iJRASET
International Journal For Research in
Applied Science and Engineering Technology

INTERNATIONAL JOURNAL FOR RESEARCH

IN APPLIED SCIENCE & ENGINEERING TECHNOLOGY

Volume: 5 **Issue:** V **Month of publication:** May 2017

DOI:

International Journal for Research in Applied Science & Engineering Technology (IJRASET)

Designing and Groove Shaft Analysis under Different Loading Condition

Mr. Sudhanshu Mishra[1] , Shani Kumar[2], Sangam Kumar[3] Mr. Sumit Verma[4]
[1,2]M. Tech Student, [3]Assistant Professor [4]Associate professor, Department of Mechanical Engineering
[2]Madan Mohan Malaviya University of Technology, Gorakhpur, 273 010 (UP), India
[1,3,4]Suyash Institute of Information Technology Hakkabad Gorakhpur,273016 (UP).India

Abstract: shaft is a critical and standard component which is used normally in all kind of industries for power/motion transmission. The discontinuity present in shaft is prime cause for failure of shaft. Discontinuity results in reduction in strength of shaft. Discontinuities are stress raisers which are responsible for the cause of fracture. The objective of this study is to design the solid shaft with single discontinuity present in shaft under different loading conditions. Based on the equations of stress concentration factor and conventional peterson's curves and roark's curves the design of shaft at critical locations is done in this study. The equations of stress concentration factors for groove are presented and used for the stress calculation purpose. The programs for every loading conditions are generated using matlab and variation of groove radius is plotted which is useful to obtain stress concentration factor for any step ratio. The final plots for step ratio and stress concentration factor at limiting condition are plotted which are quite useful to obtain minimum stress concentration factors for any d/d ratio at h = r condition and finite element analysis (fea) is performed to obtain the variation of stress magnitude at critical locations of shaft. The load is applied to the fe model in ansys, and boundary conditions are applied according to the shaft mounting conditions. The analysis is done for finding critical location in shaft. Fea carried out to find maximum stress at the critical location of shaft.
Key word: stress concentration factors, fracture, shaft analysis.

I. INTRODUCTION

Design is either to formulate a plan for the contentment of a specified need or to solve a specific problem. If the plan results in the creation of impressive having a physical reality, then the product must be functional, safe, dependable, competitive, usable, manufacturable, and profitable. Design is an original and highly iterative process. It is also known as decision-making process. Decisions sometimes have to be made with too little information, occasionally with just the right amount of information, or with an excess of partially incongruous information. Decisions are sometimes made uncertainly; with the right reticent to adjust as more becomes known. The point is that the engineering designer has to be personally relaxed with a decision-making, problem-solving role.

A. Standard sizes of shafts

Typical sizes of solid shaft that are available in the market are,
Up to 26 mm 0.5 mm increments
26 to 50 mm 1.0 mm increments
51 to 100 mm 2.0 mm increments
100 to 200 mm 3.0 mm increments

B. Types of loadings on the shaft

1) The loadings that can act on the shaft are consisting of various types as shown below:
a) Axial (compression or tension
b) Bending and Torsion
c) Cyclic torsion with static(tension or compression
d) Combination of bending , torsion and axia
e) In phase multi-axial loading

II. DESIGN CONSIDERATIONS

A. Safety factor

The safety factor can be expressed by,

$$n = \frac{\sigma_{all}}{\sigma_d}$$

Where, σ_{all} = Allowable stress, (MPa) σ_d = Design stress, (MPa)

Table 1 Safety factor characteristics A, B and C [12]

Characteristic		Vg	G	F	P
A=	C=				
A = vg	vg	1.1	1.4	1.6	1.7
	g	1.2	1.5	1.8	1.96
	{ f	1.3	1.7	1.95	2.3
	p	1.4	1.8	2.2	2.46
A = g	C = { vg	1.3	1.56	1.9	2.10
	g	1.45	1.80	2.10	2.36
	f	1.6	1.95	2.35	2.70
	p	1.75	2.20	2.60	2.95
A = f	C = { vg	1.5	1.9	2.2	2.5
	g	1.7	2.10	2.5	2.76
	f	1.9	2.4	2.8	3.2
	p	2.1	2.35	3.1	3.45
A = p	C = { vg	1.7	2.20	2.5	2.85
	g	1.95	2.40	2.85	3.25
	f	2.2	2.70	3.2	3.65
	p	2.55	2.85	3.55	3.95

Here, vg = very good, g = good, f = fair, p = poor.

Where, A, B and C = Safety factor involving characteristics

A = Quality of the materials, maintenance, workmanship, and inspection B = Control over load applied to part and

C = experimental data, accuracy of stress analysis, or experience with similar devices.

