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ASPECTS OF REMNANT LIFE ASSESSMENT IN OLD STEAM TURBINES

Juan J. Laguarda Rodrígez[1], José Carbia Carril[2], Ángel Rodríguez Fernández[3], Rafael Rodríguez Valero[4]

ABSTRACT

In order to get a high operation availability and to stablish a maintenance and spares management strategy capable of develop the life extension of steam turbines, is essential suitable assessment of aging damage and remnant life for most relevant parts.

According with machine integity inspection results, life consumption and estimated remnant life results, future maintenance plans, equipment and even big components renewal, will be carried out in basis of efficience assessments data to get optimal safety, availability, reliability and efficiency operating conditions.

Therefore, in order to get these objectives, is essential to know the main aging deterioration mechanisms and realize how are affecting to different turbine components.

Only a few of all failure mechanisms that can happen in steam turbines affect directly to aging deterioration of components, and are produced by the effect of high temperatures kept for long time periods, as well as the sudden variations of these temperatures.

Keywords: life extensión, creep, termal fatigue, consumed life, remnant life.

INTRODUCTION

Most of steam turbines installed on electric power plants last years, were designed and manufactured to operate during 20 or 25 years, that is equal to 160.000 or 200.000 operating hours.

[1]Doctorando, Universidad de La Coruña, (jlaguarda@wanadoo.es). [2]Profesor Titular, Universidad de La Coruña, (carbia@udc.es). [3]Profesor Titular, Universidad de La Coruña, (arf@udc.es). [4]Profesor Titular, Universidad de La Coruña, (valero@udc.es).

But because of costs involving new units erection relate to repairing and refusbishment, most of plant owners prefer to continue existing machines operation. Generally, this life extension offers the chance of carrying out operational and enviromental changes and considering benefits of efficiency and reliabylity improvement using advantages offered by turbine new design components adoption.

All these considerations becomes manufacturers and owners to research and develop assessment technology and plant renewal, that contribute adequately to turbine extension life when applied.

Assessment consumed life and maintenance and renewal plans using these technology, are consolidated either as a basis to establish extensión life programs and as economically reliable maintenance methods.

Although plants were operated under good conditions and high efficiency without critical problems, long term operation periods can produce latent damage on turbine inner components. These damage can suddenly become and cause important mechanical problems.

Meanwhile are ligth and affecting only simple components, problems can be easily solved by means working maintenance habitual procedures. But if problems are importants or involves critical components, it can produce long time forced outage, causing important and unexpected economical waste.

OBJECTIVE

Damage produced by mechanisms of deterioration affecting turbine integrity are directly related with time, and are produced after long year turbine operation.

These mechanism produce a progressive damage on materials that turbine is manufactured with, reducing its mechanical properties. It also produce deviation and distortion on inner turbine parts, becoming to efficiency deterioration.

There are two of these mechanisms that demonstrates to be more significant to assess the life consumption of affected component, not only for its relation with time factor, but for the responsability of turbine components that they affect, such as rotors and casings, that are large and expensive and may produce disastrous damage.

The essential objective of this article is to show, according with damages produced by these two mechanisms, the aspects involving when assessing and analyze turbine remnant life in order to get life extension and recovery or improvement of efficiency.

METHODOLOGY

The methodology used has been based on experience obtained during several turbine overhauls in a power plant, wuth participation of own power plant staff, turbines manufacturer and technologists specialized on inspections and non destructive tests, using the top technology in inspection equipment.

Scopes on this kind of overhaul are often standardized for each type of turbine, although sometimes other machine requirement are considered. Scopes to be realized during overhaul period are also standardized, and for these type of failure mechanism are often the following:

— Complete inspection by means magnetic particles of rotors, specially in section change areas and sharpen radius.
— Ultrasonic testing in rotor center bore.
— Ultrasonic testing in rotor surface critical areas.
— Metallurgical inspection in critical areas on rotors, rows and casings.
— Magnetic particle inspection in steam chests.
— Magnetic particle inspection in steam inlet pipe sleeve and stationary row blades.

Obtained data from these inspections are compared with those obtained in previous inspections, thus it is possible to assess the evolution of existing or previously repaired defects.

Manufacturer experience and machine knowledge made neccesary its overhaul mediation in order to deal with technical improvement sheets and detected defects assessment, but it is also important the mediation of inspection specialized people, that can discover defects that are not easily detected due to its features.

TURBINE LIFE MANAGEMENT

Turbine life management consists of a continuous assessment of plant condition, monitoring operation conditions and improvement of operation and maintenance procedures. Also, a plan based on assessment of component condition and the need of restoration or replacement of critical components is required.

