Design Optimization and Manufacturing of shoulder fillet in Waveguide LR 260 Upper of Transmission System for Stress Concentration: Case Study in VTPL industry, Mumbai

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Abstract— **In this paper frequent failure of a Waveguide LR Upper 260 Transmission member employed in a VFD Motor is studied. Considering the Transmission system, forces and the torques acting on the shaft, Maximum stresses occurring at the failure section are determined. Optimal Notch geometry is obtained for Shaft subject to bending & torsion loading condition. Geometrical features such as notches and corner give increases stress concentrations effect. Notches make localized stress concentration effects that can affect various failure mechanisms, in particular the initiation and expansion of small cracks under fatigue loads, considerably reducing the strength of Transmission components. For obtaining useful variable-radius notches, Mattheck Method is used to Reducing Von-mises stress, Max shear stress and Stress concentration by converting constant radius notch into variable radius notch. . Natural structural members, such as tree bones and branches, after million years of development have learned to utilize variable tip radii as an alternative of the fixed radius. Our results show that stress concentration factors, von misses stress and Max shear stress are effectively reduced by variable radius notch. From components manufacturing point of view, CNC Program is generated for Variable radius notch by using Master cam-x4.**

Keyword - Notch Optimization, Waveguide LR Upper 260, Stress concentration, Mattheck Method, Variable radius notches, Master cam X4

I. INTRODUCTION

A shaft is a rotating member, typically of circular cross-section for transmitting power. It is supported by bearings & supports gear, sprockets, wheels as well as rotors. It is subjected to torsion, or bending loads, acting in single or in combination. Usually shafts are not of regular diameter but are stepped to provide shoulders for locating gears, pulleys and bearings, [14]. In this case study, Waveguide LR Upper 260 Transmission member employed in a VFD Motor is studied. Optimal Notch geometry is obtained for Shaft subject to bending & torsion loading condition. Notches are typically required for operational, structural, or industrialized reasons, or else to reduce weight, so they are in fact a practical require. However, if not correctly designed they can much trouble the local stress and strain field around them, locally rising or concentrating the nominal stresses. Such localized stress concentration effects depend on the notch geometry and on the loading environment, and can much reduce the real component strength.

A material independent stress concentration factor (SCF) defined by

$$kt = \frac{\sigma max}{\sigma nom}$$

Where; σmax is the maximum stress acting at the notch and σnom is the nominal stresses that would act there if the notch had no effect on the stress field that surroundings it. The aim of this study is first to compare the efficiency of both constant radius and variable radius for design better notch profiles, and then to analyze the SCF improvement attainable by optimizing the variable radii of notches for torsion and bending load applications, using the FEA method.

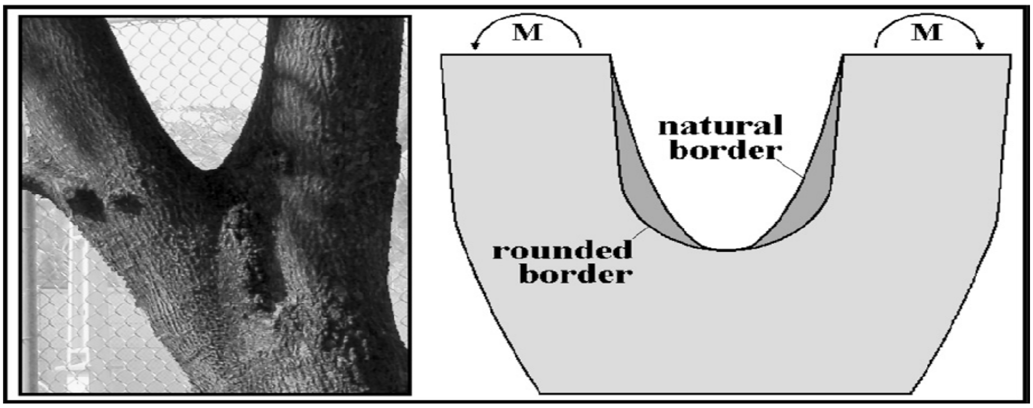

Fig.1. Natural notches usually do not have a constant radius tip to minimize stress concentration, [12], [13]

Mattheck projected an easy process for getting better notch shapes based on the apparent self-growth mechanism of tree branches. SCF can be reduce by shaping their contour to alternative bent force lines around their borders by stretched ones, the major plan behind the ‘‘method of tensile triangles” for getting better the notch geometry. So, according to Mattheck, sharp corner-like notches can be reinforced by a tensile triangle to decrease its local stresses, creating two fresh notches with less dangerous larger angles, and be further smooth starting with a 45 degree rectangular triangle and them consecutively adding obtuse isosceles triangles to strengthen the weaker notch side. The extra triangles, the lesser are the Kt of the better notch. [12], [13], [15]

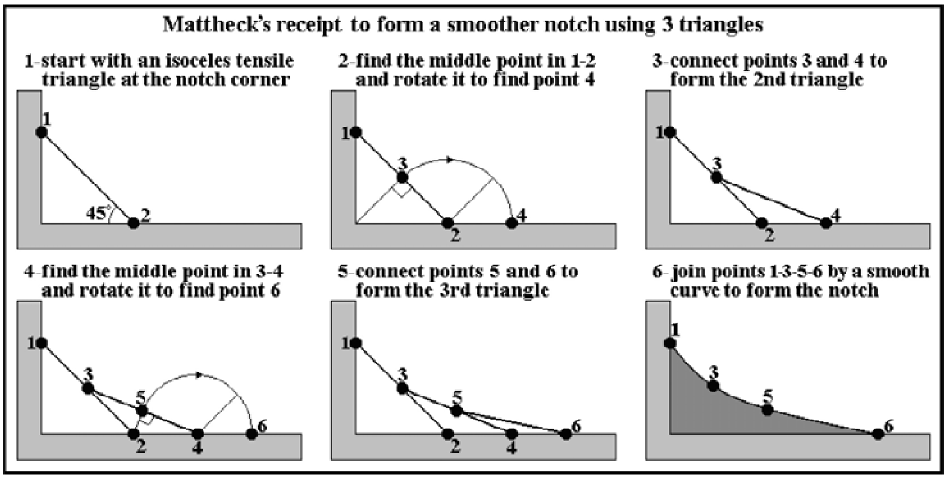

Fig. 2. Mattheck's tensile triangle method to improve notch profiles [12], [13]

II. PROBLEM DESCRIPTION

The Waveguide LR 260 Upper is used as transmission member in VFD motor. Transmission Member is fails due to combine effect of bending and torsion during working. During operation it was observed that the Waveguide LR 260 upper is bending and/or twisted after certain periods of working cycle. So the industry has to replace the Failure Waveguide LR 260 Upper which is not cost effective.

Fig -3.Waveguide LR Upper 260 rejected part sample

III.BASIC ARRANGEMENT OF DRIVE SYSTEM

The shaft examined in this paper failed after 35 to 45 days of installation. The basic drive arrangement is shown in (Fig. 4), applied with 5600 N force in vertical direction. The drive arrangement consists of a 7.1 Kw servo motor and belt drive for power transmission. The function of this arrangement is to rotate the Load drum. The load drum rotates at 900 rpm. The geometry of the shaft with its dimension is shown in Fig. 5. The material of the shaft under consideration is stainless steel 304. Since the shaft is working in a USA the operating temperature is at around Avg. 21.5°C. It can be seen that the failure has taken place at the shoulder fillet that has been provided in the shaft were the stress concentration would be more due to abrupt change in cross section of the shaft.

