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Test Protocol for Accelerated Creep and Transient Thermo-Mechanical Testing of Super Critical Steam Turbine Rotor

Bishnu Pada Pal [1], M Naga Suresh[2], K Ramakrishna[3], A Narayana Teja[4], Supak Pore[5], Prabhat Kr. Hensh[6], Somnath Nandi[7], Maneesh Batrani[8]

[1, 2, 3, 4, 5, 6, 7] BHEL, Corporate R&D Division, Hyderabad-500093, India

[8]BHEL, Heavy Electrical Equipment Plant, Haridwar-249403, India

Abstract: ***Rotor constitutes one of the critical components of steam turbine. Steam turbine rotor undergoes severe thermal transients during start up and shut down. When steam turbine starts up, temperature of rotor surface gradually raises with the incoming steam temperature. During this process temperature rise of central part lags behind that of rotor surface, resulting in extreme compressive stresses developed within the rotor. Similarly the rotor surface is subjected to tensile stresses during the cooling down in the shutdown process. After a certain number of cycles, the surface or core of rotor may produce cracks due to Low Cycle Fatigue (LCF) phenomena. For a Steam turbine unit which works under very high pressure and temperature, the working environment of rotor is severe. It is required to test the rotor in rotating condition in a high temperature spin rest rig before putting into actual operation. To test the turbine rotor in a high temperature test rig, it is required to simulate the radial temperature gradient in order to achieve the required thermal stress in the rotor. No standard test protocol exists for the accelerated LCF testing for the rotor. A test protocol is developed for Accelerated Creep and Transient Thermo-mechanical testing of steam turbine rotor at high temperature taking clue from the existing ASME standard. This paper presents a specially formulated accelerated LCF test protocol, developed with inspiration from ASME standard [1] and creep parameters derived based on Larsen Miller Parameter (LMP) [2]. The test protocol arrives at temperature difference between rotor surface and bulk mean temperature in the solid rotor considering equivalent cold starts obtained from actual cold, warm and hot starts. It is considered that this simulates damage as envisaged in the life time of actual rotor but within a short/feasible time duration. Test protocol aims to arrive at a temperature difference between test rotor surface and bulk mean temperature during transient thermal loadings. On arriving at steady-state temperature the test rotor undergoes creep phenomena. The principal aim of test protocol is to arrive at a suitable test cycle consisting of transient thermal loading, steady thermal loading followed by transient cooling. This paper describes a test protocol for given number of cycles and operating temperature and equivalent cold starts such that the accelerated testing is completed within a reasonable time frame. This requirement is to address the need for validation of rotor for high temperature super critical steam turbine for higher efficiency.***

Keywords: Rotor, Steam Turbine, higher efficiency, high temperature, test protocol, creep, fatigue

1. Introduction

A special accelerated LCF test protocol is developed in line with ASME [1] and creep parameters derived based on Larsen Miller Parameter (LMP). The test protocol arrives at temperature difference between rotor surface and bulk mean temperature in the solid rotor considering equivalent cold starts obtained from actual cold, warm and hot starts. It is considered that this simulates damage as envisaged in actual rotor. Test protocol aims to arrive at a temperature difference between test rotor surface and bulk mean temperature during transient thermal loadings. On arriving at steady-state temperature the test rotor undergoes creep phenomena. Hence the principal aim of protocol is to arrive at a suitable test cycle consisting of transient thermal loading, steady thermal loading followed by transient cooling. Following sections describe the details of test protocol for given number of cycles and operating temperature and equivalent cold starts such that the accelerated testing is completed within two years.

2. Flow chart for the test protocol for Accelerated Creep and Transient Thermo-mechanical testing of steam turbine rotor at high temperature

Flow chart of the developed test protocol with all the steps is shown in the fig 1.

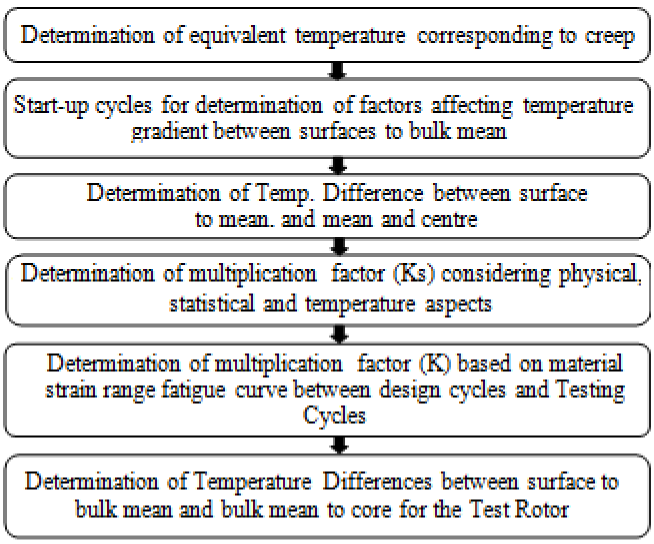

Figure 1: Flow chart of the test protocol for Accelerated Creep and Transient Thermo-mechanical testing of steam turbine rotor at high temperature

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3. A Typical Example to Demonstrate the Calculation Methodology