The process is concentrating on components whose failures have greater importance on safety, availability or maintenance

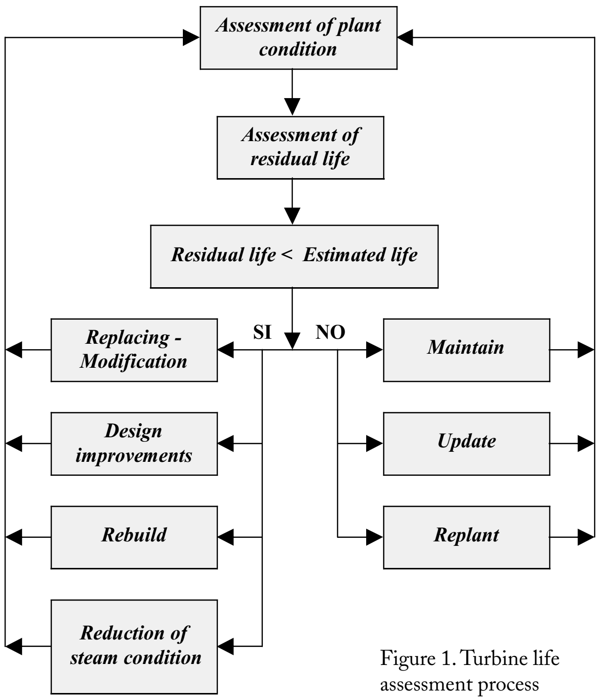

Figure 1. Turbine life assessment process

costs and on mechanisms affecting those components. The process is showed in schematic way on figure 1.

Creep

The phenomenon of creep consists of dislocations on grain boundaries of steel caused by high temperature working under the action of high centrifugal forces, so that a permanent deformation is produced on the afected component, as is showed on figure 2.

It can be deduced that more sensitive turbine parts to suffer from creep are those involved in rotatory movement, as rotors, pressure first HP and IP blade rows, center bore, serrations and blade roots, transition ratio between different rotor stages, balance holes, etc...

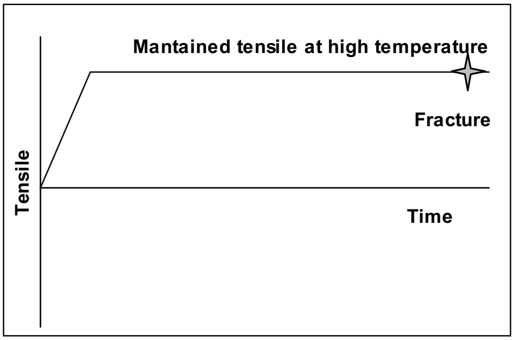

Figure 2 .- Creep mechanism

Low Cycle Fatigue

Low cycle fatigue is produced because of changes ocurring in stresses and temperatures during start, shutdown and load changes when unit is working according with load demand (see figure 3).

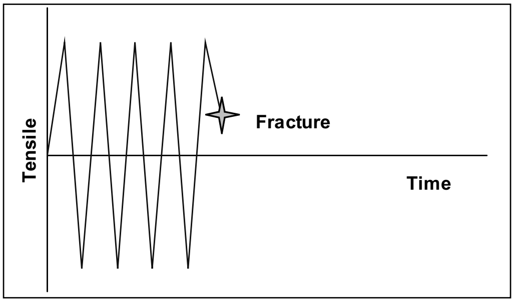

Figure 3. Fatigue mechanism

Turbine components with thick section construction, such as cylinders or high pressure casings, working under high pressure and temperature action, are affected by creep-low cycle fatigue, produced by combination of thermal fatigue caused by load changes and creep caused during high temperature and pressure operation. Schematic representation of mechanisms is showed on figure 4.

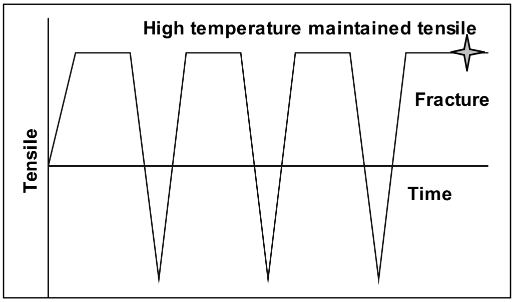

Figure 4. Creep-fatigue mechanism

REMANENT LIFE ASSESSMENT

Steam turbine components work under severe conditions, such as high temperature and pressure.

Will be then needed to know the amount of life consumed for the component until a failure was produced, in order to assess and calculate the remaining time until deffect produces failure. So that, a high machine availability level will be reached, because is possible to know, with certain grade of accuracy, the moment in which the component failure is going tobe produced.

The análisis is carried out for each failure mechanism, based in progress data records of produced deffects.

Creep Life Assessment

Creep life is assessed in basis of operation temperature and pressure stresses calculations at the critical areas using creep rupture strength tables for each material.

Creep under steady operation conditions is calculated taking into account both the centrifugal stresses set up by the rotation of the rotor an the thermal stresses set up by the temperature gradient in the rotor body. Additional loads imposed by the blades are also taken into account.

