RESEARCH ARTICLE

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Effect of Change in Startup Cycle Time on Transient Thermal Stresses in Steam Turbine Rotor

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Abstract

During the turbine start stop period, occurring at brief intervals, the temperature of the steam turbine rotor varies. Because of thesignificant temperature gradients, this shift in temperature causes transitory thermal stresses in the rotor. Changes in material parameters such as thermal expansion coefficient, modulus of elasticity, thermal conductivity, specific heat, Poisson's ratio, and others cause transient stresses. Before designing the rotor, it is necessary to estimate transient thermal stresses. The finite element study of a 210 Megawatts steam turbine rotor for transient thermal loading is investigated in this paper. 30Cr1Mo1V was used to make the rotor. CAD software (PRO-E) was used to create the high-pressure turbine model. Since the steam turbine shaft (rotor) was symmetric about the axis along which it rotates, a 1 Degree slice model was used. Ansys 11 was used for the investigation of the Finite element model for transient thermal stresses for reduced cold startup cycle. The analysis was carried out for startup cycles of 560, 440, and 320 minutes, respectively. The comparative results were studied, and it was discovered that although the thermal stresses (transient) in the reduced cold startup cycle of 440 minutes are higher as compared to the transient thermal stress value for the actual cold startup cycle (560 minutes) but are within safe limits of yield and tensile strength. The thermal stress values for the other reduced cold startup cycle (320 minutes) are quite higher than the allowable yield and tensile strength of the material. As a result, it cannot be considered a feasible startup cycle. Thermal stresses in transient states were significantly higher than steady state thermal stresses.

Keywords: Ansys., FEM, Startup cycle, Turbine rotor, thermal stresses.

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Introduction

Rotating elements have traditionally been the subject of research and study because of their widespread use in industry. One such example is the rotor of a steam turbine. During the startup cycle, turbine rotors of the power plants are exposed to high temperatures. Material parameters such as modulus of elasticity, thermal conductivity, specific heat, and density change over time because the turbine rotor is subjected to huge temperature variations. In such circumstances, if the turbine rotor isn't properly designed to account for the transitory nature of the effect, the turbine rotor is prone to failure.

The transient behavior of these turbines is just as significant as the steady state behavior. It takes a while for the rotor to reach an equilibrium temperature. The temperature changes over time during this interim phase and the rotor is said to be in a transitory condition. While in this unstable form, it is exposed to a variety of temperature gradients. The thermal shocks that are induced in the rotor are identified via transient thermal analysis.

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Literature Review

Y Maruthi Prasad(2007)[1] This paper employs finite element analysis to evaluate transient thermal stresses generated during startup and shut-down cycles. Thermal gradients are the most important sources of thermal stress generation in a steam turbine rotor, which is primarily caused by improper turbine rotor loading. Finite element analysis with ANSYS was performed using linear quadrilateral axisymmetric elements, and the time-dependent variables such asdensity, temperature, thermal conductivity, and so on were used. Under power plant operational conditions, a method of evaluating remnant turbine life is discussed.

Nagendra Babu et al. (2008)[2] In this paper, the stress concentration factor (SCF) in a steam turbine rotor was investigated using a finite element model because it is important in the design of geometrically complex such as a steam turbine rotor. ANSYS workbench was used to discretize and fine mesh the model using a two-dimensional ax symmetric element. Von Mises stress was determined to be 144 MPa, and the SCF was determined to be 1.34, both of which are within the safe limits. The results were compared to the theoretical SCF charts developed by Peterson.

Chunlin Zhang et al. (2010)[3] this article addresses the thermal shock assessment of a 600 MW turbine rotor in a hot startup cycle using ANSYS. The steam turbine rotor was thermomechanically finite and elementally analyzed using actual operational data. They indicated that the highest stresses develop at the beginning of the turbine casing. The results of the finite element model can be used to optimize the monitoring of turbine parameters.

Yong Li et al. (2010)[4] The alteration in thermal stresses in various governing modes of a 600 MW steam turbine is discussed in this paper. In the governing and first stage of the high-pressure turbine, two-dimensional analyses are performed on the High Pressure-Intermediate Pressure Turbine Model. The analysis is being carried out with the nozzle governing and the throttles governing in mind. The findings indicate that the axial stresses in nozzle governing are greater than those in throttle governing. The analysis results can be used as a reliable source for online monitoring of thermal stress variation.

