Finite Element Analysis of Steam Turbine Rotor of 210 MW Power Plant

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ABSTRACT

The steam turbine rotor is subjected to temperature variations in during start up and stop cycle which occurs in short intervals of time. This variation in temperature induces transient thermal stresses in the rotor due to large temperature gradients. The transient stresses occur due to change in the material properties like Young's modulus of elasticity, coefficient of expansion, thermal conductivity, Poisson's ratio, specific heat etc. The estimation of transient thermal stresses becomes essential before designing the rotor. The turbine rotor is subjected to thermal as well as mechanical stresses.This paper investigates finite element analysis of 210 MW steam turbine rotors for transient thermal loading. The rotor was made of 30Cr1Mo1V.The high-pressure turbine model was prepared using CAD software (PRO-E). The model was solved using a 1 Degree slice model as the rotor was Axis-Symmetric about the axis of rotation. The FE modelwas analyzed for transient thermal stresses using Ansys 11. The Von misses stresses were highest at the 11^{th} stage(Groove) at a time step of 12000secs (200 min) during 9 hours cold start up cycle. The transient thermal stresses were found to be much higher than the steady state thermal stresses.

Keywords-*Steam turbine rotor, transient thermal stresses, FEM, Ansys*

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INTRODUCTION

Rotating discs are historically, areas of research and studies due to their vast utilization in industry. Steam turbine rotor is one of the examples to name. In steam turbines, rotating discs are simultaneously subjected to mechanical and thermal loads. Transient thermal analysis is the thermal analysis wherein boundary conditions and properties change with time. This means that the constraints such as ambient temperature, thermal coefficient, material properties, etc., are time-dependent. Transient thermal analysis is important in analyzing models that are subjected to material properties and boundary conditions that vary with time and temperature.

Turbine rotors used in power plants are subjected to high temperature during start up cycle as discussed by SukhvinderKaurBhatti et.al(2006). Since the turbine rotor is subjected to largetemperature variations, the material properties such as specific heat, enthalpy, density, and Young's modulus vary with time. Under such conditions, there is the probability of failure of turbine rotor if the turbine rotor is not designed taking into consideration the transient effect as discussed in literature by ZvonimirGuzoviæet.al(2011).

There are many finite element packages available for conducting the transient thermal analysis. Some of the packages are NASTRAN, ABAQUS, ANSYS, NISA, etc. These packages allow the designer to vary the ambient temperature with time, vary the convective heat transfer coefficients and heat flux with time/temperature, and also allow heat generation to be applied. The present study aims at carrying out transient thermal analysis of a 30cr1Mo1V(Yong Liet.al(2010)) steam turbine rotor. This turbine receives high temperature steam up to 540 °C.. Transient thermal and analysis is conducted using ANSYS 11 by making use of the heat transfer coefficients as boundary conditions. In turbines of this nature, the transient behavior is also an important factor as the steady-state behavior. The rotor takes some time before it attains an equilibrium temperature. During this interim period, the temperature varies with time and the rotor is said to be in a transient state.Duringthisunsteady state, it is subjected to different temperature gradients. The various fibers tend to expand differently according to the individual temperatures and coefficients of expansion. The temperature gradients that can be established in the transient state are generally higher than those that occur in the steady state.SukhvinderKaurBhattiet.al(2006) Discussed in their the Transient heat transfer characteristics,centrifugal and the thermal stresses arising in the disk. Interesting results obtained in terms of maximum operational radial stress, maximum operational hoop stress, maximum operational Von misses stress, the temperature field etc. So the disk is expected to perform well in spite of all the stringent operating conditions. ZvonimirGuzoviæet.al(2011). prepared an algorithm and the results of non-stationary thermal stresses modeling in steam turbine rotor by means of the users software package were obtainedFinite Element Analysis Of Steam Turbine Rotor Of 210 MW Power Plantstationary thermal stresses are stipulated by pressure change on turbine exit. The results of non-stationary thermal stress calculations (i.e.ofthe numerical modeling) show on several characteristic regions of the rotor thermal stressed state: a) the rotor central bore; b) the low-pressure rotor; c) the disc of the last turbine stage, and d) the rear-end labyrinth gland. As in the low-pressure part of the rotor the high gradients of thermal and mechanical quantities (temperature, heat flux, deformation, stresses) are determined, so this region of steam turbine rotor is analyzed in detail. Chunlin Zhang et.al(2010) used a600MW supercritical steam turbine rotor as the research object, and analyzed the variation of thermal stress in the warm starting up process. In this paper the analysis based on the operating data measured from actual operation and calculate the variations of the temperature field and stress field during the process of warm starting-up with the method of thermal-structure direct interaction analysis by Ansys. By analyzing the results, it proves that the maximum stress of the rotor is in the first stage of the intermediate pressure casing, and it is the main factor restricting the velocity of steam turbines starting-up process. This result which calculated and analyzed by finite element method can be a theoretical basis for the optimal operation and online monitoring of turbine units.

Deepak Dhar et.al(2004) discussed life estimation of a turbine blade taking into account the combined effects of centrifugal stresses, vibratory stresses and thermal stresses. The stresses are determined by accounting for the rotor acceleration. The blades are subjected to aerodynamic excitation force obtained from thin cambered aerofoil theory under incompressible flow. The thermo-elastic forces are obtained from the three-dimensional non-linear heat transfer equations using the finite element analysis. The fatigue life is estimated using two well known theories, from the number of cycles in various blocks during start-up and shut-down periods of the turbine operation when the stresses peak.

Yong Li et.al(2010) In this paper thermal stresses in 600MW steam turbine in different governing modes isdiscussed. The computational object is intercepted from hp-ip rotor of the 600MW steam turbine in this paper.The distribution of the temperature and thermal stress in the governing stage and the first stage of high pressure cylinder is calculated two-dimensionally. The changes of the temperature and stress in different governing modes are analyzed. The result of calculation provides a reliable basis for on-line monitoring of the rotor thermal stress, which ensures the security and economical operation of steam turbine units. Transforming the primary throttling governing into nozzle governing reduces the throttling loss with the part load, but increases the steam temperature variation in the same part of steam turbine. The temperature field and stress field after the governing stage of 600MW steam turbine are calculated, which comes to a conclusion that the axial thermal stress ofnozzle governing is larger than that of the throttling governing. The results of calculation provide a reliablebasis for on-line monitoring of the rotor thermal stress and determining the most critical area in the system of life time management. In order to Ensure the security and the economical efficiency of the units after transformingthe primary throttling governing into nozzle governing; it is necessary to re-account the velocity of load change.