The method involves complex calculations, and an example ofresults can be saw in table 1. Calculations were made using following data, as well as a polynomial expression for temperature variation with respect to radius [1]

Bore diameter = 101 mm
Rim mean diameter = 919.5 mm
Blade centrifugal force at 3000 r.p.m. = 47.0 kN/Blade
Young modulus, E = 172900 MPa
Coef. Of thermal expansion = 0.1625×10^{-4} / ºC
Cylinder outside diameter = 711 mm
Disc otside diameter = 876 mm
Number of blades = 110
Density = 7833 kg/m^3

Calculations are based on Von Mises criterion or Von Mises-Hencky theory, that stablish that failure on ductile materials occurs when the energy of distortion by volume unit reaches the same level or exceeds the distortion energy level of the same material when traction test yield strength is reached. This theory considers the energy associated at changes of material shape and is appropiate for ductile materials, because it shows very well the triaxial stress status at the component.

Von Mises theory is expressed as:

$$\sigma_v = \sqrt{\frac{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2}{2}} \qquad [6]$$

where $\sigma_1, \sigma_2, \sigma_3$, are the principal stresses acting on component.

Centrifugal force stress in cylinder				
Radius (mm)	Stress(MPa) Axial	Tangential	Radial	Radial displacement (mm)
50.5 355.5	10.25 –10.25	128.5 38.21	0.00 21.68	0.0366 0.0715

Centrifugal force stress in disc				
Radius (mm)	Stress(MPa) Axial	Tangential	Radial	Radial displacement (mm)
355.5 438		48.55 42.79	45.89 32.37	0.0715 0.0838

Termal stress in cylinder					
Radius (mm)	Stress (MPa) Axial	Tangential	Radial	Radial displacement (mm)	Temperature °C
50.5 355.5	22.32 –22.32	25.4 –20.98	0.00 1.51	0.005 0.031	526.7 537.9

Termal stress in discs					
Radius (mm)	Stress (MPa) Axial	Tangential	Radial	Radial displacement (mm)	Temperature °C
355.5 438.0		–13.71 –15.81	3.19 0.00	0.031 0.050	537.9 540.0

Table 1. Example of rotor creep results calculation

The assessment is subjected to uncertainties resulting from several factors, as:

– Accuracy of the predicted stress and temperature
– Material properties, such as creep toughness.
– Deviation of the operating temperature from design.
– Degradation of material properties due to service exposure.
– Deviation from design either at the manufacturing stage or by subsequent modifications.

Therefore, the creef life of rotors is assessed in stages, each successive stage being more detailed as the remnant life margin is reduced.

The first stage is the initial assessment based on global service data from the same family design rotors. The stresses at critical regions are calculated and the life at each region is assessed on the basis of the generic creep data for the rotor steel.

The assessed life of the rotor is taken to be the minimum obtained for the regions considered, and will be termed as "Assessed rotor creep life"(ACRL). (See table 2)

The rotor is allowed to run to 50% of the ACRL without further actino. If the perceived operating life of the rotor is grater than 50% ACRL, it is necessary to proceed to later stages.

Second assessment stage considers individual rotors, that requires the determination of creep average operating temperature for the rotor and examination of the records of the inspections carried out during manufacture of the rotor to establish the size of defects. Besides, the records of steel composition and heat treatment during manufacture are used to estimate the creep toughness and ductility of the rotor (see table 2).

Creep effect	1º stage blade fixing			
	Stress (MPa)	Temperature (ºC)	Life to rupture (h)	50 % ARCL (horas)
Rotor type 110	94	542	2.6×10^5	130000
Rotor type 94	89	542	1.8×10^5	90000

Tabla 2. Example of assessed rotor creep life calculation (ARCL)

The rotor life is then assessed using stresses derived from design calculations as in stage 1, but using the creep average operating temperature derived from the operating records and the creep data obtained from the manufacturing records.

Creep average temperature is increased for operation at off-design condition. This is particulary important on the high pressure turbine, where operating at different load to design rated load can change the steam entalphy and, as a result, temperature after governing valves. Temperature has an important effect on creep, because, as an example, a 10 ºC rise in the nominal steam temperature of 540 ºC will increase creep strain and creep rupture by a factor of about 2.

Third stage requires removal of the rotor from the turbine in order to take material samples and to carry out non-destructive testing (NDT). The procedure is similar to that of stage 2, but the creep strength and ductility are determined directrly by acelerated tests at higher than operating temperatures on samples of steel removed from the cooler part of the rotor, which has not been subjected to high temperature.

Magnetic particle inspection (MPI) is applied at the bore surface and at regions on the outer surface where creep or termal fatigue damage may be expected., and ultrasonic examination is carried out where this is posible. Replication of the surface in critical areas is used to check for creep cavitations or micro cracking which may be due to creep.

