.1. Hole of 4mm diameter at the root of the Gear

5.2. Edges of the gear teeth tapered by an angle of 20^0

5.3. Groove at central portion of Gear Dia 636 &Dia 675

Hole of 4mm diameter at the root of the Gear

A hole of 4mm diameter was made at the root of all the Gears. The drawing of the modified gears and 3D assembly model of the gears is shown in the following figures.

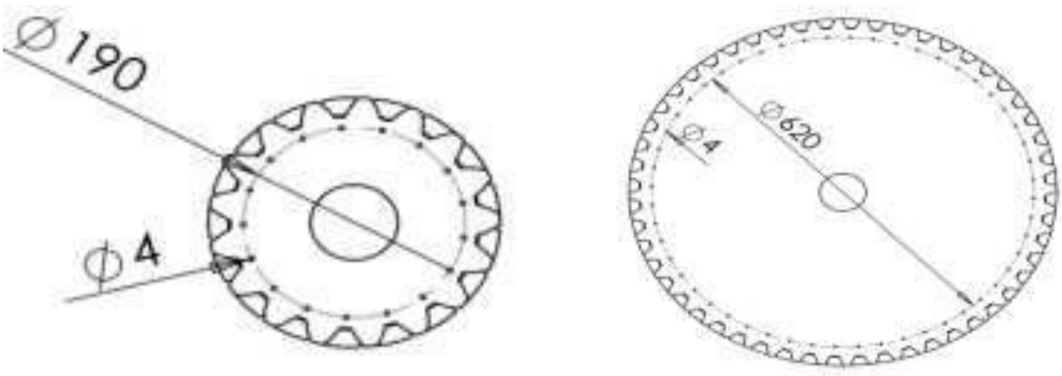

Fig. 5.1: 2D Drawing of Gear with Hole Module 12 Gear Dia 234 &Dia 675

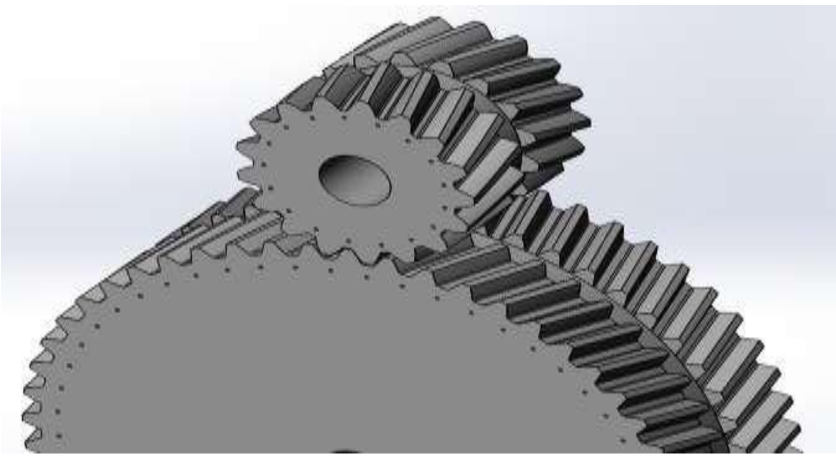

Fig. 5.2: 3D Model of Gear Assembly with hole of Module 12 Gear Dia 234 &Dia 675

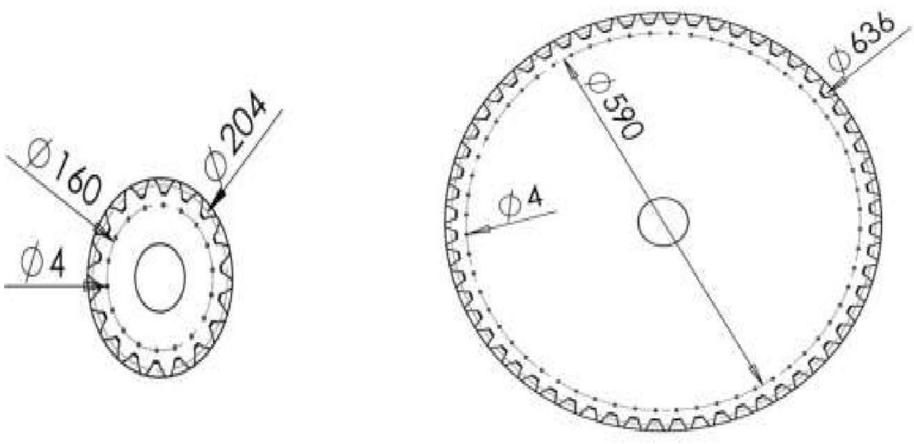

Fig. 5.3: 2D Drawing of Gear with Hole Module 10 Gear Dia 204 &Dia 636

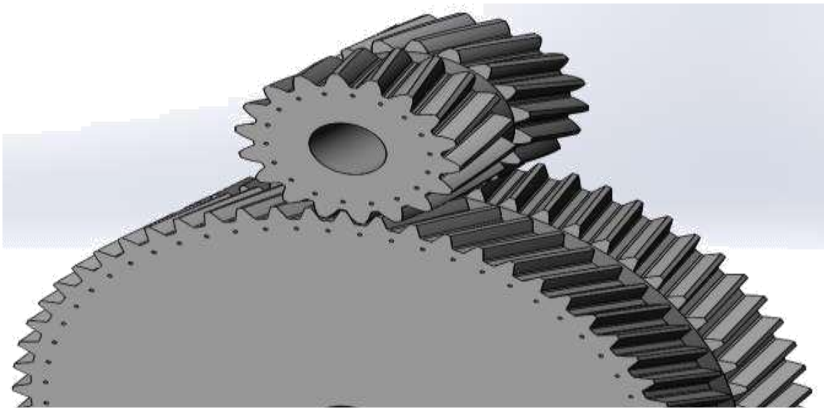

Fig. 5.4: 3D Model of Gear Assembly with hole of Module 10 Gear Dia 204 &Dia 636
Edges of the gear teeth tapered by an angle of 20^0

The edge of the gear teeth were made taper by making a cut of 20^0 on both sides and for all the gears. The detail of the cut is shown in the figures below. The analysis was performed again on the modified geometry. The results will be presented in the later chapters.