Sudheendra V.S. et.al(1989).The Paper estimates the effects of the transient thermal stresses in the turbine rotor using Finite Element Analysis. A typical turbine rotor in the form of bladed disc called Blisk is considered for Transient Thermal Analysis using the Numerically Integrated Structural Analysis(NISA) package. The Blisk material is considered as MAR M247 and the properties of the material is updated to the model. The appropriate boundary conditions depicting the actual environment in which the turbine rotor works in an aero engine are also updated. The result show that the maximum temperature occurs at the tip of the blade and decreases as we go into the disk. The Transient Thermal Analysis was successfully completed and theresults were obtained. The thermal stresses due to large temperature gradients are higher than the steady state stresses. The large thermal stresses occur before reaching the steady state value.R. NagendraBabu et.al.(2008)In this paper Finite Element Analysis (FEA) with extremely fine mesh in the vicinity of the blades of Steam Turbine Rotor is applied to determine stress concentration factors. The most comprehensive source of stress concentration factors for commonly encountered geometries has been compiled by Peterson (1953, 1974). However in these references,the stress concentration factors for only filleted shafts are available and are onlyapproximations based on photo elastic results for two-dimensional strips. The relation between two and three dimensional stress concentration factors is made by assuming an analogy exists between a circumferential fillet and a circumferential groove. Thisis thelimitation of the Peterson Graphs for estimation of the stress concentration factors. The numerical techniques are most effective due to advancement of high and large memory computers. These techniques can beapplied for any minor change in the problem, which reduces the cost and time required for manufacturing and testing of several prototypes.

LucjanWitek.(2006) This paper presents the failure analysis of the turbine disc of an aero engine, installed in a certain type of aircraft. From the visual examination of the fractured surface, it was possible to observe beach marks, typical of fatigue failure. A non-linear finite element method was utilized to determine the stress state of the disc/blade segment underoperating conditions. High stress zones were found at the region of the lower fir-tree slot, where the failure occurred. A computation was also performed with excessive rotational speed. Attention of this study is devoted to the mechanisms of damage of the turbine disc and also the critical high stress areas.

S.Barella et.al.(2011) This paper investigates the rotor turbine failure of a 60 MW unit of a thermalpower plant. The rotor was made of CrMoNiV steel, and the failure occurred after approximately 10 years in operation. Several different analyses were carried out in order to identify the failure's root cause: visual examination, SEM fractography, micro-hardness measurement, and micro structural characterization.The origin of the fatigue phenomenon can be traced to thestress field generated on the rotor surface by both the frequent start-up cycles and the blade fixing method.

S.R. Holdsworth et.al (2007) This paper examines the results from a more recent test campaign conducted toexamine the effect of varying the magnitude of the thermal transient in the TMF cycle on deformation and endurance characteristics. The evidence of physical damage accumulated during test by the specimens in this campaign is considered with reference to output from detailed finite element simulations of the series of TMF experiments. As part of an activity to examine the effectiveness of state-of the-art creep-fatigue assessment procedures, a series of service cycle thermo-mechanical (TMF) tests were carried out to provide a collation of well characterized deformation and endurance data, for a single heat of 1CrMoV rotor steel, gathered under anisothermal conditions intended to be representative of operation in service .

ANALYSIS

This study investigates transient thermal stresses in210 MW High Pressure steam turbine (HPT) rotor. The steam turbine operating conditions are very severe ranging fromregions of low temperature and moderate stress to regions of very high temperature and high stress (SudheendraV.S. et.al(1989)). Close dimensional control must be maintained which requires consideration of deformation and rigidity. The associated problems are service life, size, and weight, cost of materials and processing methods of materials. The life, size and weight are dictated by the functional requirements. It is necessary to know for a given temperature and a given expected life, the maximum amount of stress that can be carried withinthe assigned criterion for failure. The criterion for failure could be rupture or specified allowable deformation. The nature of the loads also should be considered as static loadslead to the consideration of static strength andfluctuating loads require consideration of endurance or fatigue strength. The three-dimensional flow around a guide vane should be studied numerically. The fluid flow and heat transfer in the region of the end plates should be the focus of the investigation. The purpose of this work should be to understand the influence of criticalparameters on the platform heat transfer.The finite element model used in this analysis the time steps are varied and the results are stored in load step files. ANSYS allows a coupled field analysis wherein structural and thermal analysis is carried out for the same mesh. This approach is taken. The analysis is run assuming a uniform initial metal temperature of 45 deg Cfor cold start and 200 deg C for warm start. The heat transfer coefficients are applied on the model along with the steam bulk temperatures. The initial time step size for transient analysis is quite important from the accuracy and the convergence standpoint. We have taken a timestep size of 600 sec.

The transient effects due to the sudden heating of the disk are studied. As the time progresses, the various portions of the disk are subjected to different thermal gradients. These gradients at various time intervals give raise to different stresses. The stressesincrease gradually, attains maximum values and thenstabilize. To verify the results the equation is made useto calculate the stresses with the knowledge of the boundary conditions.

$$\frac{d}{dr}\left[\frac{\sigma_t}{E}\right]-\frac{d}{dr}\left[\frac{\mu\sigma_r}{E}\right]+\frac{d}{dr}\left[\alpha\Delta T\right]-\frac{(1+\mu)(\sigma_r-\sigma_t)}{E_r}=0$$

The heat transfer coefficients across disk/rotor faces are calculated by making use of the turbulent local Nusselt number for an isothermal rotating disk.

Using the Dittus - Boelter equation [10]:
Nur = 0.020 Pr exp.0.333 Re exp.0.8 = hD/k
Where,
Nur - Nusselts Number = hD/k
Pr - Prandtls Number [10]
Re - Reynolds Number = (ρ.Vr.D)/μ
Where,
h-Convective heat transfer coefficient W/mm2 deg C k-Thermal Conductivity W/mm/deg C
ρ- Density of material Kg/mm3
μ-Poisson’s Ratio
D = 4*Area/2*Perimeter
V_r = Relative velocity calculated from velocity triangle mm/sec
Before carryingout the FE analysis, this study assumes:
1. Rotating shaft is considered to be isothermal.
2. Shaft material is completely elastic at stress distribution induced by mechanical forces and thermalloads.
3. Stress is proportional to strain.
4. Shaft material is assumed as 30Cr1MO1V 5. Nozzle angle is assumed to be 10^0.
6. Blade is considered as rectangular instead of aerodynamic profile.
7. Instead of complete model only 1 degree model is considered.
8. Tetrahedral element is considered for meshing.
9. A time step of 10 minutes i.e. 600 sec is assumed. Total no. of time steps=56.Time step of 1 minute 0r 1 second can also be

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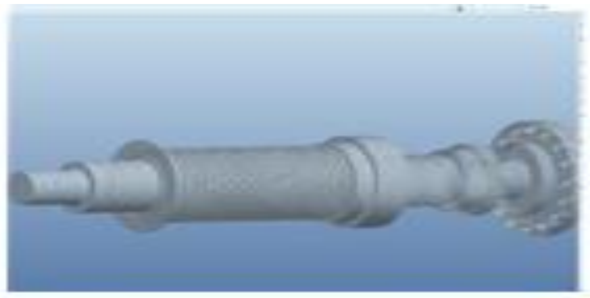
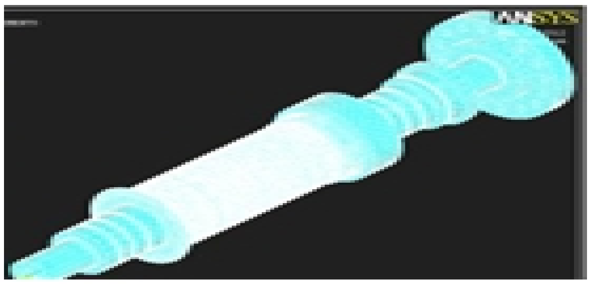
taken.