III. DESIGN OF SHAFT WITHOUT DISCONTINUITY

A. Case-[A] shaft is subjected to torsional moment or torsion only,

The shaft is subjected to torsion as it rotates. It is the case which is seen in approximately every application. The torsional moment is constantly present in shaft due to its connections with gearbox or any other devices which produces torques. So, in this case the torsional shear stress is generated and it is determined by

$$\tau = \frac{16T}{\pi D^3}$$

Where, T = Torque transmitte = Shear stress D = shaft diameter

B. Case-[B] Shaft is subjected to bending moment only,

First in this case from given load the bending moment is determined at desired point of interest and on the basis of that bending moment the bending stress determined by,

Volume 5 Issue V, May 2017
ISSN: 2321-9653
International Journal for Research in Applied Science & Engineering Technology (IJRASET)

$$\sigma_b = \frac{32M}{\pi D^3}$$

Where, *M* = Bending mo = Bending stress , D = shaft diameter

C. Case-[C] Shaft is subjected to axial load only

When shaft is subjected to the axial load only, it will be either tensile or compressive. therefore, the axial stress is determined by,

$$\sigma_a = \frac{4F}{\pi D^2}$$

Where, *F* = Axial forc = Axial stress D = Diameter of shaft

D. Case-[D] Shaft is subjected to combined bending and torsional load,

The combined load equation will contain the provisions regarding to bending and torsional loads only, the equation of the combined stress or effective stress due to both loads is determined by

$$\sigma_{eff} = \frac{32}{\pi D^3}\sqrt{M^2 + \frac{3}{4}T^2}$$

Where, M = Bending moment, T = Torque transmit = Effective Von Mises stress, D = Diameter of shaft

This equation is based on Distortion energy or Von Mises failure theory. It is used for ductile materials. The more simplified form of above equation is given by [33]

$$\sigma_{eff} = \frac{4}{\pi D^3}\sqrt{(8M)^2 + 48T^2}$$

This equation is also presented in the form of principal stresses except some modification will take place as axial load is not present here so axial stress will be diminished. Therefore, the conventional equation of principal stresses reduces to given

$$\sigma_1, \sigma_2 = \frac{\sigma_b}{2} \pm \sqrt{\left(\frac{\sigma_b}{2}\right)^2 + \tau^2}$$

Where, $_1$ = First principal stress and

$_2$= Second principal stress

Now, from above principal stresses the effective stress can be determined by,

$$\sigma_{eff} = \sqrt{{\sigma_1}^2 - \sigma_1\sigma_2 + {\sigma_2}^2}$$

E. Case-[E] Shaft is subjected to combined axial, bending and torsional load,

, the effective stress is determined,

$$\sigma_{eff} = \frac{32}{\pi D^3}\sqrt{\left[M + \frac{FD}{8}\right]^2 + \frac{3}{4}T^2}$$

above equation is based on The Von Mises theory of failure and it is used for ductile material. The more simplified mathematical form of above equation is given by,

$$\sigma_{eff} = \frac{4}{\pi D^3}\sqrt{(8M + FD)^2 + 48T^2}$$

The combined loading equation is also represented in the form of Principle stresses. The equations of principal stresses can be derived from the Mohr's circle. So, the equations of principal stresses are given by

$$\sigma_1, \sigma_2 = \left(\frac{\sigma_a + \sigma_b}{2}\right) \pm \sqrt{\left(\frac{\sigma_a + \sigma_b}{2}\right) + \tau^2}$$

www.ijraset.com
IC Value: 45.98
Volume 5 Issue V, May 2017
ISSN: 2321-9653
International Journal for Research in Applied Science & Engineering Technology (IJRASET)

F. Stress concentration

$$K_t = \frac{\sigma_{max}}{\sigma_{nom}}$$

$$k_t = \frac{\tau_{max}}{\tau_{nom}}$$

G. Design of the shaft with discontinuity

1) Case-[A] Shaft is subjected to axial load only,: The maximum stress is the stress concentration factor of shaft due to axial force *P*

$$\sigma_{max1} = K_{tn1}\sigma_{nom}$$

Here nominal stress is replaced such as

$$\sigma_{max1} = K_{tn1}\frac{4p}{\pi d^2}$$

Where, P= Axial force $_{n\,1}$ = Axial stress D= Diameter of shaft

2) Case-[B] Shaft is subjected to bending moment only : The maximum stress corresponding due to the bending moment is

$$\sigma_{max2} = K_{tn2}\sigma_{nom}$$

Here nominal stress is replaced such as,

$$\sigma_{max2} = K_{tn2}\frac{32M}{\pi D^3}$$

Where, M = Bending moment, $_2$ = Bending stress, D = Diameter of shaft

3) Case-[C] Shaft is subjected to torsional moment or torsion only : The maximum torsion stress due to the torque T is

