Cause and Countermeasure of Crack Initiation in a 50Mw Steam Turbine Rotor

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Abstract

In this paper, the cause of crack initiation in 50Mw steam turbine rotors of “S” power station and a countermeasure for preventing it were studied. Many years ago, “S” power station restructured pipe paths to supply steam to turbines. The aim of the restructure was to save the heavy oil burning for startup. Since the restructure upto now, the turbines were started in a startup method different from standard. That is, the restructure made the current startup method different from the standard. Unfortunately, after having restructured them, cracks initiated in the rotors of the turbines in the case of starting in current startup method every certain period. The present study has been conducted to explain why the cracks initiated, and to establish a countermeasure for it.

In the cases of starting the turbines in accordance with both the current and the standard startup methods, the change of convection heat transfer coefficients (CHTCs) and transient temperature and thermal stress were analyzed. Through these analyses, the cause of crack initiation in the case of current startup method was explained. And the low-cycle fatigue life until the cracks initiated was calculated and compared with experimental data. A new reasonable startup method was suggested as a countermeasure of a crack initiation. The new method not only could prevent crack initiation in the steam turbine rotors but also gave economic benefit.

Keywords: steam turbine; thermal stress; low-cycle fatigue life; startup; rotor

1. Introduction

The exact analysis of the transient thermal stresses in steam turbines during startup and stop is one of the most important problems. The transient thermal stress in the rotor during startup and stop of a turbine is much higher than the stress during steady operation. These can exceed even the yield stress. Due to cycles of this stress during startups, fatigue cracks can initiate in the rotor of a turbine.

In Ref. [1] the transient temperature and stress fields in a 600Mw steam turbine during startup and stop were analyzed using the continuum damage mechanics model. The analysis results indicated that the transient thermal stress of a steam turbine rotor was nearly 6 times higher than the value during steady operation. It is because the change of temperature of a steam turbine rotor during startup and stop is rapid. CHTCs influence deeply the change of the temperature in a turbine rotor. It is vital to exactly calculate CHTCs of rotor surfaces for correct temperature analysis. In Ref. [4, 5] an experimental formula to calculate CHTCs of rotor surfaces in a steam turbine was suggested. Employing the formula, the changes of CHTC with time in some places of a turbine rotor during startup were gotten.

In Ref. [3] fatigue-creep interaction of 30Cr1Mo1V steel was studied applying equivalent strain method. The fatigue-creep interaction of 30Cr1Mo1V steel was experimented at temperatures of 540℃ and 565. From the experiment data, a cyclic stress-strain relation was derived in the case that the cyclic strain range remained unchanged.

In Ref. [4] temperature and thermal stress fields of a 100Mw steam turbine rotor in three different startup methods were analyzed by means of the finite element analysis program ANSYS. From the analysis results, a startup method in which the stress level is the lowest was chosen to apply to actual startup. In this case, the low cycle fatigue life in dangerous places of the turbine rotor was calculated.

In Ref. [5] transient temperature and thermal stress fields of high and medium pressure steam turbine rotors during startup and stop of a 300MW steam turbine were calculated. The lives of the turbine rotors were estimated using three different low cycle fatigue life estimation methods. The theoretical study on a reasonable operation method in which the startup time of the turbines shortened and the safety of the operation was ensured was carried out.

In Ref. [6] calculation methods of temperature and thermal stress fields and life loss in dangerous places of steam turbine rotors were described in detail, and a reasonable startup method of turbines was studied. In Ref. [7] analysis of temperature and thermal stress fields of a 200 Mw steam turbine rotor was performed. The investigation on the geometry and operation condition of the turbine resulted in finding the position of maximum thermal stress. Variations of temperature and thermal stress of inside and outside of the rotor during different startups and sudden stop were simulated.

In Ref. [8] a formula and an algorithm for calculating CHTCs at several places of a steam turbine rotor were proposed. Analysis of temperature and stress fields of the turbine rotor during startup and stop was performed using finite element analysis program ANSYS. In Ref. [9] nonlinear behavior of temperature of a steam turbine during turbine startup using piecewise linear superposition method was modeled. And a reasonable startup method to reduce the life loss as much as possible was investigated.

In Ref. [10] a mathematical model for life loss of steam turbine rotors was introduced. Behavior of temperature and thermal stress fields of inside and outside of turbine rotors with time during startup of a 300MW steam turbine rotor was analyzed. In Ref. [11] analysis of influence on brittle crack initiation of rotors given by startup and overspeed experiment of steam turbines was carried out.

In Ref. [12] using data for startup, stop and load-varying experiments of 125MW steam turbine rotors, life loss values in dangerous places of rotors were calculated. A study to reduce the life loss remarkably was performed, too. In Ref. [13] under several startup conditions, distribution of temperature and thermal stress of 300MW steam turbine rotors was analyzed by means of the finite element method. From these results, factors influencing the thermal stresses and lives of rotors were studied in detail.