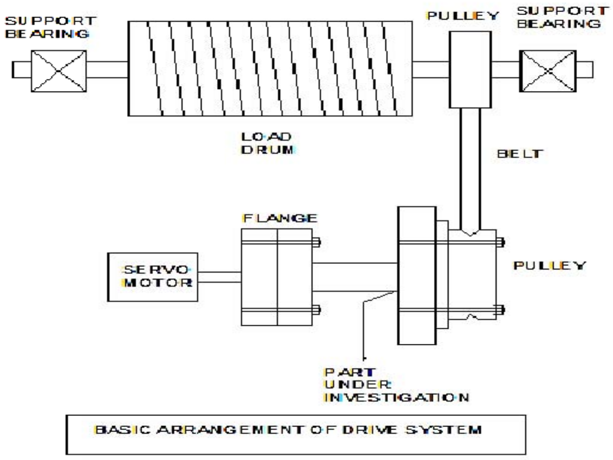

Fig.4. Basic Arrangement of Drive System

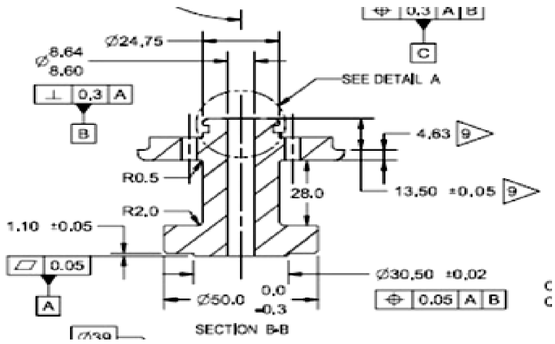

Fig.5. Shaft and its Dimensions

IV.METHDOLOGY

- Observing and comparing the Von-Misses stress, shear stress and Stress concentration of Constant Radius fillet with Variable Radius fillet.
- Comparing the Stress analysis results by using the Theoretical and FEA.
- Mattheck Method is used for converting Constant Radius into Variable Radius to reduce the effect of stress concentration.
- Master cam X4 is used to generate CNC Programmed for manufacturing point of view.

V. DESIGN AND OPTIMIZATION

Design of shaft is quite key aspect in any machines or mechanical system. Geometry of shaft plays a significant function and its mounting conditions are also considered while designing the shaft. The strength of shaft is a key aspect; it will decide that how much load a shaft can hold up. Based on loading stresses are calculate and then using those standard equation of stresses the design is completed. Another aspect to design the shaft is on rigidity basis. A shaft is a beam that deflects transversely and it is a torsion bar that deflects torsionally. Both mode of deflection are analyzed. The allowable angle of twist is to be calculated while designing the shaft on torsional rigidity bases.

A. *Design of shaft*

From industrial data: P=7.1 kW ; N=900 RPM; Material and their properties: Material of shaft of gear motor is Stainless steel: Sut/Syc =515Mpa; Sys =205Mpa; (as considering Factor of Safety i.e. F.S = 1), [2], [4], [6]

Therefore Tensile /Bending stresses in shaft, [2], [4], [6]

$$\sigma b = \frac{sut}{f.S} = 515\ mpa$$

And Shear stresses in shaft

$$\tau = \frac{sys}{f.s} = 205\ mpa$$

Servo motor Power, [2], [4], [6]:

P= $\frac{2\pi NT}{60}$

7.1 = (2π×900×T)/60

71.6 N-M

T= 71600 N-mm

Torque on pulley: Ratio for belt tension for pulley is 2.5

$\frac{FB\ 1}{FB2}$ =2.5

FB1+FB2 =5600N

T= (FB1-FB2) × (d/2)

71600= (2.5FB2-FB2) × (59.7/2)

71600=1.5×FB2×29.85

FB2=1599.10 N

FB1=3997.77N

Maximum bending moment of shaft:

M= F×L

=5600×41.5

M=232400 N-mm

Failure of shaft under twisting moment, [2], [4], [6]:

AS shaft is a step shaft having maximum diameter 60 mm and minimum diameter of 24.75 mm, so we consider minimum diameter for failure i.e. 24.75 mm

Equivalent torque;

$$Te = \sqrt{(M^2 + T^2}$$

$$Te = \sqrt{232400^2\ + 71600^2}$$

Te=243179.60 N-mm

$$\tau = \frac{16 \times Te}{\pi \times (do^3) \times [1 - k^4]}$$

Where; di = inner diameter & do=outer diameter

K= $\frac{di}{do}$

$$\tau = \frac{16 \times 243179.60}{\pi \times (24.75^3) \times [1 - 0.35^4]}$$

Shear stress.. $\tau = 82.93\ mpa$

Failure of shaft under Bending, [2], [4], [6]:

$$\sigma b = \frac{32 \times M}{\pi \times (do^3) \times [1 - k^4]}$$

$$\sigma b = \frac{32 \times 232400}{\pi \times (24.75^3) \times [1 - 0.35^4]}$$

Bending stress.. $\sigma b = 158.51\ mpa$

Now, using Maximum Principle /normal stress Theory, [2], [4], [6]:

$Max.\sigma b = \frac{\sigma b}{2} + \frac{1}{2}\sqrt{\sigma b + 4\tau^2}$

$Max.\sigma b = \frac{158.51}{2} + \frac{1}{2}\sqrt{158.51 + 4(82.93^2)}$

$Max.\sigma = 193.96\, mpa$

Minimum principle stresses are, [2, 4, and 6]:

$Max.\sigma b = \frac{\sigma b}{2} - \frac{1}{2}\sqrt{\sigma b + 4\tau^2}$

$Max.\sigma b = \frac{158.51}{2} - \frac{1}{2}\sqrt{158.51 + 4(82.93^2)}$

$Min.\sigma = -35.46\, mpa$

Thus the corresponding maximum shear stress is, [2], [4], [6]:

$Max.\tau = \frac{1}{2}\sqrt{\sigma b + 4\tau^2}$

$Max.\tau = \frac{1}{2}\sqrt{158.51 + 4(982.93)^2}$

$Max.\tau = 114.71\, mpa$

Von- Misses Stresses are, [2], [4], [6]:

$\sigma\ mises = \sqrt{\sigma 1^2 - \sigma 1 \times \sigma 2 + \sigma 2^2}$

$\sigma\ mises = 213.9\, mpa$

B. *Design for stress concentration*

Diameter of larger shaft section=D =40 mm

Diameter of smaller shaft section=d=24.75

Radius=r=2mm

Bending moment=M=232.4 N.m

Torque=T=71.6 N.m

Height =h=12.62 mm

Stress concentration factor in bending, [3], [5], [7]:

For ration, 2.0 < h /r < 20.0

$C_1 = 1.232 + 0.832\sqrt{\frac{h}{r}} - 0.008\ \frac{h}{r}$

$C_2 = -3.813 + 0.968\sqrt{\frac{h}{r}} - 0.260\ \frac{h}{r}$

$C_3 = 7.423 - 4.868\sqrt{\frac{h}{r}} + 0.869\ \frac{h}{r}$

$C_4 = -3.839 + 3.707\sqrt{\frac{h}{r}} - 0.600\ \frac{h}{r}$

$Kt = C_1 + C_2\,(2h/D) + C_3\,(2h/D)^2 + C_4\,(2h/D)^3$

$\sigma_{nom} = \frac{32\,M}{\pi\, d^3}$

$\sigma_{max} = Kt \times \sigma\ nom$

Stress concentration factor = kt =1.89

Nominal tension stress at shaft = σ_{nom} = 156.14 mpa

Maximum tension stress due to bending = σ_{max} = 294.57 mpa

Stress concentration factor in torsion, [3], [5],[7]:

