Influence of Start-up Management on the Residual Life of a Large Steam Turbine Shaft

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Abstract

The liberalization of electricity market in Europe led to a growing competition between energy producers, making crucial the ability to optimize the management strategies of power plants. Combined Cycle Power Plants (CCPP) have to operate in a flexible way, with frequent and rapid variations of the power produced, in order to quickly adapt to the frequent changes in load imposed by the demand. Nowadays, they typically operate in cycling mode with daily start-up and shut-down. The components of the plant are subjected to great cyclical variations in temperature, which induce stresses on materials, especially during the start-up phases. The present activity concerns the assessment of life consumption -caused by these operations- on the rotor of the steam turbine of the CCPP (800 MW) inside the Tirreno Power thermal plant located in Vado Ligure, Italy. The aim is to draw a set of curves representing the percent life expended per cycle as a function of rate of steam temperature change and magnitude of the overall temperature increase. These curves are called Cyclic Life Expenditure curves (CLE). In the future, the developed methodology will be used to reduce the start-up times, keeping under control the life consumption of the rotor and optimizing the maneuvers that generate thermal transients.

Keywords

Steam Turbine's Start-Up – Turbine rotor – CLE Curves – Thermoelastic stress

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INTRODUCTION

The progressive liberalization of electrical energy and natural gas in Europe introduced a growing competition between energy producers, making fundamental the ability to optimize management strategies and control of the plants to reduce production and service costs. This process must also take into account strict environmental regulations on pollutants emission.

The ability of the operators in optimally managing the facilities is essential to be competitive in the liberalized market. The variability of the fuel cost, the instability of the electricity demand and its price, the continuous increase of power production from renewable sources, create complex problems to operators and force them to develop operational models for improving management systems in terms of efficiency, flexibility and reliability.

All these aspects characterize the modern control strategies of the power generation units; in fact, a management method which can operate the plant in a highly discontinuous and irregular manner is crucial for:

- Meeting the users' demand and the severe regulations on emissions of macro pollutant (mainly CO and NOx)
- Safeguarding the stability of the electrical grid
- Maximizing profits

The power plant should be able to produce more power during peak hours, where the gain is higher, whilst reduce the power supplied to the grid or turns the production unit off in the hours of low demand. Therefore, each unit must be able to operate both at high loads and to remain off or at the minimum during the night and the weekend, but also to restart in a short time. The start-up time is strictly related to the initial status of the power plant which depends basically from the standstill time. After 8 hours of standstill, an hot start up can be performed in less than 60 minutes; a warm start-up, typically after the weekend standstill (8÷48 h) takes about 80÷120 minutes, while the cold start up requests from 2 to 3 hours [1].

Gas turbines and combined cycle are more flexible than other types of thermal power plants (fuel oil or coal), and are often called to operate in a discontinuous manner characterized by frequent and rapid variations of the produced power [2]. Nevertheless, the new requirements push all plants, even those initially designed to produce as base load, to change the type of operation or switch to a more flexible management for maximizing the economic return and safeguarding the electrical grid stability [3].

A flexible management ensures higher profits in the short term, but reduces the life of critical components of the system because of the damage caused by thermo-mechanics fatigue phenomena that arise during load changes, start-ups and shut-downs. The parts subjected to high temperature as the gas turbine and the steam components suffer significant cyclical variations in temperature and pressure parameters that induce stresses on the material, thus damaging the components [3]. This means loss of profit due to frequent failures or the need of ordinary and extraordinary maintenance.

Of course, the most stressful maneuvers are the plant start-ups and shut-downs, which affect the residual life of the components and the reliability of the whole system. In order to have the best performance during the entire life of the plant, the operator must be aware of the relation between the residual life of the component and the rate of load change to provide guidance on the long-term consequences. Gas Turbine (GT) manufacturers take into account for the additional life consumption during start up by summing the Equivalent Operative Hour to the "fired" Operative Hour or setting a maximum number of allowable start-ups between two maintenances [4]. Similar counters were introduced for the Steam Turbine (ST), discriminating between the different kind of start-up (hot, warm, cold) [1], but they can miss part of the whole picture. In fact, the life consumption due to the start-up maneuver is not just related to the initial conditions (initial rotor temperature), but also to the way the temperature ramp is performed (i.e. temperature mismatch and temperature gradient). A simple way to visualize this effect is represented by the Cyclic Life Expenditure (CLE) curves, (e.g., see Figure 6.13 in Viswanathan [5]) often given by the manufacturer. Those curves represent the percent life expended per cycle as a function of the rate of steam temperature change, and magnitude of steam temperature change and can be used operatively in the start- up optimization process.