Zvonimir Guzoviæ et al. (2011)[5] In this paper, an algorithm for modeling transient thermal stresses in a 30 MW turbine rotor is proposed. The analysis shows stress calculations in various critical areas such as the central bore, turbine disc, rear labyrinth gland, and low-pressure rotor. The analysis determines time-dependent variables such as temperature, stresses, heat flux, and specific heat. The findings indicate that the equivalent Von Mises stress increasesfrom the central bore to the disc, with a maximum value of 9.7 MPa at the disc and 34 MPa in the bore.

Barella et al. (2011)[6] The failure of a steam turbine shaft of 60 MW capacity is investigated in this paper. Therotor's material was CrMoNiV steel, and the failure occurred after 10 years of successful operation. Visualexamination was one of several analyses used to determine the root cause of the failure. For failure analysis, thefollowing analyses were performed: scanning electron microscope fractography, ocular inspection, micro-hardness measurement, and characterization of microstructure. The results show that the fracture occurred at the first stage ofthe turbine as a result of frequent startup and stop cycles, the method of securing the blades, and mechanical fatigue.

Wang et al. (2016)[7] The effect of transient thermal stresses on fatigue due to creep in the rotor of a 1000 MW steam turbine is investigated in this paper. A viscoplastic constitutive model with damage was used to study the deformation caused by creep-fatigue behavior. There was finite element modelling and thermo-mechanical analysis. The results show that creep damage caused by temperature change was greater than creep damage caused by steady-state temperature.

P.A. González-Gómez(2018)[8] This paper discusses a methodology for determining the behavior of steam generation plant heat exchangers. This paper focuses on the design considerations for a proposed solar power plant. The work is done in two startup cycles, using both the analytical and finite element methods. The results show that using this methodology, real-time stress can be determined and thus used to assess the remaining life of turbine components.

Finite Element Modeling

The operation of power machinery is cyclical in nature, with the typical work cycle consisting of the installation's startup, power generation, and shut-down. Since the first and last parts of the working cycle are non-productive, they should be kept as short as possible. Alternatively, high deviations in loads happen during these duration, resulting in a higher failure risk from fatigue processes occurring during load variations, particularly thermal loads.

The cold startup cycle means the turbine starts up after a long period of shutting down, i.e., about after 250 hours of shutdown, and the warm startup cycle means that the turbine starts up after a shorter period of shutting down, i.e., about after 36 hours of shut down.

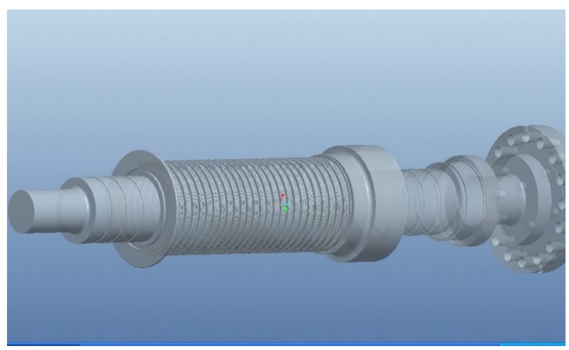

Figure 1: PRO E Model of High-pressure turbine(HPT)

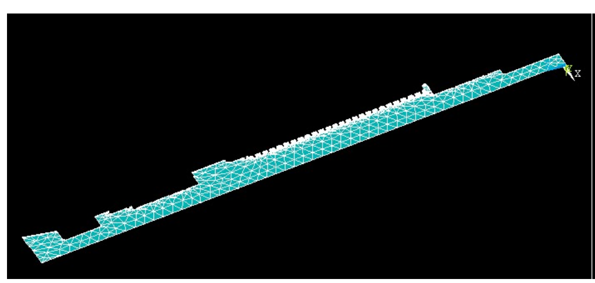

Figure 2: 1degree Model of HPT

Modeling

A computer-based modeling approach is used for drawing and analysis of the rotor. The software PRO-E was used for computer-aided modeling, and ANSYS 11 was used for analysis (Figure 1), and 1degree Model of HPT (Figure 2).