For ratio, 0.25 < h /r < 4.0

$C_1 = 0.905 + 0.783\sqrt{\frac{h}{r}} - 0.075\ \frac{h}{r}$

$C_2 = -0.437 - 1.969\sqrt{\frac{h}{r}} + 0.553\ \frac{h}{r}$

$C_3 = 1.557 + 1.073\sqrt{\frac{h}{r}} + 0.086\ \frac{h}{r}$

$C_4 = -1.061 + 0.171\ \sqrt{\frac{h}{r}} + 0.086\ \frac{h}{r}$

$Kt = C_1 + C_2\,(2h/D) + C_3\,(2h/D)^2 + C_4\,(2h/D)^3$

$$\tau_{nom} = \frac{16 \times T}{\pi\, d^3}$$

$\tau_{max} = Kt \times \tau\ nom$

Stress concentration factor = kt =1.51

Nominal shear stress at shaft = τ_{nom}=24.05 mpa

Maximum shear stress due to torsion = τ_{max} =36.27 mpa

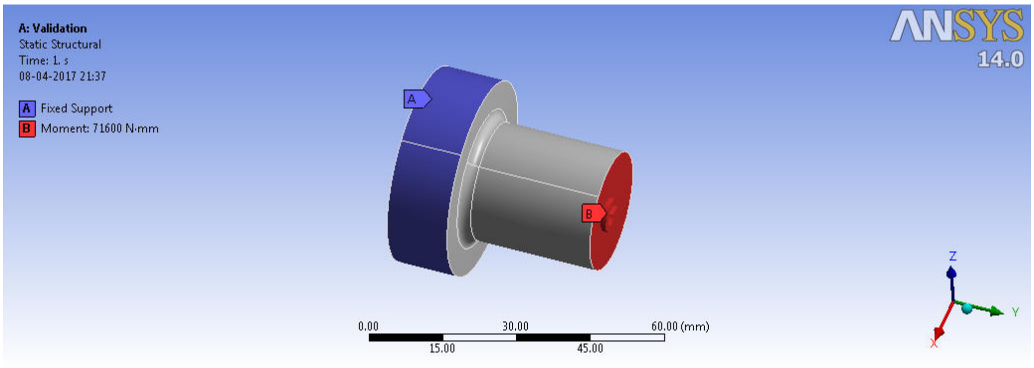

Fig.6. Boundary condition- Software Validation

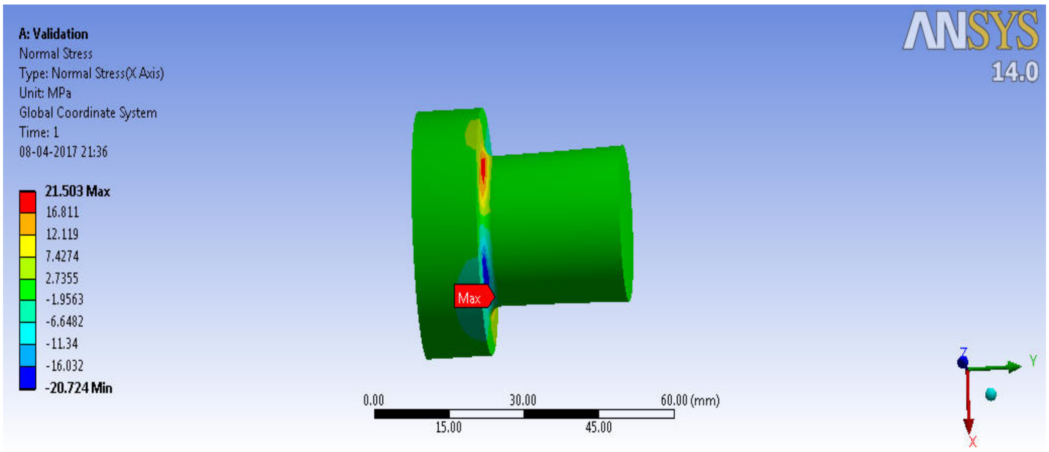

Fig.7.Normal Stresses- Software Validation

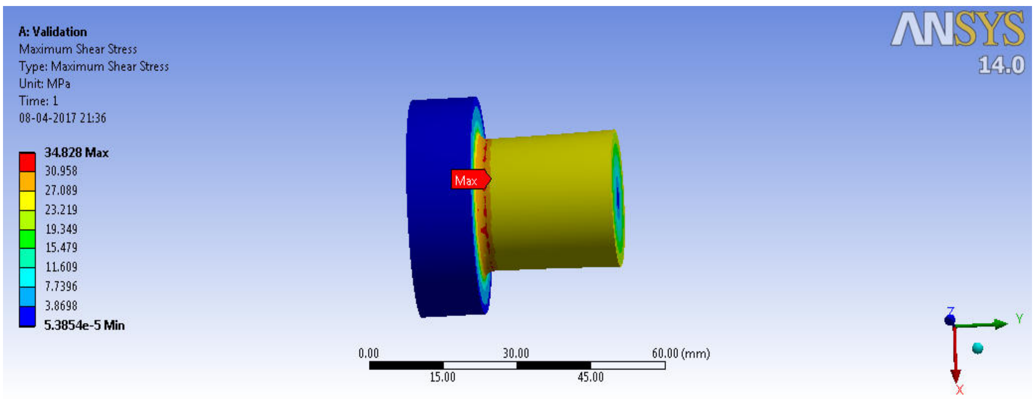

Fig.8.Maximum shear stress- Software Validation

VI. FEA ANALYSIS

A. *Constant radius geometry*

First, we prepare a model of shaft in Catia v5 software and save as .IGES file format for Analysis of shaft in ANSYS WORKBENCH 14. Import .IGES model in ANSYS Workbench simulation module. Material type:- stainless steel 304; Poisson ratio: - 0.3; Modulus of Elasticity: - 193 GPA; Modulus of rigidity G=74.23 Gpa =E/ (2(1+v))

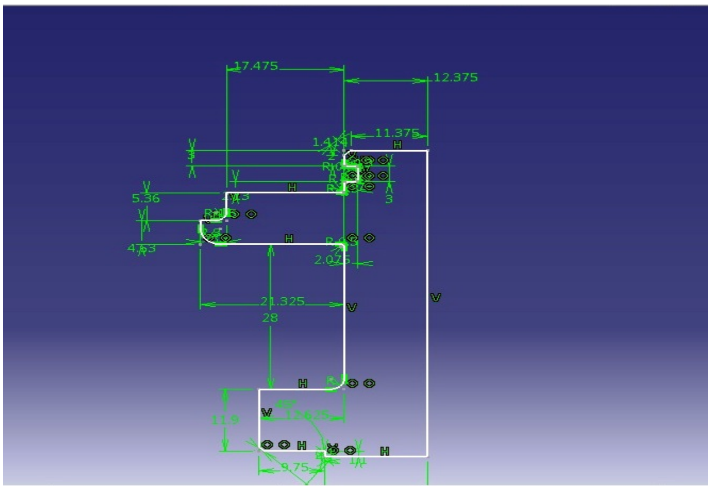

Fig.9.Catia v 5 geometry of waveguide LR 260

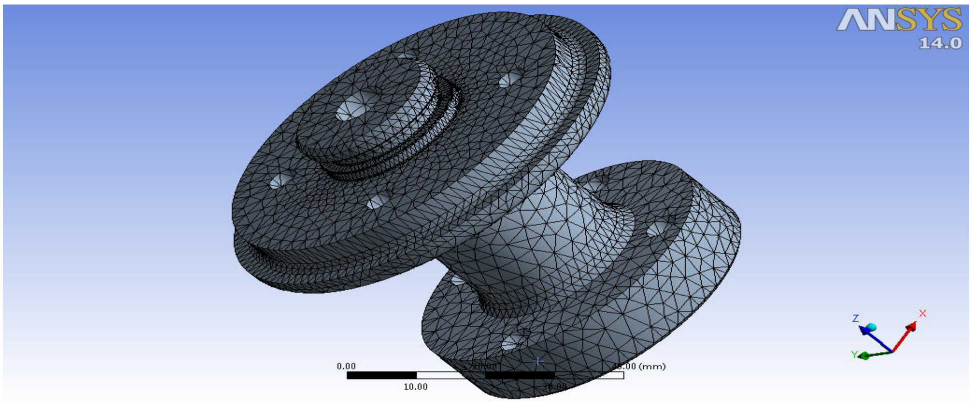

Fig.10.Meshing

Define boundary condition for analysis:

Fixed support applies on one side of shaft, Force (5600 N) Tangential is apply on one side of shaft & Moment (71600 N.mm) is apply on one side of shaft.

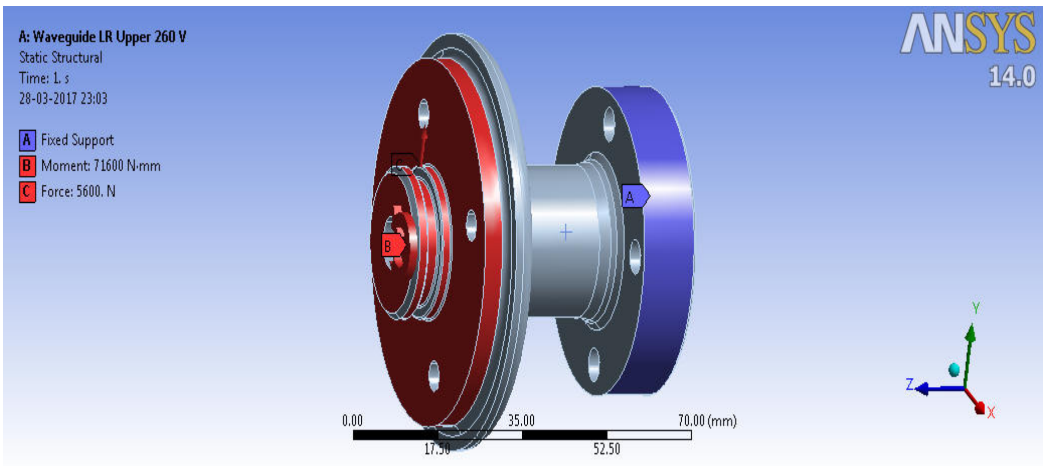

Fig.11.Define boundary condition for analysis

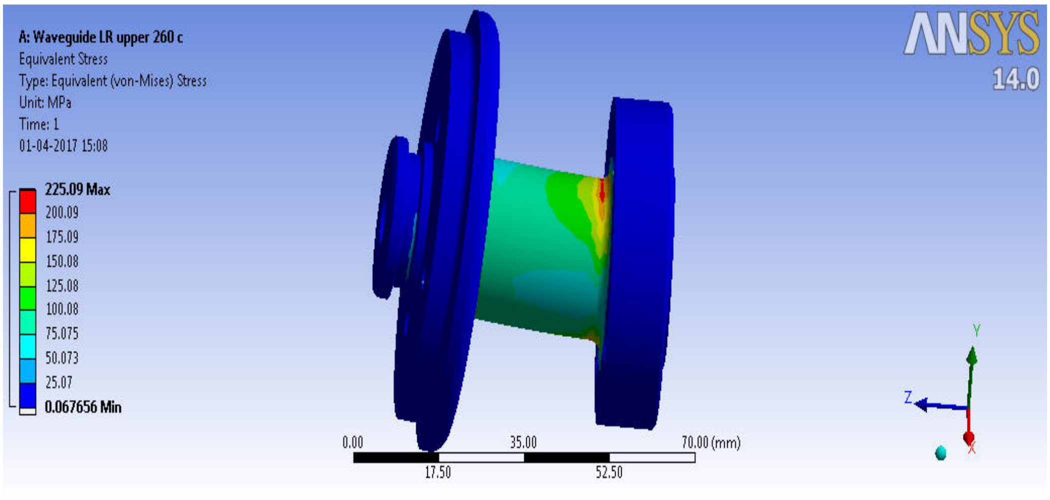

Fig.12. Von-Misses Stress analysis

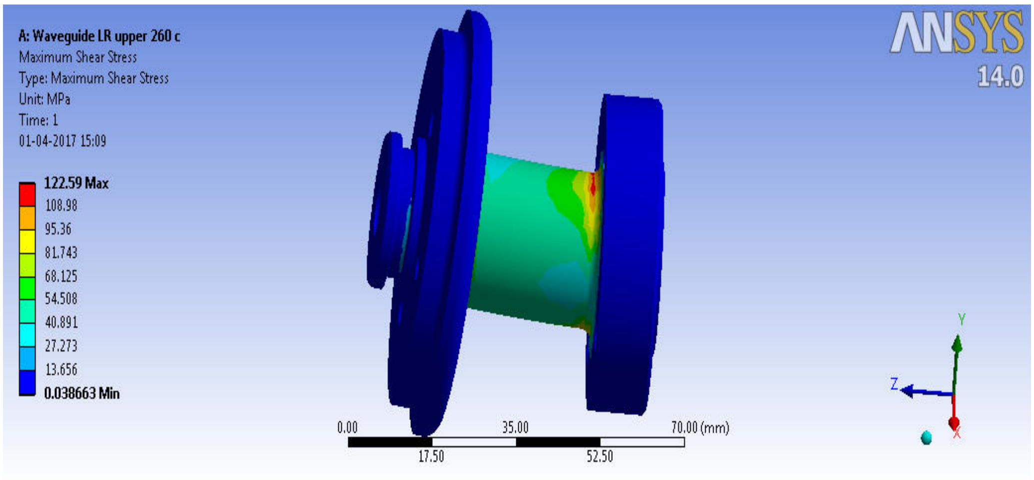

Fig.13. Shear stress analysis

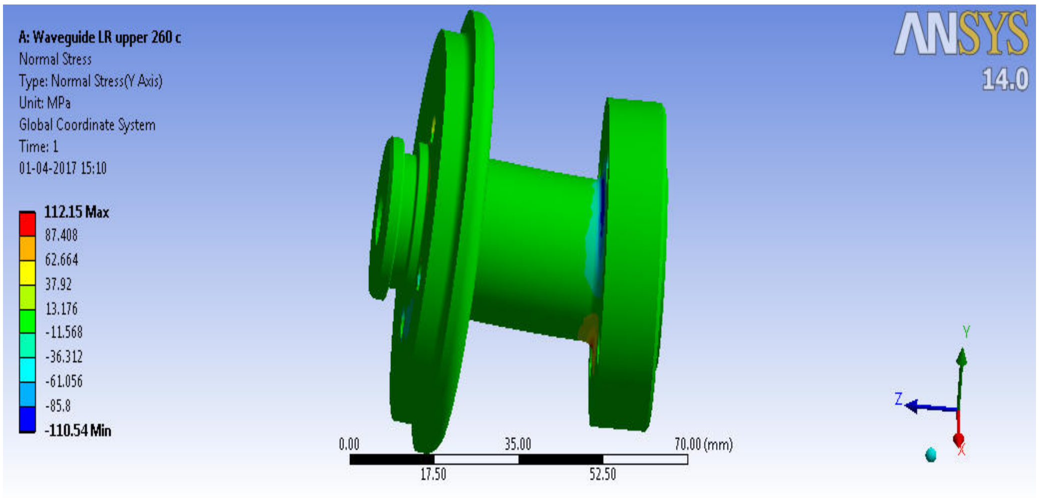

Fig.14.Nominal Stresses

TABLE I. Comparison between Theoretical and FEA Analysis stress (Mpa) values

Sr. No.	Types of stress	Theoretical stress	FEA Analysis	% error
1	Von-Misses stresses (mpa)	213.9	225.09	4.97%
2	Shear stresses (mpa)	114.71	122.59	6.42%
3	Kt σ	1.34	1.42	5.63%
4	Kt τ	1.38	1.47	6.12 %