$$\tau_{max3} = K_{tn3}\sigma_{nom}$$

Here nominal stress is replaced such as,

$$\tau_{max3} = K_{tn3}\frac{16T}{\pi D^3}$$

Where, T = Torque transmitted, τ_{max3} = Shear stress D = Diameter of shaft

The maximum stresses occur at same location, explicitly at the base of the groove, and the principal stresses are determined by using the familiar formulas

$$\sigma_1 = \frac{1}{2}\left(\sigma_{max1} + \sigma_{max2}\right) + \sqrt{\frac{1}{2}\left(\sigma_{max1} + \sigma_{max2}\right)^2 + 4\tau_{max3}{}^2}$$

$$\sigma_2 = \frac{1}{2}\left(\sigma_{max1} + \sigma_{max2}\right) - \sqrt{\frac{1}{2}\left(\sigma_{max1} + \sigma_{max2}\right)^2 + 4\tau_{max3}{}^2}$$

4) Von Mises Criterion

$$\sigma_{eq} = \sqrt{\sigma_1{}^2 - \sigma_1\sigma_2 + \sigma_2{}^2} \qquad = \sqrt{\left(\sigma_{max1} + \sigma_{max2}\right)^2 + 3\tau_{max3}{}^2}$$

Material of shaft	=	(G92550) hot rolled annealed steel
Yield stress	=	580MPa
Factor of safety (n)	=	1.2
Major diameter (D)	=	70mm
Minor diameter (d)	=	49mm
Total length (L)	=	1000mm

Volume 5 Issue V, May 2017
ISSN: 2321-9653

International Journal for Research in Applied Science & Engineering Technology (IJRASET)

Radius of Groove (rk)	=	7mm
Depth of groove in shaft (h)	=	10.5mm
Bending moment	=	1KN.m
Torque	=	2.5KN.m

Factor of safety is taken 1.5. Here the factor of safety is selected as described per Pugsley method .Taking factors from table 1 and selection of the factor of safety is done. As per selection the factor of safety is 1.44 but here 1.5 is considered. So, based on the factor of safety the allowable stress is determined. So, allowable stress is denoted such

$$\sigma_{all} = \sigma_y\ n = 537\ 1.5 = 358 Mpa$$

IV. STRESS CALCULATION AT THE CRITICAL SECTION

D = 70 mm M = 1 KN.m R = 7 mm D/d = 1.43, T = 2.5 KN.m h = 10.5 mm d = 49 mm r/d = 0.143 Bending stress is determined by taking Kt from Peterson's curve. Here D/d = 1.43 and r/d

= 0.143 so based on this data the value of the stress concentration factor for bending is taken from the Peterson's curves

Taking kt = 1.82 from Peterson's curves and determining the bending stress

$$\sigma_b = \frac{32M}{\pi d^3} = 32\times1\times10^3 / 3.14\times0.049^3 = 86.58 \text{ MPa}$$

$$\sigma_{max} = k_t \frac{32M}{\pi d^3} = 1.82\times(32\times1\times10^3 / 3.14\times0.049^3) = 157.6 \text{ MPa}$$

Now, for same section stress concentration factor Roark's So, the determined yields, found by using equation of to = 1.824. By using this,

$$\tau_{max} = k_t \frac{32M}{\pi d^3} = 1.824\times(32\times1\times10^3 / 3.14\times0.049^3) = 157.9 \text{ MPa}$$

Torsion stress is determined by taking from Peterson's curve. Here D/d = 1.43 and r/d =

0.143 so based on this data the value of the stress concentration factor for torsion is taken from Peterson's curves.

Taking = 1.46 from Peterson's curves and determining the Torsion stress.

$$\tau = \frac{16T}{\pi d^3} = 16\times2.5\times10^3 / 3.14\times0.049^3 = 108.2 \text{ MPa}$$

$$\tau_{max} = k_t \frac{32M}{\pi d^3} = 1.46\times(16\times2.5\times10^3 / 3.14\times0.049^3) = 158.0 \text{ MPa}$$

Now, for same section the stress concentration factor is found by using Roark's equation. So, the determined yields to kt = 1.416. By using this, calculating

$$\tau_{max} = k_t \frac{32M}{\pi d^3} = 1.416\times(16\times2.5\times10^3 / 3.14\times0.049^3) = 153.2 \text{ MPa}$$

After finding individual the maximum stresses, now combined stresses are needed to be determined. respectively for torsional shear, bending so, the principal stresses for these are determined as,($\sigma_{b=157.6\ MPa}$),

τ =158.0 MPa

$$\sigma_1, \sigma_2 = \frac{\sigma_b}{2} + \sqrt{\left(\frac{\sigma_b}{2}\right)^2 + \tau^2}$$

