2012

A Finite Element Approach to Stress Analysis of Face Gears

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A FINITE ELEMENT APPROACH TO STRESS ANALYSIS OF FACE GEARS

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Bachelor of Mechanical Engineering

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May, 2008

submitted in partial fulfillment of requirements for the degree

MASTER OF SCIENCE IN MECHANICAL ENGINEERING

at the

CLEVELAND STATE UNIVERSITY

May, 2012

This thesis has been approved
for the Department of **MECHANICAL ENGINEERING**
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Department/Date

ACKNOWLEDGEMENT

Foremost, I would like to express my sincere gratitude to my advisor Dr. Majid Rashidi for the continuous support of my thesis, for his patience, motivation, enthusiasm, and immense knowledge. His guidance helped me in all the time of my research and writing of this thesis. I could not have imagined having a better advisor and mentor for my thesis.

Besides my advisor, I would like to thank my thesis committee members: Dr. Rama Gorla and Dr. Asuquo Ebiana, for their encouragement, insightful comments, and helpful questions.

My sincere thanks also goes to Dr. Claude Gosslin for helping me with his HyGears software, which is the best software I have ever seen. I could not imagine how tough it would be without his support.

I thank my friends Sowmya Reddy Narala, Deepak Ravipati, Prabhu Saran Konduru for the stimulating discussions, and their continuous push and for all the fun we have had in the last two years.

Last but not the least, I would like to thank my family: my parents Rama Mohan Rao and Lavanya Sundari, for giving birth to me at the first place and my brother Bhargav Venkata Manohar for supporting me throughout my life.

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ABSTRACT

Face Gears have alternative gear-teeth configuration compared to Bevel Gears. Face Gear have a standard spur pinion as opposed to a bevel Gear. This work concentrates on modeling of a Face-gear, Meshed with a spur gear, using CAD Software and Finite Elements Analysis. The bending stress developed at the gear teeth is determined. Numerical results are validated using the bending stress developed between two involute spur gears.

TABLE OF CONTENTS

Page

LIST OF FIGURES

Figure **Page**

CHAPTER I

INTRODUCTION

Face gears are the kinds of gears which can mesh with either spur or helical pinions having cylindrical base surfaces and the terminology of Face Gear set is shown in Figure 1. The axes of the gear and the pinion generally, but not necessarily, intersect at the right angles. When the axes intersect at right angles the gear set is called “On-Center” face gear or else the gear set is called “off-set” face gear as shown in Figure 2.

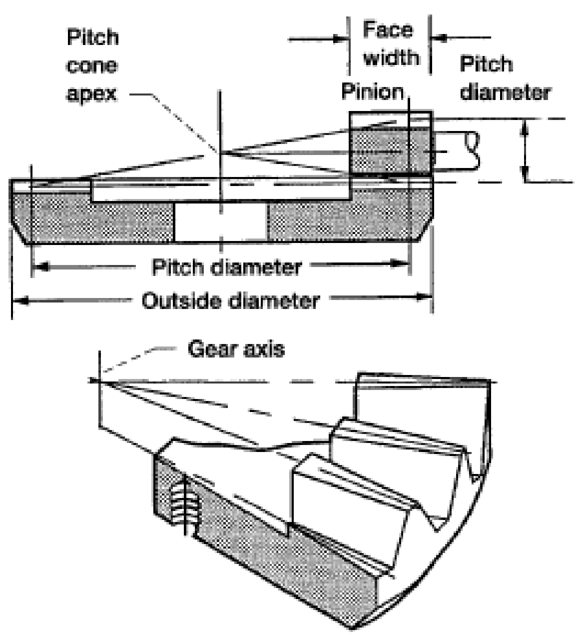


Figure 1: Face gear set terminology

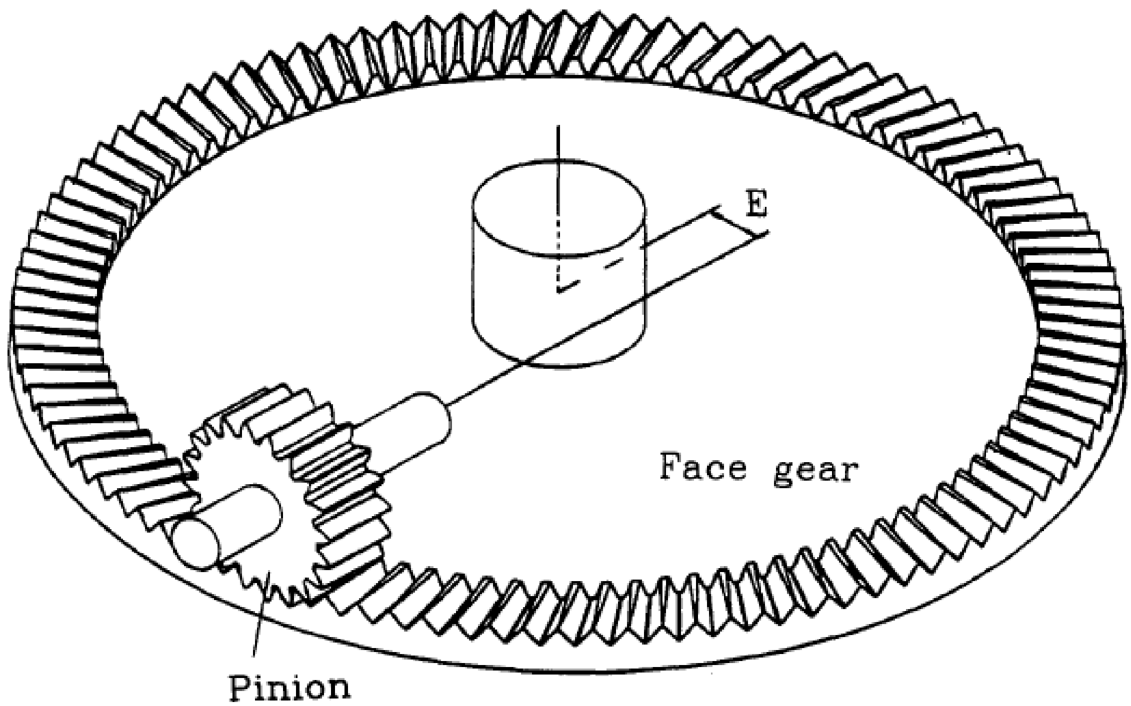


Figure 2: Off-set Face gear set

In this thesis we will be dealing with the On-center gear set, where the axes intersect and are perpendicular with each other.

Face Gear set consists of a spur or a helical pinion that is in mesh with a face gear. The pinions are really not any different from their parallel axis counterparts except for the fact that they are in mesh with a face gear and involute spur gear. Face gears have teeth cut into the blank such that the axis of the teeth lies in a plane that is perpendicular to the shaft axis. The mating pinion is either a spur or a helical gear. The pinion and face gear axes most often form a 90° are possible. In operation, this type of gear is similar to an equivalent set of straight gears.

The face gear tooth changes shape from one end of the tooth along the radius of the face of the gear as shown in Fig 3. The load capacity of face gearset, compared with that of bevel gears, is less, thus they are used mostly used as motion transmission gears rather than as power gears. Face gearsets arerelatively easy to make and less expensive compared to the bevel gears. But the calculations involved in designing the face gear set are more complex in nature, because of the change in face width, undercutting and pointing. There were some initial works done in 1950's and 1960's in Russia [1] [2]. Also a mojor contribution was done by Dudley in 1960's giving the formulaes which will be discussed in the section 1.1.

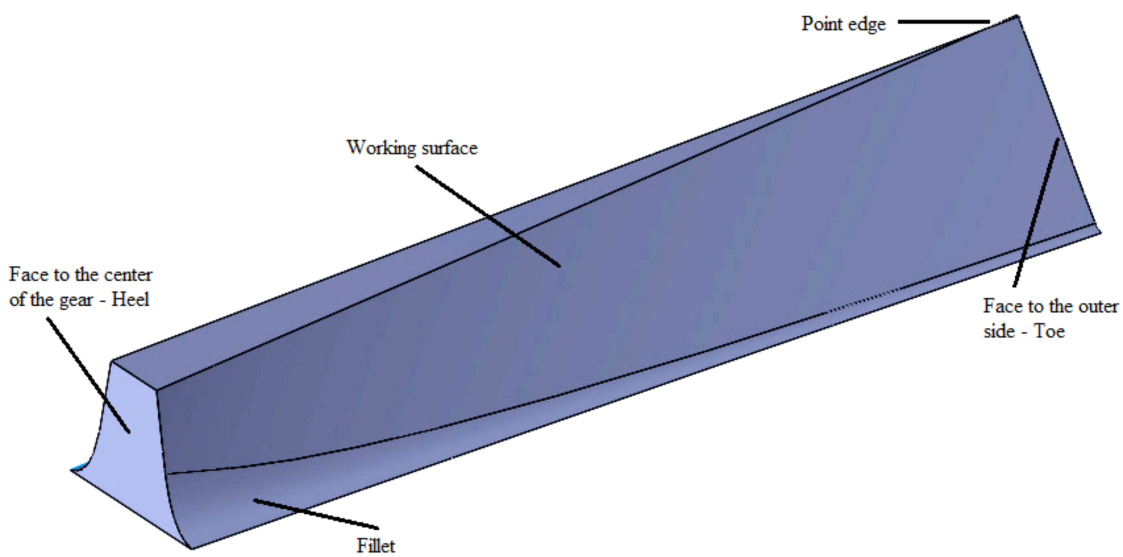


Figure 3: Face gear teeth

1.1 Previous Works

Face gears were used for centuries in wagons by Chinese and power generation by Romans [3, 4]. The invention if the modern type of face gear set was an invention of the Fellows Company. The importance of face gears was recognized when it is proposed to use in helicopter transmission because of its split torque and its less weight, which is a major advantage for the flying objects.