B. *Modified variable radius geometry by Mattheck method*

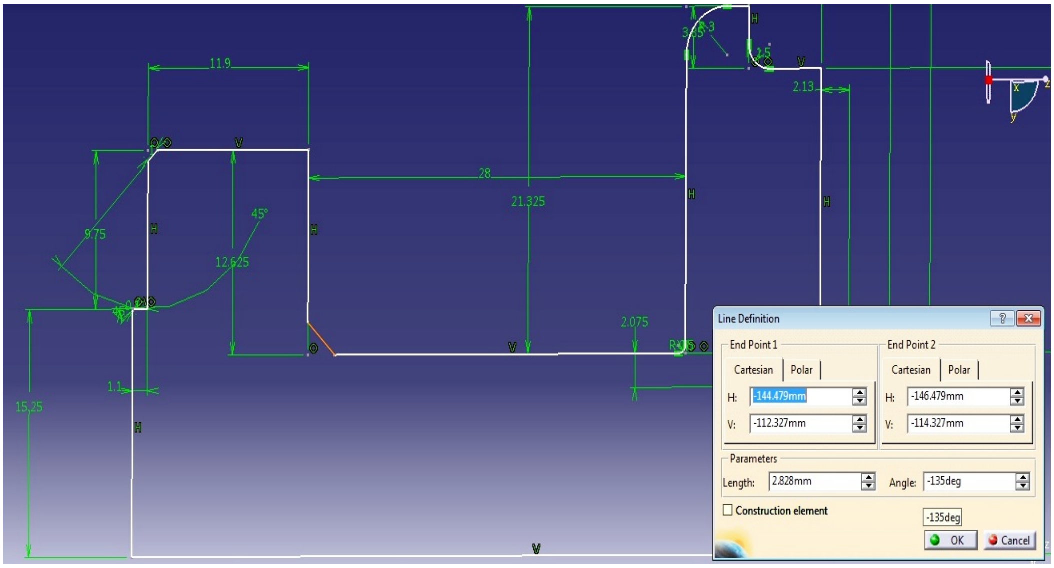

Fig.15.Variable Radius geometry in catia v5-a

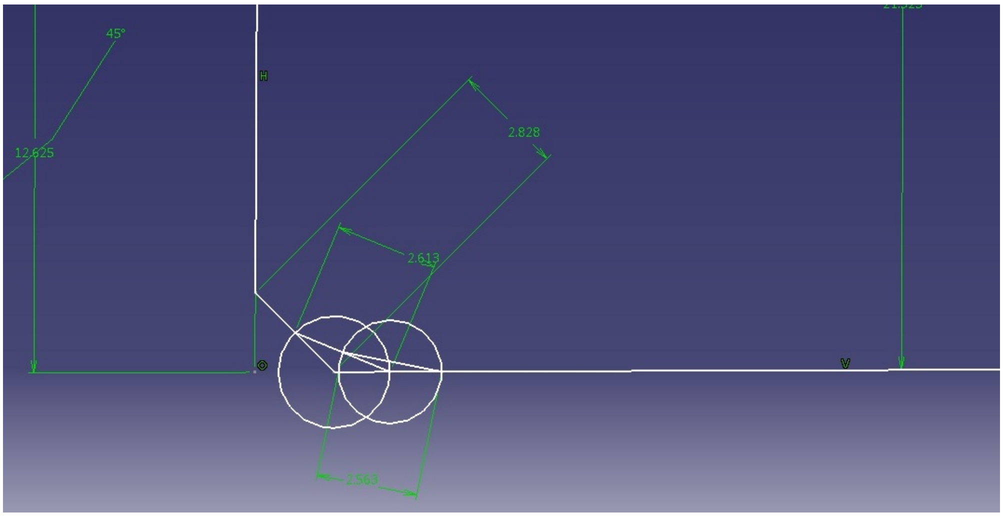

Fig.16. Variable Radius geometry in catia v5-b

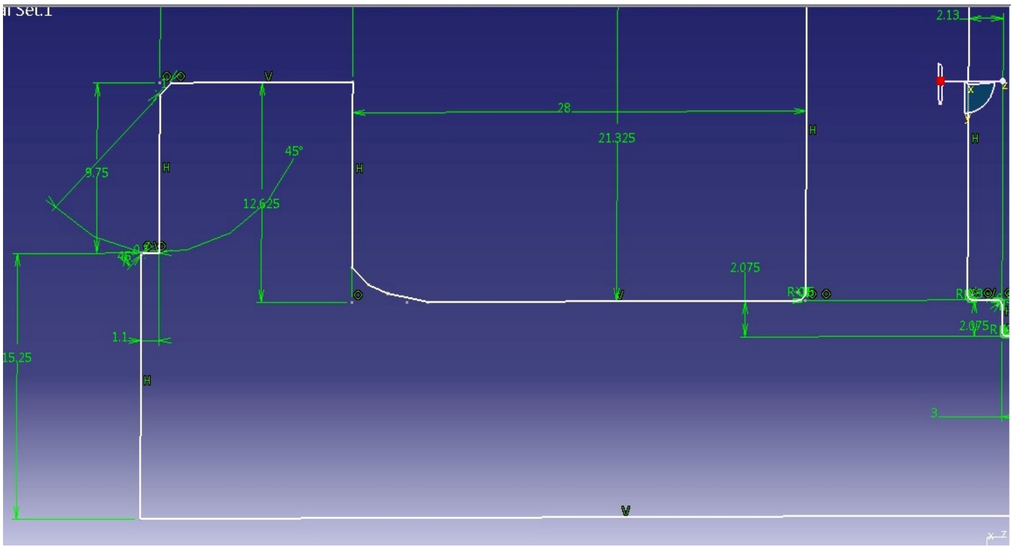

Fig.17. Variable Radius geometry in catia v5-c

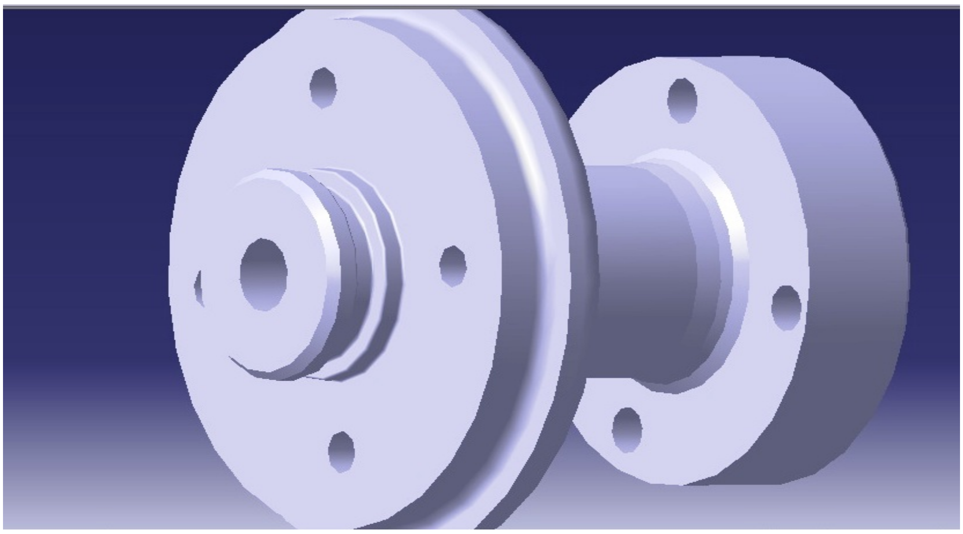

Fig.18. Variable Radius geometry in catia v5-d

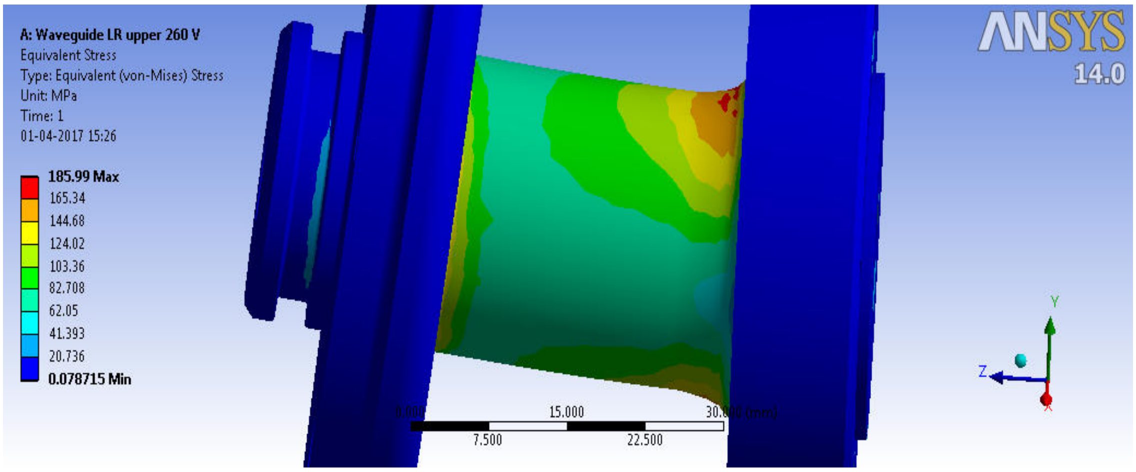

Fig.19. Von-Misses Stress analysis- Variable Radius geometry

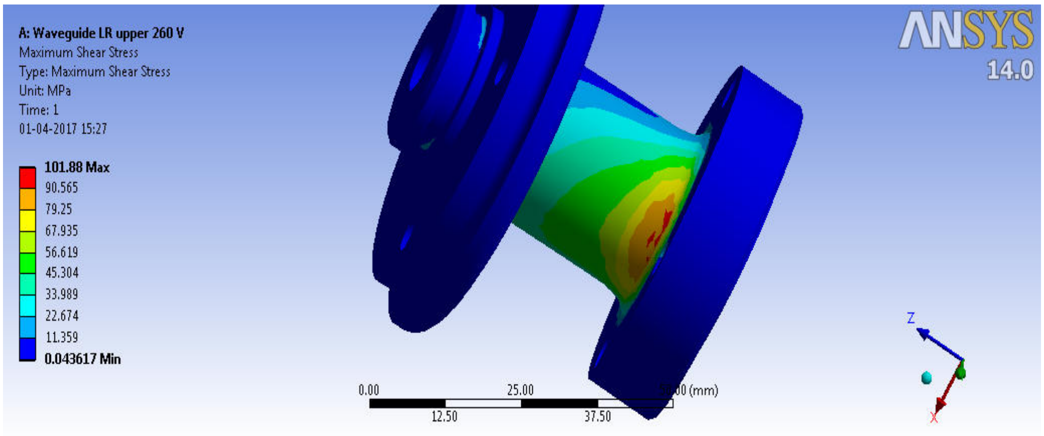

Fig.20.Maximum shear stress analysis- Variable Radius geometry

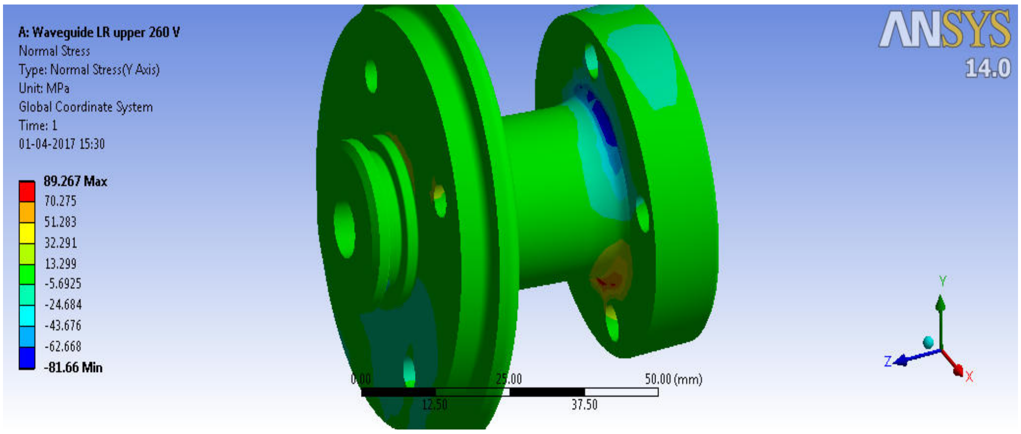

Fig.21.Nominal Stresses- Variable Radius geometry

TABLE II. Comparison between Constant Radius and Variable Radius fillet by FEA Analysis

Sr. No.	**Types of stress**	**Constant Radius Fillet**	**Variable Radius fillet**	**% stress reduced**
1	Von-Misses stresses (mpa)	225.09	185.99	17.37 %
2	Maximum Shear stresses (mpa)	122.59	101.88	16.89%
3	Kt σ	1.42	1.17	17.60 %
4	Kt τ	1.47	1.22	17.00%

VII. MANUFACTURING

Master cam x4 is computer-based computer-aided design / computer-aided manufacturing (CAD/CAM) software. Master cam X4 CNC Software's main product, started as a 2D CAM system with CAD tools that use by machinists design virtual components on a computer screen as well as guided computer numerical controlled (CNC) machine tools in the manufacture of components.