Type of Element selected for F.E. Analysis-Solid98 (A Structural-Thermal coupled field Element)

TRANSIENT THERMAL STRESS ANALYSIS

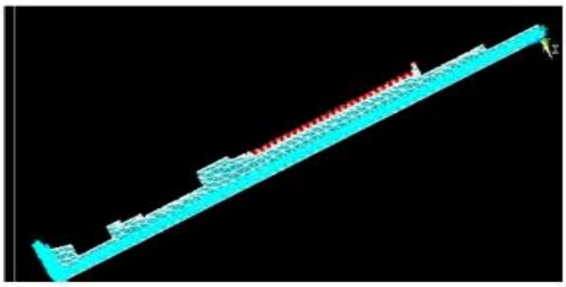
The thermal analysis has a single degree offreedom at each node, the temperature. Ax symmetric boundary conditions in the form of displacement constraints and thermal gradients are applied on the model. Thermal analysis is performed on the disk. The Thermal analysis is carried out for cold start up cycle. For cold start cycle the ambient temperature was 45 deg C and maximum temperature was 540 deg C for cycle duration of 560 minutes (33600 sec). The blades are not modeled in this analysis. The thermal stresses are obtained from this analysis are the radial, hoop and Von misses stresses.

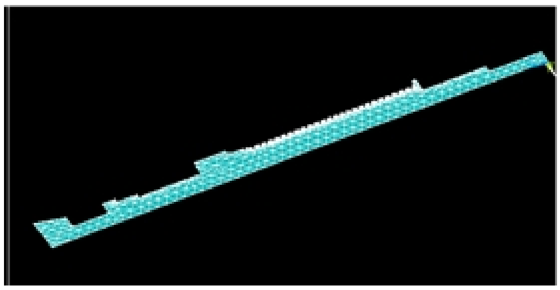




CAD Model HPT

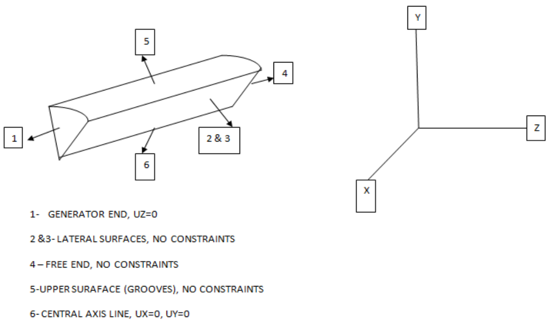
FE Meshed Model HPT





FE Model Constrained and Thermally Loaded

1 Degree HPT FE Meshed Model



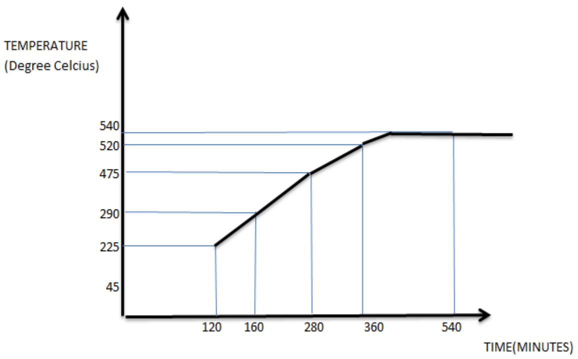


Fig.4.2 Structural Boundary Conditions

Fig.4.3 Thermal Loading

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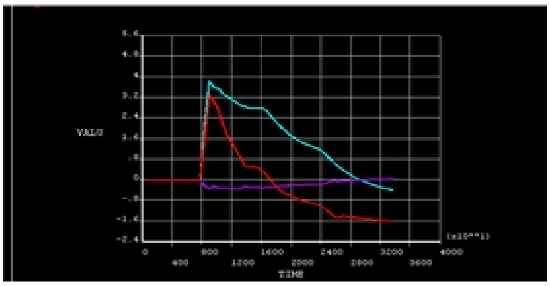


