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Design Optimization and Analysis of a Steam Turbine Rotor Grooves

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Abstract: *A large variety of turbo-machinery rotor groove geometries are used in industry. However, there is no specific attempt made to compare them on the basis of performance. This paper provides an attempt to fill that gap aiming to find an optimum geometry. The main objective of the investigations is to reduce fillet stress concentration factor and associated deformations. The present work carries out the design modification of fillet of a Steam Turbine Rotor. Finite Element Analysis is performed using ANSYS Workbench that is used to determine the fillet stresses effectively to modify the blade rotor grooves. The low pressure steam turbine rotor blades have usually a history of stress failure. They suffer from tensile and bending stresses partly due to the centrifugal force as a result of high rotational speeds and partly due to high pressure, temperature and speed steam loading. The centrifugal force is one of the problems that face the designers of turbine blades especially the long ones. The designer always aims at reducing these stresses. One way to do so is by the reduction of stress concentration factor which takes place between the rotor and its groove. That is to make the rotor blade of variable cross section instead of straight. This paper presents the method of reducing stress concentration factor as well as reduction in the deformation of the rotor.*
Keywords: *Optimization, Stress concentration factor, Turbine rotor grooves, Maximum principal stress.*

I. Objective Of The Work

The turbine rotor grooves are most rigid elements in high speed steam and gas turbines used in land based and aerospace applications. Due to rotation, the blade root gets tightened in the rotor grooves and transmit centrifugal load. The mating surface could be small in case of high pressure turbine blades but may be six times or more for low pressure turbine blades. While the average stress in the mating areas of these bladed disks is fully elastic and well below yield, the peak stress at singularities in the groove shape can reach yield values and into local plastic region. Last stage LP turbine blades and first stage LP compressor blades are the most severely stressed blades in the system. Usually these are the limiting cases of blade design allowing the peak stresses to reach yield or just above yield conditions. Failures usually occur with crack initiation at the stress raiser location and propagation. Optimization has become a necessity in the recent years to achieve an optimal design in stress or strain, stiffness and weight etc. In earlier practices, dedicated codes are developed to achieve a specific optimization problem. For example, Bhat, Rao and Sankar (1982) used the method of feasibility directions to achieve optimum journal bearings for minimum unbalance response. Failures can occur with crack initiation at the stress raiser location and propagation, two cases can be cited. The last stage blades in an Electricite de France B2 TG Set failed in Porcheville on August 22, 1977 during over speed testing Frank (1982). On March 31, 1993 Narora machine LP last stage blades suffered catastrophic failures. These blades have stresses well beyond yield. An elasto plastic analysis for the same case showed that the peak stress is 1157 M Pa well beyond the yield. While it is not possible to eliminate the yield and keep the structure fully elastic to achieve the last stage blades in limiting cases, it will be advisable to achieve the yield conditions to be as low as possible. One of the major factors which influence the life of the steam turbine rotor blade is stress concentration factor. One way to do so is by the reduction of stress concentration factor which takes place between the rotor and its groove. In this paper ANSYS workbench as an optimization tool to optimize the grooves by reducing or altering its fillet inner dimensions and observing the various stress concentration factors. This techniques and optimization enables to minimize the peak stress values so as to improve the structural integrity of the grooves.

II. Computational Methodology

The two dimensional geometry of the steam turbine rotor obtained from B.H.E.L Turbines and Compressors Department is made to revolve in solid works to obtain the cylindrical profile of a rotor with a groove as shown in Fig. 1.