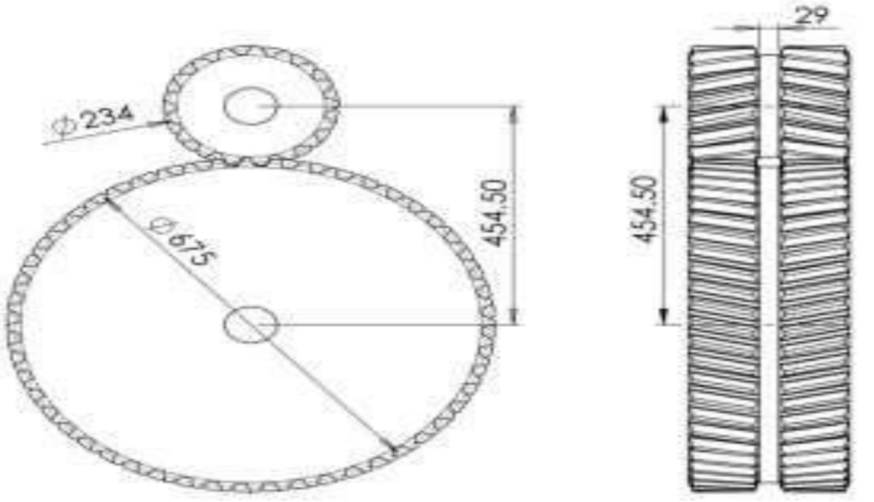

Fig. 5.5: 2D Drawing of Gear with Taper Edges Module 12 Gear Dia 234 &Dia 675

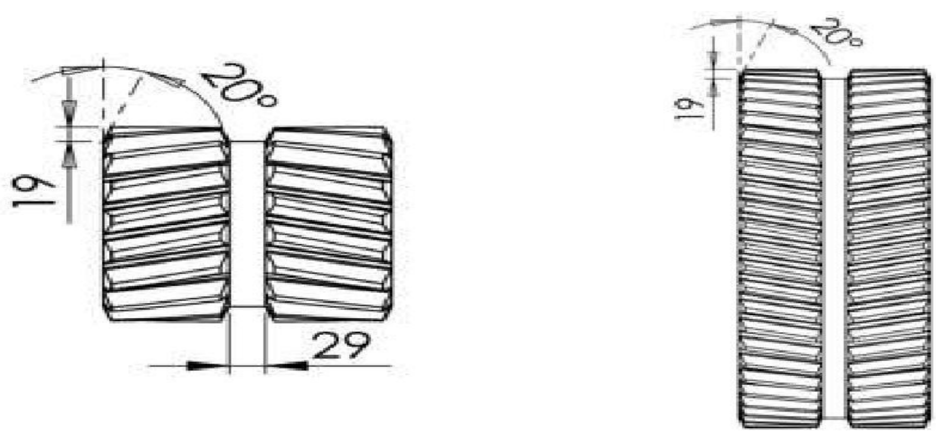

Fig. 5.6: 2D Detail Drawing of Gear with Taper Edges Module 12 Gear Dia 234 &Dia 675

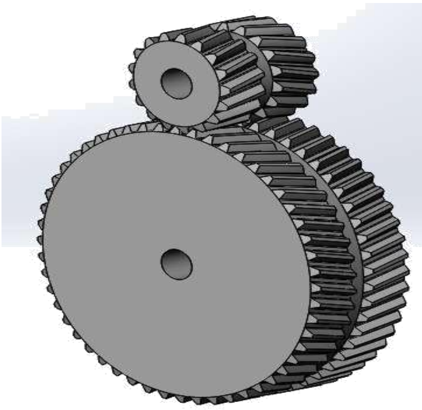

Fig. 5.7: 3D Model of Gear with Taper Edges Module 12 Gear Dia 234 & Dia 675&Dia 636

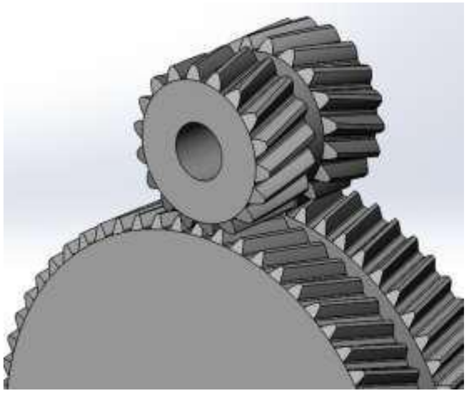

Fig. 5.8: 3D Model of Gear with Taper Edges Module 10 Gear Dia 204

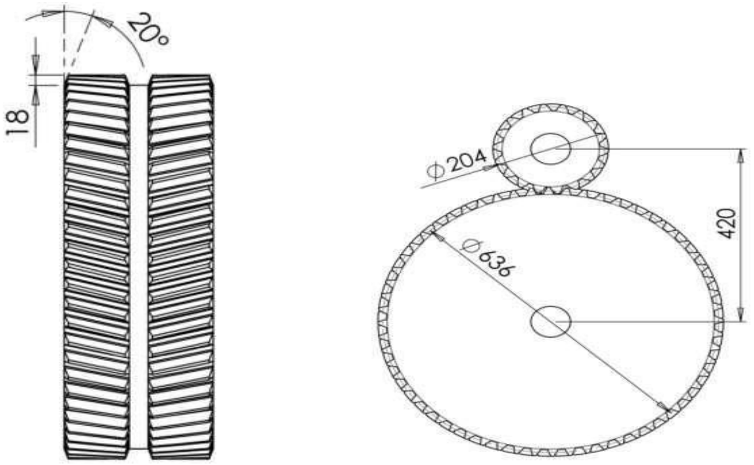

Fig. 5.9 2D drawing of gear with taper module 10gear dia. 204 and dia.636

Groove at the Gear Dia 636 &Dia 675 along with Taper at Edges

The gears were also checked for the effect of making a Groove at the centre portion of the larger gear. The shape and the details of the groove are given in the figures below. No groove was provided in the smaller gears of the assembly. The groove was provided in addition to the taper at edges of the gears.

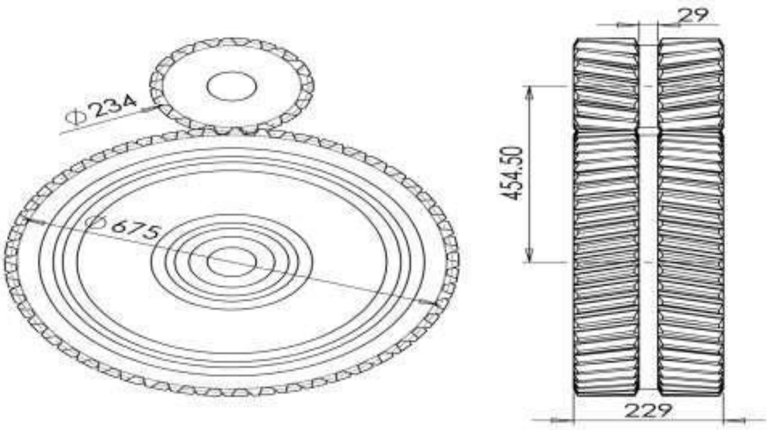

Fig. 5.10: 2D Drawing of Gear with Groove and Taper Edges Module 12 Gear Dia 234 &Dia 675