In this paper, above all, it is described why to restructure the steam supplying pipes system in "S" power station. The changes of convection heat transfer coefficients (CHTC) of the turbine rotor surface are analyzed. Considering the change of CHTC, the behavior of transient thermal stresses during current startup of the 50Mw steam turbine in "S" power station is analyzed. From these analyses, the cause of crack initiation in the turbine rotor is explained. Finally, a reasonable startup way which is appropriate to the situation of "S" power station is suggested.

2. The behavior of CHTCs during startup of the steam turbine

It is very important to exactly determine CHTC of turbine rotor surfaces for exact assessment of thermal stresses during startup of steam turbine rotors. CHTC of turbine rotors depends on temperature, pressure, rotational frequency, and geometry. Many investigations to determine CHTC have been performed over years. CHTC can be calculated as expression (1) ~ (3) bellow [4, 5]. The behavior of CHTCs of the turbine rotor in "S" power station is analyzed by the expressions.

2.1 CHTC on both sides of turbine rotor blades

CHTC on both sides of rotor blades, α is as follows

$$\alpha = \frac{Nu \cdot \lambda}{R_b} \quad (\mathrm{W/m^2\ ^\circ C}) \tag{1}$$

In Eq. (1) Nusselt number Nu can be calculated as followings according to Reynolds number:

$$\mathrm{Re} < 2.4 \times 10^5: \quad Nu = 0.675\mathrm{Re}^{0.5}$$

$$\mathrm{Re} > 2.4 \times 10^5: \quad Nu = 0.0217\mathrm{Re}^{0.8}$$

$$\mathrm{Re} = \frac{uR_b}{\nu}$$

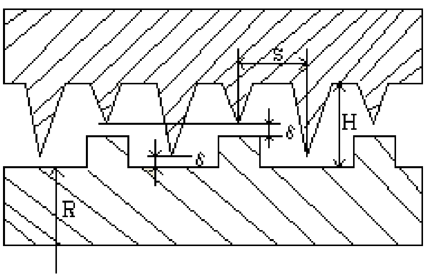

Figure 1. The geometry of labyrinth[4, 5]

where ν is kinematic viscosity(m^2/s) of steam, u is circumferential speed of the outer end point of blades, R_b is radius of the outer circle of blades, Nu is Nusselt number, Re is Reynolds number, λ_0 is coefficient of thermal conductivity of steam(W/m ℃).

2.2 CHTC on labyrinth of a turbine

The geometry of the labyrinth of a turbine rotor is shown in Figure 1.

CHTC in the labyrinth is calculated by following formula

$$\alpha = \frac{Nu\lambda_0}{2\delta} \ (\mathrm{W/m^2\ ℃}) \tag{2}$$

$$Nu = 0.043\left(\frac{\delta}{H}\right)^{0.3}\left(\frac{\delta}{S}\right)^{0.2}\mathrm{Re}^{0.8}$$

$$\mathrm{Re} = \frac{V \times 2\delta}{\nu}$$

where V is the average velocity of steam which flows through the interspace of the teeth of the labyrinth, S is the distance between the teeth of the labyrinth, δ is the distance between the surface of rotor and the teeth, and H is the distance between the surface of rotor and the bottom of the teeth.

2.3 CHTC in the smooth place of a turbine rotor

It is presented as follows

$$\alpha = Nu\frac{\lambda}{r} \ (\mathrm{W/m^2\ ℃}) \tag{3}$$

$$Nu = 0.1\mathrm{Re}^{0.68}, \quad \mathrm{Re} = \frac{ur}{\nu}$$

Where u and r are the circumferential speed and the radius at the noticed position on the surface of the rotor, respectively.

2.4 The behavior of CHTCs during the standard low temperature startup of the turbine in "S" power station

There are two startup methods to start turbines in "S" power station: low temperature startup and high temperature startup. In low temperature startup, temperature of whole rotor during startup before supplying operation steam to turbine is below 300℃. And in the high temperature startup, the temperature is higher than 300℃.

Before supplying operation steam first to the turbine, the temperature and pressure of the steam in "S" power station are 250℃ and 1.0MPa, respectively. After supplying the operation steam, the rotational frequency of the turbine rotor increases upto 3 600rpm through two steps of keeping 500rpm and 1 000rpm. It takes 4h 30min from the time of having supplied the operation steam until the power generator operates at full capacity.

Considering that $R_b = 0.377m$, $u = 2\pi R_b \cdot \frac{f}{60}$ (m/s)(Here f is the rotational frequency (rpm)) on the left side of the control stage blade(See Fig. 1) and $r = 0.282m$ at the flange of the labyrinth on the left of the control stage, the CHTCs with variation of time in these two places can be calculated according to Eqs. (1) ~ (3).