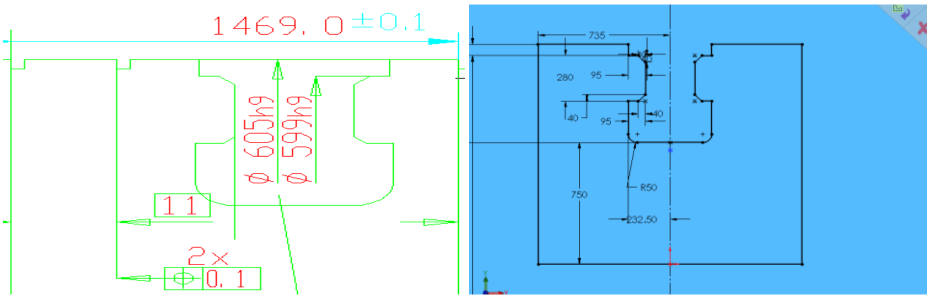

Fig. 1: Two dimensional sketch of a rotor groove and all dimensions are in mm only.

2.1 Geometric Modeling

The rotor geometry is revolved up to 180 degrees due to axi-symmetric nature. As it is difficult to view the inner portion of rotor groove at 360 degrees, it is profiled at 180 degrees only. The 180 degree profile results in reduction of considerable time of computations and tedious computer efforts. After drawing the rotor groove model in solid works, it is saved in an IGES file format. This IGES file is imported in ANSYS-12 Workbench for analysis purpose in the static structural ANSYS.

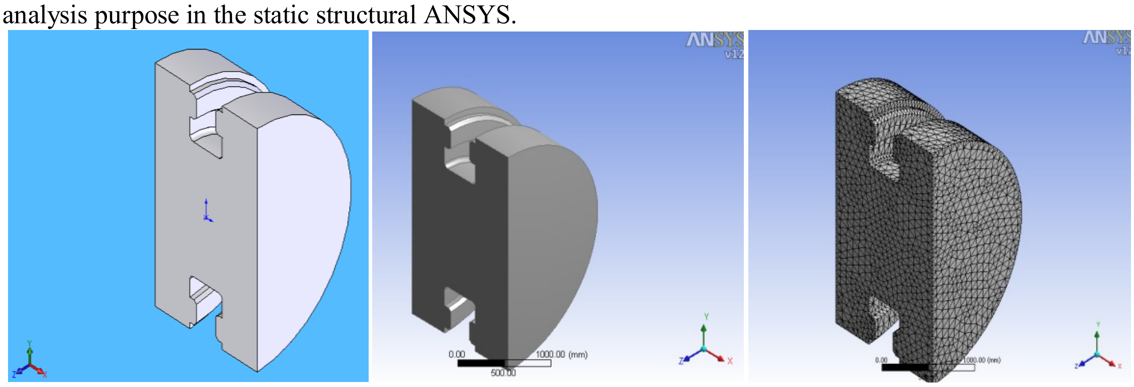

Fig. 2: Rotor modeling in solid works and Meshing of a rotor

After modeling the geometry, meshing is to be performed. Initially, a fine meshing is preferred to a total element size of up to 22809 and nodes up to 383008. The material properties are shown in table 1. Then pressure calculation is performed which is dependent upon total number of blades which is provided on the rotor, radius of groove, weight of the each blade and machine rated speed.

TABLE 1: Isotropic elasticity of material

Young's Modulus (M Pa)	Poisson's Ratio	Temperature (C)
2.e+005	0.3	constant

Steam turbine rotors shafts have blades which are mounted on the rotor groove. During shaft rotation, the blade tries to get pulled out of the groove due to centrifugal loading with certain pressure as shown in Fig. 4. This pressure exerted at rotor groove is calculated manually as shown below. The boundary conditions on a rotor model is imposed in such a way that all other surface are kept constant (as shown in Fig. 3) except the groove area that exerts pressure about 75 M Pa. The following are the calculations regarding pressure made :-

Total no of blades provided on the rotor (n) = 26

Groove surface area = 247350 mm ^{2}.

Pressure exerted at rotor groove= Centrifugal force due to blades rotation / groove surface area.

Radius of groove = 750 mm = 0.75 m.

Weight of the each blade (W) =94 N.