The test protocol as shown in the above flowchart is demonstrated for testing of a steam turbine rotor operating at 650^0C and for creep life of 100000hrs

3.1 Determination of equivalent temperature corresponding to creep

Accelerated Testing for creep damage is based on Larsen Miller Parameter (LMP) criteria considering testing of 10,000 hrs units corresponding to creep life 100,000 hrs @ 650^0C of the actual rotor.Additional time for start-up and shut down cycles for number of test cycles is extra. The following is the equation for computation of LMP

$$LMP=T(C + \log (t)) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (1)$$

Where, t is time in hours, T is temperature in degree Kelvin and C is the material constant.
Applying Eq.(1) to actual rotor, for which t=100,000 hrs,
T= 650+273=923 K and
C=20 [2] and LMP= 23075.

For the test rotor, t1 is 10,000 hours and using the LMP value computed above, Test temperature (T1) is obtained as 688.5^0C and rounded off to (T1+1% of T1) adjusting T1 as T1=696^0C. This is the temperature to be achieved at the end of each heating cycle.

3.2 Start-up cycles for determination of factors affecting temperature gradient between surface to bulk mean

For arriving at equivalent cold starts to be used for simulating damage of the actual rotor, the following design cycles are used.

- Cold start-ups: 200;
- Warm start-ups:500;
- Hot start-ups: 1300;

Total no of start-up cycles considered is 2000.

For this the equivalent cold starts are arrived as 925 cycles considering a cumulative damage of 0.8 of the actual rotor. (Where no. of equivalent cold stars = Cumulative damage * No of allowable cycles for the cold starts)

The design cold start up curve is used for transient thermal analysis to find out temperature distribution in the rotor sections. This distribution is further used to arrive at bulk mean temperature in the metal and thereafter to estimate the difference between surface and bulk mean temperatures. This difference is to be multiplied by factors to account for surface finish, size, test temperature etc., defined in the following sections to arrive at simulation temperature during heating cycle.

For accounting of different factors like surface finish, size, test temperature, no. of replicate tests and no of test cycles, a procedure is proposed in line with ASME [1]. However it is to be noted that the ASME code is strictly applicable to materials undergoing subcritical thermal loading.

3.3 Determination of Temp. Difference between surface to mean, and mean and centre

Transient thermal analysis of the Rotor given in Fig.2 based on the design cold start up cycle, is done using axis symmetric FE model of rotor segments considering actual geometry. Mean temperature for the rotor portion of 1 meter length is calculated at various axial sections. Temperature difference between surface to bulk mean and bulk mean to core for 1 meter length of rotor portion is obtained between inlet groove and first blade groove. Figure 2 describes the results of transient thermal analysis using FE.

Temperature difference between surface to bulkmean (ΔTs-m) = 80^0C
Temperature difference between bulk mean to core (ΔTm-c) = 50^0C

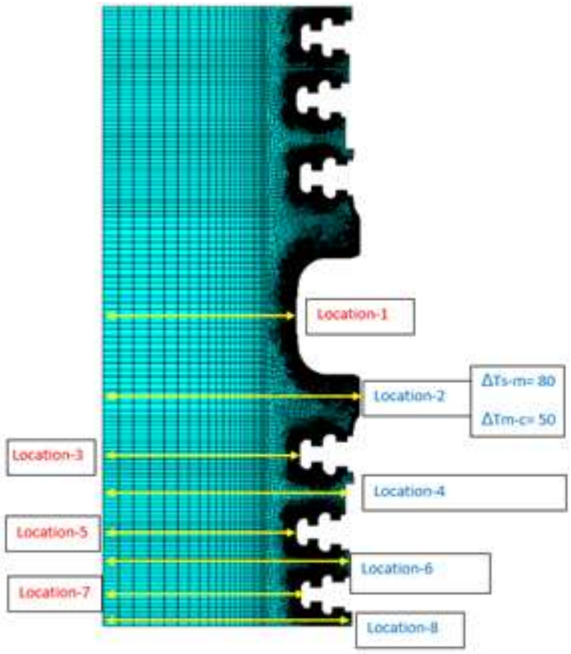

Figure 2: Temperature distribution for the test rotor segment

3.4Determination of multiplication factors considering physical, statistical and temperature aspects

The temperature difference reported in section 4, between surfaces to bulk mean (ΔTs-m) is to be multiplied by factors on temperature as per ASME code of experimental testing of full scale prototype testing. Multiplication factor is as per ASME[1] which accounts for the effect of size, cyclic rate, temperature and the number of replicate tests performed.