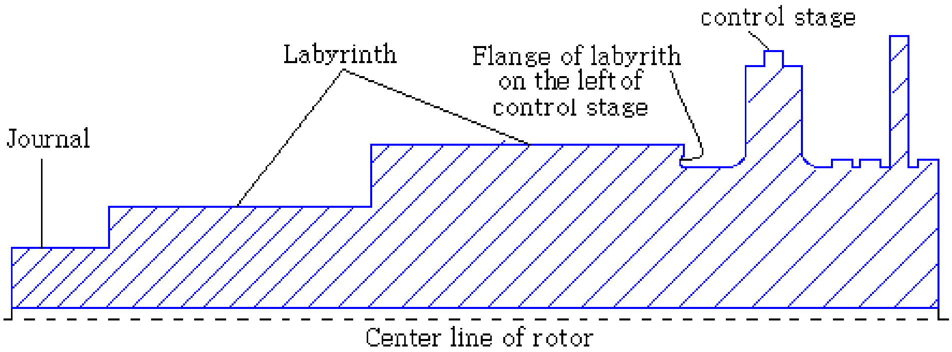

Figure 2. The geometry of a 50Mw turbine rotor in "S" power station

In labyrinth, considering that $R = 0.317m$, $\delta = 0.5 \times 10^{-3} m$, $S = 0.005m$, $H = 0.006m$ and $V = \frac{M}{\rho A}$ (m/s), the CHTC with time can be calculated. Here, ρ is density of the steam (kg/m³), $A = \pi((R+\delta)^2 - R^2)$ is the area of the section (m²) to which the steam flows through the labyrinth and M is the mass flow of the steam (kg/s) which flows through the labyrinth per second. Usually, steam flow rate through the labyrinth is $\frac{1}{200}$ of steam amount which comes to the turbine cylinder through nozzle. The flow rate of the steam which comes to the turbine cylinder through nozzle is 5t/h when the power generator is not connected, 40t/h when it works at the load of 10Mw, and 200t/h when it works at the load of 50Mw, respectively. Therefore the steam flow rate going through labyrinth is 25 kg/h, when the power generator is not

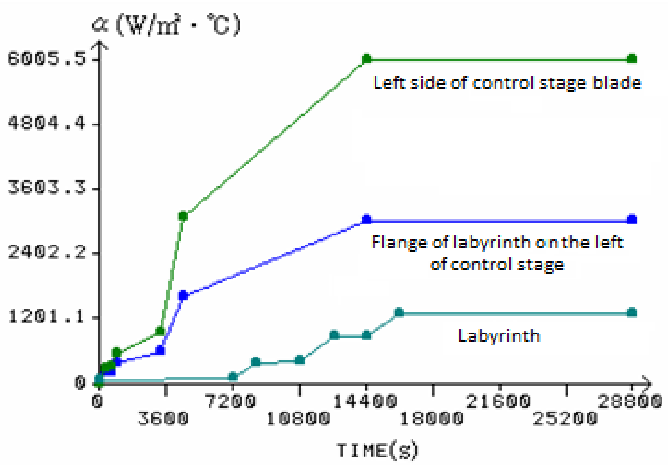

Figure 3. The behavior of CHTCs during the standard startup

connected, 200 ㎏/h, when it works at the load of 10Mw, and 1t/h, when it works at the load of 50Mw, respectively.

The behavior of CHTCs with time during the standard startup is shown in Figure 3.

As it can be seen, the CHTC on the left side of the control stage blade is compara tively high and that on the labyrinth is low. The rotational frequency of the rotor gives the biggest influence on the change of CHTC among the rotational frequency of rotor and the temperature and pressure of steam. Especially, in the period during which the rotational frequency increases up from 1 000rpm to 3 000rpm, the CHTC s on the left side of control stage blade and at the flange of labyrinth on the left of the control stage changes rapidly.

3. Calculation of low-cycle fatigue life of steam turbine rotors

3.1 The state of crack initiation in steam turbine rotors in "S" power station

The temperature and pressure of steam to supply turbines of "S" power station are 535℃ and 9.0MPa, respectively. Cracks have found several times in these turbine rotors. During the over hauling for the turbine in July 2008, circumferential cracks were found at the flange of the labyrinth on the left of the control stage of "A" turbine rotor (Here, "A" is a ID of Rotor), and after that, cracks were found in other three turbine rotors, too.(Table 1)

Table 1. State of crack initiation in steam turbine rotors of "S" thermal power station

Rotor ID	Date of crack finding	Operation time(h)	Number of startup-stop	Depth of crack (㎜)
A	July 2008	76 100	753	11.5
B	March 2009	73 530	938	4.0
C	August 2011	62 050	663	8.0
D	June 2012	68 700	1 016	14.5

For the first time, it was inferred that the cracks were initiated due to the material of turbine rotors and the many startup-stop cycles. The repair was accomplished by cutting off the crack of the rotor with certain radius.