Acceleration due to gravity (g) =9.81 m/sq.sec.

Mass of the blade (m) =W/g = 94/9.81=9.582 Kg.

Machine Rated Speed (N) =3000 rpm.

Angular velocity (ω) = $2\pi N/60$= (2x3.14159x3000)/60 = 314.1593 rad/sec.

Centrifugal force = n ($mr\omega^2$) = 26(9.582 x 0.75 x 314.1593 ^{2}) = 18.44e6 N.

Pressure = Centrifugal force/ Groove surface area = 18.44e6/247350 = 74.56 M Pa ≡ 75 M Pa =75e6 N/m^2

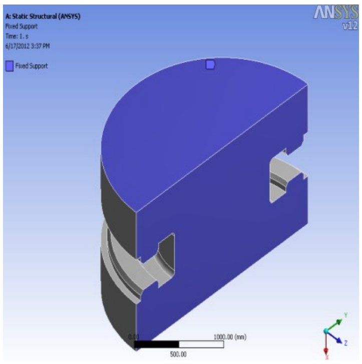

Fig. 3: Fixed Support.

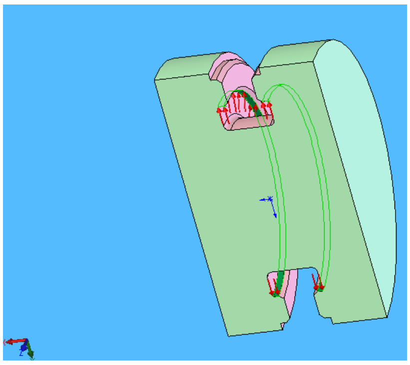

Fig. 4: Pressure applied at rotor groove

III. Results & Discussions

3.1 Case with Fillet radius, F=13 mm

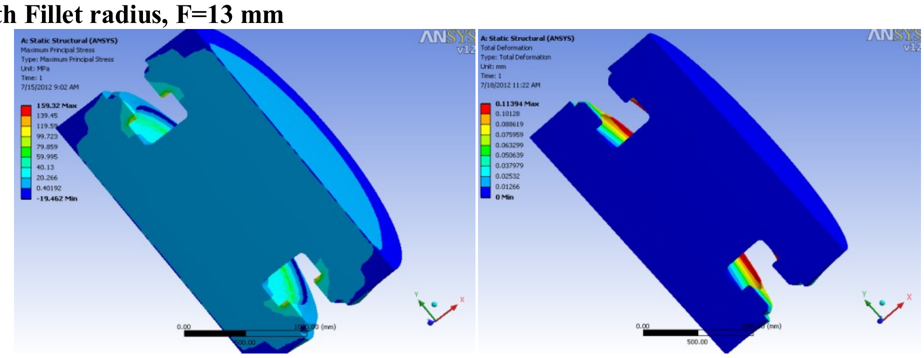

Fig. 5: Maximum Principal Stress and Deformation at rotor groove fillet radius= 13 mm

From Fig. 5, for a given nominal stress to be 107.2187 M Pa, theoretical stress concentration factor, $K_t = \sigma_{max} / \sigma_{min}$ is obtained as (159.32) / (107.2187) = 1.486.The total deformation obtained is 0.11394mm.