Master cam program for variable radius:

- Geometry Creation
- Setting the Environment
- Setting the Construction Planes
- Create the Geometry
- Create the Fillets (Radius)
- Save the Drawing
  Tool path Creation
- Define the Stock and Chuck Parameters
- Face front of the part
- Rough the outside diameter

- Finish the outside diameter
- Cut of the part
- Back plot the tool path
- Verify the tool path
- Save and updated master cam file
- Post and create the CNC code file

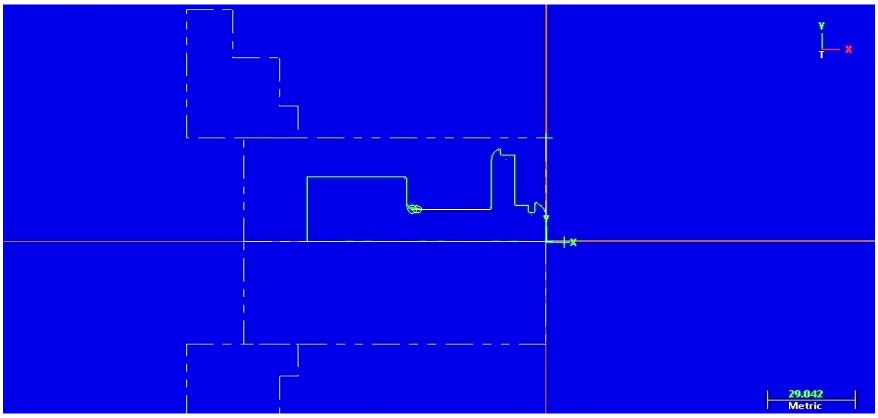

Fig.22. Geometry Creation

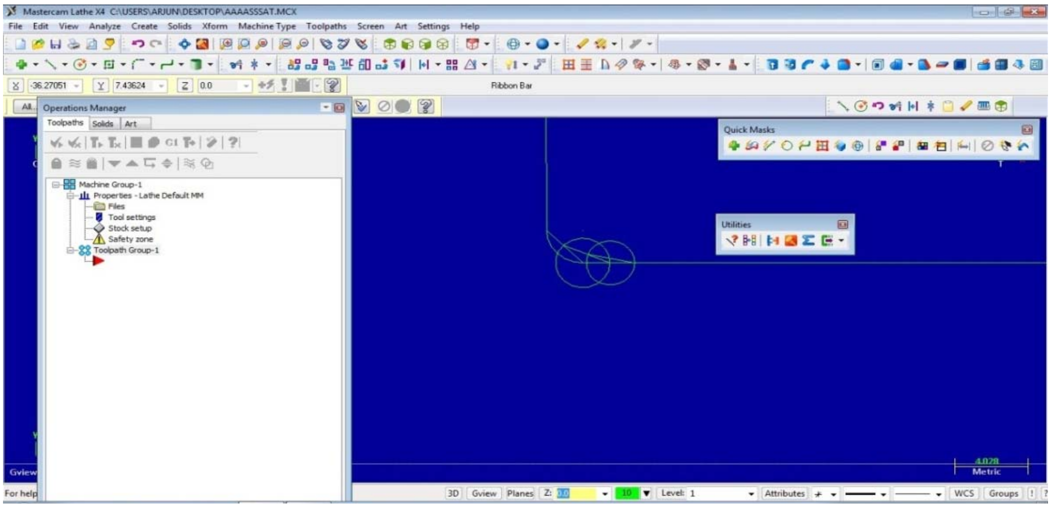

Fig.23. Variable Radius geometry in master cam x4

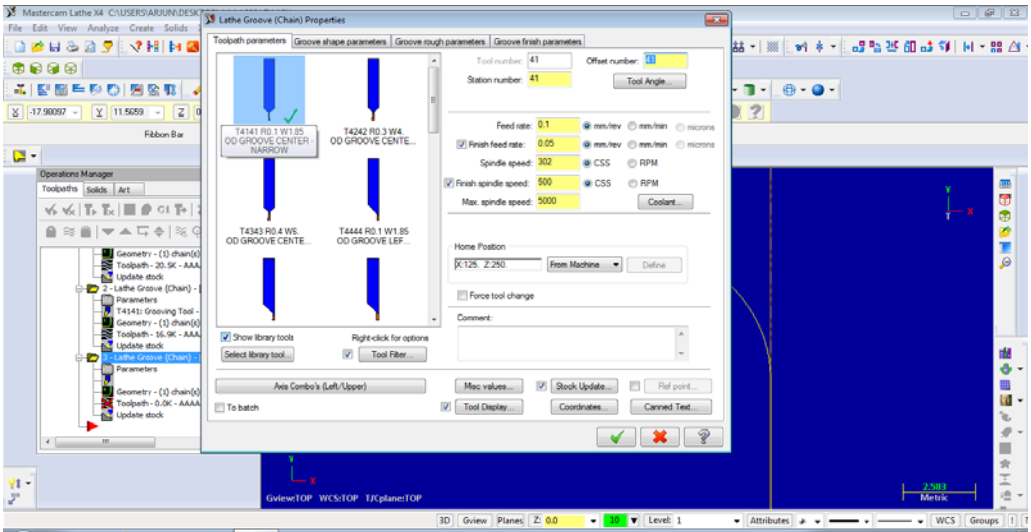

Fig.24. Tool& its path Creation

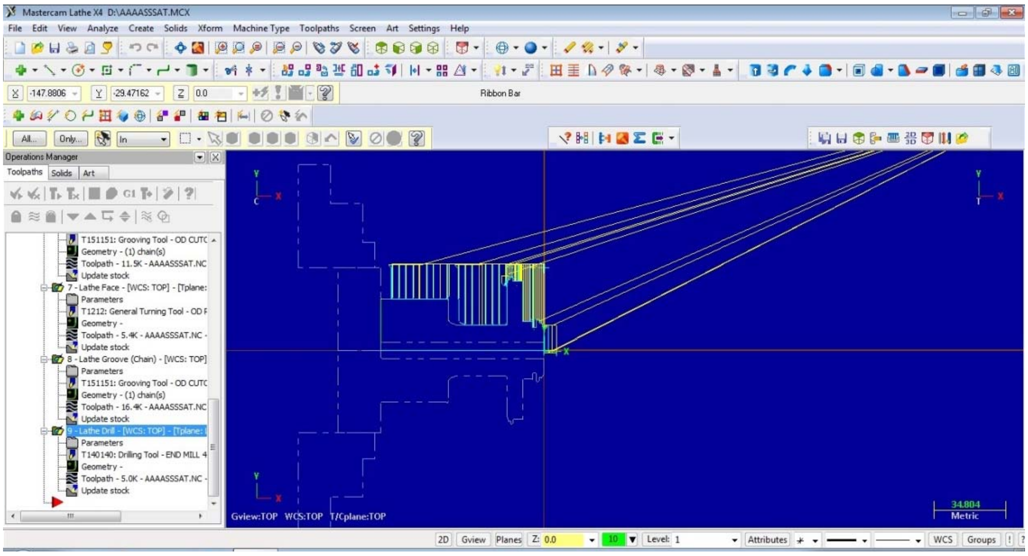

Fig.25. Back plot the tool path

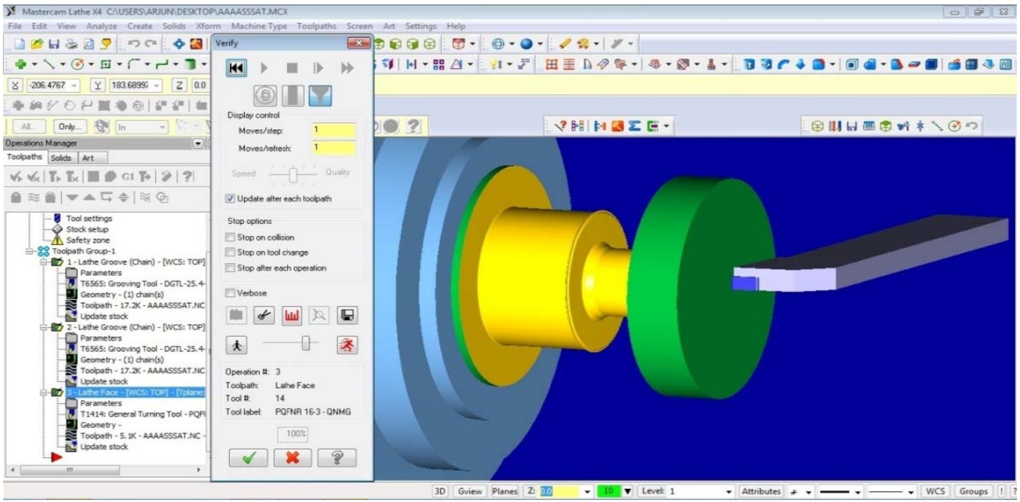

Fig.26. Verify the tool path

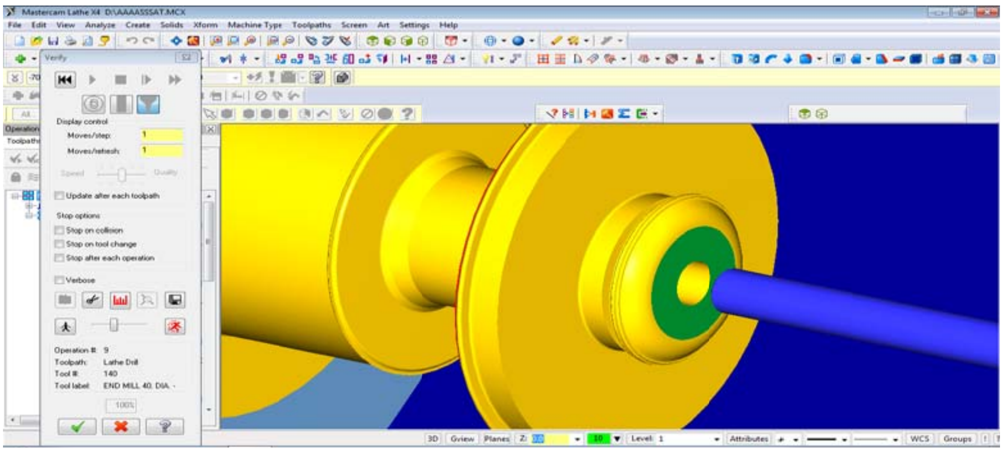

Fig.27 Post and create the CNC code file

VIII. RESULT AND DISCUSSION

The effects of Von- misses stress, Maximum shear stress and stress concentration factor (Kt) can be reduced to a great extent .Von-misses stress reduces up to 17.37%, Maximum shear stress up to 16.89% and stress concentration factor Bending and Torsion up to 17.60 & 17.00% respectively by Converting fillet radius geometry of the components, i.e. variable-radius notch as a replacement for constant-radius notch.

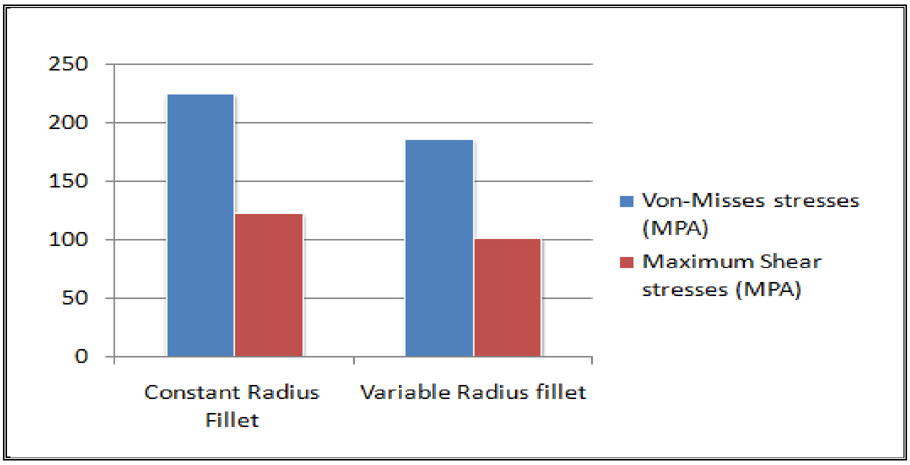

Fig.28.Von-mises stress and Max.shear stress for constant radius &Variable Radius

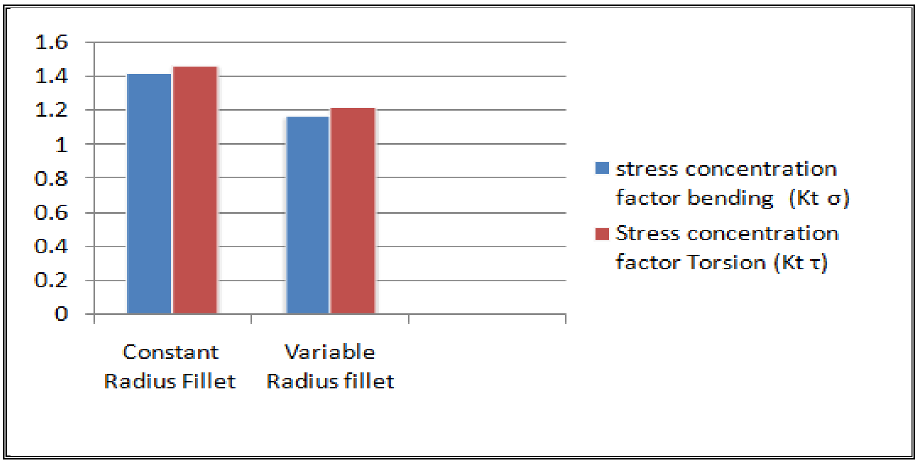

Fig.29.Stress concentration factor for constant and variable radius

IX. CONCLUSION