Several systems were installed by Original Equipment Manufacturers (OEM) of Steam Turbines or other companies to keep under control the stress of the steam turbine [3,6,7]. For example the Siemens Turbine Stress Controller allows different start-up modes which can be preselected by the operator depending on the grid requirements [1,3]. A similar approach is proposed by Ansaldo Energia: the temperature change is monitored with the so called Rotor Stress Evaluator (RSE) and the stress is maintained under some fixed thresholds limiting the GT and ST load rate [6]. The topic is not new, the Electric Power Research Institute (EPRI), developed a big study in 1994 [8], but is becoming more crucial, as shown by two recent patents developed by O&M [9,10]

In this paper the focus is on the evaluation of the stresses on the rotor shaft of a steam turbine of a combined cycle gas-steam power plant for the production of electrical power. It is inside the Tirreno Power thermal plant located in Vado Ligure, Italy: the power plant consists of a combined cycle unit (800 MW) and two coal-fired units (330 MW each).

The new combined cycle module started in 2007 and was created to replace an old unit fueled by coal and fuel oil. This choice allowed a significant increase in the efficiency of the system, not to mention that the steam turbine, its auxiliary, the condenser and the generator have been revised and updated to be reused in the new facility. This unit uses two multi-shaft gas turbines (264 MW each), fed with natural gas, and its exhaust goes into a steam turbine (273 MW). Both the gas turbine and the steam turbine were produced by Ansaldo Energia.

1 TIRRENO POWER'S PLANT MANAGEMENT AND ROTOR STRESS EVALUATOR

The control system of the steam turbine plant optimizes the time of start-up of the groups in the combined cycle.

The Rotor Stress Evaluator (RSE) is the system which limits the gradient of ST and GT load in order to reduce the fatigue on the components. It protects the rotor of the machine from exceeding the allowable thermal stress, and therefore limits the thermal stress on the rotor of the steam turbine in two ways:

- Preventing the inlet steam temperature mismatch with respect to the rotor surface
- Acting on the loading ramps

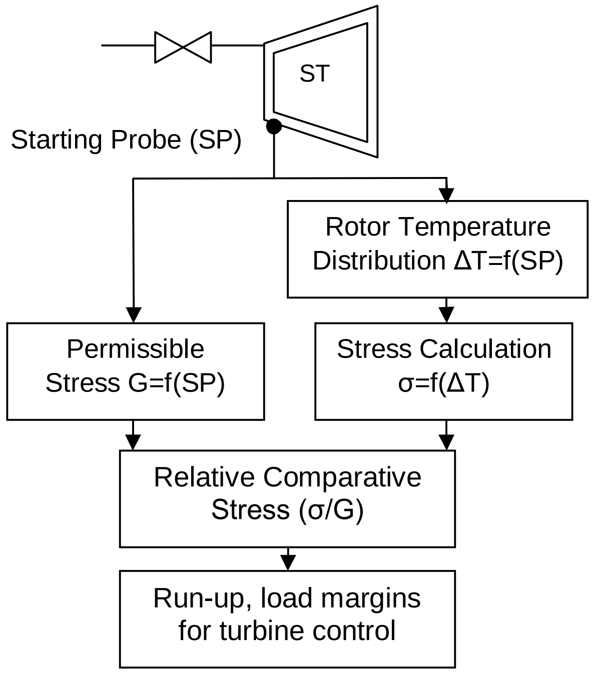

Figure 1. Rotor Stress Evaluator Calculation Scheme

The Rotor Stress Evaluator is based on an algorithm that processes the temperature of the rotor surface in the section of admission of high and medium pressure steam, an example of probe installation is provided in [3]. This temperature is considered equal to the measured stator case internal surface temperature, according to the hypothesis that the coefficients of heat exchange of the steam with the metal of the case and the rotor are quite similar (actually the temperature of the rotor is underestimated of about 40°C due to the thermal losses to the ambient).

Then, the average and the center temperature of the rotor is estimated solving -in a simplified manner- the differential equation of heat transfer in a cylindrical body, through an algorithm based on a truncated series expansion.

The stresses are then calculated on the basis of the temperature differences and the permissible stress is calculated as function of the surface temperature (prudential choice).