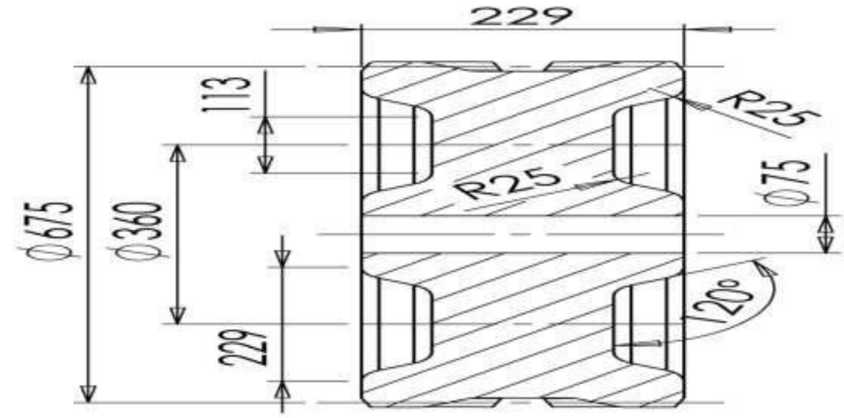

Fig. 5.11: Cut-Section of Gear with Groove and Taper Edges Module 12 Dia 675

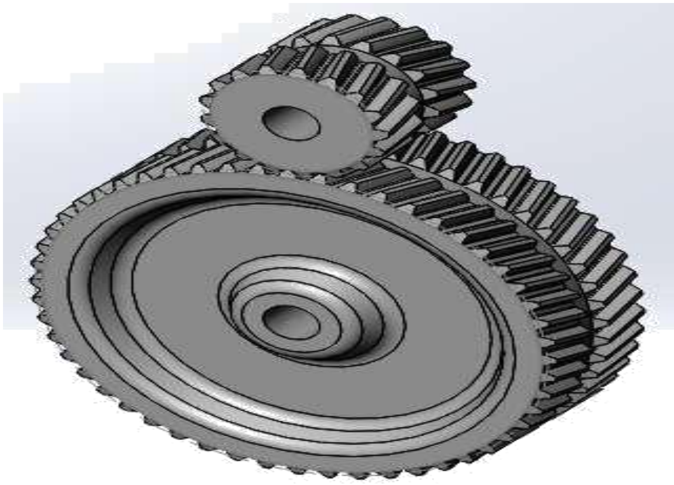

Fig. 5.12: 3D Model Assembly of Gear with Groove and Taper Edges Module 12 Gear Dia 675

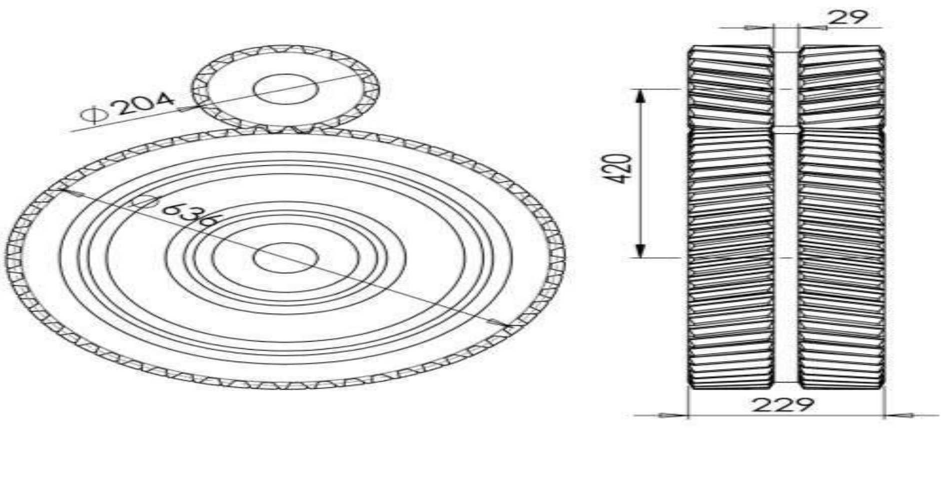

Fig. 5.13: 2D Drawing of Gear with Groove and Taper Edges Module 10 Gear Dia 204 &Dia 636

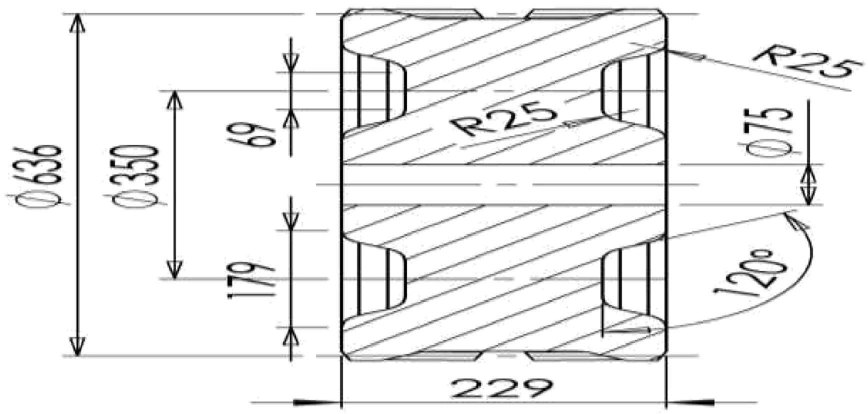

Fig. 5.14: Cut-Section of Gear with Groove and Taper Edges Module 10 Dia 636

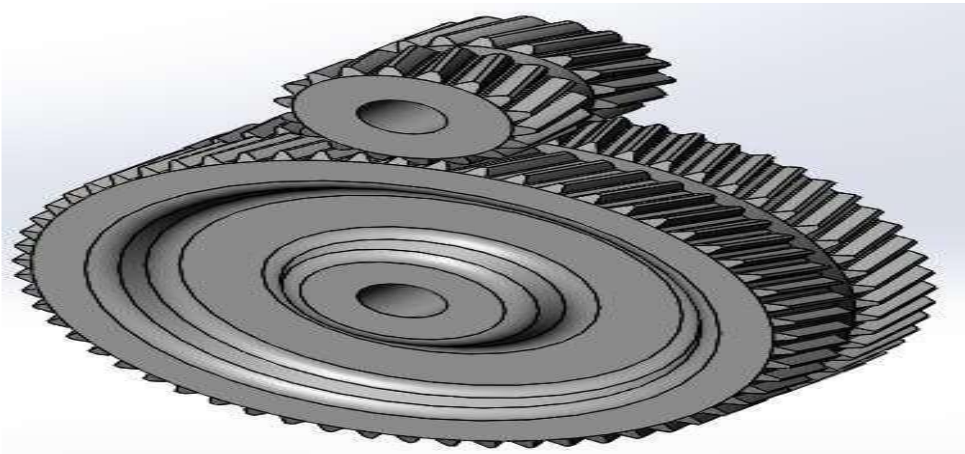

Fig. 5.15: 3D Model Assembly of Gear with Groove and Taper Edges Module 10 Gear Dia 636 &Dia 204

6. Results of Modified Gears

The analysis was run for the modified geometry of the gears. All the steps mentioned in 3.1 was performed for again for all the gear assembly. All other boundary conditions were kept same.