σ_1, = 78.8 + 176.6 = 255.4 MPa

σ_2 = 78.8 - 176.6 = -97.8 MPa

Thus the corresponding maximum shear stress is

$$\frac{\sigma_1 - \sigma_2}{2} = (255.4 + 97.8)/2 = 176.6 \text{ MPa}$$

The maximum torsional shear stress,

Volume 5 Issue V, May 2017
ISSN: 2321-9653
International Journal for Research in Applied Science & Engineering Technology (IJRASET)

Finally, the equivalent stress

$$\sigma_{eq} = \sqrt{{\sigma_1}^2 - \sigma_1\sigma_2 + {\sigma_2}^2} \quad = 315.9 \text{ MPa}$$

Based on the other equations, the effective stress is obtained,

Now σ_{eff} = 315.9 MPa $\leq \sigma_{all}$

Now, the shear, bending stresses in which stress concentration factor is taken into account by using Roark's equation are considered. Respectively for the torsional shear, bending. So, the principal stresses for these are determined as,

,($\sigma_{b=157.9 \text{ MPa}}$),$\tau$ =153.2 MPa

$$\sigma_1, \sigma_2 = \frac{\sigma_b}{2} + \sqrt{\left(\frac{\sigma_b}{2}\right)^2 + \tau^2}$$

σ_1, = 78.95 + 172.35 = 251.4 MPa

σ_2 = 78.95 – 172.35 = -93.4 MPa

Thus the corresponding maximum shear stress is

$$\frac{\sigma_1 - \sigma_2}{2} \quad = (251.4 +93.4)/2 = 172.4 \text{ MPa}$$

The maximum torsional shear stress of shaft,

Finally, the equivalent stress

$$\sigma_{eq} = \sqrt{{\sigma_1}^2 - \sigma_1\sigma_2 + {\sigma_2}^2} \quad = 311.9 \text{ MPa}$$

Based on the other equations, the effective stress is obtained

σeff = 337.64 MPa $\leq$ σall

V. ERROR CALCULATION FOR GROOVE SECTION

Here, the comparison of stresses at critical section in which stress concentration factors are calculated using the Peterson curves [14] and Roark [16] equations are presented in the table 2.

Table 2. Comparison between Peterson's eq. and Roark's eq. stress (Mpa) values

Type of load	Stress values as per Peterson's curves (Mpa)	Stress values as per Roark's equation (Mpa)	Error %
Bending	158.6	158.9	0.20
Torsion	160	154.2	3.40
Combined	258.4	269.5	4.90

VI. STRESS CALCULATION AT THE CRITICAL SECTION

The dimensions for this section are,

D = 70 mm d = 49 mm

M = 2.5 KN.m T = 1 KN.m

R = 10.5 mm h = 10.5 mm

As to find the combined stresses first individual stresses are calculated.

Here D/d = 1.42 and r/d = 0.2142 so based on this data, value of the stress concentration factor for torsion is taken from Peterson's curves.

Taking Kt = 1.32 from Peterson's curves and calculating the torsional shear stress

$$\tau = \frac{16T}{\pi d^3} \quad =16\times2.5\times10^3 /3.14\times0.049^3= 108.2 \text{ MPa}$$

Now, for same section stress concentration factor is found by using equation ,So, the calculation yields to Kt = 1.318.] using this, calculating,

Volume 5 Issue V, May 2017
ISSN: 2321-9653
International Journal for Research in Applied Science & Engineering Technology (IJRASET)

$$\tau_{max} = k_t \frac{32M}{\pi d^3}$$ = 1.46× (16×2.5×10^3 /3.14×$0.049^{3)}$ = 158.0 MPa

Now, for same section stress concentration factor is found by using equation of the Roark the calculation yields to Kt = 1.31. using this, calculating

$$\tau_{max} = k_t \frac{16T}{\pi d^3}$$ = 1.31×16×2.5×10^3/ 3.14×0.049^3= 141.74 MPa

Bending stress is calculated from Peterson's curve by taking Kb ,Here D/d = 1.42 and r/d = 0.214 so based on this data, value of the stress concentration factor for bending is taken from Peterson's curves. Taking Kb = 1.6 from Peterson's curves and calculating bending stress

$$\sigma_b = \frac{32M}{\pi d^3}$$ =32×1×10^3 /3.14×0.0493= 86.58 MPa

Now, for same section, stress concentration factor is found by using equation. So, the calculation yields to Kt = 1.318. using this, calculating,

$$\sigma_{max} = k_t \frac{32M}{\pi d^3}$$ = 1.61×32×1×10^3 /3.14×0.0493= 139.39 MPa

Now, for same section stress concentration factor is found by using equation ,So, the calculation yields to Kb = 1.6001. using this, calculating,

$$\sigma_{max} = k_t \frac{32M}{\pi d^3}$$ = 1.6001×32×1×10^3 /3.14×0.0493= 138.53 MPa