Fellows Company has invented the face gear drive and it is initially called “Cylkro Gear” by a Dutch company [4]. Darle W. Dudley[6][5], F.L.Litvin [2] [6] [7] [8] [4], Bloomfield B. [11] [12] has contributed a lot in the development of Face Gear in early stages. Which are followed by J.Chakraborty [13] [14], B.S Bhadoria [14] [13], David G. Lewicki [15], Aleksandar M. Egelja [10]. The advanced research on Face Gears was done by Crown Gear B.V. Company in Netherlands. The research was followed in United States, Germany and Japan.

Litvin has published many works on Face Gear includingdesign of Face Gear cutters, theoretical study of face gears by dividing into elemental parts and also designed and performed stress analysis on symmetric and asymmetric face gear drives [9].Litvinalso designed computer programs to perform design and analysis on any kind of symmetric or asymmetric Face Gear drives.

Claudio Zanzi and Jose I. Pedrero [16] has also developed a modified geometry and performed tooth contact analysis and stress analysis on the Face Gear set with a double crowned pinion. These analyses are performed to find the bearing contact and stresses throughout meshing of the pinion with the Face Gear, but during the analyses the tilt angle is ignored. And the main advantage of this kind of design is the position and orientation of the bearing contact can be modified or controlled using the parameters of the modified pinion.

McDonnell Douglas Helicopter Company, NASA and U.S. Army Research Laboratory have done extensive research on the applications of Face Gear set in Aerospace Applications and have been successful. Mr Dudley and Mr. Drago in the early 1980's came up with the use of Face Gear set in power transmission between two perpendicular axes. Its importance has been found in the Helicopter transmission because of its split torque and less weight compared to the bevel gears. So the bevel gears are replaced by face gears and other advantages will be discussed in Sec.1.1.2. Three different arrangements of Face Gear sets were arranged in the aerospace gearbox using ten pairs of gears. The Aim of this study is to find the functional ability and failure mode of the Face Gear set.

1.2 Advantages of Face Gears over Bevel Gears

- Bevel Gears can be replaced by face gears at high power and load transmissions in aerospace environment.
- The manufacturing of Face Gear is easy compared to the straight Bevel Gears. But the calculations are very complex in nature compared to other kinds of gear.
- The misalignment in the Face Gear set is very low and also reduces the stiffness in the bearing support. Litvin proposed a Face Gear set with less number of teeth in the pinion than the cutter.

- Due to less weight if the Face Gears over bevel gears, the use of Face Gear has been increased in the helicopter transmission.

1.3 My Contribution

As there are no standard formulae to obtain the Face Gear geometry named HyGears has been used to generate the geometry of the Face Gear set. Using these geometrical values from the HyGear, Face Gear tooth has been designed in Solidworks software. After designing the Face Gear tooth the load was applied to the tip of the tooth and bending stress was obtained using Solidworks simulation. The numerical results are validated according to AGMA standards.

CHAPTER II

VALIDATION OF RESULTS FOR BENDING STRESS OFFACE GEAR

2.1 Validation Of Bending Stress Results Using Lewis Formula And Solidworks Simulation As Per AGMA Standards

2.1.1 Calculating of bending stress using Lewis Formula

Bending Stress, $\sigma = W^t K_o K_v K_s \frac{1}{F\,m} \frac{K_m\,K_B}{J}$

Tangential Load, W^t = 10000N

Overload factors, K_o = 1, for uniform power source

Dynamic factor, $K_v = \left(\frac{A+\sqrt{200V}}{A}\right)^B$

Where, A = 50 + 56(1-B) ; B= 0.25 $(12-Q_V)^{2/3}$

Q_V = 8 for precision gearing

Pitch line velocity, V = $\frac{\pi d_n}{60}$ = 15.7 m/sec

Therefore $K_v = 1.444$

Size factor, $K_s = 1$

Face width, F = 0.06m

Module, $m = \frac{D}{T} = \frac{60 * 10^{-3}}{18}$

Load distribution factor, $K_m = 1 + C_{mc}(C_{pf}C_{pm} + C_{ma}C_e)$

$C_{mc} = 1$ for uncrowned teeth

$C_{pf} = \frac{F}{10\, d} - 0.0375 + 0.0125F = 0.06325$

$C_{pm} = 1$ for straddle mounted pinion

$C_e = 1$

$C_{ma} = A + BF + CF^2 = 0.06826$

For precision enclosed unit A= 0.0675; B = 0.0128; C = $-0.9626*10^{-4}$

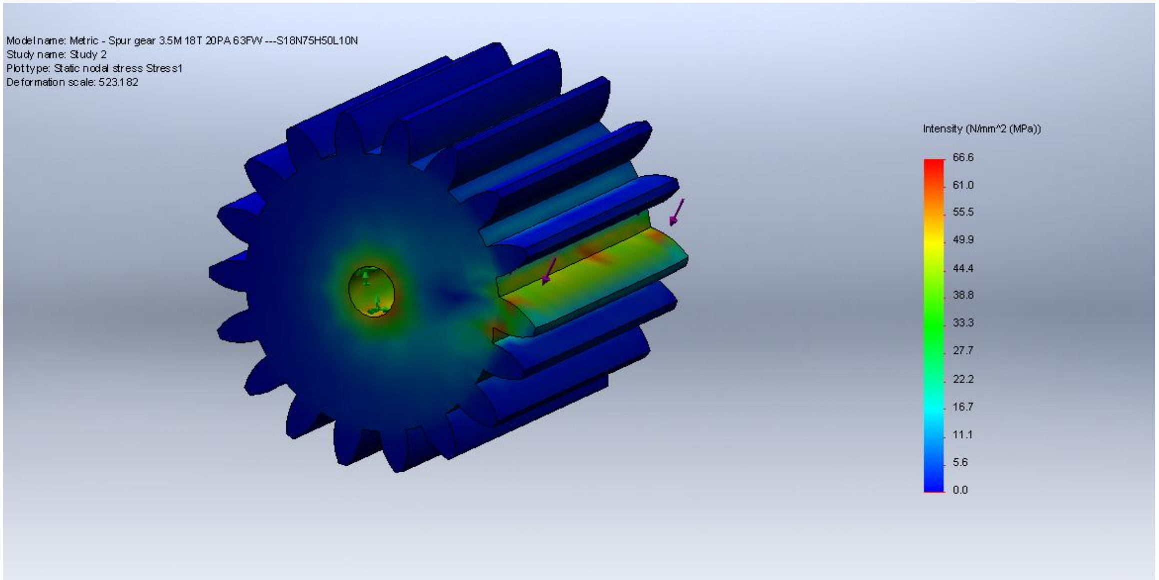
So, $K_m = 1.13151$

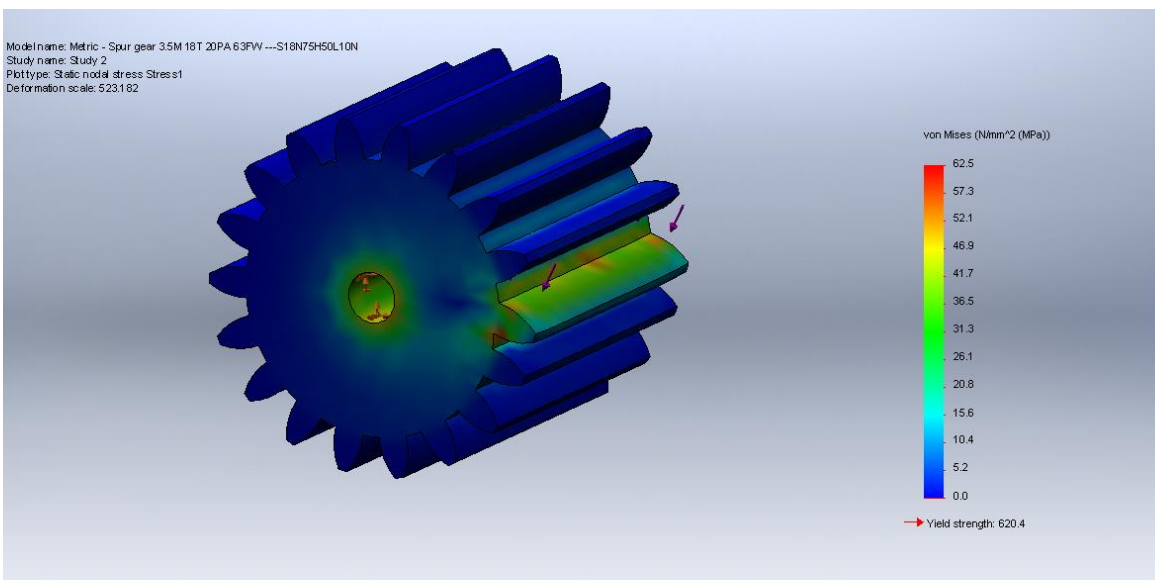
Rim thickness factor, $K_B = 1$

Bending strength geometry factor, J = 0.31

Bending stress calculated, $\sigma = 62$MPa

2.1.2 Results from Solidworks Simulation





Maximum intensity stress from solidworks simulation = 66.6Mpa

2.1.3 Error percentage

Percentage error between the stress calculated from Lewis formula and Solidworks Simulation,

Error Percentage = 7.4%

CHAPTER III

HYGEARS

3.1 Introduction To HyGears

HyGears is the Gear Processor(The High Performance Software for Gear Engineering, Analysis and Production Control)

HyGears is an advanced and a 30 year old user friendly, 3D gear modeling software that has been extensively tested in industry. Developed by Claude Gosselin, Ph.D president of Involute Simulation Software Inc. and was initially developed in September 1982 on a text only display main frame. It was first released to public in 1994 in Japan.

