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Research Article

DETERMINATION OF OPTIMUM MESH SIZE TO MEASURE TOOTH ROOT STRESS OF SPUR GEAR USING FINITE ELEMENT ANALYSIS

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ABSTRACT
Gears are one of the most important power transmission elements. High speed gear and gear which has high loading capacity are necessary for engineering applications. Gears should be analyzed theoretically according to design criteria and tested. Theoretical analysis is generally performed by using finite element methods through simulating gears. In this study, spur gears at definite module and teeth number are modelled by Catia. They are analyzed by using SimXpert and Marc softwares for different loads on pitch circle. Tooth root bending stress values are recorded after analysis. Meshing, mesh size selection and mesh type are one of important steps for FEM. Different meshes (fine and coarse) are used for tooth root and other regions of gear profiles in this study. At tooth root, different mesh sizes are experienced and tooth root stress values are obtained. These values are compared with analytical calculation results to state optimal mesh size.

Keywords: *Finite Element Analysis, Spur Gears, Mesh Size, Tooth Root Stress*

1. INTRODUCTION

Gears are the one of the main power transmission element. Commonly, Studies related to gears have been carried out via theoretical studies and these studies have been supported via experimental studies to evidence. Reason of this, theoretical studies have been conducted by researchers via some of presumptions and predictions. These accuracies or false of presumptions and predictions (hypothesis) have been authenticated with experimental studies. However, today high investment costs are needed for gear test rigs, equipments and instruments. Parallel to the very rapid development of technology, gears can be tested and analyzed with simulation softwares on computers very quickly. Due to this reason, the locations of expensive testing machines and equipment have been changed by simulation programs. Simulation programs mainly consists of the finite element programs. As required, finite element programs may be individual or commercial. Main purpose is which program can be supplied requires of users. Nowadays, finite element programs have been developed quickly as requires of user and these programs are compensated them easily.

Using gears on the finite element programs mainly tooth root stress, contact stress, fatigue, vibration and the others have been studied. Pawara, Utpatb (2015) studied about spur gear FEM analysis. In addition, Chena, Zhaia, Shaob, Wanga, Sunc, (2016) analyzed spur gears in Anys., If gears used in many critical applications such as automotive and aerospace sector are only studied theoretically, it requires non-possible risks confirmed by the designer and manufacturer. Due to these reasons, some countries and institutions about the development of gear design and manufacturing technology have been carried out scientific and technological R&D studies to be authenticated and controlled them many years. Main countries are USA(AGMA), Germany (FZG), England (BGA), France (CETIM), Italy and Japan. They've lost the most time during these studies while applying mesh. Reason of this is difficulties of mesh process. Researchers used the finite element program don't anticipate that they use which element type, which mesh size, mesh type and the other parameters.

In this study, Parametric spur gear are designed using Catia, and spur gear is analyzed for bending stress using finite element program to lead for the researchers in order to optimize of meshing. Researchers will not spend their time by trying different mesh types, mesh sizes, element types and other parameters due to this study. They will determine easily for future works.

2. MATERIAL AND METHODS

2.1. Spur Gear Computer Aided Design in Catia

There are many three dimensional softwares used to create spur gear geometry. Catia, Solidworks, AutoCAD, Pro-Engineer are most important and useful. In this study, parametric spur gear model is obtained by using Catia. Uctu, (2015) described the process in his master thesis in attaches. The following steps are applied to obtain spur gear geometry:

- Formulas need to create spur gear geometry are stated (Table 1). It is worked by Babu and Tsegaw (2009).