Now, finding the individual maximum stresses, combined stresses are needed to be calculated., the principal stresses for they are calculated as,

,($\sigma_{b=139.39MPa)}$,τ =142.82 MPa

$$\sigma_1, \sigma_2 = \frac{\sigma_b}{2} + \sqrt{\left(\frac{\sigma_b}{2}\right)^2 + \tau^2}$$

σ_1, = 69.7 + 158.9 = 228.6 MPa

σ_2 = 69.7 – 158.9 = -89.2 MPa

Thus the maximum shear stress is

$$\frac{\sigma_1 - \sigma_2}{2}$$ = (228.6+89.2)/2 = 158.9 MPa

Finally, the equivalent stress

$$\sigma_{eq} = \sqrt{\sigma_1{}^2 - \sigma_1\sigma_2 + \sigma_2{}^2}$$ = 313.9 MPa

Based on other equations ,obtain the effective stress is,

Now,

σeff = 283.92MPa ≤ σall

Now, by using shear, bending stresses in which the stress concentration factor is taken into account using the Roark's equation are considered. These are shown for torsional shear, bending. So, the principal stresses for they are calculated

,($\sigma_{b=139.39MPa)}$,τ =142.82 MPa

$$\sigma_1, \sigma_2 = \frac{\sigma_b}{2} + \sqrt{\left(\frac{\sigma_b}{2}\right)^2 + \tau^2}$$

σ_1, = 78.8 + 176.6 = 227.02 MPa

σ_2 = 78.8 - 176.6 =-88.49 Mpa

Thus the maximum shear stress is

$$\frac{\sigma_1 - \sigma_2}{2}$$ =227.02+88.49 2= 157.75 MPa

Volume 5 Issue V, May 2017
ISSN: 2321-9653

the maximum torsional shear stress of shaft.

Finally, the equivalent stress

$$\sigma_{eq} = \sqrt{\sigma_1{}^2 - \sigma_1\sigma_2 + \sigma_2{}^2} \qquad = 281.88 \text{ MPa}$$

Based on other equations ,obtain the effective stress is,

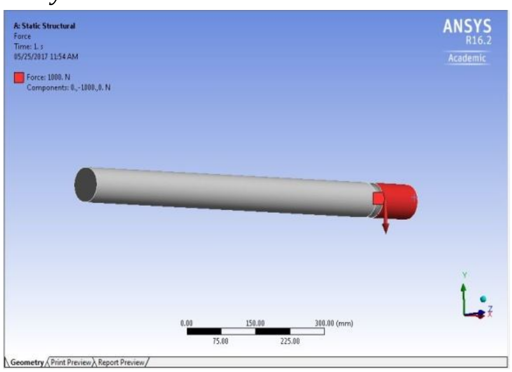
Now,

$$\sigma_{eff} = 281.88\text{MPa} \le \sigma_{all}$$

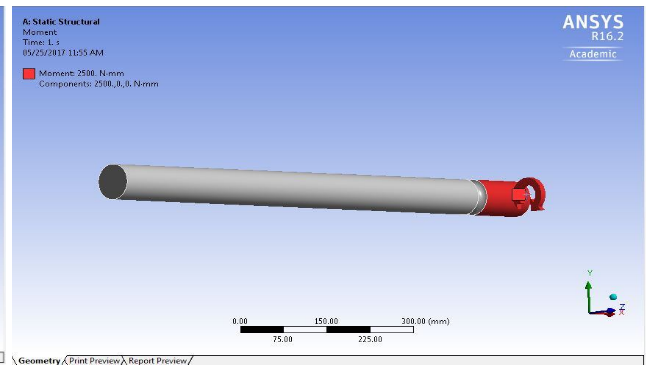
A. Need of Finite Element Analysis

, we prepare a model of shaft in the Solid work software and save as .IGES file format for Analysis of shaft in ANSYS WORKBENCH 16.2. Import .IGES model in ANSYS Workbench simulation module.