Fig 1(a)Radial Stress Distribution at different parts of Rotor

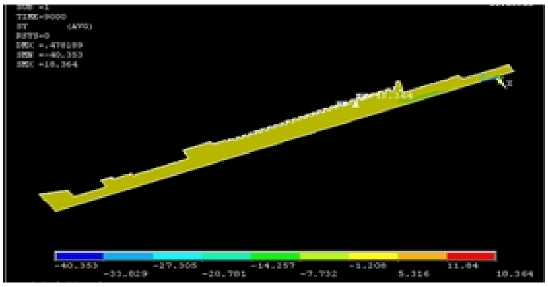


Fig 1(b)Radial Stress Distribution at different parts of Rotor

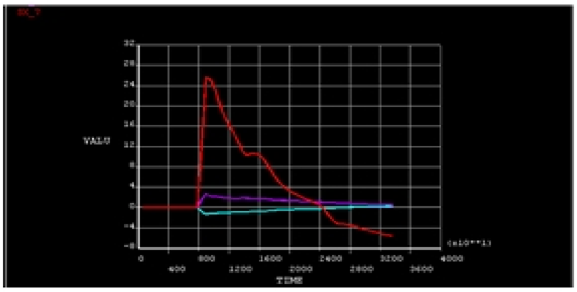


Fig 2(a)Hoop Stress Distribution at different parts of Rotor

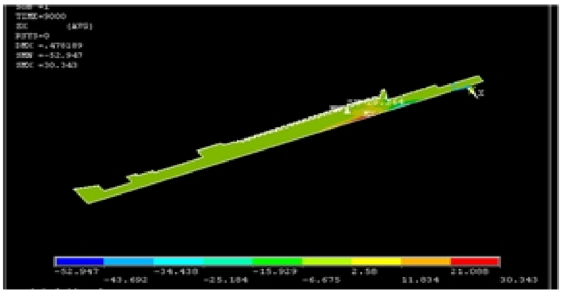


Fig 2(b)Hoop Stress Distribution at different parts of Rotor

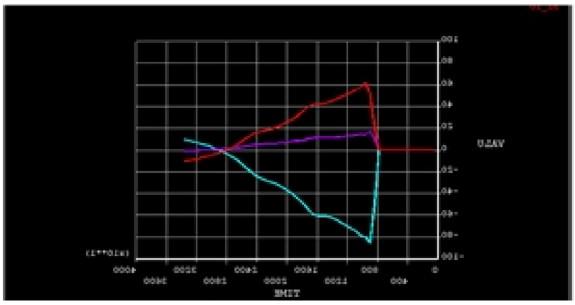


Fig 3(a)Axial Stress Distribution at different parts of Rotor

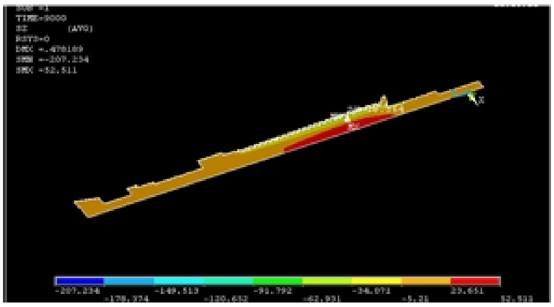


Fig 3(b)Axial Stress Distribution at different parts of Rotor

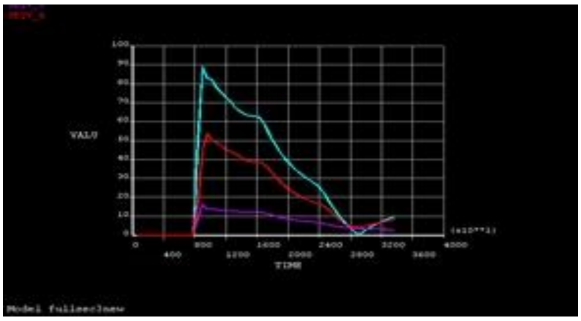


Fig 4(a)VonMissesStress Distribution at different parts of Rotor

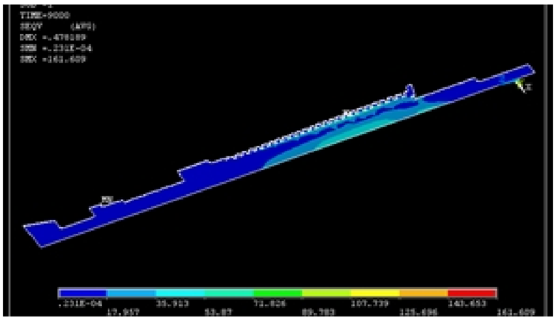


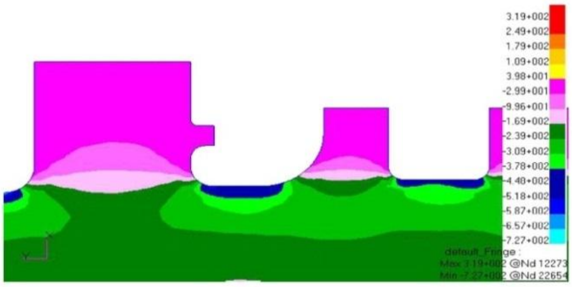
Fig 4(b)Von Misses Stress Distribution at different parts of Rotor

•Fig.1 (a) & 1(b) Shows radial stress variation with respect to time in different parts of rotor, the maximum transient radial stress for the selected 3 nodes occurs at 9000 sec (150 min) time step. The maximum radial stress at this time step is found to be 18.364 Mpa which is at 6th Stage (Groove) of HPT.

•Fig.2 (a) &2(b) Shows hoop stress variation withrespect to time in different parts of rotor, the maximum transient hoop stress for the selected 3 nodes occurs at 9000 sec (150 min) time step. The maximum hoop stress at this time step is found to be 30.343 Mpa which is at Axis line below last stage of HPT.

•Fig 3 (a) & 3(b) Shows axial stress variation with respect to time in different parts of rotor, the maximum transient axial stress for the selected 3 nodes occurs at 9000 sec (150 min) time step. The maximum axial stress at this time step is found to be 62.855 Mpa which is at Axis line below first 20 stages of HPT

•Fig 4 (a) & 4(b). Showing Von misses stress variation with respect to time in different parts of rotor, the maximum transient Von misses stress for the selected 3nodes occurs at 12000 sec (200 min) time step. The maximum Von misses stress at this time step is found to be 80 Mpa which is at 11th Stage (Groove) of HPT.



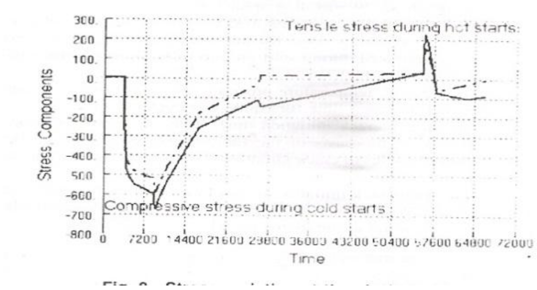


fig.5(a) high compressive stress at the surface during cold start

fig.5(b) stress variation at the shaft surface

stress relaxation under high operating temperature will take place. A stress relaxation run was carried out using a steady state Norton creep law

$\varepsilon' = A\,\sigma^{n} t^{m}$ …………..(4)

where ε'is the uniaxial equivalent creep strain rate,σ is the stress from the centrifugal load, t is the total time running hours, A is the Norton law constant, n is the stress index, and m=0 for a steady state creep.

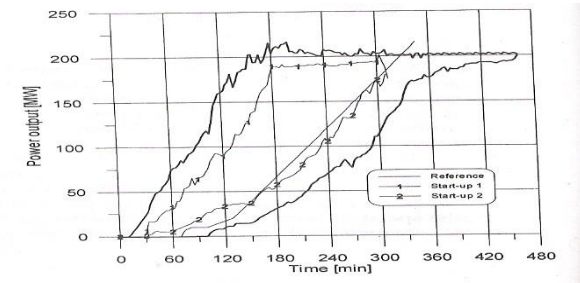


fig.6(a) start-up curve power output vs time

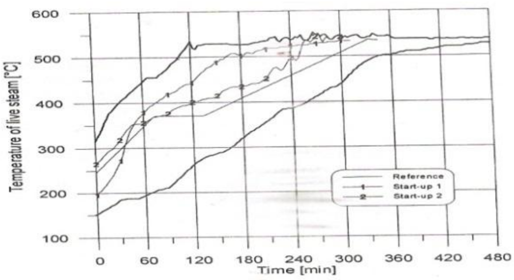


fig.6(b) start-up curve steam temperaturevs time

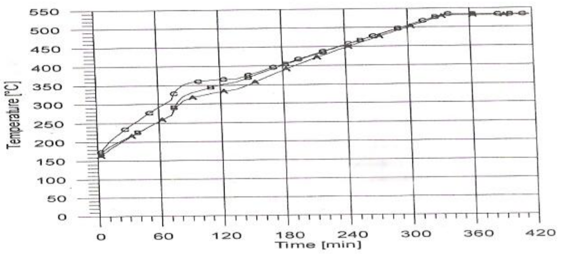


fig.7(a) reference start-up curve cut-off valve temperaturevs time

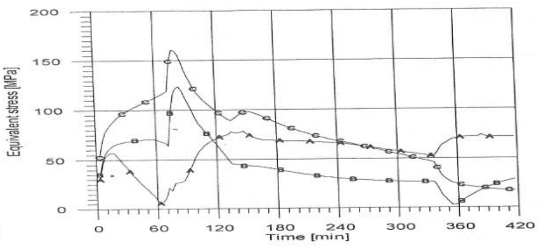


fig.7(b) reference start-up curve equivalent stress vs time

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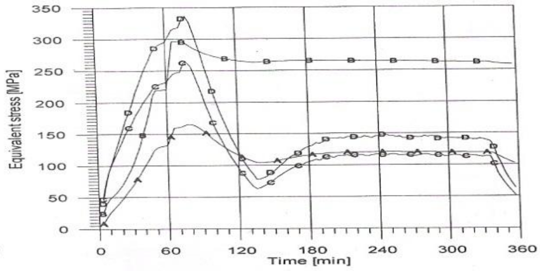