6.1. Results for M12 Gear Assembly with 4mm hole at the root of the gear

The stress contours for M12 gears with hole at the root of the gear has been shown in the figures below. Equivalent von-Mises stress was plotted. The maximum stress value is 1076.0 MPa which occurs on Gear Dia 675. The stress value on Gear Dia 234 comes to be 1003.1 MPa. Clearly the maximum stress value on the gears reduces due to the presence of the holes at the roots.

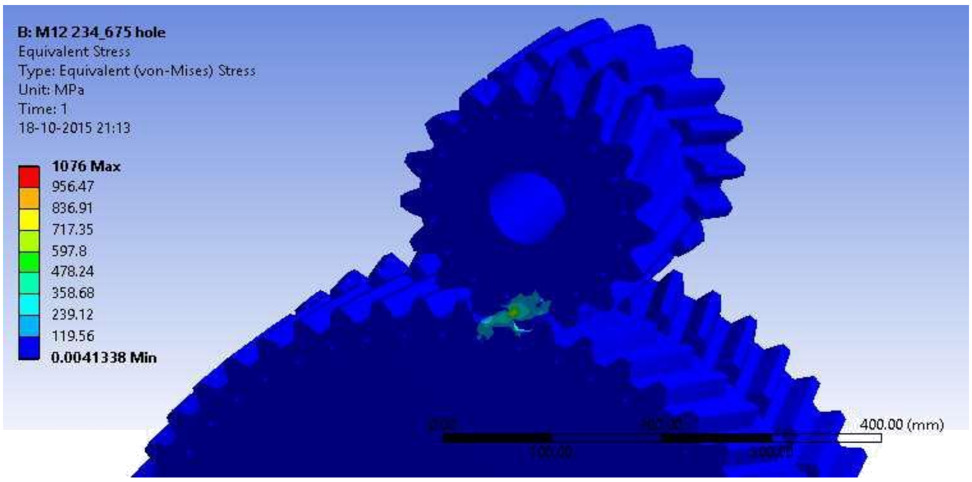

Fig. 6.1: Equivalent Stress Contours for M12 Gear Assembly with hole at the root

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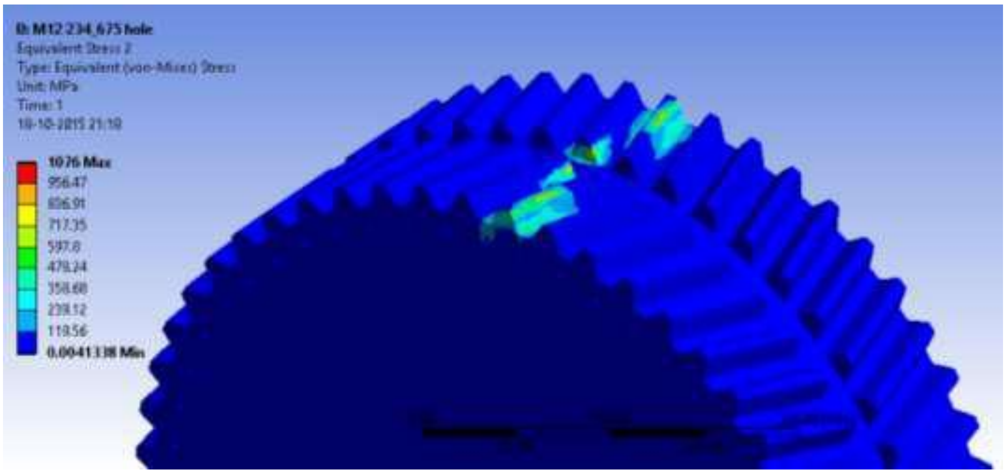

Fig. 6.2: Equivalent Stress Contours for M12 Gear Dia 675 with Hole at the root

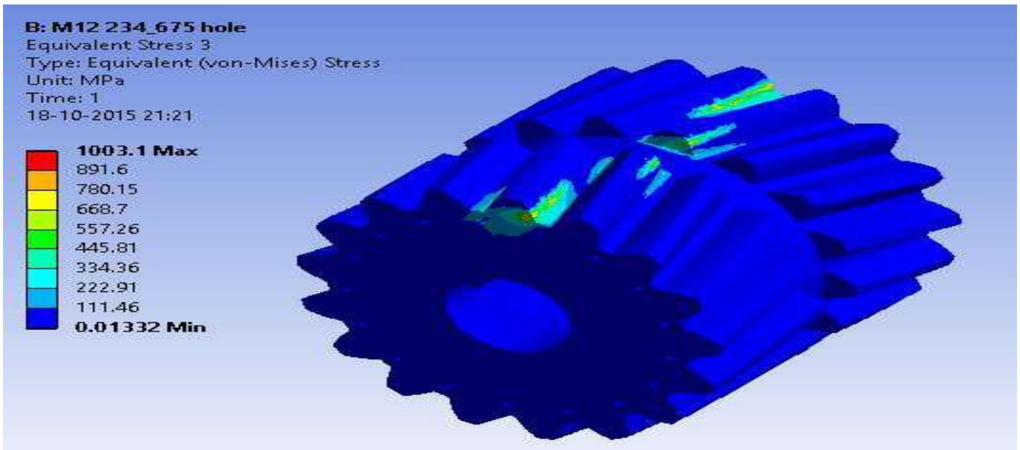

Fig. 6.3: Equivalent Stress Contours for M12 Gear Dia 234 with Hole at the root

6.2. Results for M12 Gear Assembly with Edges of the gear teeth tapered

The stress contours for M12 gears with edges of the gear teeth tapered by 20^0 has been shown in the figures below. Equivalent von-Misses stress was plotted. The maximum stress value is 1394.4 MPa which occurs on Gear Dia 675. The stress value on Gear Dia 234 comes to be 1226.6 MPa