For Carrying out transient thermal analysis with Finite Element Method(FEM) of turbine components of 210 MW capacity steam turbine, the following methodology is used:

- Modeling the steam turbine rotor using CAD software PRO-E.
- Finite Element Modeling using Ansys 11.
- Transient Thermal Stress Analysis for different Cold startup cycles of 560, 440, and 320 minutes respectively

Details and Design Criteria

Material

High-temperature gradients are applied to the turbine rotor and centrifugal forces simultaneously, necessitating a unique combination of material properties such as high strength, good fracture toughness, and low crack growth rate. The material chosen for the application is 30Cr1Mo1V,[3] a chromium, molybdenum, and vanadium alloy. This material was chosen for its high-temperature strength, corrosion resistance, and relative ease of fabrication. At high temperatures, this material maintains its strength.

Mechanical Properties

The elastic modulus, Poisson's ratio, coefficient of thermal expansion, and other physical properties of material changes with temperature. The nonlinear nature of the properties should be considered into account because the rotor is subjected to strong heat gradients. Even across a wide range of temperatures, the density of the material remains rather constant throughout the disc. The Poisson's ratio has a negligible impact on the stress distribution. As a result, correct stress estimates would necessitate accurate data on the disc material's elastic properties. The material's density is assumed to be 8220 kg/m^3.[1]

Design Criteria

The operating conditions for steam turbines are quite harsh, ranging from low temperatures and mild stress to extremely high temperatures and heavy stress. Close dimensional control is required, which necessitates consideration of deformation and rigidity. Service life, size, and weight, as well as material cost and processing processes, are all issues that need to be addressed. The functional requirements determine the life, size, and weight ofthe creature. It is vital to know the maximum amount of stress that can be carried within the designated failure criterion for a certain temperature and predicted life. Rupture or a specified permitted deformation could be used as a failure criterion. The nature of the loads should also be considered, as static loads necessitate the evaluation of static strength, whilst variable loads necessitate the inclusion of endurance or fatigue strength. Numerical analysis of the multi-flow around a guiding vane is required. The examination should concentrate on flow and heat transfer in close proximity to the plates. The goal of this project should be to figure out how important parameters affect platform heat transmission.

Temperature Gradient

Ambient temperature, working conditions, cooling flow, thermal barrier coating (TBC) quality, and other factors influence the temperature distribution of hot segment parts. Centrifugal and tangential loads, as well as temperature gradients, are used to create steady stresses in rotating blades. The disc is exposed to fast steam heating. The primary flow of steam heats the rim of the disc and the blades. The steam flowing out of the nozzle has a temperature of roughly 540 degrees Celsius. The ambient temperature is 45 degrees Celsius for a cold start cycle and 200 degrees Celsius for a warm start cycle. The analyses were carried out with these temperatures in mind.

Finite Element Analysis

In this analysis, the finite element model was employed, and the time steps were varied, with the results saved for each time step. ANSYS supports structural and thermal analysis to be performed on the same mesh. The analysis is

The transient effects of the rotor's sudden heating are investigated. As time passes, various temperature gradients are applied to different parts of the disc. These temperature variations at different time intervals cause various stresses. The stresses rise gradually, reach their apex, and then level off. The following equation is used to compute the stresses with the knowledge of the boundary conditions in order to confirm the results (Figure 3).

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

Table 1: Material Properties[3]

Temperature (°C)	30	100	200	300	400	500	600
Young's Modulus (E x 1000 Mpa)	214	212	205	199	190	178	…….
Poisson's ratio (μ)	0.288	0.292	0.287	0.299	0.294	0.305	…….
Thermal Conductivity (K), W/m deg C	…….	…….	44.8	42.8	40.3	37.5	35.3
Specific Heat (Cp), J/Kg deg C	…….	…….	599	624	666	720	804
Thermal Diffusivity(α), 10e-6), 10e-6 mm/mm deg C	…….	11.99	12.81	13.25	13.66	13.92	14.15