Once creep remnant lif is assessed for these stage, tue rotor is allowed to operate to 95% of ARCL.

Low Cycle Fatigue Life Assessment (Thermal Fatigue)

Thermal fatigue phenomenom is common on high temperature turbines, in this case, low cycle fatigue, that occurs mainly due to severe temperature reached during start up and initial stages of load generation. It is also afected by a combination of factors, sucha as deviations from the stipulated start-up procedures, poor designs, poor machining and poor material properties.

Excessive temperature increase during start up induces large stresses that could excede the yield limit of the material.

A typical stress-strain cycle during one cycle of start-up and shutdown is showed at figure 5 [2].

Considre a component subjected to an alternating tensile/comprensive load cycle in which the total stress variation is "Δσ". This load introduces within the component stresses ranging from the maximum comprensive effect "–Δσ" to the maximum tensile effect "+Δσ", that produce the subsequents strains "–Δε" and "+Δε". Normally, these loads are of equal magnitude in the tensile and comprensive direction.

The center point "O" represents the first put in service of the component, where it has not still experienced cycling load. When an initial tensile load is applied, a tensile stress is induced that produces a linear elastic strain "$\Delta\varepsilon_{ie}$" up to condition A.

After condition A, the extensión with increasing load continúes but no longer follows a linear relationship, being a plastic strain "$\Delta\varepsilon_{ip}$" and, therefore irrecoverable. Therefore, the total strain is:

$$+\Delta\varepsilon = \Delta\varepsilon_{ie} + \Delta\varepsilon_{ip}$$

At condition B, the load has reached its maximum tensile value. Now, as the load is reversed, the tensile stress reduces to zero and strain

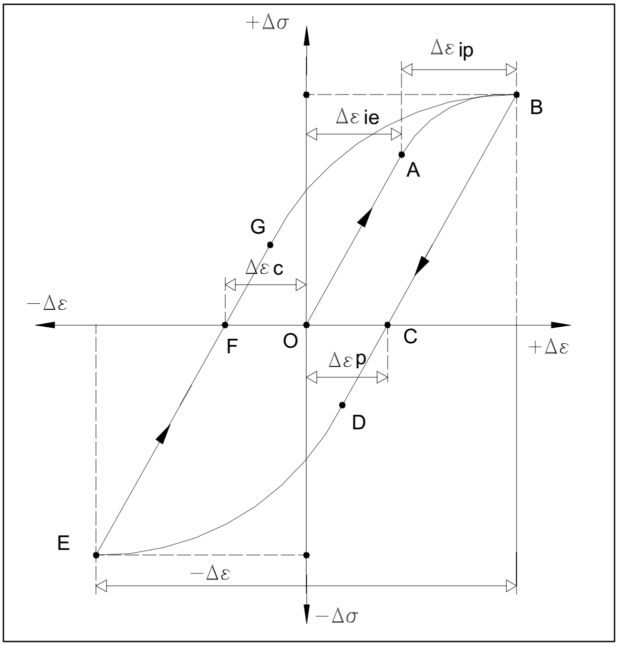

Figure 5. Typical stress-strain cycle for a turbine component material

reduces, but does not reach zero (C condition). A plastic strain "$\Delta\varepsilon_p$" remains at the component. At this condition, the direction of the load is reversed, the load becomes comprensive, initially following a linear (elastic) relationship to condition D, where the strain becomes plastic and follows a curved relationship from D to E, untill comprensive stress reach its maximum value at E, "$-\Delta\sigma$" . As a result of this, a total elastic and plastic strain "$-\Delta\varepsilon$" is produced.

Next, the compresive load is reduced from a maximum value, and so the compresive stress is reduced reaching zero at condition F, where there is still a residual strain remaining "$-\Delta\varepsilon_c$". The tensile load is again applied with the initial strain being elastic to G and the final strain being plastic to B.

This series of stress and strain are repeated causing the material of the component undergoes a cyclic loading and the accumulation of plastic strain continues to deform the component.

If comprensive and tensile stresses are of the same magnitude, the curve will be symmetrical about the point of origin O.

The total strain range of the component is the sum of "$+\Delta\varepsilon$" and "$-\Delta\varepsilon$", shown at the diagram as the strain range from B to E, that consists of two elements, the elastic "$\Delta\varepsilon_{ie}$" and the plastic "$\Delta\varepsilon_{ip}$" portions.

The elastic portion is represented by the ranges OA, BD and EG, but since the range OA is present only during the initial strain cycle, it can be neglected. The plastic strain ranges are from AB, DE and GB, and again, the range AB can be neglected.