Multiplication factor is determined as follows

$$K_S = \text{Max}\left[(K_{SC}{*}K_{Sl}{*}K_{Sf}{*}K_{st}{*}K_{ss}),\ 1.25\right] \quad \ldots..(2)$$

Where, Ks_C = Factor for difference in design fatigue curve at various temperatures,
K_{sl} = Factor for the effect of size on fatigue life,
K_{sf} = Factor for the effect of surface finish,
K_{st}= Factor for effect of test temperature,
K_{ss}= Factor for statistical variation in test results
Where,

$$K_{sc} = \text{Max}\left[\left(\frac{\text{Sa (N,Tc)}}{\text{Sa (N,TD)}}\right) \text{x} \left(\frac{\text{Sae (TT)}}{\text{Sae (Tc)}}\right), 1.0\right] \quad \text{...........(3)}$$

Where,

S_a (N,Tc) = Stress amplitude from the applicable design fatigue curve for *N* cycles evaluated at Tc (Component temp. i.e. Operating Temperature)
S_a (N,T_D) = Stress amplitude from the applicable design fatigue curve for N cycles evaluated at T_D (design temp.)
S_{ae} (T_T) =stress amplitude from the applicable design fatigue curve at the maximum number of cycles defined on the curve evaluated at T_T (Test Temp.)
S_{ae} (Tc) = stress amplitude from the applicable design fatigue curve at the maximum number of cycles defined on the curve evaluated at Tc (component temp.)
Using Stress amplitude vs Cycles curve for 617M Super alloy of ORNL data [8], shown in fig.3 at 650^{0}C and 696^{0}C for 925 cycles and maximum number of cycles of 30000, the following are determined
S_a (N, Tc) = Sa (N, T_D) =420 MPa,
S_{ae} (T_T) = 200 MPa ,
S_{ae} (T_C) =220 MPa

Using the above, different factors cited above are determined as follows:
K_{sc}=1

$$K_{sl}=\text{max } [(1.5\text{-}0.5 \text{ x } R_{LP}), 1]\text{................ (4)}$$

Size of test rotor is same as component, K_{sl} = 1

$$K_{sf}=\text{max } [(1.175\text{-}0.17R_{SF}), 1]\text{................ (5)}$$

Surface finish of test rotor is same as component, K_{sf}=1

$$K_{st}=\text{Max}\left[\left(\frac{Sa\,(N,TT}{\text{Sa (N,TD}}\right), 1.0\right] \quad \text{................ (6)}$$

Where,
Sa (N, T_T) =stress amplitude from the applicable design fatigue curve for N cycles evaluated at test temp.(T_T)= 360.11 MPa

As per Stress amplitude vs Cycle curve at 696^{0}C

K_{st} =1

$$K_{ss}= \text{max } [(1.470\text{-}0.044N_{RT}), 1]\text{...............(7)}$$

Where, N_{RT} is the number of replicate tests =0
K_{ss}= 1.47

K_S Multiplication Factor
= Max [(K_{SC} * K_{Sl} *K_{Sf} *K_{st} *K_{ss}), 1.25]
= Max [(1*1*1*1*1.47), 1.25]
=1.47

3.5 Determination of multiplication factor based on material strain range fatigue curve between design cycles and Testing Cycles:

Multiplication factor based on material strain range fatigue curve between design cycles and Testing Cycles is computed as follows

No. of test cycles (N_T) is decided to be 200 (10% of total number of cycles i.e. 2000)

No. of design cycles (ND) = No. of equivalent cold starts= 925

Multiplication factor is calculated based on the equation

$$(K) = \frac{\Delta \varepsilon NT}{\Delta \varepsilon ND}\text{.........................(8)}$$

Where, $\Delta\varepsilon_{NT}$ = strain range corresponding to no of test cycles (N_T) = 200 as per the strain range vs Cycles curve taken from ASME B&PV Draft Code Casefor Alloy 617 [7]
$\Delta\varepsilon_{NT}$ = 0.065

Where, $\Delta\varepsilon_{ND}$= strain range corresponding to no of design cycles (ND) = 925
$\Delta\varepsilon_{ND}$= 0.0045

Hence, Multiplication factor (K)
= 0.065/0.045= 1.45

3.6 Determination of Temperature Differences between surface to bulk mean and bulk mean to core for the Test Rotor:

Temperature differences between surface to mean and mean to core for the Test Rotor is calculated by multiplying the temp. Difference as obtained for the actual rotor (as per section 3) with the above factors.

Hence, ΔTs-m for test rotor
=Ks*K*ΔTs-m
= 1.47* 1.45*80 = **170^{0}C and**

ΔTm-c for test rotor
= Ks*K*ΔTm-c
= 1.47* 1.45* 50= **106^{0}C**

- Maximum temperature difference between surfaces to bulk mean for the test rotor during the heating is calculated as **170^{0}C** and mean to core is calculated as 106^{0}C.
- Maximum temperature difference between surfaces to bulk mean during the cooling is taken as 80% of temperature gradient during heating. i.e.184*0.8= **136^{0}C.**
- Heating cycle and cooling cycle for the test rotor are evaluated based on the above temperature differences between surface to mean and mean to core.

4. Conclusions

A test protocol suitable for accelerated testing is designed and the salient factors computed. These values form the basis for designing the heating, steady state and cooling parts of overall cycle.

Following are the criteria for the selection of duration of testing during heating/Cooling (LCF criteria) and steady state (creep criteria)

- Number of test cycles : 200
- Optimum duration for rotor testing Two years
- Accelerated creep test : 10000 hrs
- Optimum steady state test cycle = 50 hrs
- Optimum Heating cycle : </= 15 hours
- Optimum cooling cycle : </= 15hours

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