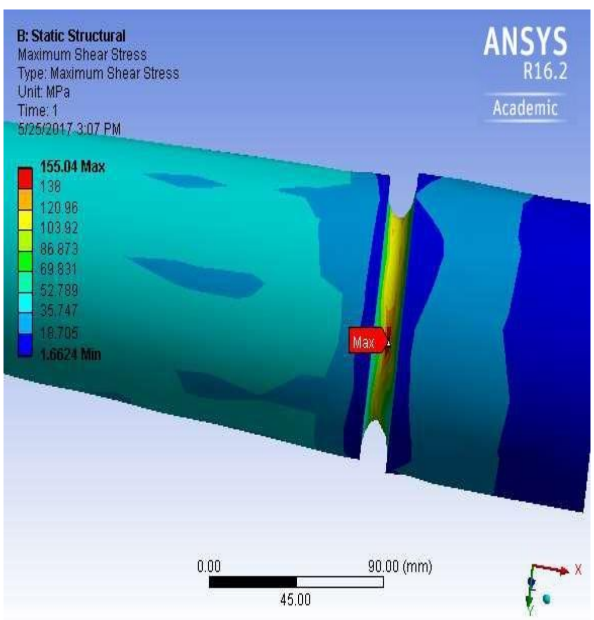
1) Type of Analysis: Static structural



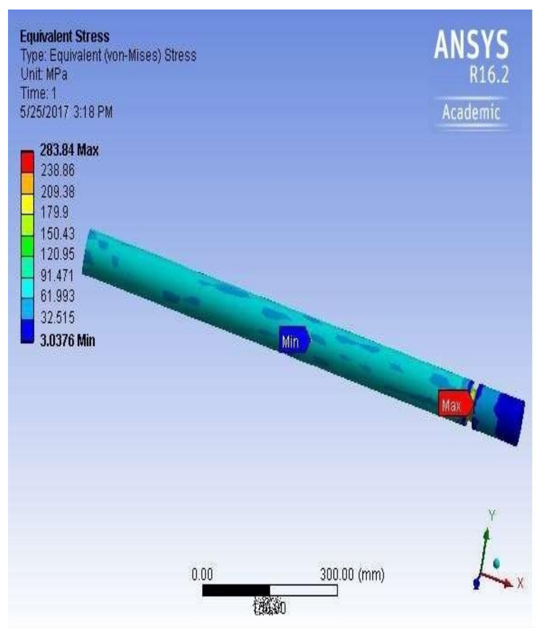
Apply tangential force (3a)



Shear stress analysis(3b)



Shear stress analysis(3c)



Von-Misses Stress analysis(3d)

VII. ERROR CALCULATION FOR GROOVE SECTION

Here, the comparison of the stresses at section in which stress concentration factors are calculated by using Peterson curves [14] and Roark [16] equations are presented in table 3.

Volume 5 Issue V, May 2017
ISSN: 2321-9653
International Journal for Research in Applied Science & Engineering Technology (IJRASET)

Table 3.Comparison between Peterson curves and Roark's eq. stress values

Type of load	Stress values as per Peterson's curves (Mpa)	Stress values as per Roark's equation (Mpa)	Error %
Bending	140.39	137.35	0.74
Torsion	144.82	140.74	0.76
Combined	238.60	230.02	0.70

VIII. RESULTS

By using MATLAB software the variation of the groove radius and stress concentration is plotted. Programs are generated for various loading condition and for various groove radius equations and based on that programs curves are developed. Here. From this graph it can be seen that the sudden rise is taking place when S.C.F value reaches at 0.7 and the peak value is obtained at 1 which is 660 mm. This groove radius is not possible and even as it is known that value of the stress concentration factor can never be less than 1. When Kt reaches at 1.1 the sudden fall is observed. Even when Kt =1.1 the value of groove radius is 52.8 mm. So, it can be seen that to obtain value of the stress concentration nearly one it is necessary to keep the groove radius large. This is basically not possible. The similar patterns are obtained in all the equations in which higher values of the groove radius are obtained for keeping the stress concentration factor 1.1 or nearly equal to 1.1

Table 4 Comparison between Peterson curves and Roark's eq. stress (Mpa) values

Stresses reduction in groove (Mpa)

r =7mm	r =10.5	% Reduction
315.9	280.88	10.76
337.64	282.92	12.91

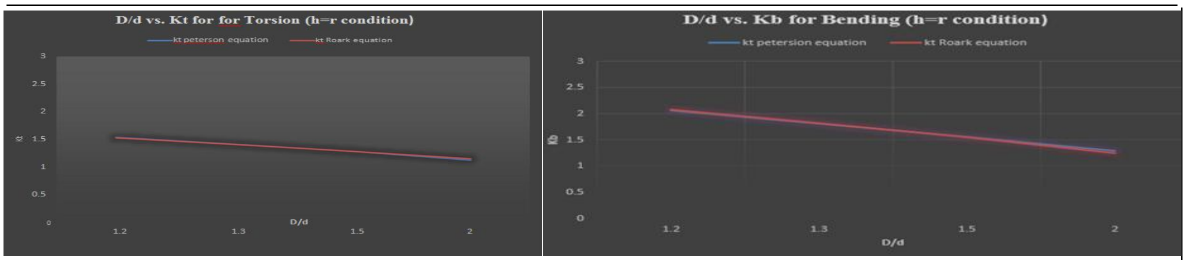


Fig. D/d ratio vs. S.C.F for torsional load (h = r), D/d ratio vs. S.C.F for bending load (h = r)

IX. CONCLUSION

this study it can be said that for any D/d ratio the minimum stress concentration factor is obtained when groove radius kept equals to groove height for circular groove radius. For, further reduction the stress concentration factor the groove radius should be kept more than groove height which results into unbalancing of bearings due to higher transition region.