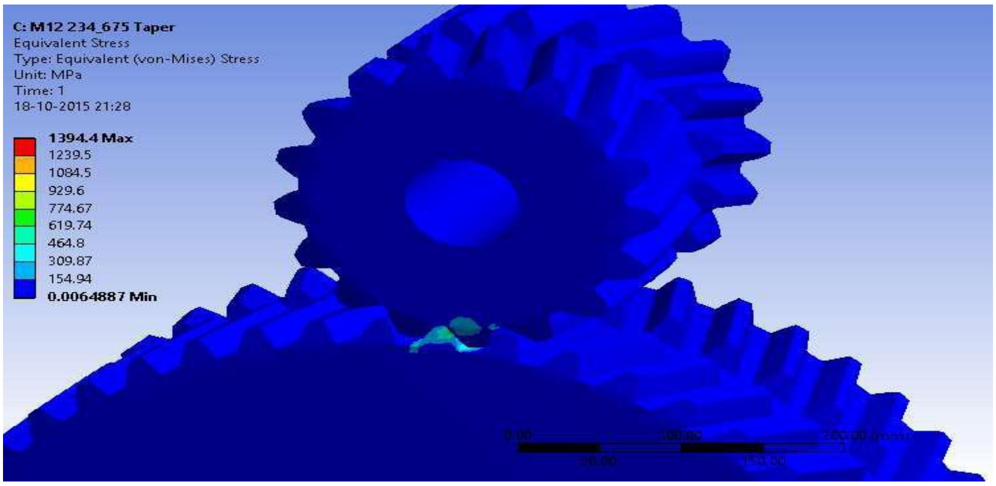

Fig. 6.4: Equivalent Stress Contours for M12 Gear Assembly with edges tapered

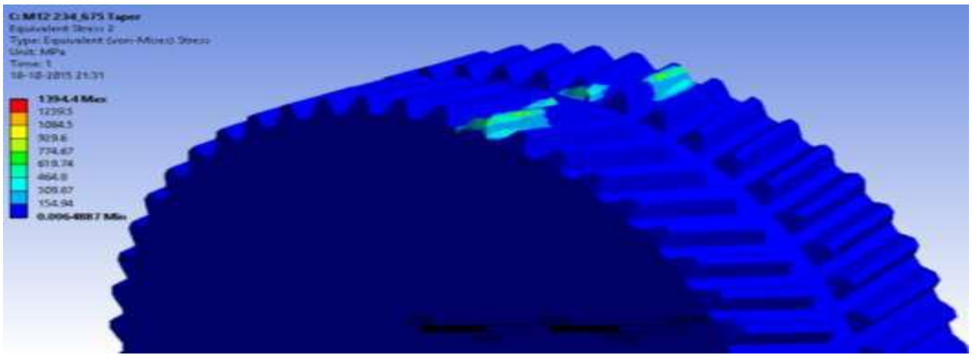

Fig. 6.5: Equivalent Stress Contours for M12 Gear Dia 675 with edges tapered

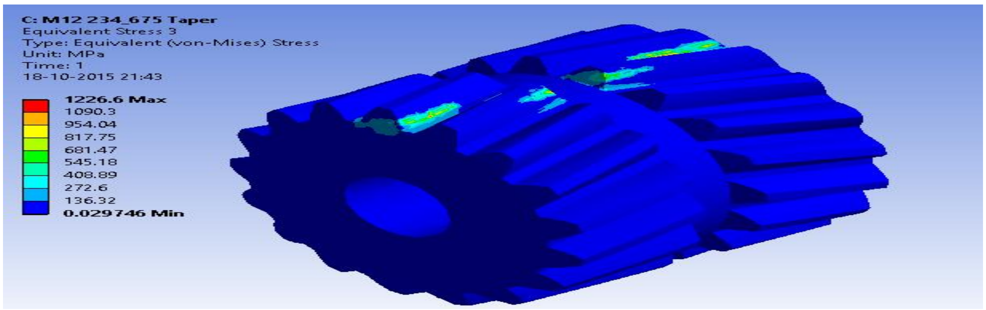

Fig. 6.6: Equivalent Stress Contours for M12 Gear Dia 234 with edges tapered

6.3. Results for M12 Gear Assembly with Groove and Taper Edges

The stress contours for M12 gears with Groove at larger gear and edges of both the gear teeth tapered by 20^0 has been shown in the figures below. Equivalent von-Mises stress was plotted. The maximum stress value is 1238.9 MPa which occurs on Gear Dia 675. The stress value on Gear Dia 234 comes to be 1186.2 MPa.

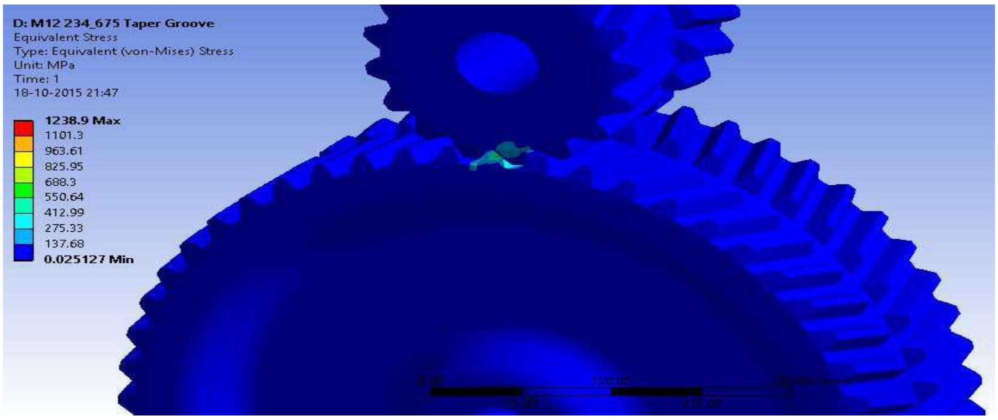

Fig. 6.7: Equivalent Stress Contours for M12 Gear Assembly with Groove and Edges tapered

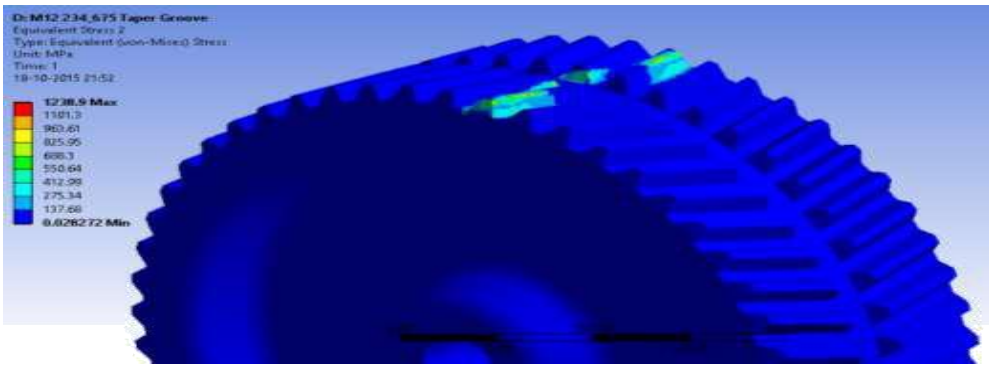

Fig. 6.8: Equivalent Stress Contours for M12 Gear Dia 675 with Groove and Edges tapered