Then, there is a relationship between the total and elastic strains for any material, that is taken by the expression:

$$+\Delta\varepsilon = B \cdot \Delta\varepsilon \cdot \Delta\varepsilon_p \cdot g$$

where

$\Delta\varepsilon$ is the total strain range

$\Delta\varepsilon_p$ is the plastic strain range

B and *g* are constants determinated for the material

This relationship is only approximate and applicable primarily to cycles of intermediate strain ranges.

The accumulated damage will lead to crack initiation as the number of start up increase. Thereafter, crack will grow with each additional start.

The remnant life is assessed as the time for the crack to grow to a depth at which the steady state stress could cause rupture in a relatively short time. It should be noted that the procedure for estimating the thermal fatigue carck growth rate includes the effect of the steady state stresses (for example, at the creep state).

Termal fatigue life is not assessed in a easy procedure, as is creep life. A similar first stage assessment can be considered using typical starting and shutdown procedures for a family of turbines. Calculations are then made to predict the stresses set up at the critical regions of cold, warm and hot start conditions due to the rate of rise of steam temperature. The calculated stresses are then used to predict the number of cycles required for crack initiation for each type of start.

An example of start up data and the induced termal stresses at the critical locations of a HP turbine rotor is given in figure 6.

The life expired is the sum of the number for each type of start accumulated to date divided by the number of cycles calculated for cold, warm and hot starts. That is to say if the calculated number of starts are N_f, N_t and N_c, and the number of accumulated starts to date are n_f, n_t y n_c

$$RL = 1 - \left(\frac{n_f}{N_f} + \frac{n_t}{N_t} + \frac{n_c}{N_c} \right)$$

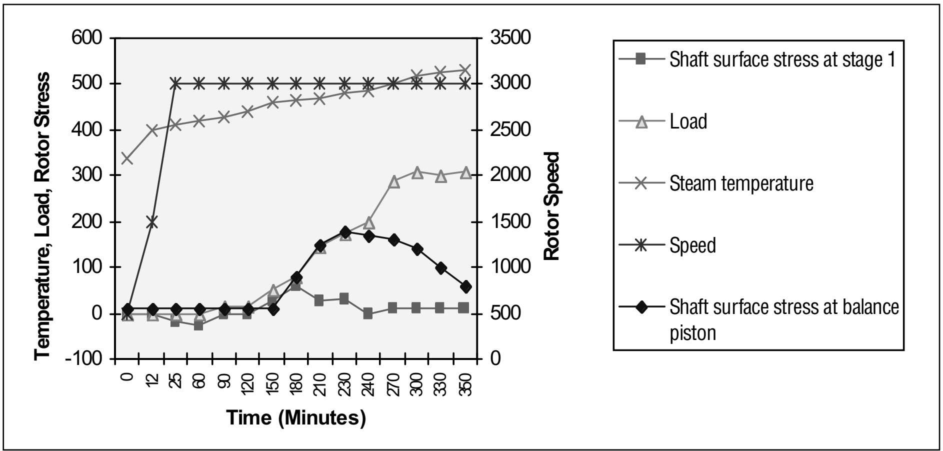

Figure 6. Termal stresses set up on rotor during start-up

Generally, cold starts are defined when the metal temperature falls below 200 ºC. A start following a weekend shutdown can be classified as warm, and after an a trip is considered as hot.

Experience shows that normally cold starts induce about three times more damage than warm or hot starts. Therefore, an equivalent number of starts may be used to calculate the allowable safe number of starts before an inspection becomes necesary.

$$Equivalent\ starts = cold\ starts + \frac{1}{3}\left(warm\ starts + cold\ starts\right) \qquad [3]$$

High pressure and intermediate pressure rotors, valve chests or steam chests and cilindres must be inspected before they reach the number of equivalent starts.

Normally, the magnitude of the start up termal stress decreases with depth and crack growth may stop. However, propagation to a critical size where it could grow due to thermal cyclic stresses needs to be considered. Therefore, is advisable to adopt a policy against crack initiation when manufacturing, and the procedure of determination of crack growth rate by means monitoring it, at future inspections.

The calculated termal fatigue life indicates the risk of damage at rotor and regions of the rotor where damage is most likely to occur. An example of this is given in table 3 [4].