Table 1. Formulas for spur gear parametric design

P	Formula	Description/ Units used for construction
a	20deg	Pressure angle: technologic constant (10deg ≤ a ≤ 20deg) / Angular degree
m	—	Modulue / millimeter
Z	—	Number of teeth (5 ≤ Z ≤ 200) / Integer
p	m * π	Pitch of the teeth on a straight generative rack / millimeter
e	p / 2	Circular tooth thickness, measured on the pitch circle / millimeter
ha	m	Addendum = height of a tooth above the pitch circle / millimeter
hf	if m > 1.25 hf = m * 1.25 else hf = m * 1.4	Dedendum = depth of a tooth below the pitch circle. Proportionally greater for a small modulus (≤ 1.25 mm) / millimeter
rp	m * Z / 2	Radius of the pitch circle / millimeter
ra	rp + ha	Radius of the outer circle / millimeter
rf	rp - hf	Radius of the root circle. / millimeter
rb	rp * cos(a)	Radius of the base circle /millimeter
rc	m * 0.38	Radius of the root concave corner. (m * 0.38) is a normative formula / millimeter
t	0 ≤ t ≤ 1	Sweep parameter of the involute curve / floating point number
yd	rb * (sin(t * π) - cos(t * π) * t * π)	Y coordinate of the involute tooth profile, generated by the t parameter / millimeter
zd	rb * (cos(t * π) + sin(t * π) * t * π)	Z coordinate of the involute tooth profile / millimeter
ro	rb * a * π / 180deg	Radius of the osculating circle of the involute curve, on the pitch circle / millimeter
c	sqrt(1 / cos(a)^2 - 1) / PI * 180deg	Angle of the point of the involute that intersects the pitch circle / angular degree
phi	atan(yd(c) / zd(c)) + 90deg / Z	Rotation angle used for making a gear symmetric to the ZX plane / angular degree

- "Generative Shape Design" Module of Catia is opened. Then, "Parameters and Relations" (under Properties) is made active to see it in Product Tree and "With value and with Formula" is selected.
- "Formulas: Partbody (f(x))" property is used to enter gear parameters respectively. Selection of parameters type is very important. Parameter type is entered as a characteristic length for module and degree for pressure angle.
- After adding input parameters, all parameters are selected in main menu. Then in same menu, "add formula" function (Table 1) is entered. Parameters such as pitch circle radius, base radius, outside circle, root circle and fillet radius can be calculated. If they are calculated or not should be checked in product tree (Figure 1).

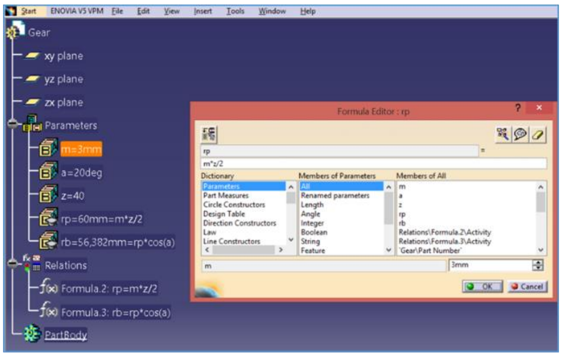


Fig. 1. Checking of Parameters

- Involute gear profile is drawn with respect to Y and Z Cartesian coordinates. Law editor in Catia window is opened by using fog icon. Then, the following path "yd > select ok > add parameters, t – select real, and x - length select their types and apply > ok" is pursued. yd equation is:

yd: yd=rb*(sin (t*PI*1rad)-cos (t*PI*1rad) * t*PI) (1)

Same procedure is applied to obtain zd. Equation of zd is:

zd: zd= rb*(cos (t*PI*1rad) +sin (t*PI*1rad) *t*PI) (2)

These equations are used to obtain involute curve. These two curves (yd and zd) combine involute curve.

- Yd(t) and Zd(t) equations are used to provide involute gear coordinates. In YZ plane, five points are obtained. The following steps are applied. The points are used to create involute by using spline command:

Insert > Wireframe> Point> Select point type: on plane>Select YZ plane > for position Y and Z, right click and Edit formula. In order to ordinate, double click the Relations\Zd. evaluate (0). Extrapolate command is used to involute is moved to gear center.