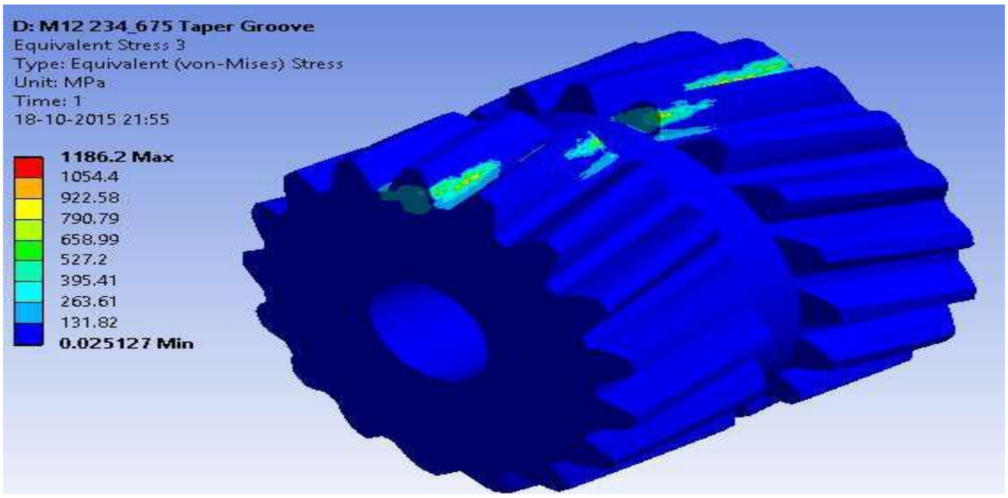

Fig.6.9:Equivalent Stress Contours for M12 Gear Dia 234 with Edges tapered mating with Gear Dia 675 having Groove and edges tapered

6.4. Results for M10 Gear Assembly with 4mm hole at the root of the gear

The stress contours for M10 gears with hole at the root of the gear has been shown in the figures below. Equivalent von-Mises stress was plotted. The maximum stress value is 392.85 MPa which occurs on Gear Dia 636. The stress value on Gear Dia 204 comes to be 376.87 MPa.

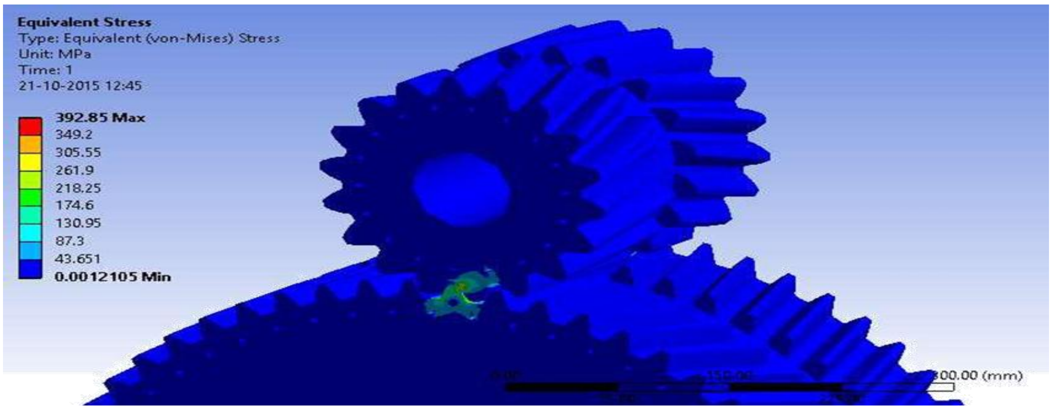

Fig. 6.10: Equivalent Stress Contours for M10 Gear Assembly with hole at the root

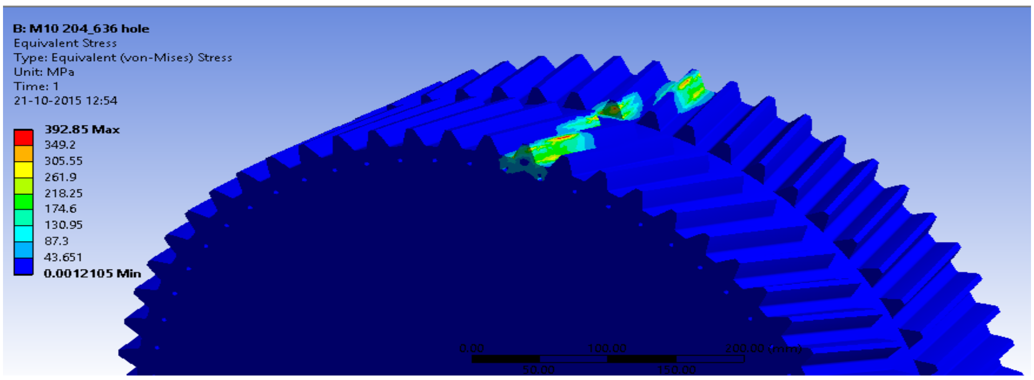

Fig. 6.11: Equivalent Stress Contours for M10 Gear Dia 636 with Hole at the root

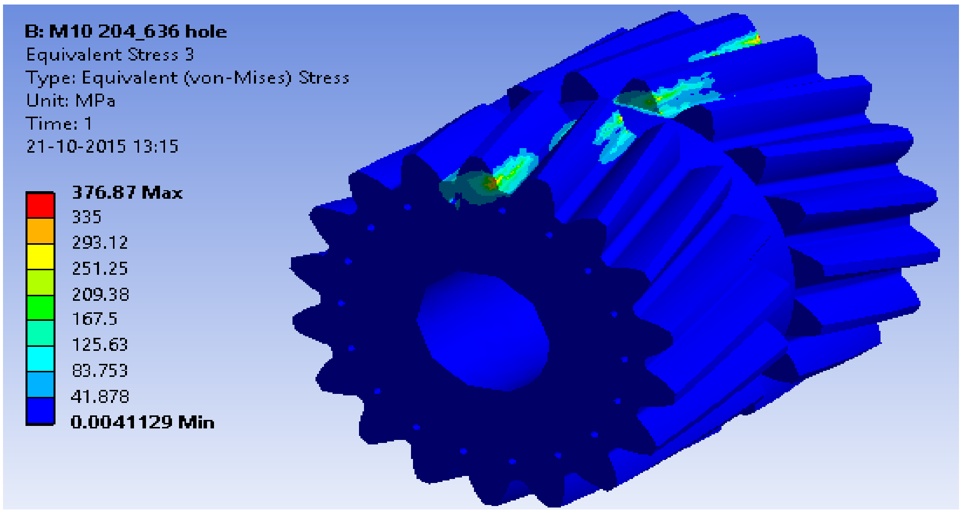

Fig. 6.12: Equivalent Stress Contours for M10 Gear Dia 204 with Hole at the root

6.5. Results for M10 Gear Assembly with Edges of the gear teeth tapered

The stress contours for M10 gears with edges of the gear teeth tapered by 20^0 has been shown in the figures below. Equivalent von-Mises stress was plotted. The maximum stress value is 451.46 MPa which occurs on Gear Dia 636. The stress value on Gear Dia 234 comes to be 336.29 MPa.