3.2. Case with Fillet radius, F=15 mm

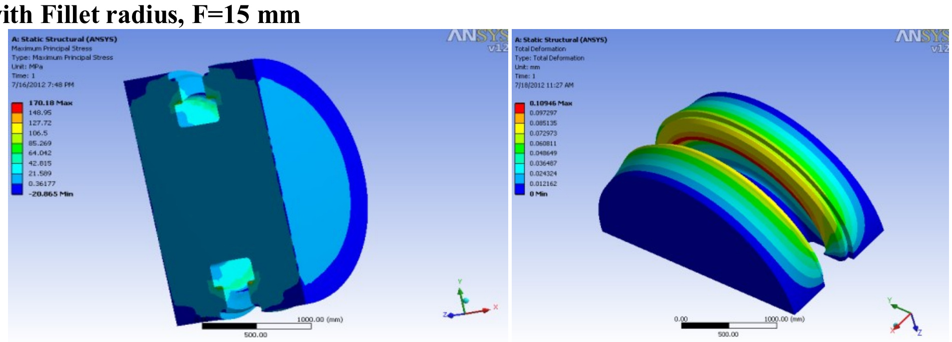

Fig. 6: Maximum Principal Stress and Deformation at rotor groove fillet radius= 15 mm

From Fig. 6, for a given nominal stress to be 107.2187 M Pa, theoretical stress concentration factor, $K_t = \sigma_{max} / \sigma_{min}$ is obtained as (170.18) / (107.2187) = 1.587.The total deformation obtained is 0.10946 mm.

3.3. Case with Fillet radius, F=20 mm (Base Line Model)

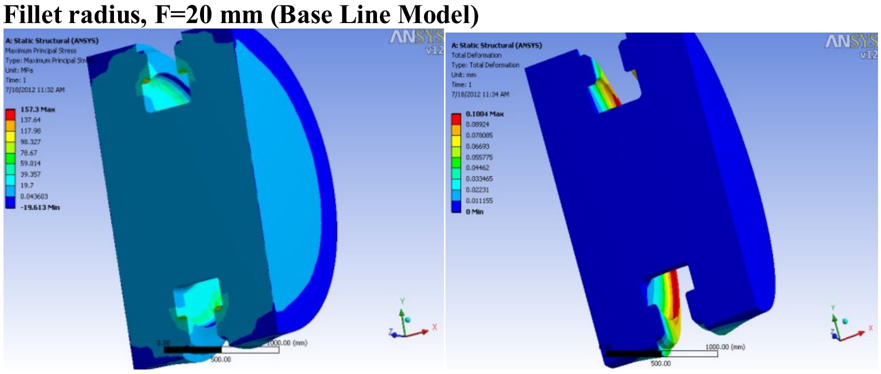

Fig. 7: Maximum Principal Stress and Deformation at rotor groove fillet = 20 mm

From Fig. 7, for a given nominal stress to be 107.2187 M Pa, theoretical stress concentration factor,

$K_t = \sigma_{max} / \sigma_{min}$ is obtained as (157.3) / (107.2187) = 1.467 .The total deformation obtained is 0.1004mm.

3.4. Case with Fillet radius, F = (25, 20) mm

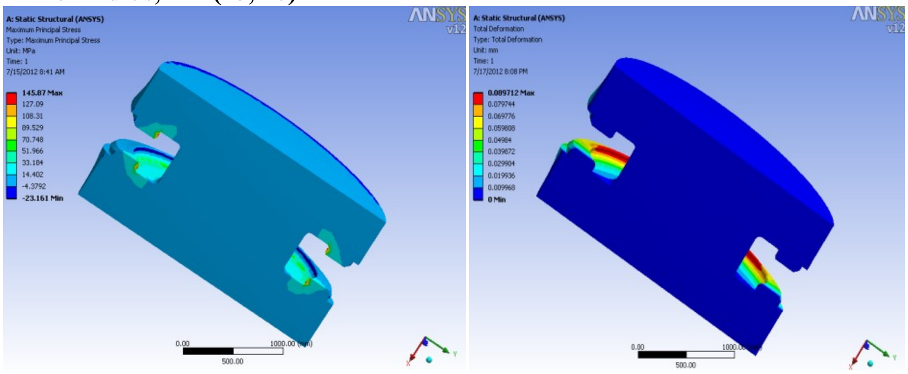

Fig. 8: Maximum Principal Stress and Deformation at rotor groove = (25, 20) mm

From Fig. 8, for a given nominal stress to be 107.2187 M Pa, theoretical stress concentration factor,
$K_t = \sigma_{max} / \sigma_{min}$ is obtained as (145.87) / (107.2187) = 1.36. The total deformation obtained is 0.089712 mm.