- ZX Plane is used as symmetry axis and curve is rotated for an exact angle. Before angle rotation, c parameter and formula is added (c=sqrt(1/(cos(a)*cos(a))-1)/PI). C parameter is real. For angle, "Phi =atan (Relations\yd. Evaluate(c)/Relations\zd. Evaluate(c)) +90deg/z" equation is generated and rotation is applied. Reference axis and points are hided.
- Root and tip circles are created.
- At the intersection point of root circle and involute curve, fillet is created and this fillet radius is stated as rc parameter. It is used also for next tooth of gear.
- Involute curve, tip circle diameter and fillet radius are divided by split command. The symmetry of curve is generated. The excess parts of curve are trimmed.
- Gear tooth profile is arrayed by number of teeth around gear center.
- Face width is entered to created three-dimension model.
- Tooth geometry is arrayed around the center of gear about gear numbers and linked with teeth number parameter.
- Finally, gear is padded to give thickness and finished. (Figure 2)

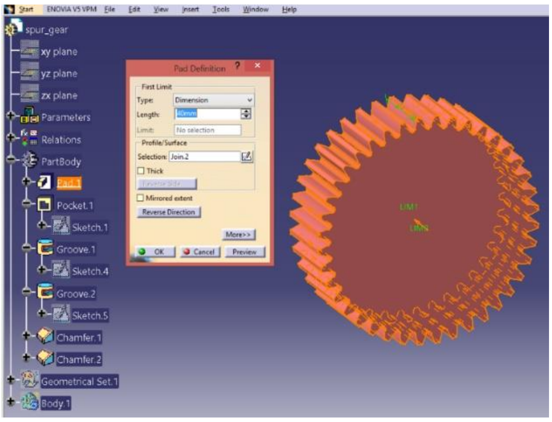


Fig. 2. Final Gear Model

2.2. Application of Mesh

The most important step in finite element analysis is meshing of model. The selection and application of suitable mesh provides easy analysis of models for users. Before meshing, some geometrical arrangement is made. Generally, only one tooth is analyzed by cancelling rest of gear teeth. But the result is not very accurate for one tooth application. So, three teeth model is used to analyze in this study as shown in Figure 3.

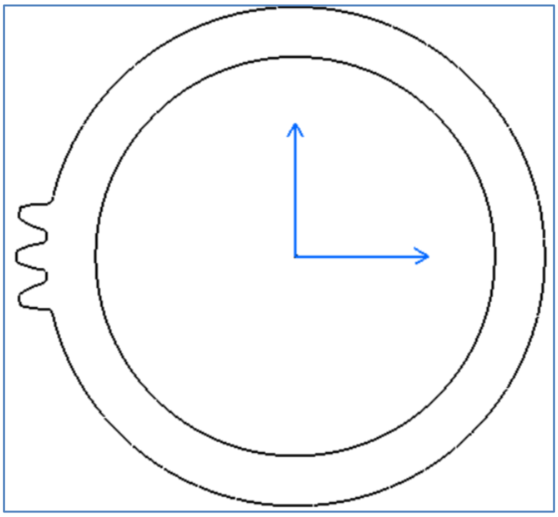


Fig. 3. Three Teeth Gear Model

In this study, Simxpert and Marc finite element software are used. These softwares have advantages and disadvantages. Meshing is easy and fast in Simxpert. Simxpert training can found guide (MSC Softwares,2015). Load application on gear is more effective in Marc. Marc training can found guide (MSC Softwares,2016). So, meshing is applied for surface in Simxpert's structure module and face width is added. Gear geometry which is loaded is offset by 2-3 mm from gear center. The reason is to apply fine mesh through involute and trochoid. Other regions are meshed less finely. Application of fine mesh for all regions on gear profile is possible but this increase analysis time. Sometimes software faults due to less performance of computer. Meshing process variables are mesh shape, elements size and mesh method. Mesh type is not changed. Element shape is used as Quad 4 and Quad 8 (figure 4).