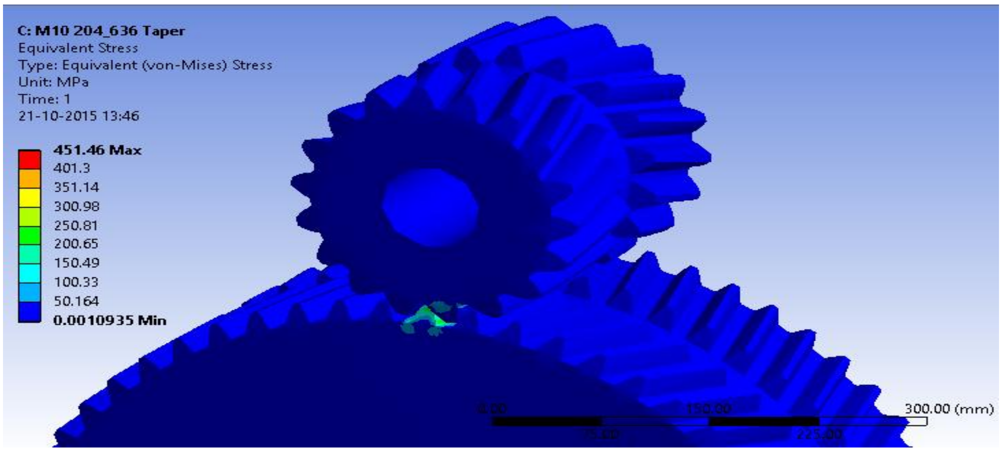

Fig.6.13: Equivalent Stress Contours for M10 Gear Assembly with edges tapered

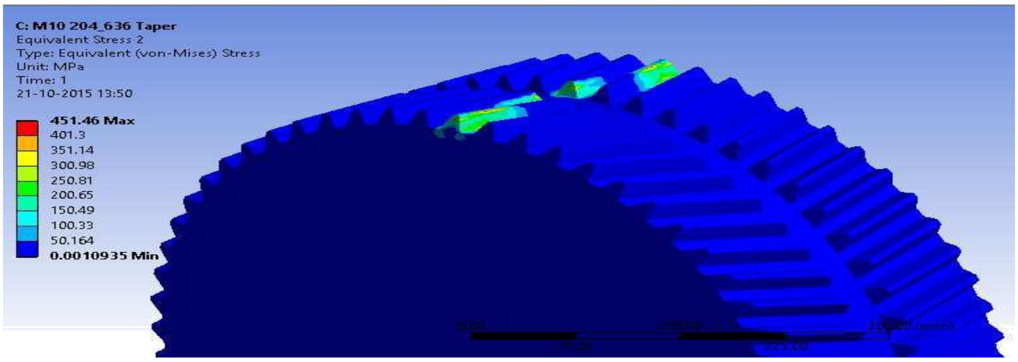

Fig. 6.14: Equivalent Stress Contours for M10 Gear Dia 636 with edges tapered

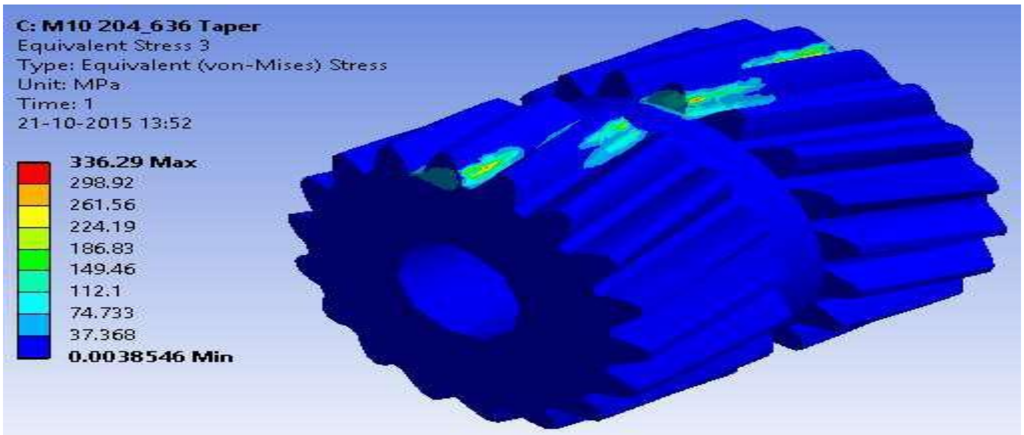

Fig. 6.15: Equivalent Stress Contours for M10 Gear Dia 204 with edges tapered

6.6. Results for M10 Gear Assembly with Groove and Taper Edges

The stress contours for M10 gears with Groove at larger gear and edges of both the gear teeth tapered by 20^0 has been shown in the figures below. Equivalent von-Mises stress was plotted. The maximum stress value is 441.12 MPa which occurs on Gear Dia 636. The stress value on Gear Dia 204 comes to be 303.97 MPa.