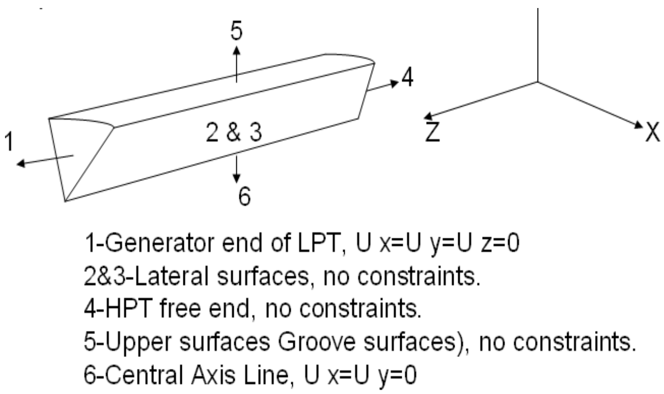

Figure 3: Structural boundary conditions

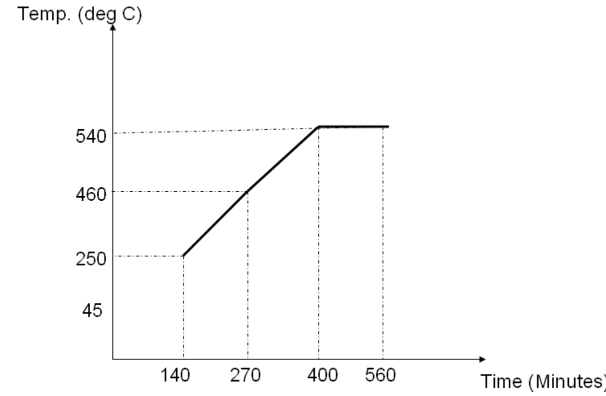

Figure 4: Thermal loading (560 minutes cycle)

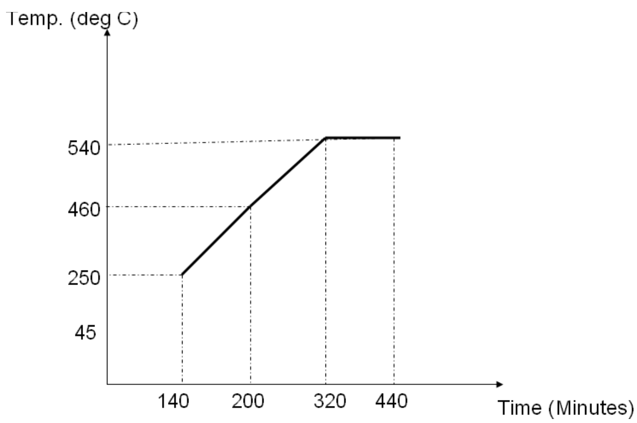

Figure 5 Thermal loading (440 minutes cycle)

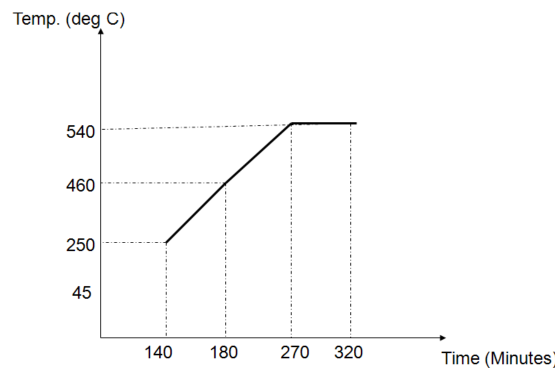

Figure 6: Thermal loading (320 minutes cycle)