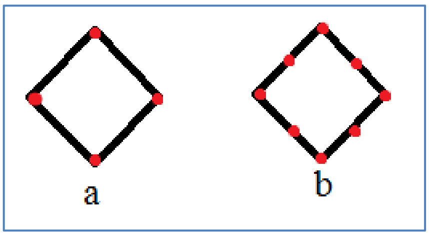


Fig. 4. Element Shape (a: Quad4, b: Quad8)

Table 2. Element Size which is studied for Mesh Size

Element Size (mm)	**0,5**	**0,4**	**0,3**	**0,2**	**0,1**	**0,05**	**0,02**	**0,01**

Element sizes used in analysis are given in Table 1. Meshing method are automatic mesh, and mapped mesh used in this study. Mesh model is given in Figure 5.

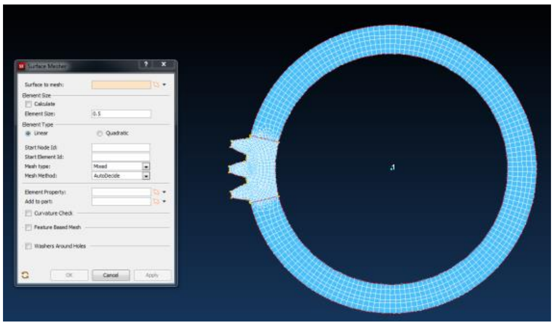


Fig. 5. Meshed Model of Gear

Applied mesh sizes between 0,5mm and 0,01mm are presented in Figure 6. After mesh application, models are saved as .bdf extension format to provide opening by Marc. For each mesh size, a new file is created.

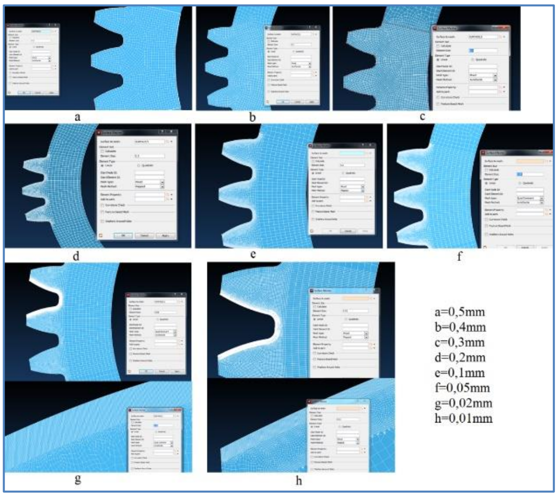


Fig. 6. Mesh Model with Their Sizes

2.3. Application of Loads and Analysis of Spur Gear

Another important step is application of load after meshing. This is very easy in Marc. The files created in .bdf file extension by Simxpert software can be opened easily by Marc. A point is created at the center of gear and all nodes are tightened to this point (Figure 7). This makes analysis steps easier. Then, the intersection point of pitch circle diameter and involute curve is stated. The force is applied at this point by using edge force command (Figure 8). Load application for Quad4 and Quad8 is shown in Figure 7a and 7b.

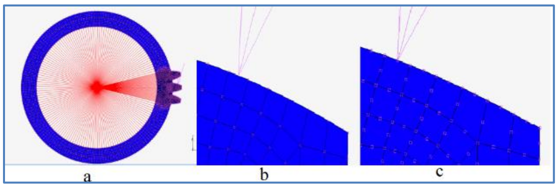


Fig. 7. Nodes Tightening (a: General view, b: Quad4, c: Quad8)

These steps are applied for other meshes respectively. Force values are given in Table 2.

Table 3. Force Values

Force (N)	**1000**	**2000**	**3000**	**4000**	**5000**

For each force value, 8 different mesh sizes and 2 different mesh shapes are used as general. For each force 16 analyses are applied. 80 different analyses are conducted for pitch point of one tooth of gear. The result of analysis for 1000N and 5000N, Quad4, 0,5mm mesh size is given in Figure 8.