<table>
<tr><th rowspan="2">Component</th><th rowspan="2">Location</th><th rowspan="2">Stress (MPa)</th><th colspan="3">Nº of starts to cracking</th><th rowspan="2">Inspect at</th></tr>
<tr><th>Cold</th><th>Warm</th><th>Hot</th></tr>
<tr><td rowspan="2">HP Rotor</td><td>Balance piston</td><td>2.5</td><td colspan="3">Little risk of cracking below 10000 starts</td><td>Major overhau</td></tr>
<tr><td>Reaction stage</td><td>5</td><td colspan="3">Little risk of cracking below 10000 starts</td><td>Major overhaull</td></tr>
<tr><td rowspan="2">HP Cylinder</td><td>Inlet piping</td><td>1.5</td><td colspan="3">Little risk of cracking below 7000 starts</td><td>Major overhaul</td></tr>
<tr><td>Diaphragm groove</td><td>3</td><td>1700</td><td></td><td></td><td>Inspect before 2000 equiv. starts</td></tr>
<tr><td>IP Rotor</td><td>First Reaction stage</td><td>4</td><td>900</td><td>900</td><td>3300</td><td>Inspect before 1000 equiv. starts</td></tr>
<tr><td rowspan="2">HP Cylinder</td><td>Inlet piping</td><td>1.5</td><td colspan="3">Little risk of cracking below 10000 starts</td><td>Major overhaul</td></tr>
<tr><td>First stage stationary blade fixing</td><td>3</td><td>3000</td><td></td><td>2500</td><td>Major overhaul</td></tr>
</table>

Table 3. Estimated termal fatigue life and recommended inspections [5]

Figures 7 and 8 show the distribution of temperatures and thermal stresses at first HP blade grooves during a hot start.

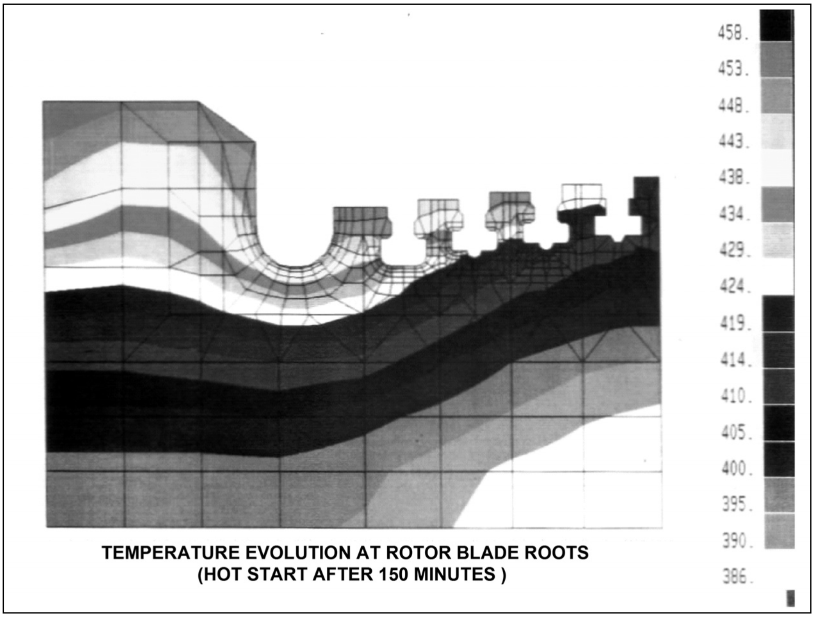

Figure 7. Rotor temperature distribution during a hot start

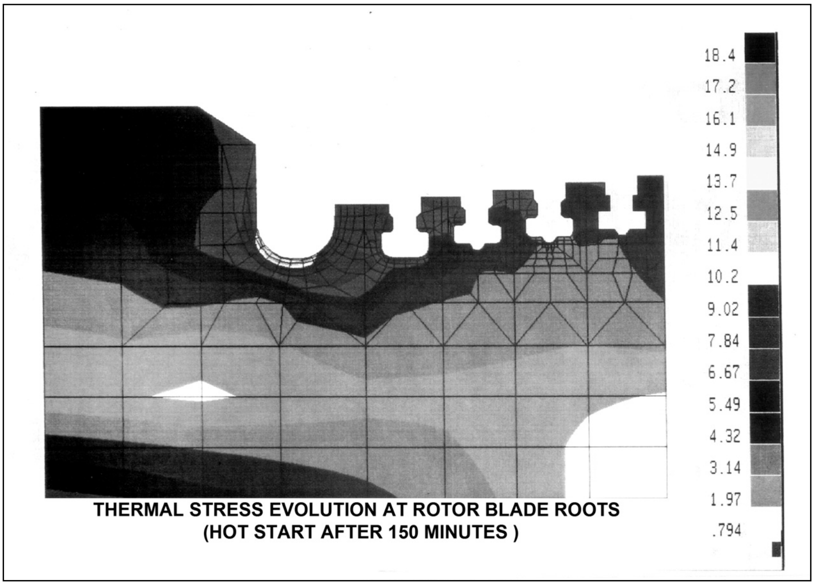

Figure 8. Rotor thermal stress distribution during a hot star

CONCLUSIONS

In order to get a suitable remnant life assessment of steam turbines and to guarantee its reliable operation, is essential to establish an inspection program affecting at least the mos critical parts of the machine, that are those sensitive to produce stress concentration, such as seal fin grooves, blade root serrations, balance holes regions, changing section regions, etc...