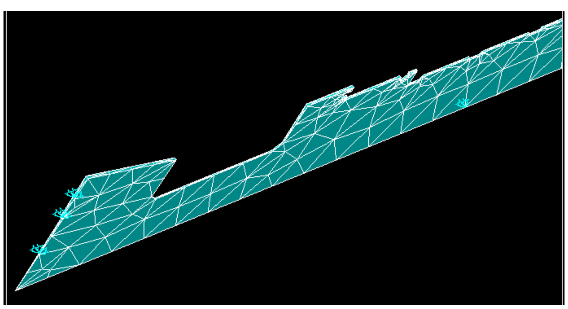

Figure 7: Structural constraints

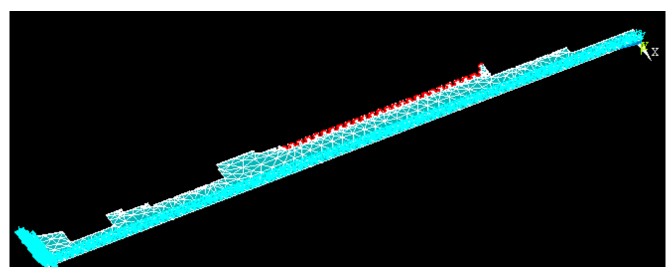

Figure 8: Thermal loading

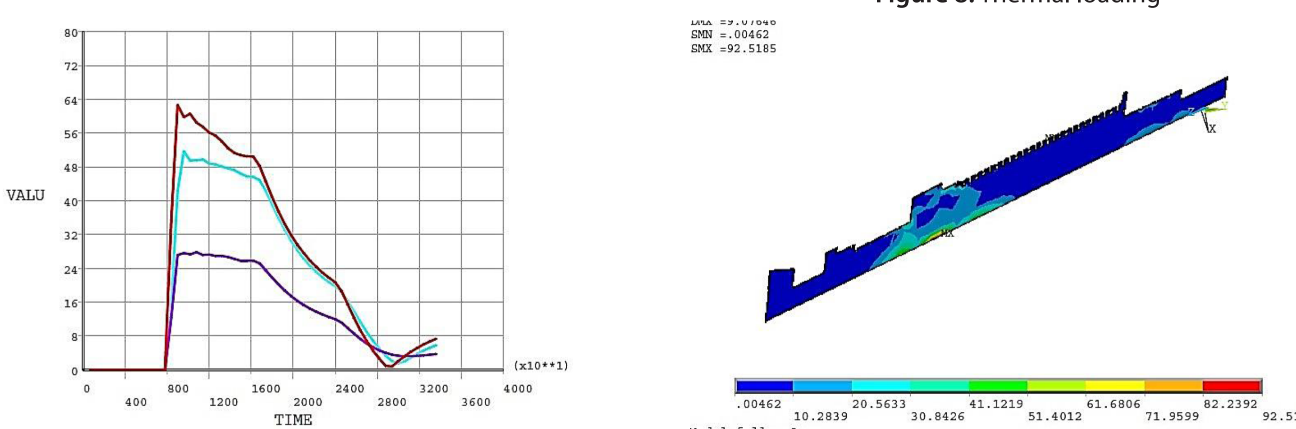

Figure 9: Von mises Stress variation and distribution in different parts of a rotor with respect to time (560 minutes)

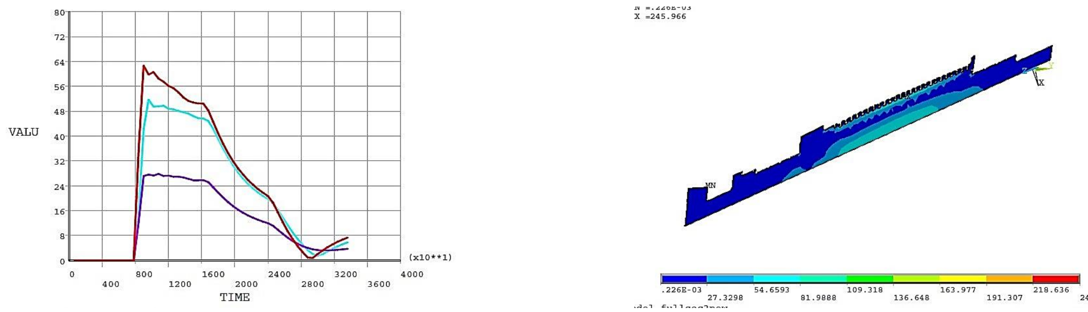

Figure 10: Von mises Stress variation and distribution in different parts of a rotor with respect to time (440 minutes)