3.5. Case with Fillet radius, F=25 mm

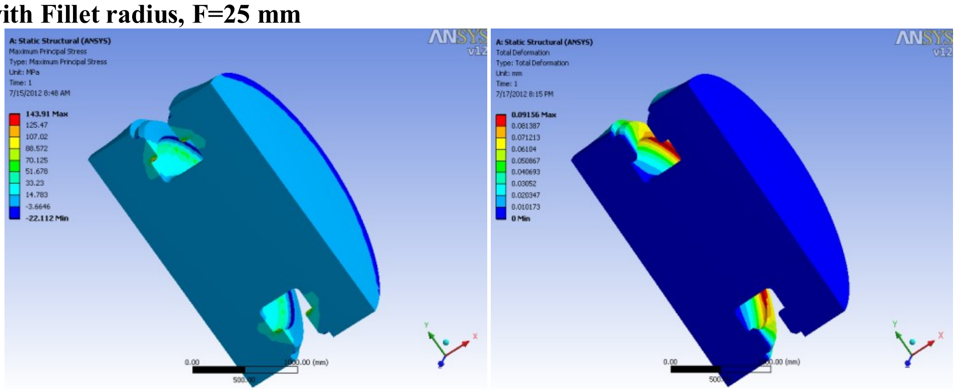

Fig. 9: Maximum Principal Stress and Deformation at rotor groove fillet radius= 25 mm

From Fig. 9, for a given nominal stress to be 107.2187 M Pa, theoretical stress concentration factor,
$K_t = \sigma_{max} / \sigma_{min}$ is obtained as (143.91) / (107.2187) = 1.342.The total deformation obtained is 0.09156 mm

3.6. Case with Fillet dimensions, F= (25, 30) mm

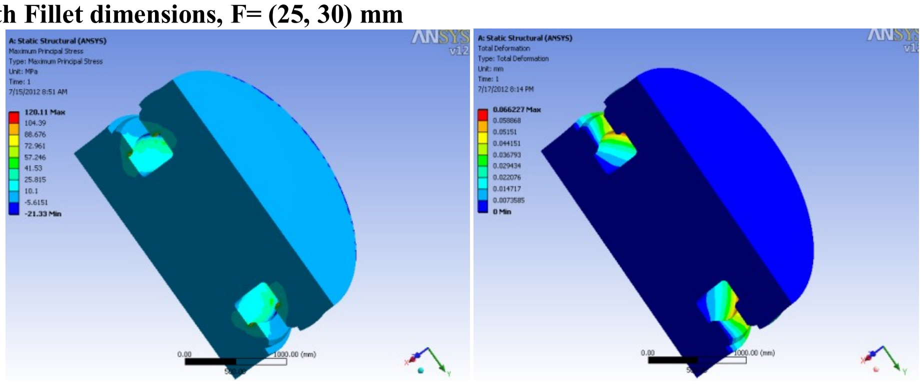

Fig. 10: Maximum Principal Stress and Deformation at rotor groove radius= (25, 30) mm

From Fig. 10, for a given nominal stress to be 107.2187 M Pa, theoretical stress concentration factor,
$K_t = \sigma_{max} / \sigma_{min}$ is obtained as (120.11) / (107.2187) = 1.12. The total deformation obtained is 0.066227 mm.

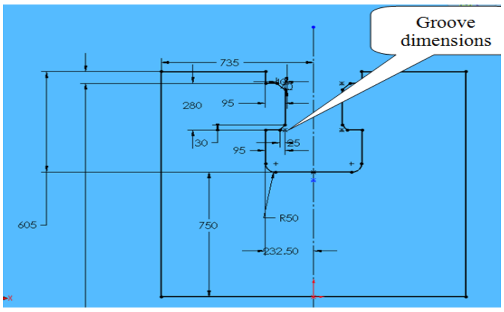

Fig. 11: Groove model at radius of (25, 30) mm.