HyGears supports :

- Spiral and hypoid gears for the Face Milling - Fixed Setting, Duplex Helical, Modified Roll, Spread Blade, Formate and Helixform - cutting processes (all, The Gleason Works)
- It can also model the Face Hobbing process (Klingelnberg and Gleason)

- Straight bevel gears (external and internal); straight bevels which can be ‘forged’ and thus used to create forging dies
- Spur and helical gears, external and internal, in planetary or offset arrangements
- Involute splines and face gears

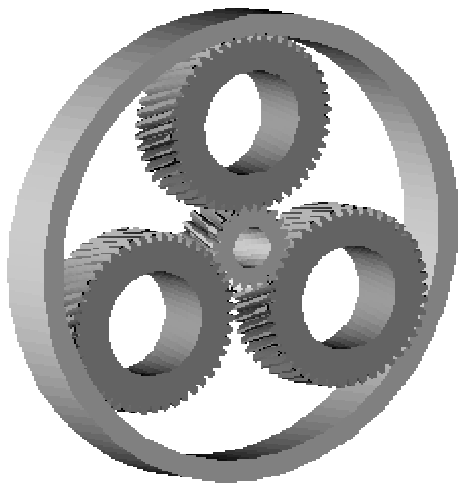


Figure 4: Planet gear developed using Hygears

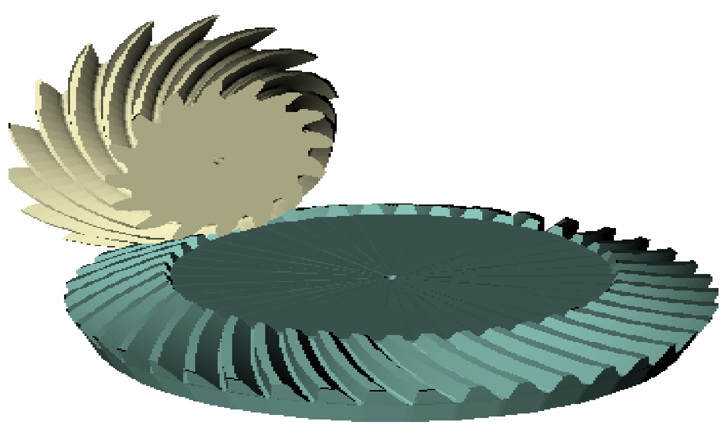


Figure 5: Helical Bevel Gear

3.2 Gear Design

HyGEARS allows the design, analysis and optimization of gear sets through functions such as Tooth Contact Analysis (TCA) and Loaded Tooth Contact Analysis (LTCA).

The initial design of a gear set requires very few data input as numerous default values can be computed as a starting point; these default values can easily be modified later during the design process. The gear set is generated and displayed in WYSIWYG (What You See Is What You Get) format as shown below.

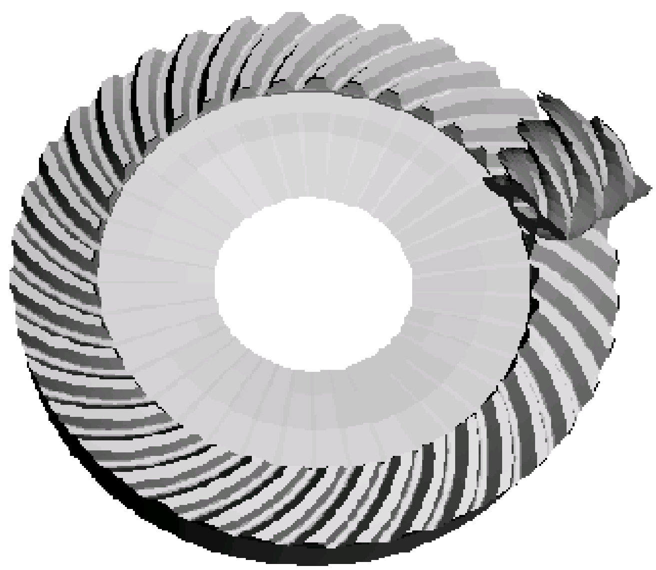


Figure 6: Bevel gear compared to the real manufactured gear with the gear generated using hygear

TCA ANALYSIS

The Transmission Error (TE) and the actual extent of the Bearing Pattern (BP), unloaded and loaded, are readily accessed in graphic form, as shown below. E, P, G, runout, alignment and shaft angle values may also be changed to see how a gear pair will behave in different operating positions as imposed by manufacturing tolerances or housing deformation under load.

In addition, E, P, G, alignment and shaft angle values may be modified in pairs to produce a “map” of the expected behavior of a given gear pair over, say, a given tolerance range.

Shown below is such a grid where E and P are modified in combinations. Obviously, the E-P combinations along the diagonal from top-left to bottom-right are not a problem; however, the E-P combinations along the top-right to bottom-left diagonal are to be avoided.

LTCA ANALYSIS

The load and torque transmitted by neighbouring tooth pairs in mesh can be established through the LTCA optional module. The associated bending and contact stresses, bearing reactions, thermal-EHD oil film thickness, temperature increase and scoring factor can be calculated and assessed in real time to ensure a sound design.

The axial and radial positions of meshing gear pairs, their alignment and shaft angle can be modified to analyze "worst case" conditions, or to automatically produce grid-like projections of the loaded behavior of a gear pair.

For example, in the figure below, the E and P values of a gear pair are modified to produce E-P combinations allowing prediction of the Hertz contact stresses as the pinion and gear relative position, because of gearbox deflection, manufacturing tolerances, or else, displaces the contact pattern.

This Grid-like output can be applied to several combinations of relative positions for gear pairs under load (LTCA). Support bearings and gearbox housing stiffness can be accounted for in the LTCA such that the TE, BP, contact and fillet stresses are calculated at the actual location under load.

CMM, CORRECTIVE MACHINE SETTINGS AND REVERSE ENGINEERING

Target files for Coordinate Measurement Machines (CMM) can be defined and outputted in Zeiss Ram/Rfd, Gear Bevel, Hoeffler and Klinglenberg-P formats (other formats can easily be added).

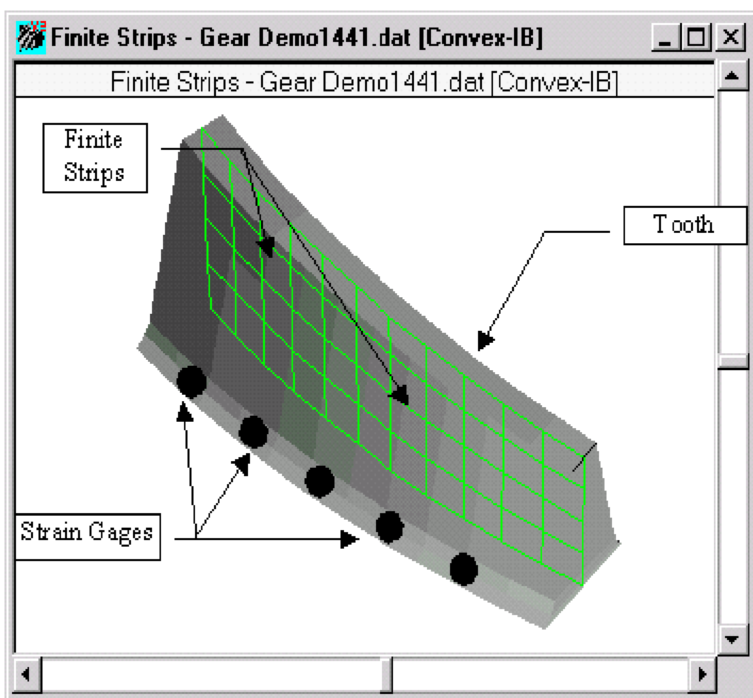
When the diameter of the CMM probing sphere is given, the sphere (white ball in the figure below) can be displayed and animated along the target CMM grid to visually detect possible interference with the opposite tooth flank, or the toot root, as measurement proceeds.

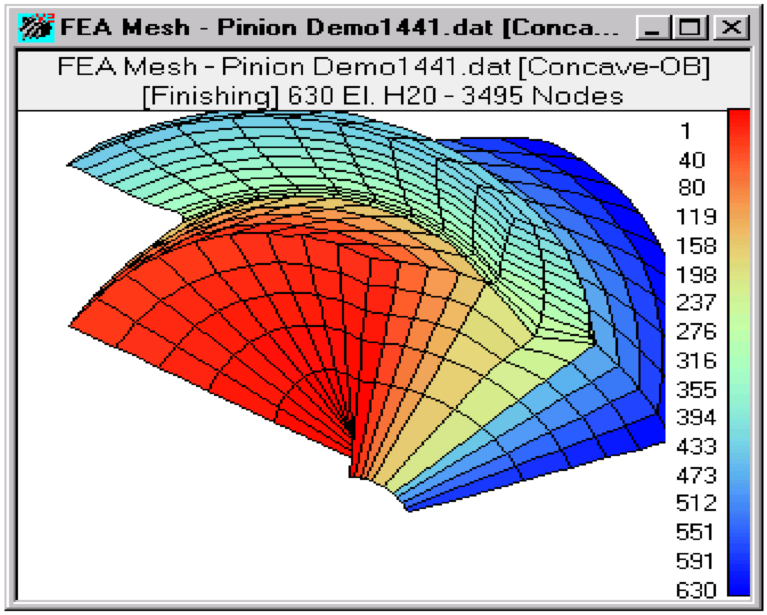
HyGEARS can import any text based CMM output file to calculate Corrective Machine Settings, or to Reverse Engineer existing gearsets.