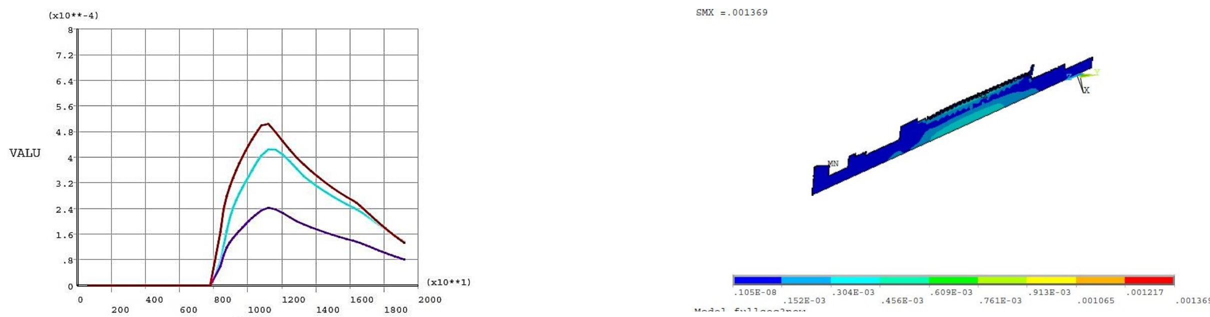

Figure 11: Von mises Stress variation and distribution in different parts of a rotor with respect to time (320 minutes)

Table 2: Comparative transient thermal stresses for various cold startup cycles

	560 minutes Cycle			*440 minutes Cycle*			*320 minutes Cycle*		
Particulars	Value	Time Step	Physical Location	Value	Time Step	Physical Location	Value	Time Step	Physical Location
Max.Von Mises Stress	161.61 Mpa	9000 secs (150 mins)	9th Stage (Groove)	247 Mpa	7800 secs(130 mins)	14th Stage (Groove)	281 Mpa	7800 secs(130 mins)	14th stage (Groove)
Max. Hoop Stress	30.343 Mpa	9000 secs(150 mins)	Axis line below last stage	49 Mpa	7800 secs (130 mins)	Axis line below last stage	54 Mpa	7800 secs(130 mins)	Axis line below last stage
Max. Radial Stress	18.364 Mpa	9000 secs(150 mins)	6th Stage (Groove)	24 Mpa	7800 secs (130 mins)	6th Stage (Groove)	27 Mpa	7800 secs(130 mins)	6th stage (Groove)
Max.Axial Stress	62.855 Mpa	9000 secs(150 mins)	Axis line below first 20 stages	90 Mpa	7800 secs (130 mins)	Axis line below first 20 stages	98 Mpa	7800 secs(130 mins)	Axis line below first 20 stages

Thus maximum thermal von mises stresses occurs at the time step of 7800 seconds (130 minutes) for the cold start cycle(440 minutes and 320 minutes). At 130 minutes the thermal load applied was 250°C and the ambient temperature was 45°C. The von mises stress was higher at this time step since the temperature gradient at this time step was higher (Table 2).

Table 3: Comparative Transient thermal strains for various cold startup cycles

	560 minutes Cycle			*440 minutes Cycle*			*320 minutes Cycle*		
Particulars	Value	Time Step	Physical Location	Value	Time Step	Physical Location	Value	Time Step	Physical Location
Max.Von Mises	7.62E-	9000secs (150mi	9th Stage	1.22E-	7800 secs (130	14th Stage	1.35E-	7800 secs (130	14th Stage
Strain	04	ns)	(Groove)	03	mins)	(Groove)	03	mins)	(Groove)
Max.Hoop	5.93E-	33600s secs (560	1st stage(entry	5.83E-	26400 secs (440	1st stage (entry	5.35E-	19200 secs (320	1st stage (entry
Strain	03	mins)	point)	03	mins)	point)	03	mins)	point)
Max. Radial	5.93E-	33600s secs (560	1st stage	5.83E-	26400 secs (440	1st stage (entry	5.35E-	19200 secs (320	1st stage (entry
Strain	03	mins)	(entry point)	03	mins)	point)	03	mins)	point)
		33600s			26400			19200	
Max.Axial	5.93E-	secs (560	1st stage	5.83E-	secs (440	1st stage (entry	5.35E-	secs (320	1st stage (entry
Strain	03	mins)	(entry point)	03	mins)	point)	03	mins)	point)

The maximum thermal strains occur at the last time step for each Cold Startup cycle, i.e., at 560, 440, and 320 minutes (Figure 4 to 6). These values are maximum at the first stage of (HPT) where the steam enters the turbine for all three cold startup cycles (Table 3) and Figure 7, 8).