The theoretical stress concentration factor K_t can be minimized to 1.12 by obtaining the radius of dimensions (25, 30) mm which is within the safety limits. But practically it is difficult to obtain such kind of grooves as there will be a difficulty in manufacturing especially in mass production.

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3.7. Case with Fillet radius, F=30 mm

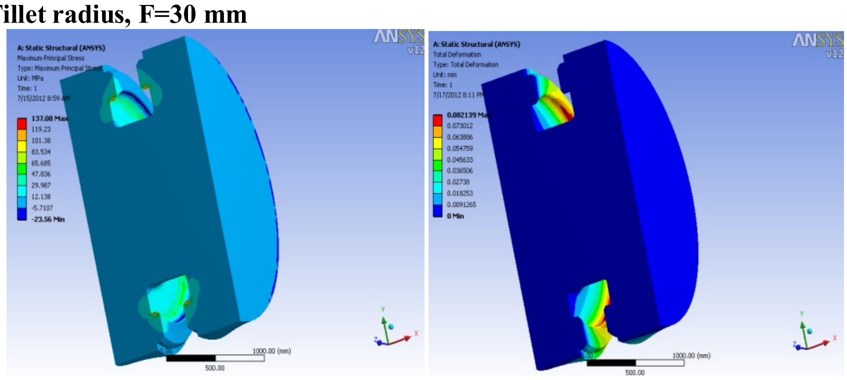


Fig. 12: Maximum Principal Stress and Deformation at rotor groove fillet radius= 30 mm

From Fig. 12, for a given nominal stress to be 107.2187 M Pa, theoretical stress concentration factor, $K_t = \sigma_{max} / \sigma_{min}$ is obtained as (137.08) / (107.2187) = 1.279.The total deformation obtained is 0.082139 mm.

3.8. Case with Fillet radius, F=40 mm

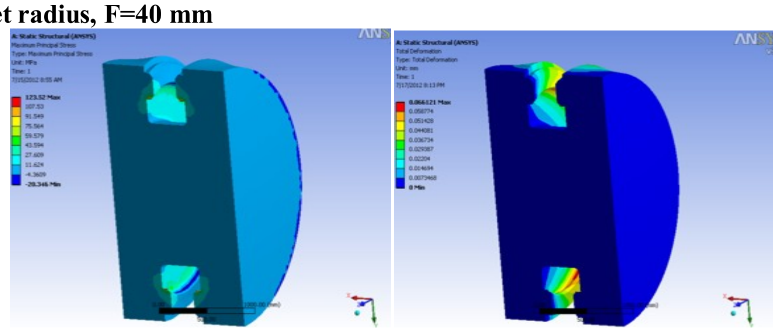


Fig. 13: Maximum Principal Stress and Deformation at rotor groove fillet radius= 40 mm

From Fig. 13, for a given nominal stress to be 107.2187 M Pa, theoretical stress concentration factor, $K_t = \sigma_{max} / \sigma_{min}$ is obtained as (123.52) / (107.2187) = 1.152.The total deformation obtained is 0.066121 mm.