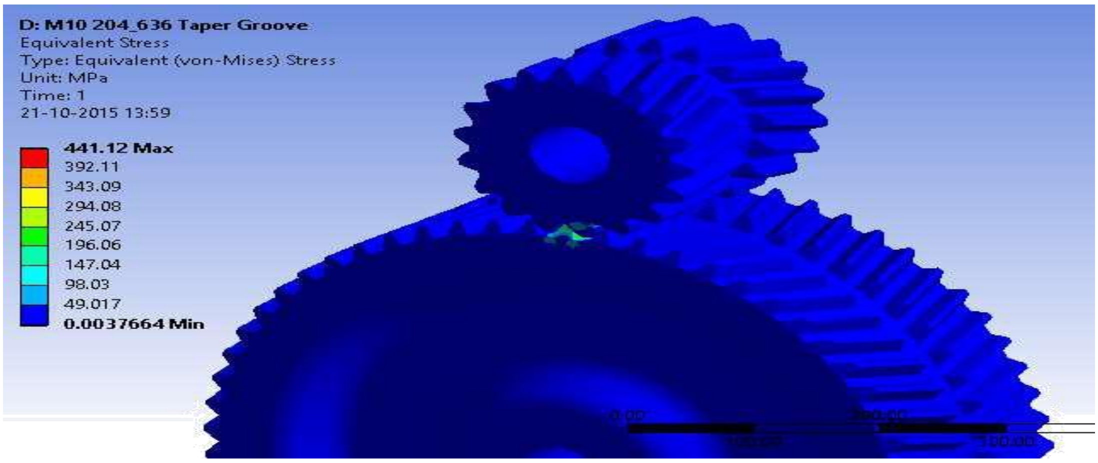

Fig. 6.16: Equivalent Stress Contours for M10 Gear Assembly with Groove and edges tapered

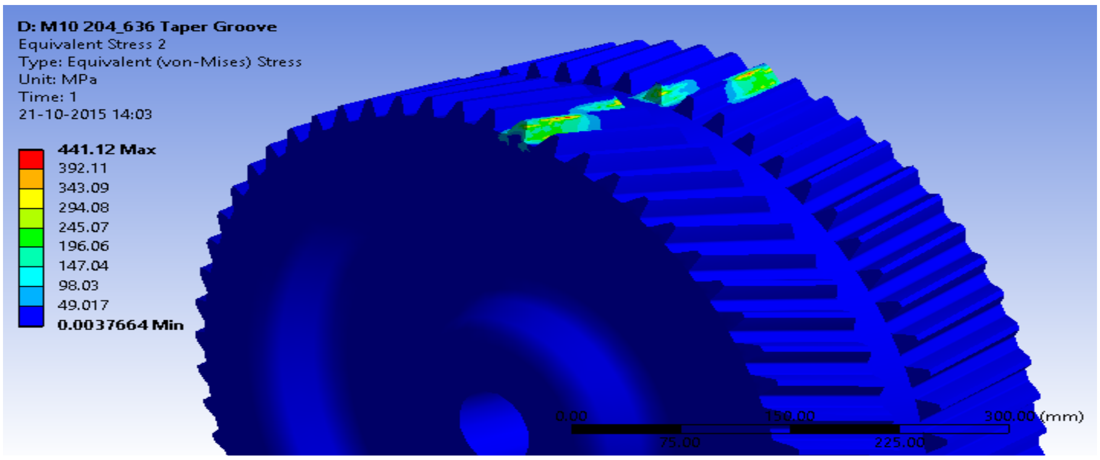

Fig. 6.17: Equivalent Stress Contours for M10 Gear Dia 636 with Groove and Edges tapered

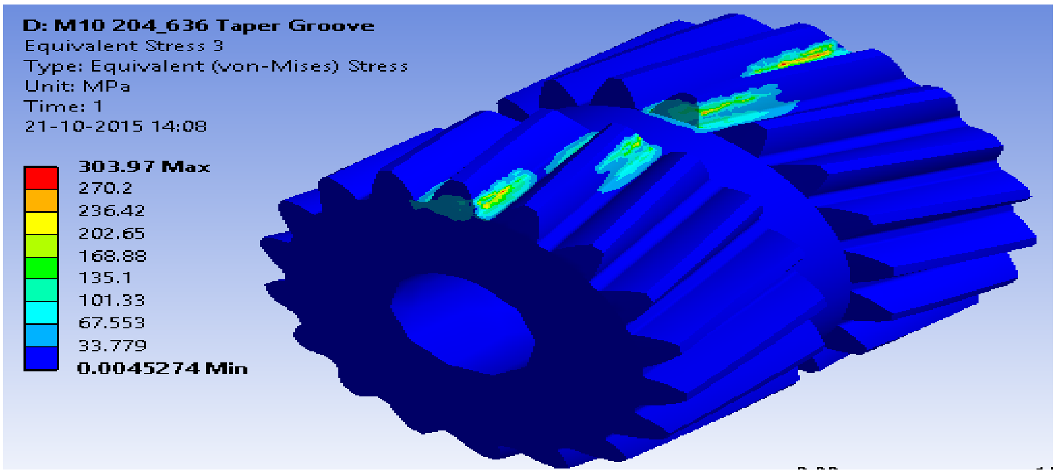

Fig. 6.18: Equivalent Stress Contours for M10 Gear Dia 204 with Edges tapered mating with Gear Dia 636 having Groove and edges tapered

7. Discussion on Results & Conclusion

7.1 Comparison Of Results:-

The results of various analysis performed on original and modified Gear geometry for a specified input loading conditions have been tabulated below.

S. No.	Gear Geometry Description	Maximum Stress on Gear Dia 675 (MPa)	Maximum Stress on Gear Dia 234 (MPa)
1	Original Double Helical Gear	1168.2	1011.0
2	4mm Dia Hole at the root of the gears	1076.0	1003.1
3	Edges of the gear teeth tapered by anangle of 20^0	1394.4	1226.6
4	Groove at central portion of Gear Dia 675 in addition to the taper at theedges	1238.9	1186.2

Table 7.2: Comparison of Results for geometry modification of Module M12 Gears

S. No.	Gear Geometry Description	Maximum Stress on Gear Dia 636 (MPa)	Maximum Stress on Gear Dia 204 (MPa)
1	Original Double Helical Gear	421.05	363.13
2	4mm Dia. Hole at the root of the gears	392.85	376.87
3	Edges of the gear teeth tapered by an angle of 20^0	451.46	336.29
4	Groove at central portion of Gear Dia. 636 in addition to the taper at the edges	441.12	303.97

Table 7.3: Comparison of Results for geometry modification of Module M10Gears

7.2 Conclusion

The stress value for M12 values were higher side as the M12 gears are used for IInd stage and the torque to be transmitted are more. The maximum stress value of the gear reduces due to the presence of holes at the root of the gear. The stress value reduces by 7.9% for M12 gears and by 6.7% for M10 gears.The stress value of the gears with edges tapered increases due to reduction in the tooth width of the gears. The stress value reduces by making a groove at the gear but does not reduce it to the value below the original gear geometry stress.

7.3 Future scope of Work

Static Analysis was performed for analyzing the stress values of the gear. The power transmission through gears is essentially dynamic in nature and the stress values must be checked by doing a Transient Analysis in which time varying loads are applied and stress is calculated accordingly. But transient analysis requires more resources and time, so a Static Analysis is performed for initial comparison of various designs. Transient analysis can be performed for gears with hole at the root.The groove shape at the larger gear can be changed for analyzing its effect on the reduction of the gear stresses.Other geometry modifications can also be checked for their effect on gear stress.

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