The variation of the groove radius equation is plotted using by MATLAB for any step ratio (D/d ratio) for all the three loading conditions. Curves are generated in form of the groove radius (r) vs. stress concentration factor. So, by select the appropriate groove radius the stress concentration factor is directly obtained for any step ratio.

For, same rotor turbine shaft the groove radius is selected equals to the groove height and stresses are calculated at the critical sections which results into reduction of stresses as compare to previous calculations. So, it can be said that by increasing the groove radius the stresses are reduced at critical section. So, the minimum stresses are calculated at critical section.

Variation of the minimum stress concentration factor is plotted in the form of curves for every possible D/d ratios in the form of step ratio (D/d) vs. stress concentration factor (kt). These curves are useful to obtain minimum stress concentration factor for any D/d ratio for all the three loading conditions.

X. FUTURE SCOPE

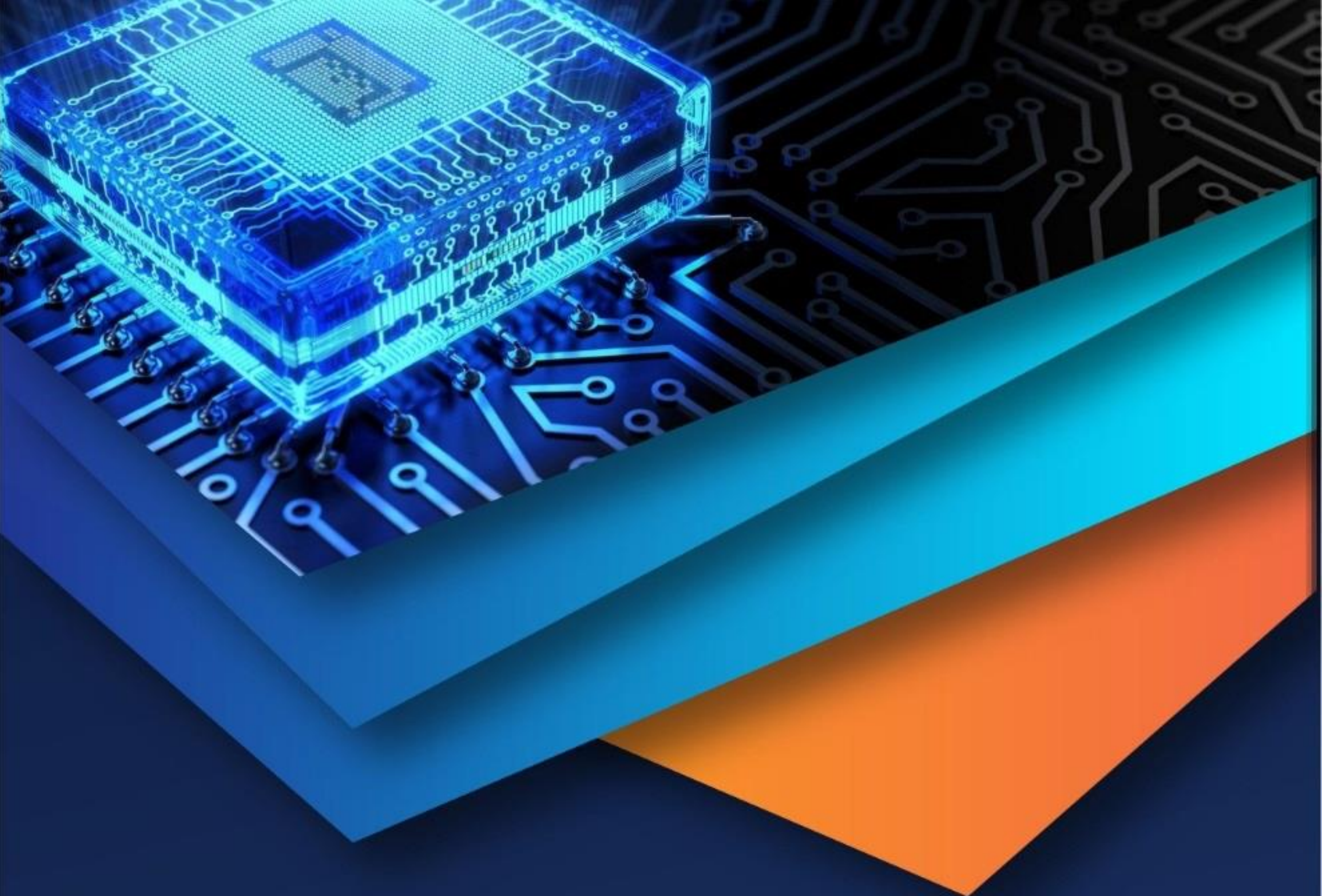
By considering the aspects presented in this thesis more work can also be done in the same