The specification and scope of neccesary inspections must be agree on manufacturer and inspections specialized people, and, as a minimum must include:

— Assessment of initial condition of component and operation and maintenance data records n order to identify critical components.
— Magnetic particle inspection of the complete rotor with particular attention to changes of section.
— Ultrasonic examination from the bore rotor. This technique, known also as Boresonic, achieves to inspect bore surface and blade fixing areas in order to look for cracking.
— Ultrasonic inspection from critical rotor surface regions, in order to complement magnetic particle or liquid penetrant tests and to get a suitable measurement from deffects found by means these techniques.
— Metallurgical examination by means replications from critical regions, in order to study the material metallurgical structure and to know its age deterioration grade.
— Ultrasonic inspection and magnetic particle inspection from steam chests. If possible, is advidable to measure the depth and length of known deffects and monitor it. Deffects found at earlier inspections, must be also measured to determine if its progress is as expected. Evaluation of chest integrity at these inspections will provide a guide to the frequency of further inspections and planning for eventual repair or replacement.
— Magnetic particle examination of the cylinder at steam inlet belt region and stationary blade rings.
— Set up a database of turbine operating statistics including hours of operation , number of starts, records of previous inspections.
— Monitoring, when posible, the termal stress and remnant termal fatigue life of high pressure and intermediate pressure, that are the ones affected by worst working conditions.

Knowing phenomenom affecting turbine integrity, evaluation of time progress of deffects caused by them, data obtained from operating records from own or another similar machines, review from critical components inspection results, are an essential tool to asses remnant life of turbine.

Assessment of this remnant life, will allow the establishment of economically capable preventive maintenance plans or improbé or renewal plans in order to guarantee availability and reliability of machines, and economically capable life extensión programs to allow to extend steam turbines operation.

REFERENCES

[1] Life Management of Steam Turbines (Turbinecare Ltd. Sept. 2000))
[2] Advances in Steam Path Technology (.John I. Cofer, John K. Reinker, William J. Summer, GE Power Systems Schenectady, NY))
[3] Inspection Manual for MHI 350 MW Steam Turbines (MHI Ltd. Nagasaki Shipyard & Machinery Works)
[4] Life Management of Steam Turbines (Turbinecare Ltd. Sept. 2000))
[5] Life Management of Steam Turbines (Turbinecare Ltd. Sept. 2000)
[6] www.en.wikipedia.org

BIBLIOGRAPHY

Inspection Manual for MHI 350 MW Steam Turbines (MHI Ltd. Nagasaki Shipyard & Machinery Works)
Life Management of Steam Turbines (Turbinecare Ltd. Sept. 2000)
Technical Bulletin, Reliability Enhancement. MHI Ltd.
Catálogos de Refurbished Turbine Components (Turbine Blading Ltd.)
Repair of Very Large Steam Turbines (Barney McLaughlin. Sulzer Turbomachinery Services)
Advances in Steam Path Technology (John I. Cofer, John K. Reinker, William J. Summer, GE Power Systems Schenectady, NY)
Páginas web: www.mhi.co.jp
www.plant-maintenance.com
www.en.wikipedia.org

ASPECTOS DE LA EVOLUCIÓN DE LA VIDA RESIDUAL EN TURBINAS DE VAPOR VIEJAS

RESUMEN

Con en fin de conseguir una alta disponibilidad de operación e implantar un plan de mantenimiento y de gestión de repuestos económicamente viable para llevar a cabo el alargamiento de vida de las turbinas de vapor, es imprescindible una adecuada valoración del deterioro por envejecimiento de las mismas, así como de la vida residual de los componentes más relevantes.

En base a los resultados de inspecciones de integridad de la máquina, consumo de vida actual y vida remanente estimada se estructuran, de acuerdo con los resultados de la evaluación del rendimiento del ciclo, los planes de mantenimiento futuros, la renovación de equipos e incluso de grandes partes de la instalación para que la planta siga operando en unas condiciones de seguridad, fiabilidad y rendimiento óptimas.

Es fundamental, por tanto, para conseguir estos objetivos, conocer los principales mecanismos de deterioro por envejecimiento y saber de qué forma afectan a los diferentes componentes de la turbina.

De todos los mecanismos de fallo que pueden producirse en las turbinas de vapor, solamente algunos de ellos intervienen directamente en el envejecimiento de los componentes de las mismas, y son los ocasionados por el efecto de las altas temperaturas mantenidas durantes largos períodos de tiempo, así como las variaciones bruscas de estas temperaturas.

METODOLOGÍA

La metodología utilizada ha estado basada en las experiencias obtenidas durante sucesivas revisiones de turbina en una central termoeléctrica en las que han intervenido, además del personal de la empresa propietaria, los fabricantes de las turbinas y otros tecnólogos especialistas en inspecciones y ensayos no destructivos utilizando las últimas tecnologías en equipos y técnicas de inspección.