The effects of Von- misses stress, Maximum shear stress and stress concentration factor (Kt) can be reduced to a great extent .Von-misses stress reduces up to 17.37%, Maximum shear stress up to 16.89% and stress concentration factor Bending and Torsion up to 17.60 & 17.00% respectively by Converting fillet radius geometry of the components, i.e. variable-radius notch as a replacement for constant-radius notch. Stresses are calculated at critical section and there is no need to compute the stresses at every position. FEA analysis shows that maximum stress concentration at radius fillet and crack / or bending initiation also occurs at fillet radius. The Mattheck Method is capable of creating variable-radius notches with reduced to a great extent maximum stresses, using a very simple approach based on the apparent self-growth mechanism of tree branches. This approach could be very helpful in industrial areas by using advance CNC Programming software like Master cam X4 from manufacturing point of view.

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NOMENCLATURE

Kt	Stress concentration factor	*Greek Symbols*	
p	Power (Kw)	σb	Bending stress (mpa)
N	Rotational Speed(rpm)	τ	Shear Stress (mpa)
F.S	Factor of safety	$\sigma\ max$	Maximum Stress (mpa)
F	Force (N)	σ_{mises}	Von –mises Stress (mpa)
T	Torque (N-mm)	τ_{Max}	Maximum shear stress (mpa0
M	Bending Moment (N-mm)	Kt σ	Stress concentration factor for Bending Stress
Te	Equivalent torque (N-mm)	Kt τ	Stress concentration factor for torsion Stress
di	Inner diameter	$\sigma\ nom$	Nominal Stress (mpa)
do	outer diameter	Sut	Ultimate strength in bending (mpa)
K	Ration of diameters	syt	Yield strength (mpa)
h	Height (mm)		
r	Radius (mm)	*Subscripts*	
D	Diameter of larger shaft section(mm)	S.S	Stainless Steel
d	Diameter of smaller shaft section(mm)		

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