However, during over hauling in August 2012, the crack initiated again at the same place of "A" turbine rotor. Since the previous over hauling, the operation time and the number of startup-stop were 21 700h and 134, respectively.

3.2 Analysis of startup condition of turbine in "S" power station

Analysis of the current low temperature startup

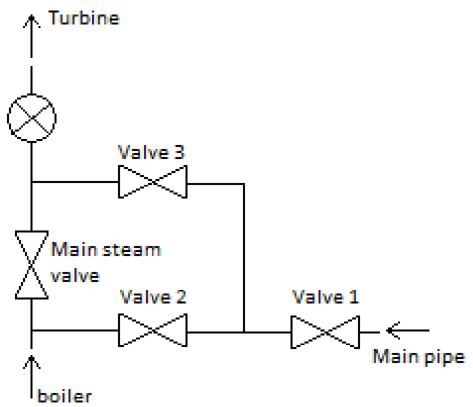

Figure 4. The reconstructed structure

The turbines of “S” power station were designed to be connected directly with only a boiler. According to the design the steam is supplied to the turbine only through the main steam valve. (See Fig. 4) However, “S” power station reconstructed the structure as shown in Figure 4, by linking the turbine with the main pipe through the valve 3 and 1, adding the valve 2 between the boiler and the main pipe. It was for saving heavy oil wasted in the boiler while pre-heating the turbine. It takes 4h to pre-heat the turbine, and 1h to pre-heat the boiler. Therefore, there must be waste of heavy oil burnt in the boiler for 3h. This is the reason why the restructure of pipes system was carried out. As the result of above restructure, steam is supplied from the main pipe to the turbine through valve 1 and 3 for 3h before the boiler is pre-heated. Then, opening the valve 2, the turbine is pre-heated by receiving steam from both the boiler and the main pipe for 1h. If the turbine operates at the normal state, closing the valve 3 and 1, the steam goes only from the boiler to the turbine.

The procedure by which the turbine in “S” power station has started up yet is as follows.

-Before supplying steam to the turbine, the temperature and pressure of the steam are kept 300℃ and 1.0MPa, respectively.

-After supplying steam to turbine, the rotational frequency of turbine is grown up to 500rpm by increasing 100rpm every minute. When reaching 500rpm, the rotational frequency remains for 5 minutes while inspecting the state of turbine such as noise and so on.

- If everything is OK, the rotational frequency continues to be grown up to 1 000rpm by increasing 100rpm every minute. The turbine is heated for 20~30 minutes while checking the state of turbine running.

- If everything is good, the rotational frequency is grown up to 3 000rpm by increasing 100rpm every minute. During this period, the rotational frequency should be rapidly grown up in the vicinity of the critical rotational frequency. The critical frequencies of power generator and turbine are 1 610rpm and 2 368rpm, respectively.

- After the rotational frequency reaches its rating, the power generator is connected to turbine and the electric load is slowly grown up to 10~15Mw. After remaining this state for 20 minutes, the electric load is slowly grown up to full capacity.

As it can be seen, since supplying steam to turbine, it takes 1h until the rotational frequency of turbine reaches its rating, and 2h until the power generator runs at full capacity, respectively. The startup period is too short. The change of CHTC with

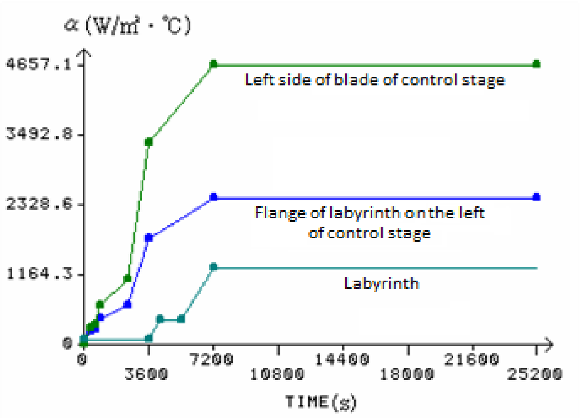

Figure 5. Variation curve of CHTC during low temperature startup

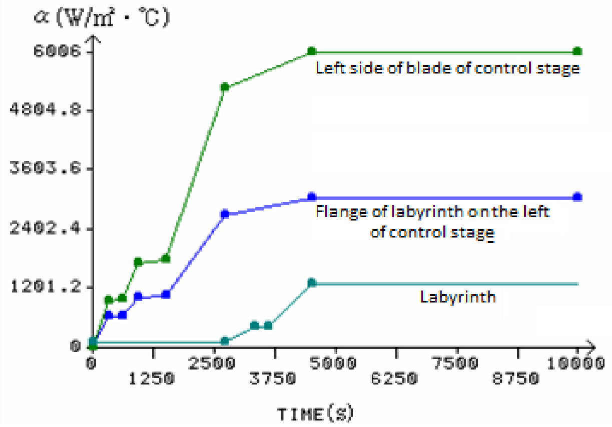

Figure 6. Variation curve of CHTC during high temperature startup

variation of time in some places of rotor is shown in Figure 5.