Los alcances de este tipo de revisiones suelen estar estandarizados para el tipo de turbina de que se trate, aunque a veces se tienen en cuenta otras necesidades de la máquina. También están estandarizados los alcances de las inspecciones a realizar durante la revisión, que suelen ser los siguientes para este tipo de mecanismos de fallo, termofluencia y fatiga de bajo ciclo:

- — Inspección completa mediante partículas magnéticas de los rotores, prestando especial atención a los cambios de sección y radios de acuerdo.
- — Inspección mediante ultrasonidos de los orificios centrales de los rotores.
- — Inspección mediante ultrasonidos de las zonas críticas de las superficies de los rotores.

— Inspección metalográfica de las zonas más críticas en rotores, coronas y carcasas.
— Inspección mediante partículas magnéticas de las cajas de vapor.
— Inspección mediante partículas magnéticas de los manguitos de entrada de vapor y álabes de coronas fijas.

Los datos resultados obtenidos mediante estas inspecciones se comparan con los obtenidos en inspecciones anteriores, y de esta forma se puede evaluar la evolución de defectos existentes o que habían sido reparados.

La experiencia y conocimiento de la máquina por parte del fabricante de la misma, hace muy necesaria su intervención en la revisión, de cara a la expedición de recomendaciones y valoración de defectos encontrados, pero también es muy importante la intervención de especialistas en inspecciones que puedan sacar a la luz defectos que por su naturaleza no siempre son fáciles de descubrir.

CONCLUSIONES

Para una adecuada valoración de la vida remanente de una turbina de vapor y para asegurar la operación fiable de la misma, es fundamental la implantación de un programa de inspecciones que afecte al menos a las partes más críticas de la máquina, que son aquéllas susceptibles de producir concentraciones de esfuerzos, como los encastres de las pletinas de cierre, encastres o serratiles de álabes, las inmediaciones de los orificios de equilibrado, las zonas de transición bruscas debidas a cambios de diámetro, etc...

La especificación y alcance de las inspecciones necesarias han de ser consensuadas con el fabricante de la máquina y con los especialistas en materia de inspección, y como mínimo han de incluir las siguientes:

— Valoración inicial del estado de los componentes y estudio de los históricos de operación y mantenimiento para identificar los componentes críticos
— Inspección completa mediante partículas magnéticas del rotor, prestando especial atención a las zonas donde se producen cambios de sección.
— Inspección mediante ultrasonidos del orificio central del rotor. Con esta técnica, conocida también como Boresonic, se consigue inspeccionar, además de la superficie del orificio, las zonas de los encastres de los álabes en busca de posibles fisuras.
— Inspección mediante ultrasonidos en zonas críticas de la superficie del rotor, con objeto de complementar las inspecciones mediante partículas magnéticas o líquidos penetrantes y, en su caso, dimensionar los defectos encontrados con esas técnicas.
— Inspección metalográfica mediante réplicas en las zonas críticas, con objeto de estudiar la estructura metalográfica del material y determinar su grado de envejecimiento.

— Inspección mediante ultrasonidos y partículas magnéticas de las cajas de vapor. Si es posible, conviene dimensionar en profundidad o longitud los defectos encontrados y realizar un seguimiento de los mismos. En cuanto a los defectos ya conocidos de inspecciones anteriores, han de ser también dimensionados para determinar si su comportamiento es el esperado. La evaluación del estado de las cajas de vapor mediante estas inspecciones proporcionará una guía de la frecuencia con que se han de realizar las inspecciones y para planificar posibles reparaciones o sustituciones.
— Inspección mediante partículas magnéticas de la carcasa o cilindro interior de la turbina en la zona de los manguitos de entrada de vapor y en las coronas de álabes fijos.
— Establecimiento de una base de datos de las estadísticas de operación de la turbina, en la que se reflejen, además de las horas de operación, número de arranques, resultados de anteriores inspecciones, etc...
— Monitorización en lo posible de las tensiones térmicas y la vida remanente por fatiga térmica de los componentes de la turbina de alta presión y presión intermedia, que son las sometidas a las peores condiciones.

El conocimiento de los fenómenos que afectan a la integridad de la turbina, así como la evaluación del progreso con el tiempo de los defectos ocasionados por ellos, los datos obtenidos de la acumulación de históricos de funcionamiento de la máquina o de otras similares, la interpretación de los resultados de inspecciones de los componentes críticos, son una herramienta fundamental a la hora de valorar la vida remanente de la máquina.

La evaluación de esta vida remanente permitirá la implantación, de manera económicamente viable, de planes de mantenimiento preventivo o planes de mejora o renovación con objeto de asegurar la disponibilidad y la fiabilidad de las máquinas, y programas de extensión de vida económicamente viables que permitan prolongar la explotación de las turbinas de vapor.