A. Area. The considerations for future work are as follows,

1) Design based on fatigue and fracture considerations and employing equations of the groove radius in the further study for fatigue calculations.
2) .value of the Von-Misses stresses that comes out from the analysis is far less than material yield stress so our design is safe and we should go for optimization to reduce material and cost.
3) Crack initiation or growth rate (da/dN) calculated for the combined loading conditions with single or multiple discontinuities.
4) Strain and life cycle estimation approach for single or multiple discontinuities with the multiple loading conditions.
5) The ranges presented in this thesis can be refined further and accurate results can be obtained based on the experimental work and elasticity concepts.

REFERENCES

[1] Shigley, J. E. and Michke, C. R. “Mechanical engineering design”, eighth edition, 2008 (Tata McGraw Hill, New Delhi, India).
[2] Norton, R. L. “Machine design: an integrated Approach”, third edition, 2006 (Pearson Education, Upper Saddle River, NJ).
[3] Shafts Unit no. 7 IGNOU. “http://www.ignou.ac.in/upload/Unit-7-60”.
[4] R.E.PETERSON, “Stress Concentration Factor”, third edition, 2011(A Wiley interscience publication).
[5] Saleh A. Al-Fozan, Anees U. Malik and Mohammad Al-Hajri “Failure Analysis of Sheared Shaft of a Brine Recycle Pump”, Troubleshooting Technical Report No. TSR 3804/05005 in August 2005
[6] E. Rusiński, J. Czmochowski, P. Moczko “Failure reasons investigations of dumping conveyor breakdown” Volume 23 Issue 1 July 2007Journal of Achievements in
[7] Deepan Marudachalam M.G, K.Kanthavel, R.Krishnaraj “Optimization of shaft design under fatigue loading using Goodman method” International Journal of Scientific &
[8] Sandeep Bhattacharyya “Failure analysis of gas blower shaft of a blast furnace” Engineering Failure Analysis 15 (2008) 349–355 Osman Asi “Fatigue failure of a rear axle shaft of an automobile” Engineering Failure Analysis 13 (2006) 1293–1302
[9] R.W. Fuller a, J.Q. Ehrgott Jr. b, W.F. Heard b, S.D. Robert b, R.D. Stinson b, “Failure analysis of AISI 304 stainless steel shaft” Engineering Failure Analysis
[10] V.S. Rocha, “Fracture failure analysis of gearbox shaft” Vol. 24, Engineering Failure Analysis, 2012
[11] Chun-Yuan Lin Jui-Pin Hung Tze-
[12] Chi Hsu, “Failure Analysis of Reverse Shaft in the Transmission System of All-Terrain Vehicles” ASM International 2008F. Vodopivec et al. “Cracking and fracture of valve shafts in the secondary steam circuit of a nuclear power plant. A case investigation”. Nuclear Engineering and Design (2001).
[13] Iradj Sattari-Far et al. “Failure of stainless digester shafts in a paper production plant” Engineering Failure Analysis (2003).
[14] Y.J. Li et al. “Fracture analysis of a castellated shaft” Engineering Failure Analysis (2007).
[15] Y.J. Li et al. “Fracture analysis of a castellated shaft” Engineering Failure Analysis
[16] A.H.V. Pavan et al. “Root cause analysis of bowl-mill pinion shaft failures” Case Studies in Engineering Failure Analysis 1 (2013) 103–109
[17] Zˇ. Domazet et al. “Failure of two overhead crane shafts” Engineering Failure Analysis 44 (2014) 125–135.
[18] Zhi-wei Yu et al. “Case internal oxidation and intergranular fracture of carburized splined-shaft” Engineering Failure Analysis 22 (2012) 141–151
[19] Dragan Trifkovic et al. “Failure analysis of the combat jet aircraft rudder shaft” Engineering Failure Analysis 18 (2011) 1998–2007
[20] Dj. Mandrino et al. “Investigation of the fracture of a car’s drive shaft” Engineering Failure Analysis 16 (2009) 1252–1261
[21] G. Urquiza at al. “Failure analysis of a hydraulic Kaplan turbine shaft” Engineering



doi crossref

10.22214/IJRASET

INDEX COPERNICUS

45.98

ISRA JIF

IMPACT FACTOR: 7.129

TOGETHER WE REACH THE GOAL

IMPACT FACTOR: 7.429

ISI

INTERNATIONAL JOURNAL FOR RESEARCH

IN APPLIED SCIENCE & ENGINEERING TECHNOLOGY