CMM results can also be used to estimate the TCA and LTCA behaviour of a *real* gear pair, and thus allows the troubleshooting and improvement of problematic gear sets.

FEA PRE-PROCESSING AND FINITE STRIPS

HyGEARS offers advanced functions such as Finite Element Analysis pre-processing (meshing and load application, left figure below), or the analysis of the gears under load using the integrated Finite Strips (right figure below), a subset of the FEA.





CONTACT ELEMENT

An advanced Contact Element is integrated into HyGEARS to allow the precise evaluation of the contact stresses at any point on the tooth surface. Tooth surface irregularities obtained from CMM measurements can be accounted for in the analysis.

UNIVERSAL 5 AXIS CNC MACHINE OUTPUT

Any (face milled) spiral-bevel gear design that is targeted for cutting on a conventional generator can be converted to a Universal 5 Axis CnC machine, given the sign convention of the axes on the CnC machine is known.

For example, below, a partial output for a Universal 5 Axis CnC machine is given for a spiral-bevel pinion manufactured with cutter tilt. In this output, Q and Roll are respectively the cradle and work-piece roll angles on the conventional generator; the X, Y and Z axis are conventional, i.e. X is horizontal and + towards the right when facing the machine, Z is vertical and + upwards, and Y is perpendicular to X and Z and + going into the machine; the B axis supports the turntable, and the C axis is the turntable holding the work piece, and thus provides the roll angle of the work-piece on the CnC machine.

"STEP" OUTPUT

HyGEARS can directly export the topography of the pinion and gear teeth to any STEP capable CAD/CAM system.

Thus, molds for plastic gears, electrodes for sintered gears, or simply CAD models, can conveniently be obtained using the *exact* tooth topography.

For example, the left figure below shows a spiral-bevel pinion tooth in the HyGEARS display; the topography of this tooth was exported in STEP format to a CAD system, which is displayed in the right figure below.

CHAPTER IV

RESULTS

<u>CASE 1</u>

INPUT DATA:

Number Of Teeth, Pinion = 17

Number Of Teeth, Gear = 94

Pressure Angle = 20^0

Pitch Diameter, Pinion = 76.5mm

Face Width, Gear = 43.09mm

Face Width, Pinion = 43.09mm

Module = 4.5

OUTPUT FROM HYGEARS

General Data

<u>Pinion</u>

Minor Diameter = 63.3290

Outside Diameter = 85.5000

Diameter Over Ball = 92.2654

Roller-Ball Diameter = 9.5250

Addendum Factor = 1.0000

Dedendum Factor = 1.1500

Fillet Factor = 0.2500

Addendum = 4.5000

Dedendum = 6.5856

Gear

Number Of Teeth = 94

Face Width = 43.0931

Inner Diameter = 398.04

Pitch Diameter = 423.0000

Outside Diameter = 498.2230

Addendum Factor = 1.0000

Dedendum Factor = 1.2500

Fillet Factor = 0.2500

Addendum = 4.4998

Dedendum = 6.3506

Blank Data

Pinion

Mean Helix Angle (Right) = 0.00.00

Mean Helix Angle (Left) = 0.00.00

Mean Press Angle (Right) = 19.59.28

Mean Press Angle (Left) = 19.59.27

Gear

Mean Helix Angle (Right) = 1.18.30

Mean Helix Angle (Left) = 1.18.29

Mean Press Angle (Right) = 29.07.19

Mean Press Angle (Left) = 29.07.19

Tooth Data

Pinion

Calculated Tooth Depths (Chordal):

Form Depth (Mid-F) = 7.0142

Whole Depth (Mid-F) = 11.6541

Calculated Tooth Depths (Circular):

Form Depth (Mid-F) = 6.8362

Whole Depth   (Mid-F) = 11.0856

Fillet Radius @ Mid-Face:

Drive – Root Diameter = 4.6138

Coast = 4.6160

Drive – Form Diameter = 12.4105

Coast = 12.4161

Fillet Radius Pressure Angle @ Mid-Face:

Drive – Root Diameter = 75.94

Coast = 76.02

Drive – Form Diameter = 0.07

Coast = 0.07

Calculated Blank Diameters:

Root Diam. (Toe) = 63.3290

Tip Diam. (Toe) = 85.5000

Calculated Chordal Tooth Thickness @Mid-Face:

Theo. Finish Thickness = 7.0592

Normal Thick. @ Mean Point = 7.5168

Trans . Thick @ Mean Point = 7.5192

Tooth Topland (Toe) = 2.9261

Tooth Topland (Heel) = 2.9261

Gear

Calculated Tooth Depths (Chordal):

Form Depth (Mid-F) = 11.6627

Whole Depth (Mid-F) = 12.6810

Calculated Tooth Depths (Circular):

Form Depth (Mid-F) = 11.6620

Whole Depth (Mid-F) = 12.6799

Fillet Radius @ Mid-Face:

Drive – Root Diameter = 1.7199

Coast = 1.7199

Drive – Form Diameter = 9.7158

Coast = 9.7150

Fillet Radius Pressure Angle @ Mid-Face:

Drive – Root Diameter = 72.84

Coast = 72.84

Drive – Form Diameter = 29.11

Coast = 29.11

Calculated Chordal Tooth Thickness @Mid-Face:

Thoe . Finish Thickness = 14.0902

Normal Thick . @ Mean Point = 8.4709

Trans . Thick . @ Mean Point = 8.4684

Tooth Topland (Toe) = 4.2355

Tooth Topland (Hoel) = 0.4490

Operating Data

Pinion

Backlash (Min) = 0.1016

Backlash (Max) = 0.1524

Backlash (Cals @M.Point) = 0.1295

Bottom Clearance (Toe) = 2.0856

Bottom Clearance (Heel) = 2.0856

Gear

Bottom Clearance (Toe) : 1.9956

Bottom Clearance (Heel) : 1.9956

Pinion Cutter Specifications

Left and Right

Helix Angle = 0.00.00

Cutter Type = Normal Shaper

Blade Angle = 20.00.00

Blade Edge Radius = 0.0443

Blade Thickness = 0.3105

Addendum = 0.3100

Dedendum = 0.1772

Machine Settings

<u>Left and Right</u>

X Factor = 0.0000

Generating Pitch Dia. = 76.50000

Roll Rate = 0.664052

Tooth Crowning = 0.0450

Crowning Order = 2

Distance To Edge = 10.7733

Gear Cutter Specifications:

<u>Left and Right</u>

Helix Angle = 0.00.00

Cutter Type = Normal Shaper

Blade Angle = 20.00.00

Blade Edge Radius = 0.0443

Blade Thickness = 0.3055

Addendum = 0.7042

Dedendum = 0.1772

Machine Settings:

X Factor = 0.0000

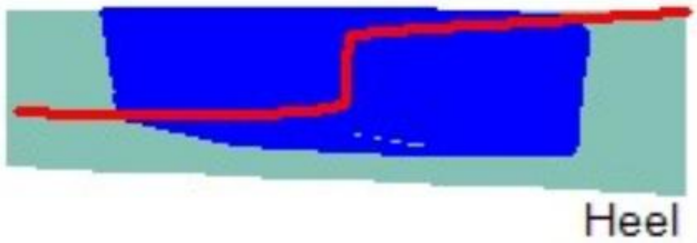
Generating Pitch Dia. = 455.1299

Roll Rate = 0.111617

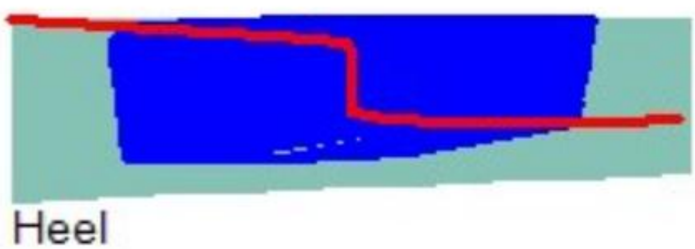
Tooth Crowning = 0.0000

Crowning Order = 1

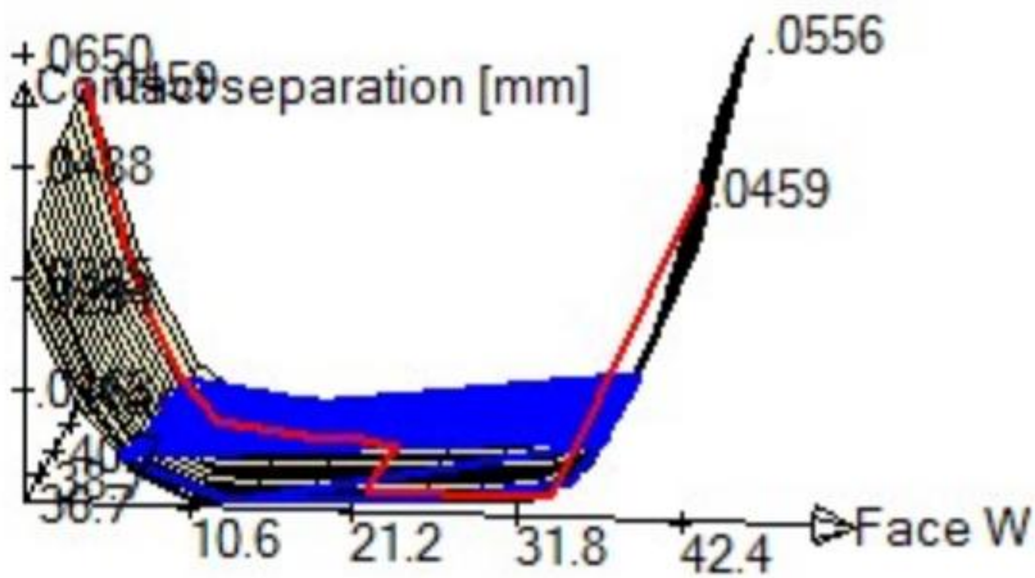
Distance To Edge = 0.0000



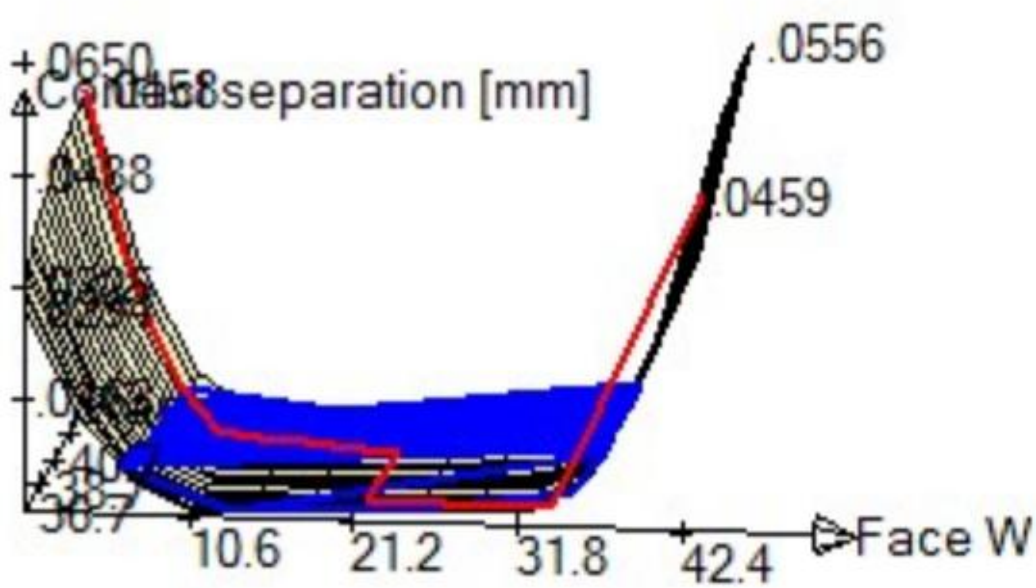
Gear [Finishing] Right [NoEr] E=0.00 P=0.00
G=0.00 [mm] B:0.134 [mm]



Gear [Finishing] Left [NoEr] E=0.00 P=0.00 G=0.00
[mm] B:0.134 [mm]



.0556
Contact separation [mm]
.0459
10.6
21.2
31.8
42.4
Face W



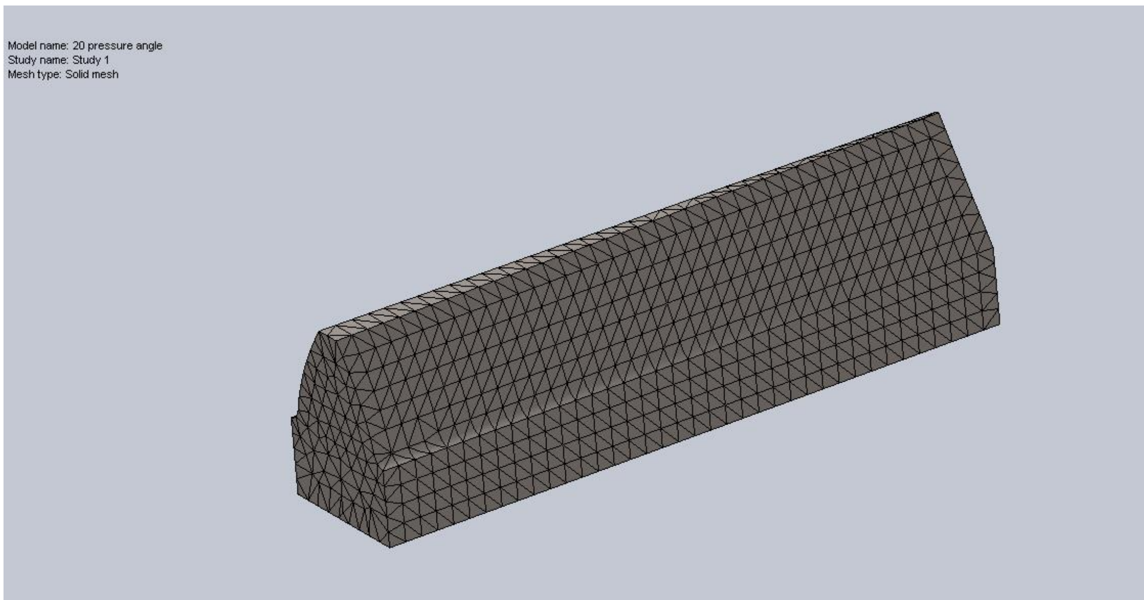
.0556
Contact separation [mm]
.0459
10.6
21.2
31.8
42.4
Face W

Using the above values from the HyGears, the FaceGear has been developed in the SolidWorks 2011 and performed Finite Element Analysis using SolidWorks Simulation.

The load has been applied on the tip of the tooth at an angle of 20^o.

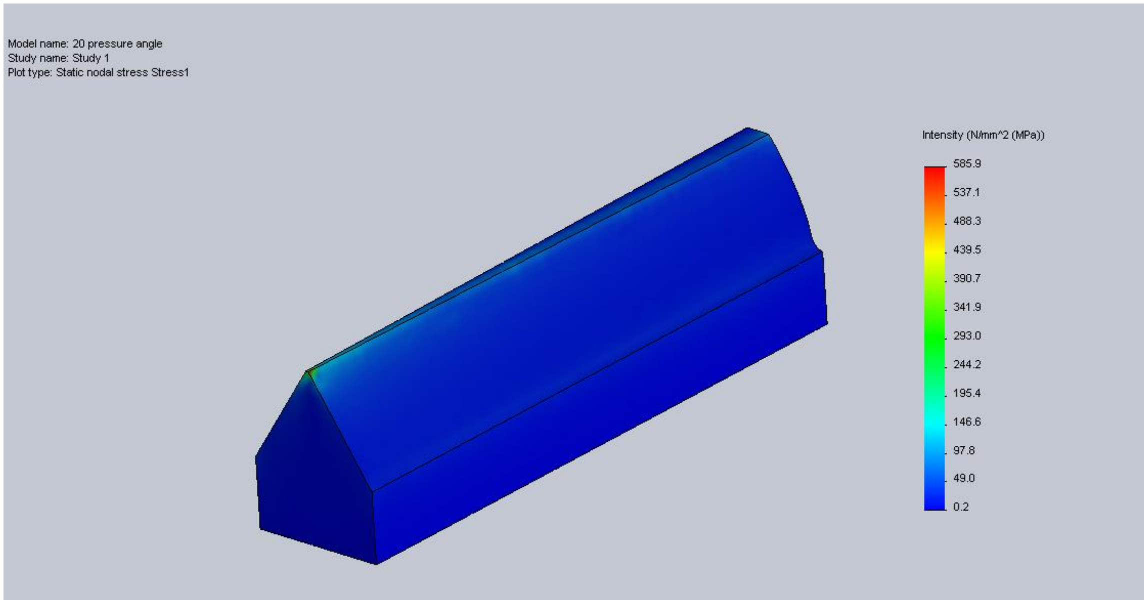
The results are:

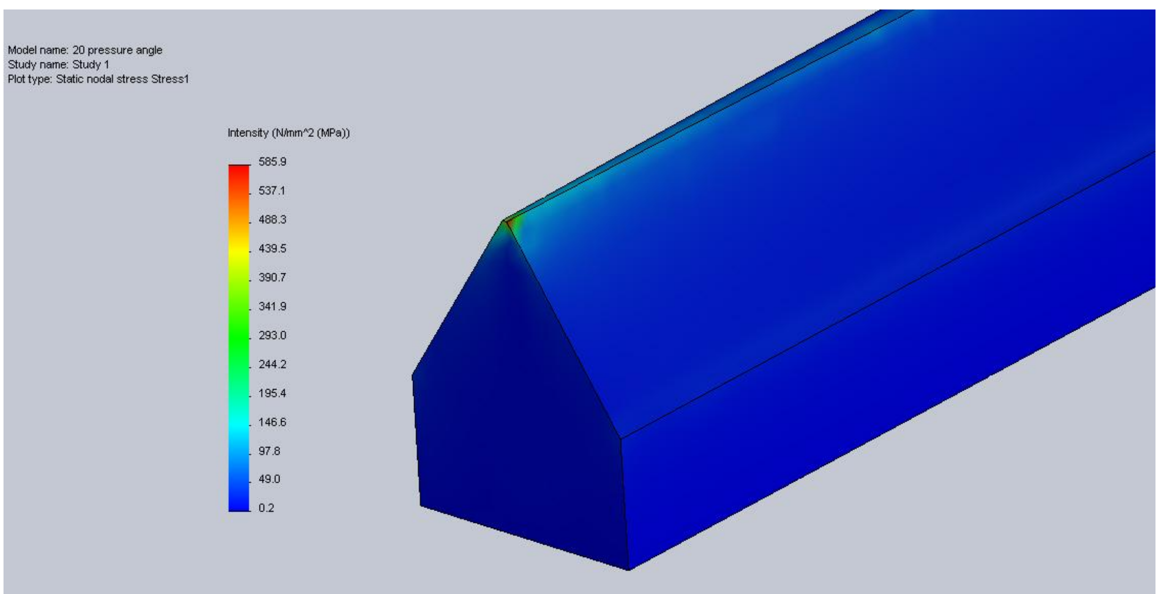
Meshing of Face Gear teeth



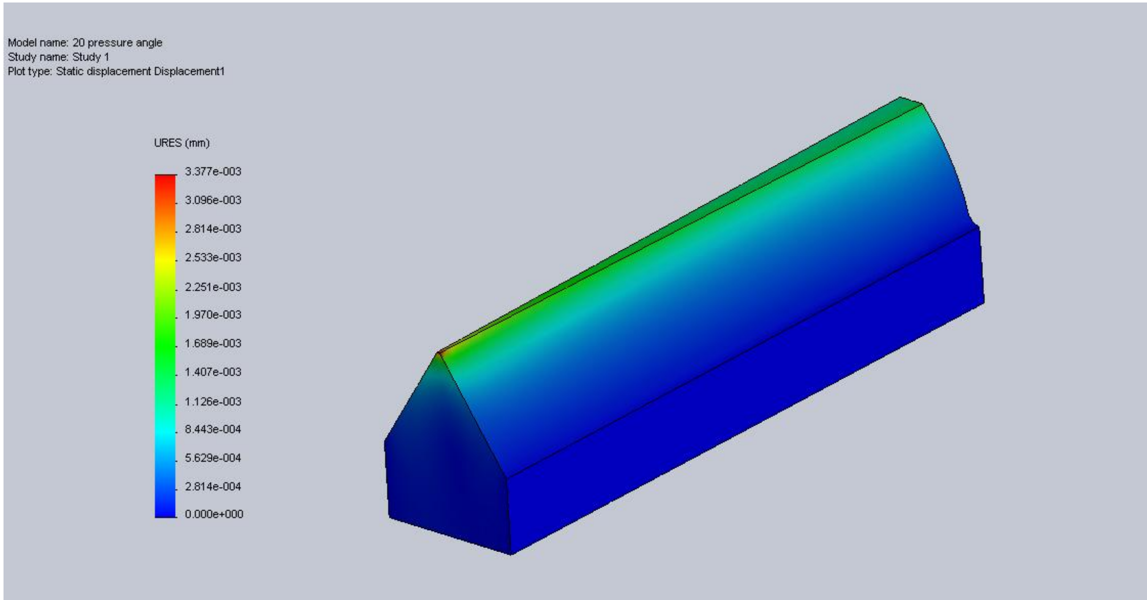
Stress Intensity results:

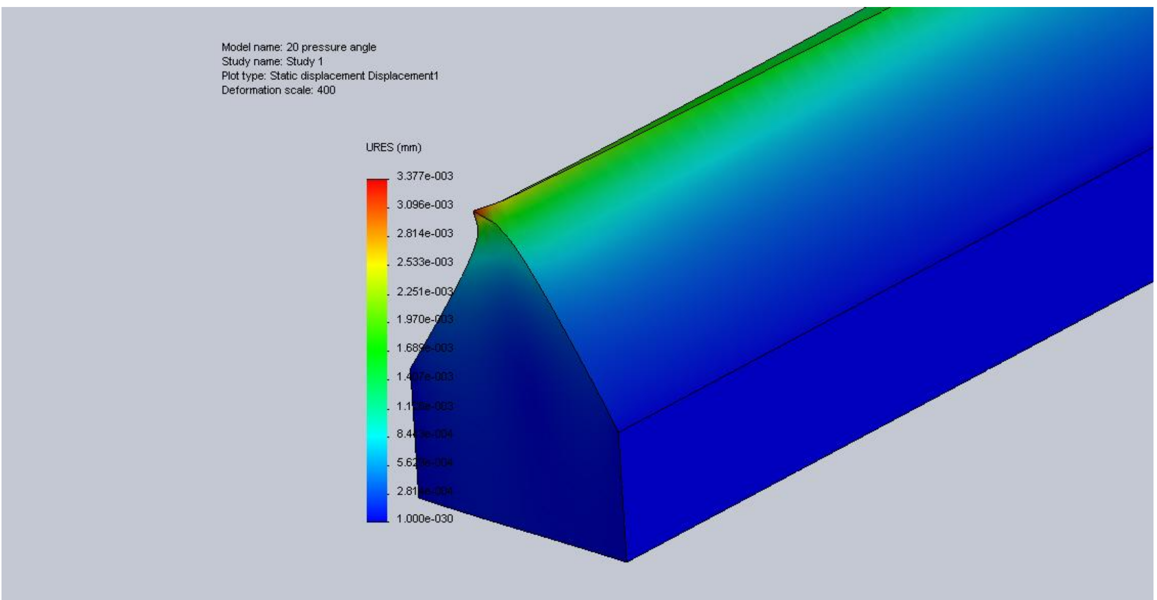
A load of 1200N of force is applied on the tip of the Face gear





Displacement results:





In the figure above 1:400 scale has been taken to have a clear understanding of the Teeth Deformation

<u>CASE 2</u>

INPUT DATA:

Number Of Teeth, Pinion = 17

Number Of Teeth, Gear = 94

Pressure Angle = 25^0

Pitch Diameter, Pinion = 76.5mm

Face Width, Gear = 43.09mm

Face Width, Pinion = 43.09mm

Module = 4.5

OUTPUT DATA FROM HYGEARS:

<u>Pinion</u>

Minor Diameter = 64.8907

Pitch Diameter = 76.5000

Out Side Diameter = 85.5000

Diameter Over Ball = 92.6465

Roller-Ball Diameter = 9.7790

Addendum Factor = 1.0000

Dedendum Factor = 1.1500

Fillet Factor = 0.2500

Addendum = 4.5000

Dedendum= 5.8048

Gear

Inside Diameter = 398.04

Pitch Diameter = 423.0000

Out Side Diameter = 484.2216

Addendum Factor = 1.0000

Dedendum Factor = 1.2500

Fillet Factor = 0.2500

Addendum = 4.5001

Dedendum = 5.4498

Blank Data

Pinion

Mean Helix Angle (Right) = 0.00.00

Mean Helix Angle (Left) = 0.00.00

Mean Helix Angle (Right) = 24.59.32

Mean Press Angle (Left) = 24.59.31

Gear

Mean Helix Angle (Right) = 0.44.48

Mean Helix Angle (Left) = 0.44.49

Mean Helix Angle (Right) = 29.38.29

Mean Press Angle (Left) = 29.38.29

Tooth Data

<u>Pinion</u>

Calculated Tooth Depths (Chordal):

Form Depth (Mid - F) = 7.8586

Whole Depth (Mid - F) = 10.8479

Calculated Tooth Depths (Circular):

Form Depth (Mid – F) = 7.6189

Whole Depth (Mid – F) = 10.3048

Fillet Radius & Mid – Face:

Drive – Root Diameter = 3.7126

Coast = 3.7096

Drive – Form Diameter = 9.8940

Coast = 9.8953

Fillet Radius Pressure Angle & Mid – Face:

Drive – Root Diameter = 71.87

Coast = 71.80

Drive – Form Diameter = 9.47

Coast = 9.47

Calculated Blank Diameters:

Root Diam. (Toe) = 64.8907

Tip Diam. (Toe) = 85.5000

Calculated Chordal Tooth Thickness & Mid – Face:

Theo. Finish Thickness = 7.0588

Normal Thick . &Mean Point = 7.5092

Trans. Thick. & Mean Point = 7.5110

Tooth Topland (Toe) = 2.1006

Tooth Topland (Heel) = 2.1006

<u>Gear</u>

Calculated Tooth Depths (Chordal):

Form Depth (Mid - F) = 10.6913

Whole Depth (Mid - F) = 11.7642

Calculated Tooth Depths (Circular):

Form Depth (Mid – F) = 10.6908

Whole Depth (Mid – F) = 11.7632

Fillet Radius & Mid – Face:

Drive – Root Diameter =1.8584

Coast =1.8584

Drive – Form Diameter = 10.1929

Coast = 10.1950

Fillet Radius Pressure Angle & Mid – Face:

Drive – Root Diameter =72.23

Coast = 72.23

Drive – Form Diameter = 29.60

Coast =29.60

Calculated Chordal Tooth Thickness & Mid – Face:

Theo. Finish Thickness =13.8215

Normal Thick . &Mean Point = 7.5528

Trans. Thick. & Mean Point = 7.5502

Tooth Topland (Toe) = 3.6221

Tooth Topland (Heel) = 0.1341

Operating Data

<u>Pinion</u>

Backlash (Min) = 0.1016

Backlash (Max) = 0.1524

Backlash (Calc&M.Point) = 0.1337

Bottom Clearence (Toe) = 1.3046

Bottom Clearance (Heel) = 1.3046

Gear

Bottom Clearence (Toe) = 1.0961

Bottom Clearance (Heel) =1.0961

Pinion Cutter Specifications

Left and Right

Helix Angle = 0.00.00

Cutter Type = Normal Shaper

Blade Angle = 25.00.00

Blade Edge Radius = 0.0443

Blade Thickness = 0.3196

Addendum = 0.3100

Dedendum = 0.1772

Machine Settings

X Factor = 0.0000

Generating Pitch Dia. = 76.5000

Roll Rate = 0.664052

Tooth Crowning = 0.0450

Crowning Order = 2

Distance To Edge = 10.7733

Gear Cutter Specifications

Left and Right

Helix Angle = 0.00.00

Cutter Type = Normal Shaper

Blade Angle = 25.00.00

Blade Edge Radius = 0.0443

Blade Thickness = 0.3146

Addendum = 0.7042

Dedendum = 0.1772

Machine Settings

X Factor = 0.0000

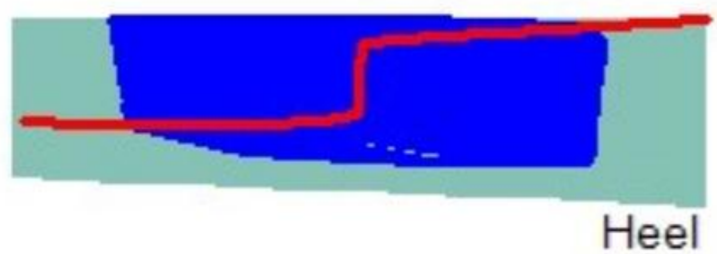
Generating Pitch Dia. = 441.1284

Roll Rate = 0.115159

Tooth Crowning = 0.0000

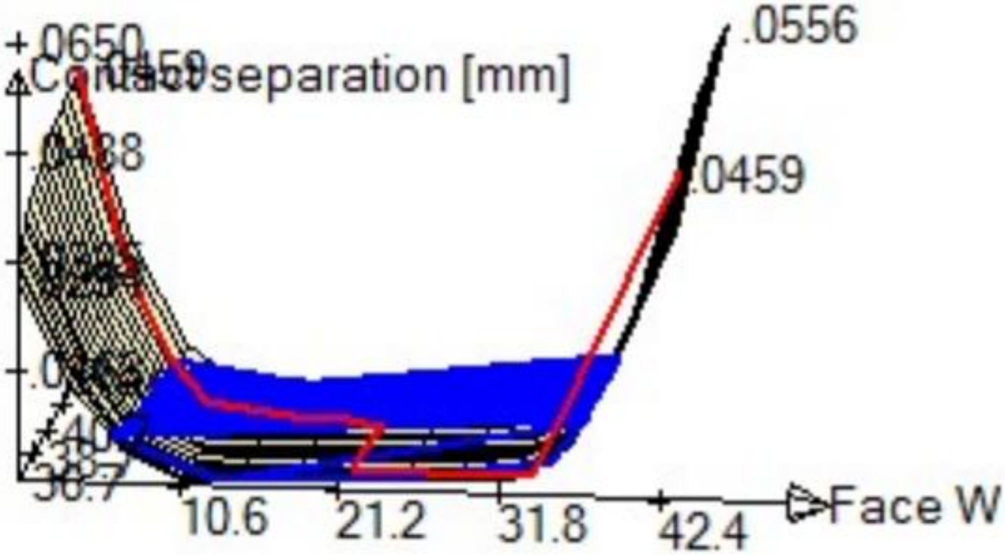
Crowning Order = 1

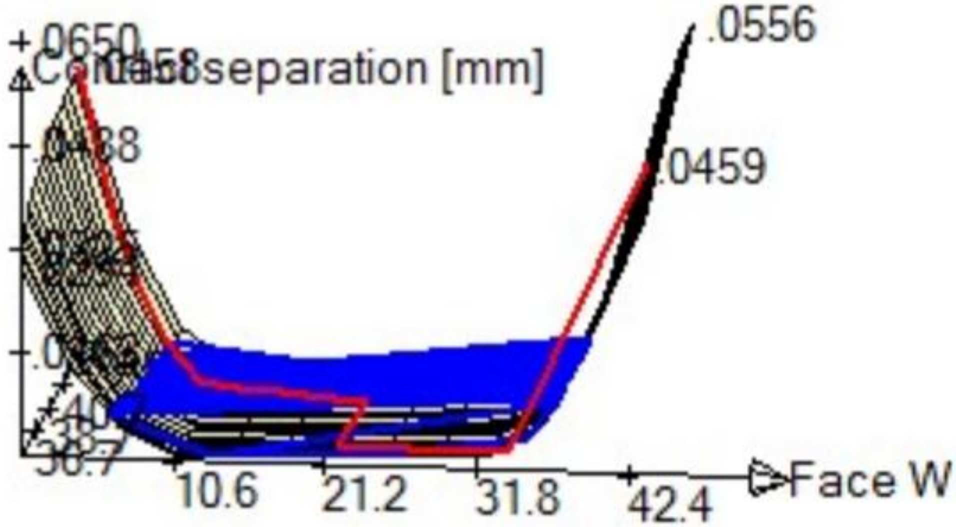
Distance To Edge = 0.0000



Gear [Finishing] Right [NoEr] E=0.00 P=0.00 G=0.00 [mm] B:0.134 [mm]

Gear [Finishing] Left [NoEr] E=0.00 P=0.00 G=0.00 [mm] B:0.134 [mm]


Contact separation [mm]
.0650
.0488
.0556
.0459
10.6
21.2
31.8
42.4
Face W

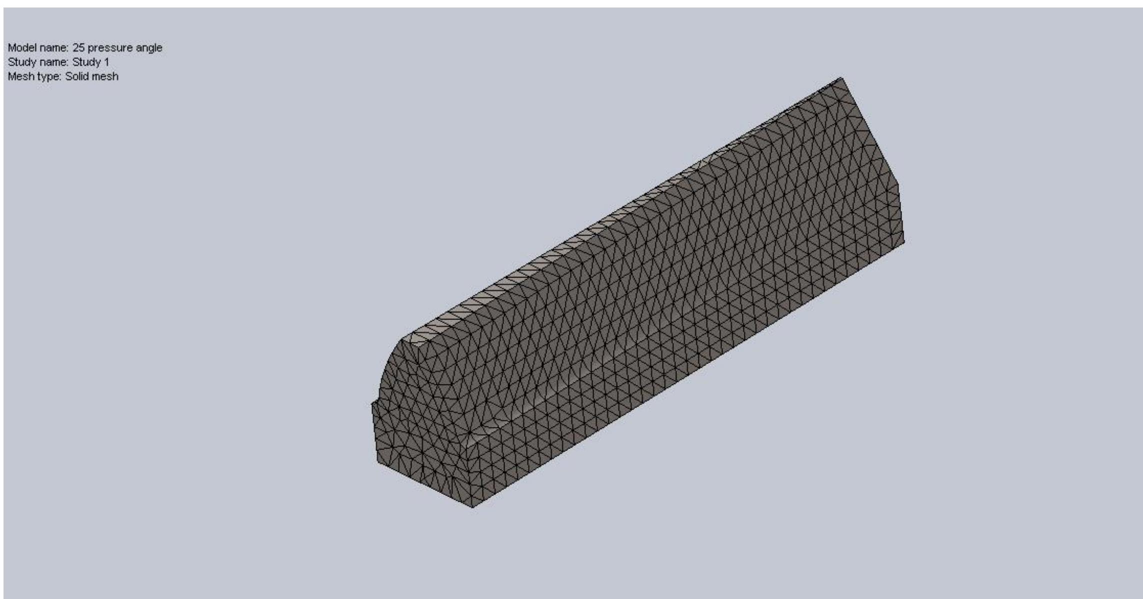

Contact separation [mm]
.0650
.0488
.0556
.0459
10.6
21.2
31.8
42.4
Face W

Using the above values from the HyGears, the FaceGear has been developed in the SolidWorks and performed Finite Element Analysis using SolidWorks Simulation.

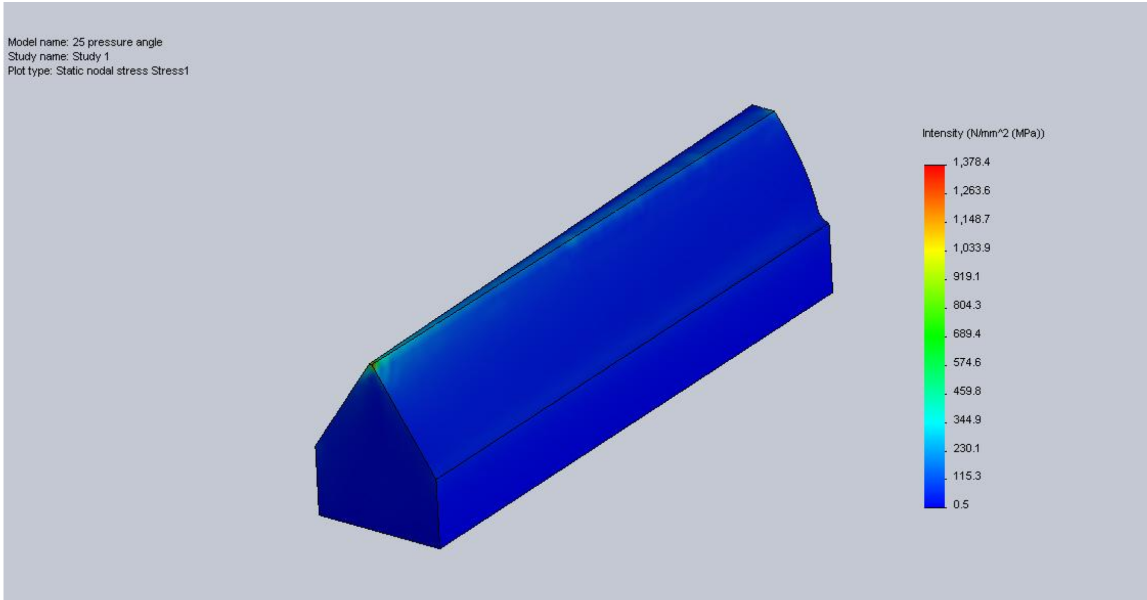
The load has been applied on the tip of the tooth at an angle of 25^{o}.