fig.8(a) reference start-up curve equivalent stress vs time

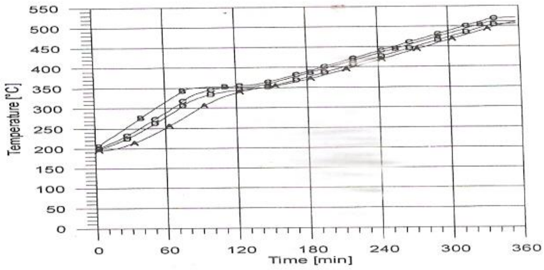


fig.8(b) reference start-up curve IP rotor temperaturevs time

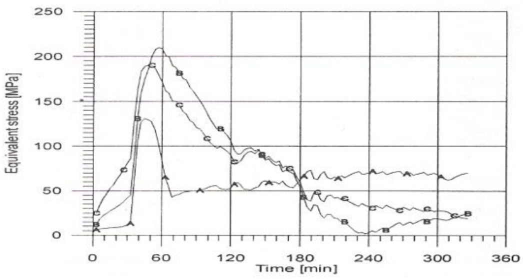


fig.9(a) equivalent stress vs time

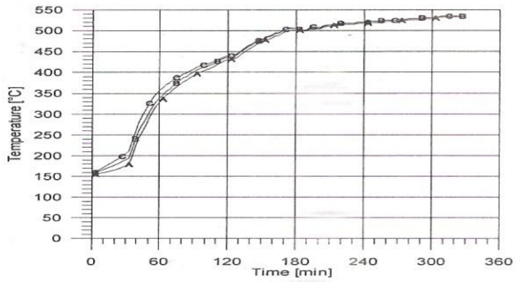


fig.9(b) HP cut-off valve temperaturevs time

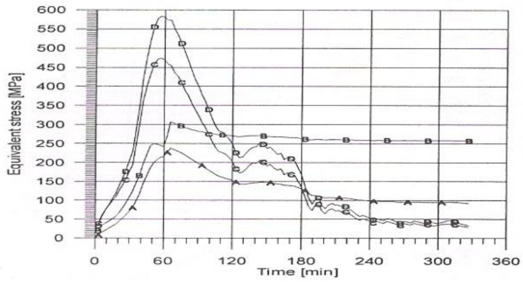


fig.10(a)equivalent stress vs time

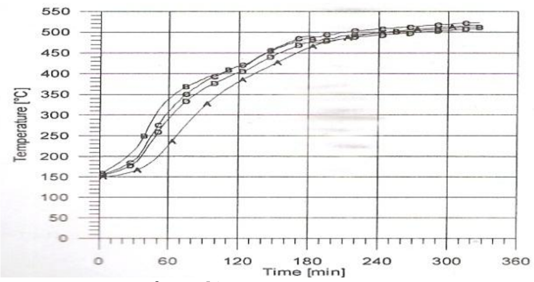


fig.10(b) IP rotor temperaturevs time

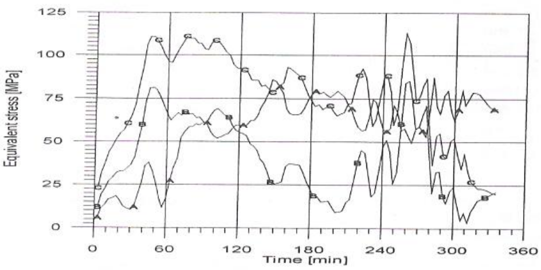


fig.11(a) equivalent stress vs time

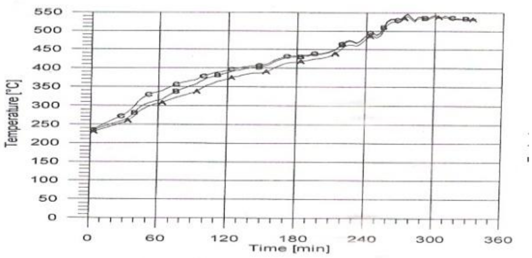


fig.11(b) HP cut-off valve temperature vs time

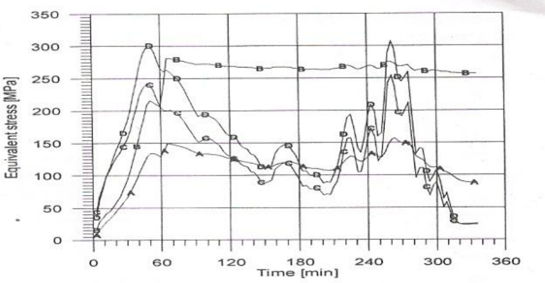


fig.12(a) equivalent stress v/s time

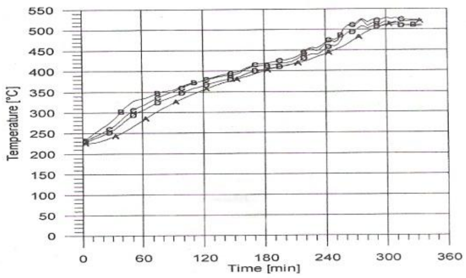
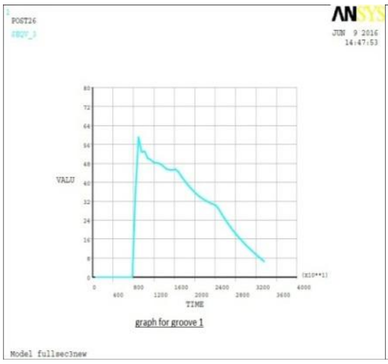
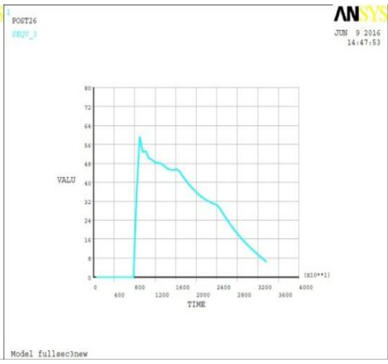
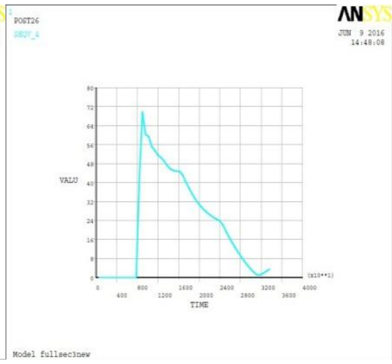
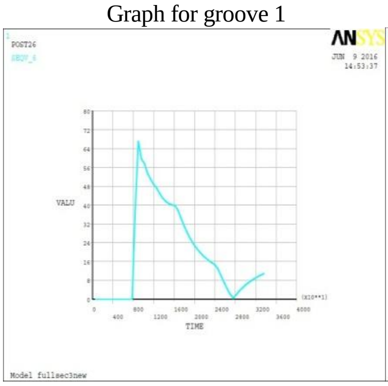