Heat Transfer in Blades and Across the Rotor

The turbulent local Nusselt number for an isothermal rotating device is used to compute the heat transfer coefficients across rotor faces. 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

Vr = Relative velocity calculated from velocity triangle mm/sec

Results

The thermal analysis(transient) results for three cold startup cycles of 560, 440, and 320 minutes are presented graphically, and a comparative summary is tabulated in Table 3 and Figure 9 to 11.

Discussion

Based on the present analysis work, the following are the various discussions on the results of transient thermal analysis obtained for 3 different cold startup cycles:

Transient Thermal Stresses and Strains

- The maximum transient radial stress for the selected 3 nodes is found to be 27 Mpa which is at the 6^{th} groove (stage) of HPT, which occurs at the time step of 7800 seconds (130 minutes) for 320 min startup cycle.(Table 2)
- The maximum transient hoop stress for the selected 3 nodes s found to be 54 Mpa which is at Axis line below last groove (stage) of HPT, which occurs at 7800 seconds (130 minutes) time step for 320 minutes startup cycle. (Table 2)
- The maximum transient axial stress for the selected 3 nodes is found to be 98 Mpa which is at the Axis line below first 20 grooves (stages) of high-pressure turbine, which occurs at 7800 seconds (130 minutes) time step for 320 min startup cycle. (Table 2)
- The maximum transient Von mises stress for the selected 3 nodes is found to be 281 Mpa which is at 9^{th} groove (stage) of HPT, which occur 7800 seconds (130 minutes) time step for 320 min startup cycle. (Table 2)
- The maximum transient strain values for the selected 3 nodes is found to be 5.93e-3, which is at 1^{st} groove(entry point stage) of high-pressure turbine which occurs at 33600 seconds (560 minutes) time step for the 560 min. startup cycle. (Table 3)
- The maximum transient Von mises strain value for the selected 3 nodes is found to be 1.35e-3 which is at 9^{th}

groove (stage) of HPT occurs at at 7800 seconds (130 minutes) time step for 320 min startup cycle. (Table 3)

Conclusion and Future Scope

From the comparative results tabulated in the Tables numbers 2 and 3 it is observe that the transient thermal von mises stress values for reduced cold and warm startup cycle are higher than the actual cold and warm startup cycle. Although the von mises stresses for reduced cold startup cycle of 440 minutes (247 Mpa) are higher as compared to the thermal stresses for the actual cold startup cycle, these values are below the allowable yield and tensile strength of the material. The thermal stress values for the other reduced cold startup cycle (320 minutes) are quite higher than the allowable yield and tensile strength of the material, and hence they cannot be deemed feasible startup cycles.

Based on the results obtained it can be concluded that although the transient thermal stresses in the reduced cold startup cycle of 440 minutes and are higher as compared to the transient thermal stress value for the actual cold startup cycle (560 minutes) ,but are within safe limits of yield and tensile strength. Moreover, the objective of reducing the time of existing startup cycles can be accomplished by reducing the time to 440 minutes for cold startup cycle.

Future Scope

The blades on the rotor grooves were believed to be rectangular in this analysis. Using actual bladeprofile, the same study may be done more precisely. Thermal stresses were also measured in this area. The analysis can be performed independently for thermal and mechanical loads and then combined for thermo-mechanical loading. Inthis study, the nozzle angle is considered to be zero; however, an accurate nozzle angle can improve the result accuracy. The ANSYS FEM software offers a robust tool for transient analysis, with time steps ranging from 0.5 milliseconds to 3600 seconds. Because of the limits of the hardware design, a time step of 600 seconds was used in thisinvestigation. There is also the possibility of fine-tuning the time step to 10 seconds or even 1 second in order to acquiremore accurate results. Because the meshing in the current study is rather coarse, a finer mesh can be used to achieve convergence of results.

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