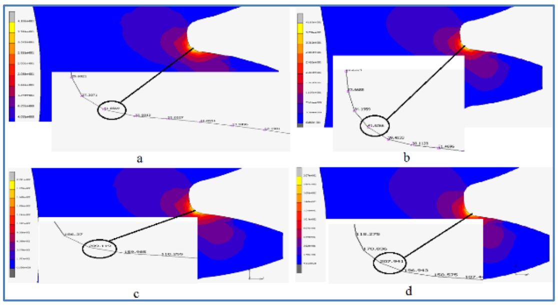


Fig. 8. Force Application (a: 1000N-Quad4, b: 1000N-Quad8, c: 5000N-Quad4, d: 5000N-Quad8)

The analysis result is given in Table 4 for 1000N and Quad4. Max gear tooth root stress is between 41,63Mpa and 43,93Mpa. Maximum stress is obtained for 0.1 mm mesh size whereas minimum one occurs 0.4 mm size.

Table 4. FEM Results in 1000 N - Quad4

Force(N)	R	Element Shape	Element Size (mm)	Number of Element	Mesh Type	Mesh Method	Maximum Rooth Stres (Mpa)
1000	60mm	Quad4	0,5mm	2499	Mixed	AutoDecide	41,86
			0,4mm	5786	Mixed	AutoDecide	41,63
			0,3mm	7909	Mixed	AutoDecide	43,33
			0,2mm	5548	Mixed	Mapped	43,69
			0,1mm	10015	Mixed	Mapped	43,93
			0,05mm	11605	Mixed	Mapped	43,70
			0,02mm	40755	Mixed	Mapped	43,49
			0,01mm	59938	Mixed	Mapped	43,45

The analysis result is given in Table 5 for 1000N and Quad8. Max gear tooth root stress is between 41,61Mpa and 48, 62 Mpa. Maximum stress is obtained for 0.4 mm mesh size whereas minimum one occurs 0.05 mm size.

Table 5. FEM Results in 1000 N – Quad8

Force(N)	R	Element Shape	Element Size (mm)	Number of Element	Mesh Type	Mesh Method	Maximum Rooth Stre (Mpa)
1000	60mm	Quad8	0,5mm	2499	Mixed	AutoDecide	46,55
			0,4mm	5786	Mixed	AutoDecide	48,62
			0,3mm	7909	Mixed	AutoDecide	47,76
			0,2mm	5548	Mixed	Mapped	43,73
			0,1mm	10015	Mixed	Mapped	44,55
			0,05mm	11605	Mixed	Mapped	43,61
			0,02mm	40755	Mixed	Mapped	44,52
			0,01mm	59938	Mixed	Mapped	43,65

The analysis result is given in Table 6 for 2000N and Quad4. Max gear tooth root stress is between 83,23Mpa and 87,84Mpa. Maximum stress is obtained for 0.1 mm mesh size whereas minimum one occurs 0.4 mm size.

Table 6. FEM Results in 2000 N - Quad4

Force(N)	R	Element Shape	Element Size (mm)	Number of Element	Mesh Type	Mesh Method	Maximum Rooth Stress (Mpa)
2000	60mm	Quad4	0,50	2499	AutoDecide	Mapped	83,70
			0,40	5786	Mixed	Mapped	83,23
			0,30	7909	Mixed	Mapped	86,63
			0,20	5548	Mixed	Mapped	87,36
			0,10	10015	Mixed	Mapped	87,84
			0,05	11605	Mixed	Mapped	87,38
			0,02	40755	Mixed	Mapped	86,96
			0,01	59938	Mixed	Mapped	86,87

The analysis result is given in Table 7 for 2000N and Quad8. Max gear tooth root stress is between 87,20Mpa and 97,22Mpa. Maximum stress is obtained for 0.4 mm mesh size whereas minimum one occurs 0.05 mm size.

Table 7. FEM Results in 2000 N – Quad8

Force(N)	R	Element Shape	Element Size (mm)	Number of Element	Mesh Type	Mesh Method	Maximum Rooth Stre (Mpa)
2000	60mm	Quad8	0,5mm	2499	Mixed	AutoDecide	93,08
			0,4mm	5786	Mixed	AutoDecide	97,22
			0,3mm	7909	Mixed	AutoDecide	95,50
			0,2mm	5548	Mixed	Mapped	87,43
			0,1mm	10015	Mixed	Mapped	89,08
			0,05mm	11605	Mixed	Mapped	87,20
			0,02mm	40755	Mixed	Mapped	89,01
			0,01mm	59938	Mixed	Mapped	87,28

The analysis result is given in Table 8 for 3000N and Quad4. Max gear tooth root stress is between 124,8Mpa and 131,7Mpa. Maximum stress is obtained for 0.1 mm mesh size whereas minimum one occurs 0.4 mm size.