3.9. Case with Fillet radius, F=45 mm

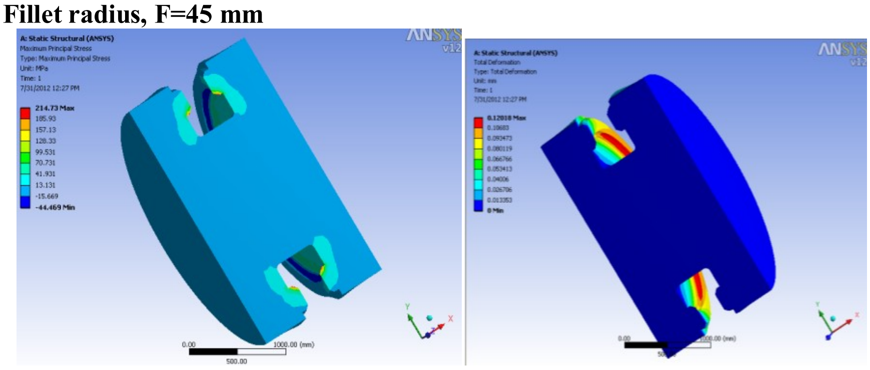
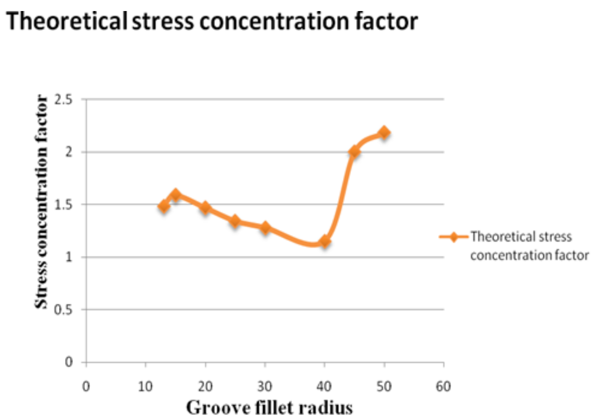


Fig. 14: Maximum Principal Stress and Deformation at rotor groove fillet radius= 45 mm

From Fig. 14, for a given nominal stress to be 107.2187 M Pa, theoretical stress concentration factor, $K_t = \sigma_{max}/ \sigma_{min}$ is obtained as (214.73) / (107.2187) = 2.0027 and the total deformation obtained is 0.12018 mm. Similarly, for F=50 mm, it has been observed that theoretical stress concentration factor,
$K_t = \sigma_{max}/ \sigma_{min}$ is obtained as (233.97) / (107.2187) = 2.1822.The total deformation obtained is 0.1206 mm.

TABLE 2: Comparison at Various Fillet Radius

S.NO	Fillet radius (mm)	Maximum Principal Stress, σ_{max}, (M Pa)	Theoretical stress concentration factor, K_t	Deformation (mm)
1.	F = 13	159.32	1.486	0.11394
2.	F = 15	170.18	1.587	0.10946
3.	F = 20	157.3	1.467	0.1004
4.	F = 25	143.91	1.342	0.09156
5.	F = 25,20	145.87	1.36	0.089712
6.	F = 25,30	120.11	1.12	0.066227
7.	F = 30	137.08	1.279	0.082139
8.	F = 40	123.52	1.152	0.066712
9.	F = 45	214.73	2.0027	0.12018
10.	F = 50	233.97	2.1822	0.1206



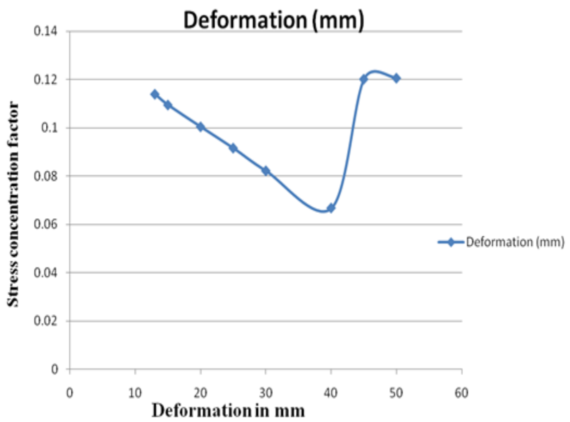


Fig. 15: Variation of theoretical stress concentration factor deformation with fillet radius with fillet radius

It can be inferred from the Fig. 15 that there is an increase in the groove fillet radius, stress concentration factor reduces and at a particular point an optimal value can be observed. On increasing the fillet radius further, the K_t increases, it is due to the pressure acting on that groove which enables the steam turbine blade to pull out of the rotor grooves and may cause failure due to an increase in fillet stresses. It can be inferred from the Fig. 15 that as there is an increase in the groove fillet radius, there is a decrease in the deformation and at a particular point the deformation is minimum. On increasing the fillet radius further, the deformation also reduces similar to K_t; it is due to the pressure acting on that groove which enables the steam turbine blade to pull out of the rotor grooves and may cause failure due to an increase in fillet stresses.