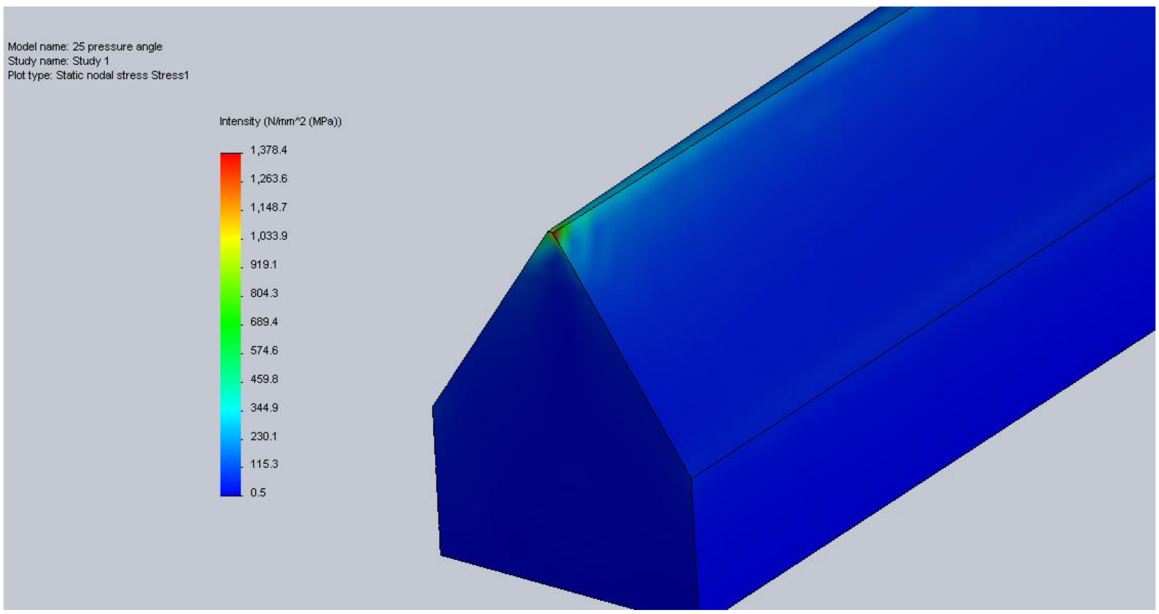
The results are:

After Meshing:

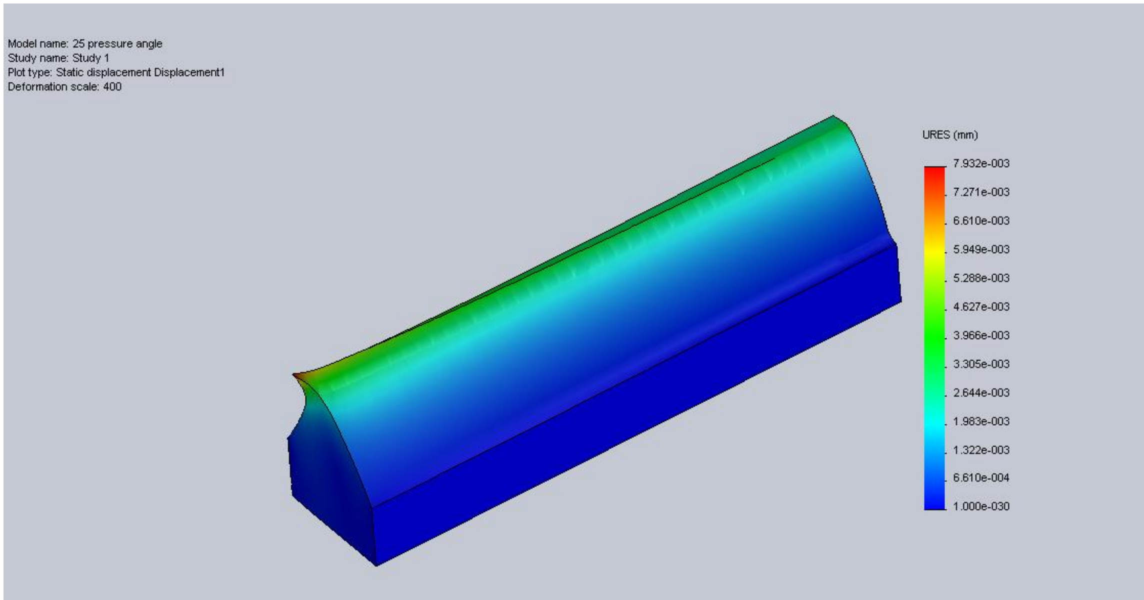


A load of 1200N of force is applied on the tip of the Face gear

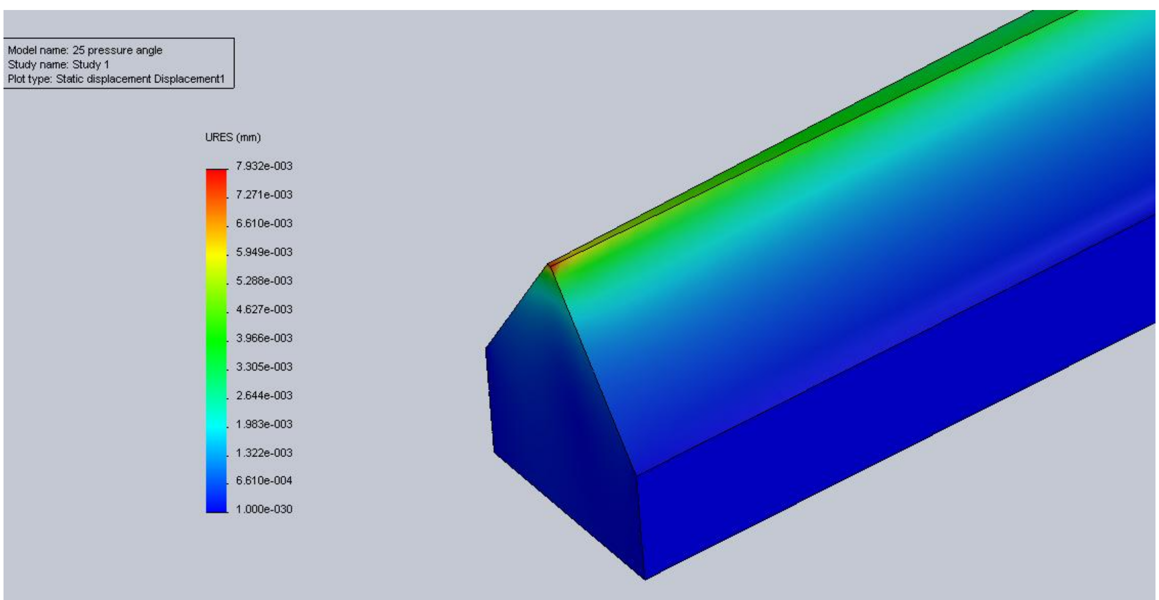




Displacement Results:



In the figure above 1:400 scale has been taken to have a clear understanding of the Teeth Deformation



CHAPTER V

CONCLUSION AND FUTURE SCOPE

5.1 Conclusion

1. Finite Element method can be successfully used to predict bending stress of a typical Face Gear set
2. The percentage error between AGMA formula and from solidworks for an involute spur gear about 7.4%
3. The geometry can be generated with one face(inner face) of a regular involute spur gear and a point edge on the other face(outer face)
4. The bending stress has been obtained from Solidworks Simulation for two different gears of 20^o and 25^o pressure angle.
5. The result of this study can be used to size a face gear set for a given operating condition (transmitted power at a given speed) and the bending stress on the tooth can be determined

5.2 Future Scope

1. Contact stress can also determined for the Facegear
2. Experimental validation of the results predicted by AGMA formula and the Finite element analysis presented in this work
3. Because of the high torque and high loads during operation, considerable amounts of heat is also dissipated. So the heat transfer condition can also be considered in the future analysis of the Facegear.

BIBLIOGRAPHY

[1] Y. Davidov, Non-Involute Gears, Mashgis, 1950 (in Russian).

[2] F. L. Litvin, Theory of Gearing, Nauka, 1968 (in Russian).

[3] S. Beermann and U. Kissling, "Face Gears: Geometry and strength," *Gear Technology*, pp. 54-61, Jan/ Feb 2007.

[4] F. L. Litvin, "Development of gear technology and theory of gearing," NASA Reference Publication 1406, 1997.

[5] "Cylkro Face Gear," Nov/Dec 2010. [Online]. Available: www.geartechnology,com. [Accessed July 2011].

[6] D. W. Dudley, Gear handbook. The design, manufacture, and the application of gears, New York: McGraw- Hill , 1962.

[7] D. Dudley, Handbook of Practical Gear Design, McGraw-Hill, 1984.

[8] F. L. Litvin, Gear Geometry and Applied Theory, Prentice Hall, 1994.

[9] F. L. Litvin, "Handbook of Face Gear Drives with a Spur Involute Pinion," NASA Final Contractor Report, 2000.

[10] L. F. Litvin and A. E. Aleksandra, "Computerized design, generation and simulation of meshing of orthogonal offset face-gear drive with a spur involute pinion with localized bearing contact," *Mech. Math. Theory*, vol. 33, no. 1/2, pp. 87-102, 1998.

[11] B. Bloomfield, "Design of Face Gears," *Mach Des.*, vol. 19, no. 4, pp. 129-134, 1947.

[12] Bloomingfield, "Alignment charts of face gears," in *Product Engineering*, 1952, p. 187.

[13] J. Chakraborty and B. S. Bhadoria, "Some studies on Hypoid Face Gears," *Mechanism and machine theory,* vol. 8, pp. 339-349, 1973.

[14] J. Chakraborty and B. S. Bhadoria, "Design of Face Gears," *Mechanisms,* vol. 6, pp. 435-445, 1971.

[15] D. G. Lewicki and R. F. Handschuh, "Evaluation of Carburized and Ground Face Gears," NASA; US ARL, Quebec, 1999.

[16] C. Zanzi and J. I. Pedrero, "Application od modified geometry of face gear drive," *Elsevier,* vol. 194, pp. 3047-3066, 2005.

[17] F. V. and S. J., "Face gear deisgn factors," in *Gear design and applications*, N. P. Chironis, Ed., McGraw-Hill, 1967.