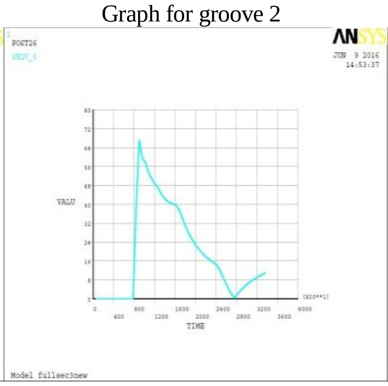
fig.12(b) IP rotor temperature v/s time

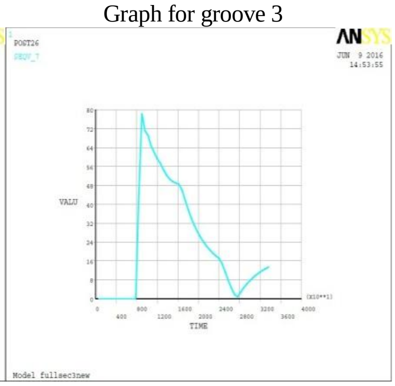

Graph for groove 1

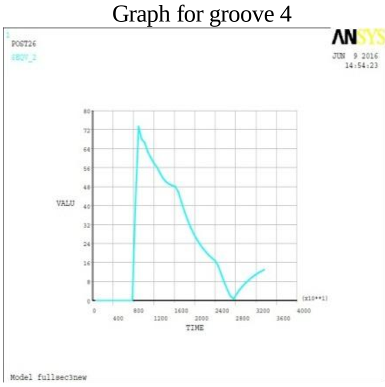

Graph for groove 2

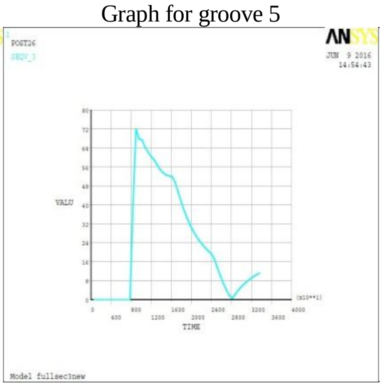

Graph for groove 3

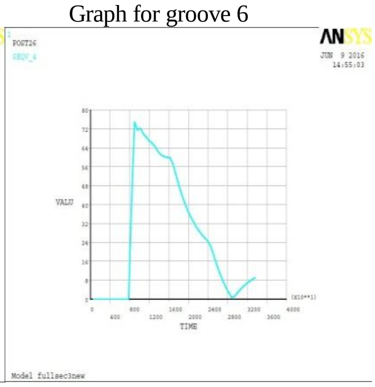

Graph for groove 4

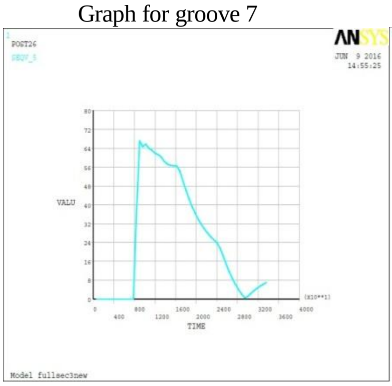

Graph for groove 5

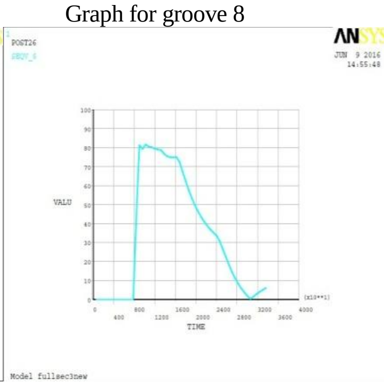

Graph for groove 6

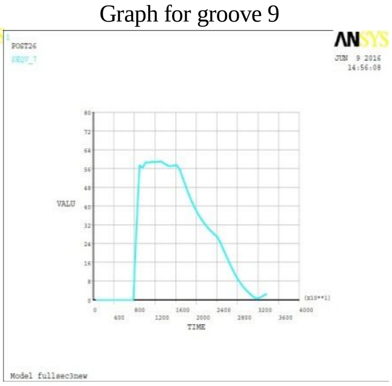

Graph for groove 7


Graph for groove 8


Graph for groove 9


Graph for groove 10


Graph for groove 11


Graph for groove 12

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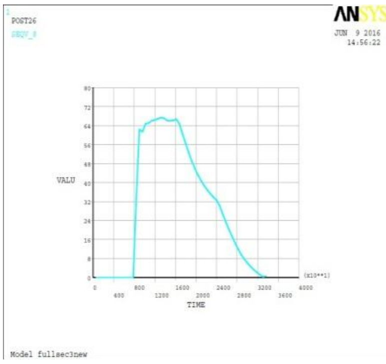
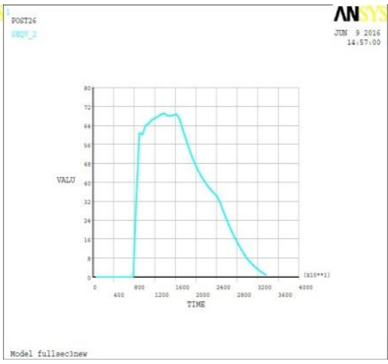
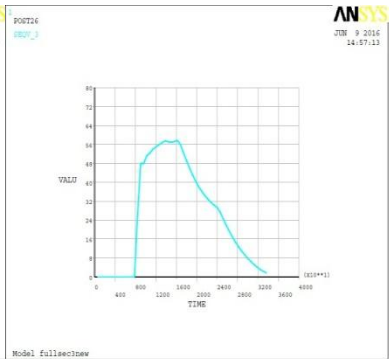