IV. Conclusions

The design considerations made to address the structural integrity of the blade and the sensitivity analysis for geometric parameters with aid of finite element analysis has resulted in the following observations during the course of study: From the original blade root of geometry with fillet radius 20 mm, the design changes were made on the geometry and an optimized blade root dimensions of radius 40 mm is obtained as the recommended design change as it gives good reduction on stress concentration factor and fillet stresses. The base line model and the modified root model were subjected to centrifugal loads and the resulting centrifugal stresses were analyses has been carried out in ANSYS workbench. It is observed that for a given fillet radius of 20 mm, the stress concentration factor (K_t) for a steam turbine rotor groove and total deflection is found to be 1.587 and 0.10946 mm. It can be inferred from the stress concentration factor plot that an increase in the groove fillet radius, reduces the stress concentration factor and at a particular point an optimal value is obtained. It can be inferred from the Deformation plot that an increase in the groove fillet radius reduces the Deformation also just similar to K_t and at a particular point an optimal value is obtained. As the fillet radius is decreased, K_t and deflection is increased. Gradually, on increasing the fillet radius to a value of 40 mm, we get a $K_t = 1.152$ and deflection = 0.066121 mm. The K_t can also be minimized further to 1.12 by obtaining the dimensions of (25, 30) mm which is within the safety limits as discussed in Fig. 11.

V. Future Scope

The project can be extended by analyzing the relief groove along with the rotor grooves as there will be an easier method to determine the crack propagation and eliminate it quickly. The crack will first initiates at the relief groove as it has more depth than other rotor grooves and can be identified easily.

References

[1] R.Nagendra Babu, K.V.Ramana and K.Mallikarjuna Rao, "*Determination of Stress Concentration Factors of A Steam Turbine Rotor by FEA", World Academy of Science, Engineering and Technology, Volume No.39, 2008, Pg. 137-41.*

[2] Dr. Arkan K. Husain Al-Taie, "*Stress Evaluation of Low Pressure Steam Turbine Rotor Blade and Design of Reduced Stress Blade", Mechanical Engineering Dept/ Univ. of Tech, Eng.Tech.Vol.26, No.2, 2008.*

[3] L. Moroz, A. Tarasov, *IJPGC 2003, International Joint Power Generation Conference. June 16-19, 2003, Atlanta, Georgia, USA on "Coupled CFD and Thermal Steady State Analysis of Steam Turbine Secondary Flow Path ".*

[4] IJEST 01/2010, International Journal of Engineering Science and Technology, November 2010 "*Prediction of Stress Concentration effect under Thermal and Dynamic loads on a High Pressure Turbine Rotor". Babu R.Nagendra, Dr.K.V.Ramana, Rao Dr.K.Mallikarjuna, Journal of Mechanical Engineering Research Vol.2(6),pp. 106-115.*

[5] Peterson.R.E, "*Stress Concentration Factors*" (John Wiley & Sons, ISBN 0-471-683329-9, Page No.57, Year 2008).

[6] Walter D. Pilkey, *Stress concentration factor* (John Wiley & Sons, I S B N 0-471-53849-3, Year 2007).

[7] ANSYS ICEM, CFD Tutorial Manual, http://www1.ansys.com/customer/content/Tutorials/PDF/itut110.pdf

[8] ANSYS, Release-12, Modeling and Meshing Guide.