Table 8. FEM Results in 3000 N – Quad4

Force(N)	R	Element Shape	Element Size (mm)	Number of Element	Mesh Type	Mesh Method	Maximum Rooth Stres (Mpa)
3000	60mm	Quad4	0,5mm	2499	Mixed	AutoDecide	125,50
			0,4mm	5786	Mixed	AutoDecide	124,80
			0,3mm	7909	Mixed	AutoDecide	129,90
			0,2mm	5548	Mixed	Mapped	131,00
			0,1mm	10015	Mixed	Mapped	131,70
			0,05mm	11605	Mixed	Mapped	131,00
			0,02mm	40755	Mixed	Mapped	130,40
			0,01mm	59938	Mixed	Mapped	130,30

The analysis result is given in Table 9 for 3000N and Quad8. Max gear tooth root stress is between 130,8 Mpa and 145,8 Mpa. Maximum stress is obtained for 0.4 mm mesh size whereas minimum one occurs 0.05 mm size.

Table 9. FEM Results in 3000 N – Quad8

Force(N)	R	Element Shape	Element Size (mm)	Number of Element	Mesh Type	Mesh Method	Maximum Rooth Stre (Mpa)
3000	60mm	Quad8	0,5mm	2499	Mixed	AutoDecide	139,60
			0,4mm	5786	Mixed	AutoDecide	145,80
			0,3mm	7909	Mixed	AutoDecide	143,20
			0,2mm	5548	Mixed	Mapped	131,10
			0,1mm	10015	Mixed	Mapped	133,60
			0,05mm	11605	Mixed	Mapped	130,80
			0,02mm	40755	Mixed	Mapped	133,50
			0,01mm	59938	Mixed	Mapped	130,90

The analysis result is given in Table 10 for 4000N and Quad4. Max gear tooth root stress is between 166,4Mpa and 175,6Mpa. Maximum stress is obtained for 0.1 mm mesh size whereas minimum one occurs 0.4 mm size.

Table 10. FEM Results in 4000 N – Quad4

Force(N)	R	Element Shape	Element Size (mm)	Number of Element	Mesh Type	Mesh Method	Maximum Rooth Stres (Mpa)
4000	60mm	Quad4	0,5mm	2499	Mixed	AutoDecide	167,30
			0,4mm	5786	Mixed	AutoDecide	166,40
			0,3mm	7909	Mixed	AutoDecide	173,20
			0,2mm	5548	Mixed	Mapped	174,60
			0,1mm	10015	Mixed	Mapped	175,60
			0,05mm	11605	Mixed	Mapped	174,70
			0,02mm	40755	Mixed	Mapped	173,80
			0,01mm	59938	Mixed	Mapped	173,70

The analysis result is given in Table 11 for 4000N and Quad8. Max gear tooth root stress is between 174,3Mpa and 194,4Mpa. Maximum stress is obtained for 0.4 mm mesh size whereas minimum one occurs 0.05 mm size.

Table 11. FEM Results in 4000 N – Quad8

Force(N)	R	Element Shape	Element Size (mm)	Number of Element	Mesh Type	Mesh Method	Maximum Rooth Stre (Mpa)
4000	60mm	Quad8	0,5mm	2499	Mixed	AutoDecide	186,10
			0,4mm	5786	Mixed	AutoDecide	194,40
			0,3mm	7909	Mixed	AutoDecide	190,90
			0,2mm	5548	Mixed	Mapped	174,80
			0,1mm	10015	Mixed	Mapped	178,10
			0,05mm	11605	Mixed	Mapped	174,30
			0,02mm	40755	Mixed	Mapped	177,90
			0,01mm	59938	Mixed	Mapped	174,50

The analysis result is given in Table 12 for 5000N and Quad4. Max gear tooth root stress is between 207,9Mpa and 219,5Mpa. Maximum stress is obtained for 0.1 mm mesh size whereas minimum one occurs 0.4 mm size.