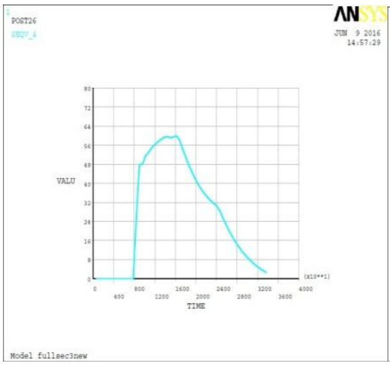
Graph for groove 13



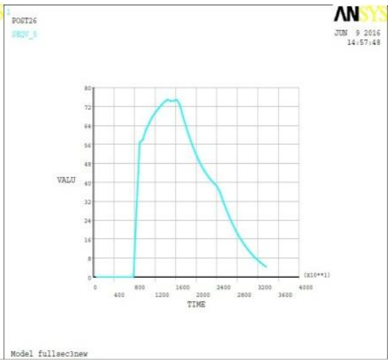
Graph for groove 14



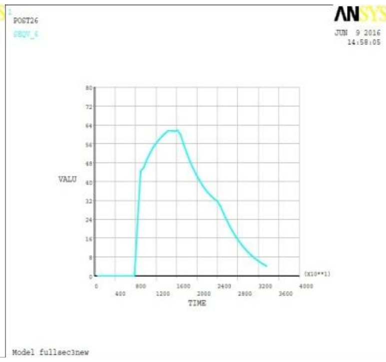
Graph for groove 15



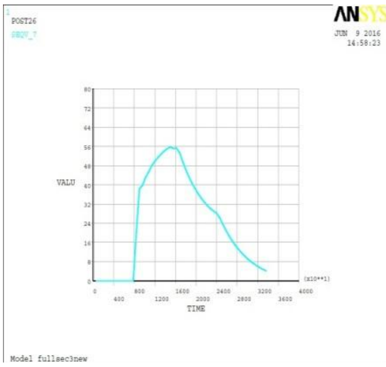
Graph for groove 16



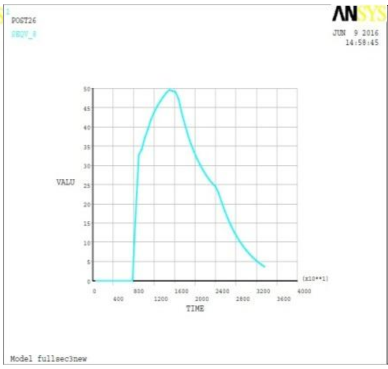
Graph for groove 17



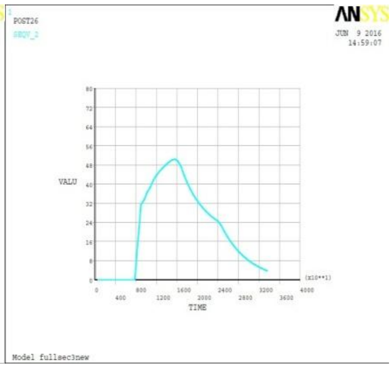
Graph for groove 18



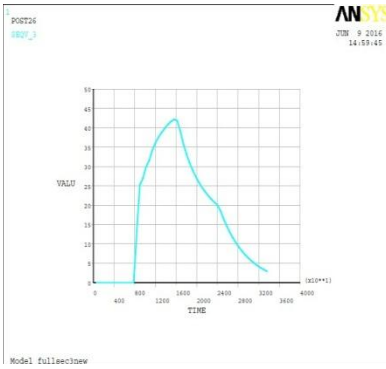
Graph for groove 19



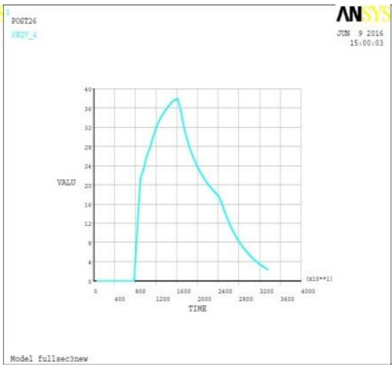
Graph for groove 20



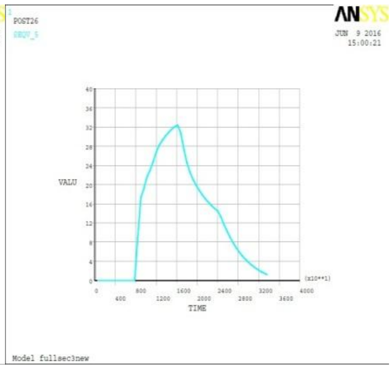
Graph for groove 21



Graph for groove 22



Graph for groove 23



Graph for groove 24

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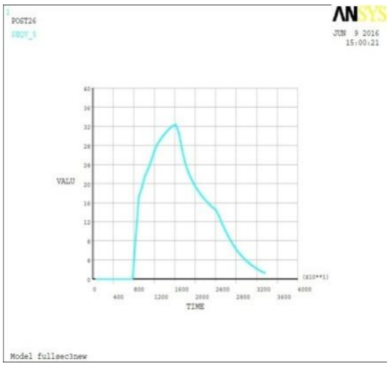

Graph for groove 25

Readings for groove no. 11.

Groove 11				
Sr.No.	Steps (Sec)	Step(Min)	Thermal Von Misses	
			Mpa	Psi
1	4000	67	0	0
2	4604	77	2	290.076
3	5208	87	6	870.228
4	5812	97	9	1305.342
5	6416	107	11	1595.418
6	7020	117	13	1885.494
7	7624	127	16	2320.608
8	8228	137	20	2900.76
9	8832	147	30	4351.14
10	9436	157	40	5801.52
11	10040	167	50	7251.9
12	10644	177	60	8702.28
13	11248	187	72	10442.736
14	11852	197	80	11603.04
15	12456	207	79	11458.002
16	13060	217	78	11312.964
17	13664	227	77	11167.926
18	14268	237	76	11022.888
19	14872	247	76	11022.888
20	15476	257	75	10877.85
21	16080	267	75	10877.85
22	16684	277	70	10152.66
23	17288	287	68	9862.584
24	17892	297	63	9137.394
25	18496	307	60	8702.28
26	19100	317	55	7977.09
27	19704	327	52	7541.976
28	20308	337	50	7251.9
29	20912	347	48	6961.824
30	21516	357	43	6236.634
31	22120	367	40	5801.52
32	22724	377	39	5656.482
33	23328	387	35	5076.33
34	23932	397	33	4786.254
35	24536	407	29	4206.102
36	25140	417	25	3625.95
37	25744	427	23	3335.874
38	26348	437	20	2900.76
39	26952	447	18	2610.684
40	27556	457	13	1885.494
41	28160	467	10	1450.38
42	28764	477	10	1450.38

43	29368	487	10	1450.38
44	29972	497	9	1305.342
45	30576	507	9	1305.342
46	31180	517	9	1305.342
47	31784	527	8	1160.304
48	32388	537	8	1160.304
49	32992	547	7	1015.266
50	33596	557	7	1015.266