The CHTC during the current low temperature startup changes more rapidly compared to that during the standard startup. Such rapid change of CHTC yields large difference between temperatures of the outside and inside of rotor during turbine startup.

Analysis of high temperature startup condition

After pre-heating, the temperature of rotor is higher than 300℃. Before supplying the steam to the turbine, the temperature and pressure of the turbine are 460℃ and 6.0MPa, respectively. Since steam has been supplied first, it takes 45min to increase the rotational frequency upto the rating value, and 1h 15min for the power generator to run at full capacity, respectively.

The variation curve of CHTC in some places of rotor during this period is as shown in Figure 6.

3.3 Temperature and stress fields during the turbine startup

Material characteristics of the rotor and the geometry for analysis

The turbine rotor in “S” power station is made of 30Cr2MoV steel, of which chemical composition is given in Tables 2.

Table 2. Chemical composition of 30Cr2MoV steel

Element	C	Mn	Si	P	S	Cr	Mo	Ni	V
Content (%)	0.137	0.66	0.36	0.016	1.64	0.66	0.66	0.11	0.23

Mechanical and thermal properties of material with above chemical composition are employed in analysis.

The rotor geometry from journal to the stage on the right of the control stage is shown in Fig 2. The geometry includes the flange of labyrinth at which crack initiated.

Behavior of temperature and stress during the current low temperature startup

The crack initiated at the flange of the labyrinth on the left of the control stage.

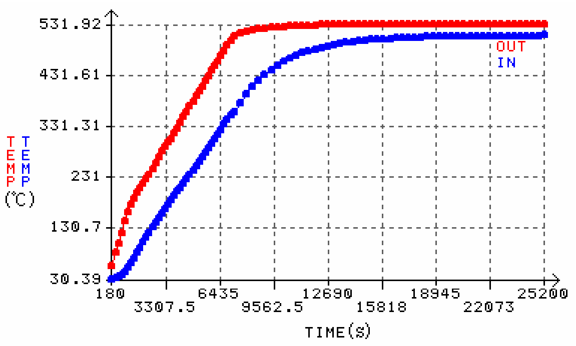

Figure 7. Temperature behavior in the inside and outside of rotor during the low temperature startup

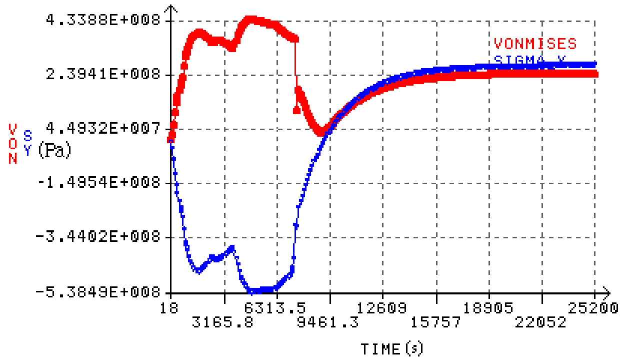

Figure 8. Equivalent stress and axial stress at the flang of the labyrinth during the low temperature startup

Before seeing the behaviors of stresses at the flange, the temperature behaviors in the rotor are shown in Fig 7. As it can be seen, the temperature difference between inside and outside of the rotor goes up beyond 120℃ 60~120 min after getting started the turbine. Especially, the maximum of the difference reaches to even 150℃ 120 min after startup of turbine. Due to these temperature differences, it is inevitable to raise stresses.

The behaviors of equivalent and axial stresses with time at the flange of the labyrinth are shown in Fig 8. As it can be seen in Fig 8, the maximum of equivalent stress at the flange of the labyrinth on the left of the control stage reaches 433MPa, which exceeds the yield stress. Thus, plastic strain must occur here. The axial stress at the flange of the labyrinth is compressive in the earlies after getting started the turbine. It is because temperature increase is rapid in the outside of the rotor and slow in the inside of the rotor in the earlies, so the temperature difference between inside and outside is big.

Behavior of temperature and stress during high temperature during startup

The temperature behavior along with time at the flange of the labyrinth on the left of the control stage and in inside of the rotor are shown in Fig 9. The temperature difference between inside and outside of rotor goes up beyond 100℃ 14~36 minutes

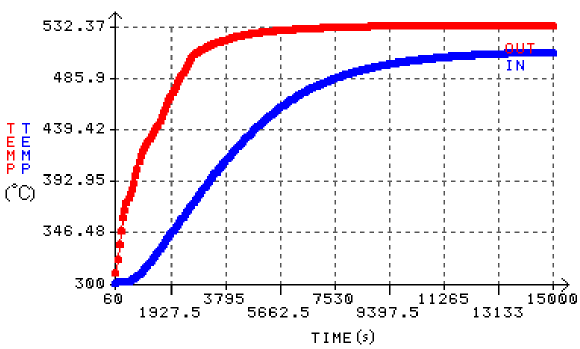

Figure 9. Temperature variation curve at the flange and in the inside of rotor during high temperature startup