Table 12. FEM Results in 5000 N – Quad4

Force(N)	R	Element Shape	Element Size (mm)	Number of Element	Mesh Type	Mesh Method	Maximun Rooth Stre (Mpa)
5000	60mm	Quad4	0,5mm	2499	Mixed	AutoDecide	209,10
			0,4mm	5786	Mixed	AutoDecide	207,90
			0,3mm	7909	Mixed	AutoDecide	216,40
			0,2mm	5548	Mixed	Mapped	218,20
			0,1mm	10015	Mixed	Mapped	219,50
			0,05mm	11605	Mixed	Mapped	218,30
			0,02mm	40755	Mixed	Mapped	217,20
			0,01mm	59938	Mixed	Mapped	217,00

The analysis result is given in Table 13 for 5000N and Quad8. Max gear tooth root stress is between 217,9Mpa and 242,9Mpa. Maximum stress is obtained for 0.4 mm mesh size whereas minimum one occurs 0.05 mm size.

Table 13. FEM Results in 5000 N – Quad8

Force(N)	R	Element Shape	Element Size (mm)	Number of Element	Mesh Type	Mesh Method	Maximun Rooth Stre (Mpa)
5000	60mm	Quad8	0,5mm	2499	Mixed	AutoDecide	232,50
			0,4mm	5786	Mixed	AutoDecide	242,90
			0,3mm	7909	Mixed	AutoDecide	238,60
			0,2mm	5548	Mixed	Mapped	218,40
			0,1mm	10015	Mixed	Mapped	222,60
			0,05mm	11605	Mixed	Mapped	217,90
			0,02mm	40755	Mixed	Mapped	224,40
			0,01mm	59938	Mixed	Mapped	218,10

2.4. Analytical Solution of Spur Gear Bending Stress

Calculation of tooth root stress of spur gear is based on Lewis equation and ISO Gear Standards. Spur gear strength is descripted (International Organization International Organization for Standardization 2006, Calculation of tooth bending strength) for Standardization 1982) Şahin, (2014) described the process in his master thesis. Equations are given below:

- The basic bending stress for gear teeth is obtained by using the Lewis formula

$$\sigma = \frac{F_t}{(b_a * m * Y)} \qquad (3)$$

F t = Tangential force on tooth
σ = Tooth Bending stress (MPa)
b_a = Face width (mm)
Y = Lewis Form Factor
m = Module (mm)

- The Lewis formula is often expressed as

$$\sigma = \frac{F_t}{(b_a * p * y)} \qquad (4)$$

Where y = Y/π and p = circular pitch

You can find modified equation in ISO 6336-3 (ISO, 2009). According to analytical formulas, the results are presented in Table 14.

Table 14. Force versus Stress Values

Force (N)	1000	2000	3000	4000	5000
Analytical Results (Mpa)	44,89	89,78	134,67	179,56	224,46

Comparison of analytical and finite element results is presented in Figure 9- 13 for different mesh size. For different force values, analyses for Quad 4 are almost same. It can be said also for Quad 8. The closest result of finite element analysis to analytical calculation is obtained 0.1 mm mesh size for Quad 4 and Quad 8 whereas the value most far from analytical result is taken at 0.4 mm mesh size.