Groove number V/S Von-misses thermal stress

Von-misses thermal stress in HP shaft

	Thermal stress (MPa)							
Time In min → Groove No. ↓	67	133	200	267	333	400	467	560
01	0	16	66	69	64	58	47	33
02	0	17	48	45	36	29	18	07
03	0	18	50	46	29	24	09	04
04	0	18	49	45	28	18	00	09
05	0	16	48	40	23	15	00	10
06	0	17	58	49	28	17	01	14
07	0	16	57	48	26	16	02	15
08	0	17	58	51	30	20	0.5	11
09	0	16	67	69	37	24	0.5	09
10	0	18	62	56	34	24	04	07
11	0	20	80	75	50	33	10	07
12	0	16	60	58	38	28	08	03
13	0	17	67	68	44	32	12	01
14	0	16	68	70	47	35	15	01
15	0	14	55	58	40	30	14	02
16	0	15	56	60	40	31	15	04
17	0	16	70	76	51	38	18	06
18	0	12	56	60	43	32	15	06
19	0	11	50	55	38	28	14	05
20	0	06	44	49	33	24	13	04
21	0	08	44	50	32	24	12	04
22	0	06	37	43	28	20	09	03
23	0	05	32	38	24	18	08	02
24	0	04	26	33	19	14	06	02
25	0	02	11	13	07	05	02	0.5

Maximum von-misses thermal stress v/s time

Groove no.	Maximum von-misses stress(MPa)	Time (Minute)
01	69	267
02	48	200
03	50	200
04	59	200
05	48	200
06	58	200

07	57	200
08	58	200
09	69	267
10	62	200
11	80	200
12	60	200
13	68	267
14	70	267
15	58	267
16	60	267
17	76	267
18	60	267
19	55	267
20	49	267
21	50	267
22	43	267
23	38	267
24	33	267
25	13	267

CONCLUSIONS

This study investigates the transient thermal stresses arising in a 210 MW high pressure stream turbine rotor. This work employed FE software (Ansys 11) to evaluate the transient thermal stresses. The resultsshow that the transient thermal stresses are higher than the steady state stresses. In addition, the stresses are calculated for cold start up cycle in which the temperature gradients are higher as compared to the warm start up cycle as the temperature difference between the initial temperature of the rotor (45^{o}C) and the final steam temperature (540^{o}C).

It is observed that at 67 Minute the value of Von-Misses thermal stress is 0 Mpa. From 1st Groove to 25th Groove.

It is observed that at 133 Minute the value of Von-Misses thermal stress is gradually increasing from 16 Mpa.to 20 Mpafrom the 1st Groove to 11th Groove and gradually decreasing to a value 02 Mpa.to a last 25th Groove.

It is observed that at 200 Minute the value of Von-Misses thermal stress is increasing from 48 Mpa.to 80 Mpa.from the 1st Groove to 11th Groove and decreasing to a value 11 Mpa.to a last 25th Groove. The maximum value of stress is 80 Mpa at 11th Groove.

It is observed that at 267 Minute the value of Von-Misses thermal stress is increasing from 40 Mpa.to 76 Mpa.from the 1st Groove to 17th Groove and decreasing to a value 13 Mpa.to a last 25th Groove.

It is observed that at 333 Minute the value of Von-Misses thermal stress is increasing from 23 Mpa.to 51Mpa.from the 1st Groove to 17th Groove and decreasing to a value 07 Mpa.to a last 25th Groove.

It is observed that onwards 400 Minute to 560 Minute initially the value of Von-Misses thermal stress is high at 1stGroove and then decreasing to a last 25th Groove.

REFERENCES

[1] G. SukhvinderKaurBhatti, ShyamalaKumari, M LNeelapu, C Kedarinath, Dr. I N NiranjanKumar,"Transient State Stress Analysis On An Axial FlowGas Turbine Blades And Disk Using FiniteElement Procedure".in *Int. Conf.on HEATTRANSFER, THERMAL ENGINEERING and ENVIRONMENT, Elounda, Greece, August 21-*

[2] ZvonimirGuzović, BranimirMatijašević, TihomirMihalić"Characteristics Of Non- StationaryThermal Stresses In The Low-Pressure Part OfThe Rotor"*15th International Research/ExpertConference TMT-2011,Prague,Czech Republic(2011)*

[3] Chunlin Zhang, Niansu Hu, JianmeiWang,Qiping,chen,Feng He,Xiaoli " Thermal StressAnalysis for Rotor of 600MW Steam Turbine"*978-1-4244-4813-5/10/&25.00c/2010/IEEE(2010)*

[4] Deepak Dhar, A. M. Sharan." Transient Stress Analysisand Fatigue Life Estimation of Turbine Blades" *Journal of Vibration and Acoustics OCTOBER2004, Vol. 126 Õ 495(2004)*

[5] Yong Li, Haoran Sun, YuhuoNie"ThermalStressAnalysis of 600MW Steam Turbine Rotor inDifferent Governing Modes" *978-1-422-4813-528-31-March2010 IEEE(2010)*

[6] Sudheendra,V.S, S.Ramamurthy,K.Murugesan, "TransientThermal Analysis Of A Turbine Rotor" *nalir.nal.res.in/8928(1989)*

[7] R. NagendraBabu, K. V. Ramana, and K. MallikarjunaRao, " Determination of Stress Concentration Factors of a Steam Turbine Rotor by FEA"*World Academy of Science, Engineering andTechnology 39 (2008)*
[8] LucjanWitek "Failure Analysis of turbine disc of anaero engine" *Elsevier, Engineering FailureAnalysis 13 9-17. (2006)* S.Barella, M.Bellogini, M.Boniardi, S.Cincera"Failure Analysis of steam turbine rotor"*Elsevier, Engineering Failure Analysis 18 1511-1519(2011).*
[9] S.R.Holdworth, E.Mazza, L.Binda, L.Ripamonti "Development of thermal fatigue damage in 1CrMoV rotor steel" *Elsevier, NuclearEngineering Design 237(2007)2292-2301(2007)*
[10] Yong Li, Haoran Sun, YuhuoNie "Thermal Stress Analysis of 600MW Steam Turbine Rotor inDifferent Governing Modes" *978-1-422-4813-5 28-31-March2010 IEEE(2010)*
[11] Xiaoling Zhang, "High Temperature Rotors: Failure Mechanisms and Remnant Life Assessment",*Feb 2009,Vol.131/011406-1,Journal of Pressure Vessel Technology,ASME.*
[12] Grzegorz Nowak, Andrej Rusin, "Lifetime Deterioration of Turbine components During Start-ups",*OMMI(Vol.3,Issue1)April 2004.*
[13] R.NagendraBabu, K.V.Ramana and K.MallikarjunaRao, "Prediction of stress concentration effect under thermal and dynamic loads on a high Pressure turbine rotor", *International e-journal of Mathematics and Engineering 67(2010),P667-678.*

NOMENCLATURE

Re : Reynolds's number

Pr :Prandtl number

Nu :Nusselt number

K : Thermal conductivity, W/m deg C

E :Young's Modulus, × 1000 Mpa

σu: Ultimate strength, N/mm2

σy :Yield strength, N/mm2

Cp : Specific heat J/kg deg C

ρ : Mass density, kg/m3
αd : Thermal diffusivity, mm/mm deg C T : Temperature deg C

Mpa : Mega pascal, i.e., N/mm2

HPT : High Pressure Turbine

μ: Poisson's ratio

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