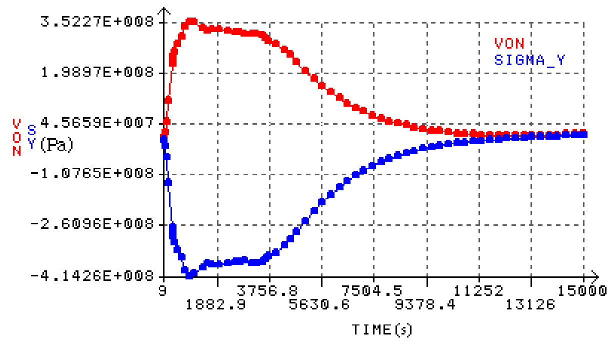

Figure 10. Equivalent stress and axial stress at the flange of labyrinth during high temperature startup

after getting started the turbine. Especially, the maximum temperature difference reaches 132℃ 23 min after getting started the turbine.

Meanwhile, the behavior of the equivalent and axial stresses at the flange of the labyrinth is as shown in Fig 9. As it can be seen, the maximum of the equivalent stress at the flange of the labyrinth is 353MPa 15 min after getting started the turbine, that is, when the rotational frequency is in the vicinity of 1 000rpm. This stress is not high enough to raise plastic strain in the rotor during the turbine startup.

Through both the analyses for the low and high temperature startups, followings can be known.

In a word, the rapid change of CHTC on the surface of the rotor in the current low temperature startup method is due to short turbine startup time, that is, due to short heating time at high temperature. Consequently, the temperature difference between inside and outside of the rotor is so big and the thermal stress goes beyond yield stress. During cycles of startup and stop, the rotor is subjected repetitively to the cyclic thermal stress. The crack was initiated at the flange of labyrinth where the thermal stress is at maximum. The low-cycle fatigue by the thermal stress cycles accounts for the crack initiation.

As for high temperature startup, the startup time is short, too. But the temperature of the whole rotor before getting started the turbine is higher than 300℃. Thus, the temperature difference in high temperature startup between inside and outside of the rotor is not as big as in low temperature startup. The thermal stress level in high temperature startup is lower than in low temperature startup. That is lower than the yield stress. So, the effect of low-cycle fatigue on crack initiation is bigger in the low temperature startup than in the high temperature startup.

3.4 Calculation of equivalent strain range

Equivalent strain range utilized for calculation of low-cycle fatigue life is equal to the sum of elastic and plastic strain ranges. [3]

$$\Delta\varepsilon = \Delta\varepsilon_e + \Delta\varepsilon_p \,. \tag{4}$$

If stress-strain relation attained from uniaxial tensile test is used for calculating the plastic strain range in Eq. (4), it results in underestimation of real damage of material. Thus, it is preferable to employ stress-strain relation under the cyclic loading of 30CrMoV steel. It is as follows:

$$\frac{\Delta\sigma}{2} = K\left(\frac{\Delta\varepsilon_p}{2}\right)^n \tag{5}$$

where parameter $\Delta\sigma$ is the cyclic stress range and $\Delta\varepsilon_p$ is the plastic strain range. K and n are the cyclic strength coefficient(MPa) and the cyclic strain hardening coefficient, respectively, which are material constants related to temperature.

Considering above expression (4) and (5), the equivalent strain range is calculated as follows:

$$\Delta\varepsilon = \Delta\varepsilon_e + \Delta\varepsilon_p = \frac{\Delta\sigma}{E} + 2\left(\frac{\Delta\sigma}{2K}\right)^{\frac{1}{n}} \tag{6}$$

Employing the formula (6), the equivalent strain range under the condition of “S” power station was calculated.

E=179GPa, K=592.7MPa and n=0.064 8 for 30CrMoV steel at 500℃. Considering that the maximum and minimum axial stresses of the rotor during low temperature startup are $\sigma_{y,\max} = 325MPa$ and $\sigma_{y,\min} = -558MPa$ respectively, the axial stress range is $\Delta\sigma = \sigma_{y,\max} - \sigma_{y,\min} = 883MPa$. From $\sigma_{y,\max} = 10MPa$ and $\sigma_{y,\min} = -414MPa$ during high temperature startup, $\Delta\sigma = \sigma_{y,\max} - \sigma_{y,\min} = 424MPa$. Assigning above values into the formula (6), the equivalent strain range is as followings:

Low temperature startup: $\Delta\varepsilon = 0.0262$

High temperature startup: $\Delta\varepsilon = 0.00237$

3.5 The low-cycle fatigue life estimation of the rotor

There exist many fatigue life estimation formulae, but there is no universal formula applicable to all cases. To estimate the life of turbine rotors of “S” power station, Coffin-Manson formula and its advanced formula are employed.