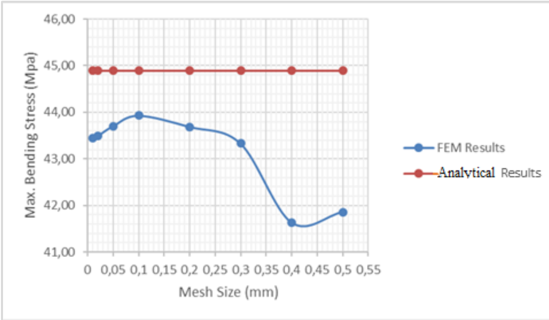


Fig. 9. Comparison of FEM Results and Analytical Solution for Quad4-1000N

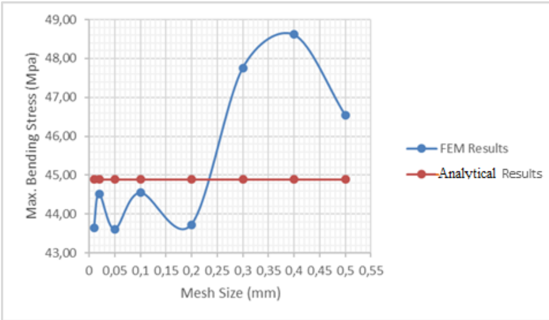


Fig. 11. Comparison of FEM Results and Analytical Solution for Quad8-1000N

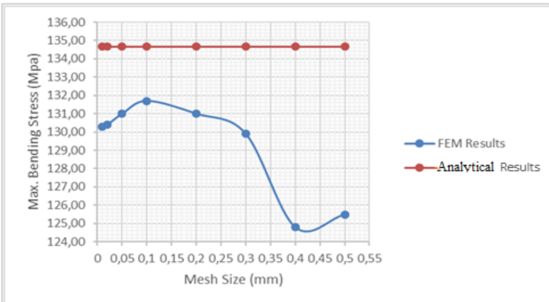


Fig. 12. Comparison of FEM Results and Analytical Solution for Quad4-3000N

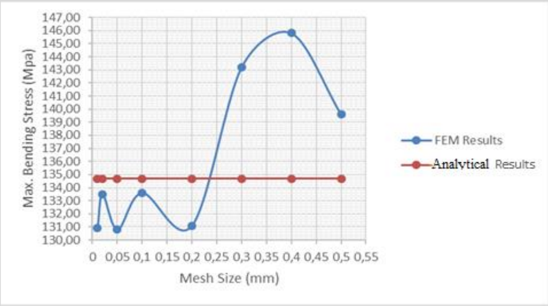


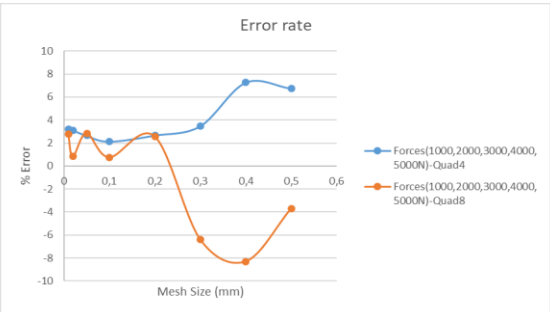
Fig. 13. Comparison of FEM Results and Analytical Solution for Quad8-3000N

The differences between finite element and analytical solutions' results is calculated according to following formulae:

$$\%E_r = \left|\frac{R_a - R_{fem}}{R_a} * 100\right| \qquad (5)$$

Where Ra is analytical result, R_{fem}=FEM results and E_r=Error rate.

Error percent are obtained for Quad 4 and Quad 8, and then graphics is drawn. Minimum error for 0.1 mm and 0.02 mm mesh is observed as about 1% from Figure 19. Maximum error is observed for 0.4 mm mesh size as 8%.



3. CONCLUSION

In this study, gear with 3 mm of module, 40 of teeth number and 20 degree of pressure angle is created by Catia. This gear model is meshed by Simxpert. Two different mesh types are used; Quad 4 and Quad 8. For this mesh models, 0.5 mm and 0.01 mm mesh sizes are used. Then five different loads (5000N, 4000N, 3000N, 2000N and 1000N) are applied by Marc. The results are presented in Tables3-Table12. Analytical calculations for gear tooth root stress are compared with Finite Element Analysis. According to results, Quad 8 mesh type and 0.1 mm mesh size are most suitable mesh parameters. 0.4 mm mesh size gives the most different results.

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