Coffin-Manson formula to estimate the low-cycle fatigue life is as follows [1]:

$$\frac{\Delta\varepsilon}{2} = \frac{\Delta\varepsilon_e}{2} + \frac{\Delta\varepsilon_p}{2} = \frac{\sigma_f'}{E}(2N)^b + \varepsilon_f'(2N)^c \tag{7}$$

where $\Delta\varepsilon$ is the equivalent strain range, $\Delta\varepsilon_e$ is the elastic strain range, E is the elastic modulus, and N is the number of cycles to fatigue fracture. σ_f', b are the fatigue strength coefficient and the fatigue strength exponent, respectively. ε_f', c are the fatigue ductility coefficient and the fatigue ductility exponent, respectively.

Assigning the equivalent strain range of low and high temperature startups and material properties at 510℃ into Eq. (7), the low-cycle fatigue life of the rotor in “S” power station is calculated. The results are followings:

Low temperature startup: N=664 cycles

High temperature startup: N=5.25×10^{6} cycles

Manson and Coffin, based on fatigue life formula (7), suggested the following advanced formula, which considers the influence of creep-fatigue interaction on life at high temperature. [2]

$$\Delta\varepsilon = AN^{a}v^{b} + BN^{c}v^{d} \tag{8}$$

According to the experiment data at 500℃ attained by a research institute, the coefficients of Eq. (8) are as follows [5]:

$$A=0.0052, B=1.18, a=-0.088, b=0.0104, c=-0.73, d=0.088$$

The turbine A of "S" power station operated for 21 700 hours from July 2008 to August 2012. During this period, there were 134 startups, which consist of 50 low temperature startups and 84 high temperature startups. During these startups, startup number v per minute is calculated as follows:

Low temperature startup: $v=1/(21700/50\times60)=3.84\times10^{-5}$/min

High temperature startup: $v=1/(21700/84\times60)=6.45\times10^{-5}$/min

Assigning the equivalent strain range and the startup cycles into Eq. (8), the low-cycle fatigue lives N are calculated as follows:

Low temperature startup: N=65 cycles

High temperature startup: N=18 480 cycles

According to the experiment data at 538℃ attained by another research institute, the coefficients of Eq. (8) are as follows [5]:

$$A=0.0094, B=0.885, a=-0.092, b=0.033, c=-0.759, d=0.034$$

Using these coefficients, the low-cycle fatigue lives are as follows:

Low temperature startup: N=84 cycles

High temperature startup: N=144 447 cycles

The results show that the second fatigue lives(65 and 18 480 cycles) are comparatively appropriate to real lives in "S" power station.

4. A reasonable startup method of the steam turbine as countermeasure of crack initiation

4.1 Selection of a reasonable startup method by quality engineering method

The aim of the study on a reasonable startup method of the steam turbine is: [9]

(1) to improve the economic effectiveness by decreasing the turbine startup time

(2) to decrease the life loss of rotor and ensure the safe operation of turbine.

Formerly, the rotational frequency increased upto 3 600rpm through the two steps of keeping 500rpm and 1 000rpm. To low down the change velocity of the CHTC, the step of rotational frequency keeping 2 800rpm adds after the step of keeping 1 000rpm.

The following calculation experiment plans are made to select a new reasonable startup method. The orthogonal experiment with 3 factors A, B, C of 3 levels is conducted. The factors and levels for the calculation experiment are as follows:

A: The time of keeping 500rpm, A_1=10min, A_2=20min, A_3=30min

B: The time of keeping 1 000rpm, B_1=15min, B_2=30min, B_3=40min

C: The time of keeping 2 800rpm, C_1=5min, C_2=10min, C_3=20min

The range of axial stress(MPa), y_1 and the startup time(min), y_2 are set up as experiment values in calculation experiment.

The orthogonal table $L_9(3^4)$ with 3 levels is utilized because there are 3 factors of 3 levels. Factors A, B, C are arranged in column 1, 3, and 4, respectively. The following table 3 with experiment data is attained by calculation experiment. (The void column 2 is made formally to organize the orthogonal table.)

Table 3. Experiment data

No. of column / No. of Experiment	A	E	B	C	Experiment value		SN ratio		
	1	2	3	4	y_1	y_2	η_1	η_2	η_3
1	1	1	1	1	513.9	190	-51.76	-53.10	-53.89
2	1	2	2	2	438.2	210	-50.72	-52.65	-53.69
3	1	3	3	3	392.0	230	-50.14	-52.62	-53.84
4	2	1	2	3	393.2	230	-50.16	-52.63	-53.85
5	2	2	3	1	410.2	225	-50.39	-52.68	-53.84
6	2	3	1	2	447.3	205	-50.83	-52.65	-53.64
7	3	1	3	2	377.8	240	-50.00	-52.71	-54.00
8	3	2	1	3	396.0	225	-50.16	-52.55	-53.74
9	3	3	2	1	397.4	225	-50.18	-52.56	-53.75

The experiment values in Table 3 were gained, by calculating the CHTCs between the rotor and the steam during startup and then analyzing the temperature field and the thermal stresses.

In this calculation experiment, the smaller axial stress range and startup time are, the better. Thus, characteristics parameter η is

$$\eta = -10\lg\left(\frac{1}{2}\sum_{i=1}^{2} y_i^2\right) \tag{9}$$

By the way, all the axial stress ranges attained from the experiment are comparably small. Therefore, it is focused to shorten startup time as much as possible. The weight coefficients are introduced to improve the economic effectiveness by shortening startup time. The bigger weight coefficient for y_2 than y_1 is set while calculating η. Then η expression can be modified as follows:

$$\eta = -10\lg\left(\frac{1}{2}\sum_{i=1}^{2}(w_i y_i)^2\right) \tag{10}$$

where $w_i\,(i=1,2)$ is weight coefficient for $y_i\,(i=1,2)$.

η_1, η_2, η_3 in Table 3 are the values of η obtained in the case that the ratio of weight coefficients $\frac{w_2}{w_1}$ is set to 1, 2, and 2.5 respectively. In these three cases, combinations from one of levels of every factor are made and optimums for the ratios of weight coefficients are obtained as follows: $A_3B_3C_3$ for $\frac{w_2}{w_1}=1$, $A_3B_2C_2$ for $\frac{w_2}{w_1}=2$, $A_2B_1C_2$ for $\frac{w_2}{w_1}=2.5$.

Table 4 shows the experimental values for selected above experimental plans. The values in brackets in column 3 of the table are the coefficients of safety of the maximum equivalent stress to the yield stress.

Table 4. Low-cycle fatigue lives corresponding to selected 3 experiment plans

Experiment plan	Startup hour (min)	$\sigma_{e,\max}$ (MPa)	$\Delta\sigma_y$ (MPa)	$\Delta\varepsilon$ (1×10^{-3})	N
$A_3B_3C_3$	250	293.0(1.60)	363.5	2.03	42 817
$A_3B_2C_2$	230	319.7(1.44)	391.3	2.19	27 050
$A_2B_1C_2$	205	362.4(1.19)	447.3	2.50	13 300

As it can be seen, the experiment plan $A_2B_1C_2$ is selected as a new startup method in the aim of shortening startup time in "S" power station. In this case, startup time is the shortest among above values as 205 min.

4.2 The new startup method

- The temperature and pressure of steam before supplying to turbine are 250℃ and 1.0MPa, respectively.
- After having supplied steam to the turbine, the rotational frequency of the turbine is grown up to 500rpm by increasing 100rpm every minute. When reaching

500rpm, the rotational frequency keeps for 20 min while inspecting the state of turbine.

- Then, the rotational frequency is grown up to 1 000rpm by increasing 100rpm every minute. At 1 000rpm, the rotational frequency remains for 15 min while inspecting the state of turbine.

- The rotational frequency is grown up to 2 800rpm by increasing 100rpm every minute. At 2 800rpm, rotational frequency remains for 10 minutes.

- The rotational frequency is again grown up to its rating, 3 600rpm by increasing 100rpm every minute.

- After the rotational frequency reaches its rating, the power generator is connected with the turbine and the electric load is grown up to 10~15Mw slowly. After having reached 10~15Mw, the electric load remains for 50 minutes.

-The electric Load is grown up to 20~25Mw slowly and then remains for 20 minutes. After that, the electric load is grown up to the rating, 50Mw for 30 minutes.

5. Conclusion

The cause of the crack initiated in 50 Mw steam turbine rotors in “S” power station was studied. The low-cyclic fatigue lives were estimated. A reasonable turbine startup method was suggested.

Firstly, the main cause of crack initiation of steam turbine rotors in “S” power station is the short startup time at high temperature.

As shown from the analysis results, the CHTC at the flange of labyrinth on the left of the control stage varies rapidly during the startup. The equivalent stress is 433MPa at maximum, which is higher than the yield stress. There must be plastic strain, thus, the low-cycle fatigue occurs.

Secondly, advanced formula (8) is more correct than Coffin-Manson formula (7) in estimating the low-cycle fatigue life of turbine rotor.

In fact, turbine A of “S” power station operated from July 2008 to August 2012. During this period there were 134 startups which consist of 50 low temperature startups and 84 high temperature startups. Number of cycles to fracture, 65 obtained from Eq. (8) is more appropriate to the real life in “S” power station than 664 obtained from Eq. (7). It is because the standard operation temperature and pressure of steam turbines of “S” power station are 535℃ and 9.0MPa, respectively, but in fact, the current values of “S” power station are lower than them.

Lastly, a new reasonable startup method which can ensure the safe of the operation of the rotor and give economic benefit is suggested using quality engineering method.

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