A parameter called "margin of load" is evaluated in the rotor region (surface or center) having the slightest difference between calculated stress and permissible stress. The margin of load conventionally produces limitations just when a threshold is overcome: a proportional action is triggered to reduce the load gradient. If the load reduction does not reduce the mechanical load or the calculated stress itself overcomes some security threshold, the RSE trips (i.e. shuts down) the turbine to avoid severe consequences for the rotor.

2 THERMAL DISTRIBUTION IN THE ROTOR

The first step leads to verify the truncated series expansion results, with respect to analytical calculation in order to use this specific part of the RSE algorithm to estimate the time evolution of rotor's temperature due to different rates of load change. In fact, this is the basis in creating the CLE curves.

So, the RSE program response is compared with the results of analytical calculations for simple input laws. The known analytical models are, in fact, limited by the need to have well-defined boundary conditions.

The geometrical dimensions and physical characteristics used in the calculations are the following ones:

- Rotor diameter, HP stage (first row of blades): 725 mm
- Rotor diameter, IP stage (first row of blades): 810 mm

The rotor hasn't a central bore and is a welded disc type. The material is a special alloy for large turbines and generators, having the composition defined as DIN 30CrMoNiV5-11, which mechanical characteristics of can be found in [11].

Therefore, the results are analyzed for the case of the rotor which undergoes a step change in its surface temperature, and a step change in the conditions of the steam lapping it; a one-dimensional model of heat exchange in the radial direction is adopted, which allows to considerably simplify the calculations without incurring in large errors.

The first comparison is made with a step change in the surface temperature, see Figure 2. The second one is obtained by a rising temperature of the rotor surface, generated by steam (in constant conditions) entering the machine, see Figure 3.
The trend calculated with the analytical model is faithfully reproduced by the RSE. The maximum temperature difference results in the center of the rotor and is about 8° C; moreover, the trend of the average temperature has been calculated correctly, as well. So, as the RSE offers results almost identical to those of the analytical model it was decided to use the RSE series expansion to calculate the temperature distribution, and its values for the calculation of thermo-elastic stresses during transients.

3 STRESS EVALUATION

The stresses in the rotor are produced by temperature gradients in the material, by the centrifugal force, by the pressure. They can be evaluated separately and combined summing the terms acting through the same axis and then converting the triaxial stresses to an equivalent uniaxial stress using the Von-Mises method [12].

First, using the elasticity equations for a unit length cylinder, the radial, circumferential and axial thermal stresses are calculated, respectively [13]:

$$\sigma_r(r) = \frac{E \cdot \alpha}{1-\nu}\left[\frac{r^2 - r_i^2}{2r^2}\bar{T} - \frac{1}{r^2}\int_{r_i}^{r} T(r) \cdot r dr\right]$$

$$\sigma_\vartheta(r) = \frac{E \cdot \alpha}{1-\nu}\left[\frac{r^2 + r_i^2}{2r^2}\bar{T} + \frac{1}{r^2}\int_{r_i}^{r} T(r) \cdot r dr - T(r)\right]$$

$$\sigma_z(r) = \frac{E \cdot \alpha}{1-\nu}\left[\bar{T} - T(r)\right]$$

The average temperature is given by:

$$\bar{T} = \frac{2}{r_e^2 - r_i^2}\int_{r_i}^{r_e} T(r,t) \cdot r \cdot dr$$

The integration is made in the radial direction.
For example, a temperature difference of about 100 °C between the surface and the average temperature, typical for warm start up, causes a thermal stress of about 360 MPa, without taking into account the concentration stress factor.

The stresses due to the rotation of the rotor spinning at 3000 rev/min (50 Hz) for the HP and IP stage are shown in Table 1 and are obtained with the following equations [14]:

$$\sigma_r = \rho\omega^2 r_e^2\left[1 - \left(\frac{r}{r_e}\right)^2\right]\frac{(3+\nu)}{8}$$

$$\sigma_\vartheta = \rho\omega^2 r_e^2\left[1 - \frac{1+3\nu}{3+\nu}\left(\frac{r}{r_e}\right)^2\right]\frac{(3+\nu)}{8}$$

Table 1. Centrifugal Rotor Stresses (MPa)

HP (d = 725 mm)	Surface	$\sigma_\theta(r_e) \simeq 18$
HP (d = 725 mm)	Center	$\sigma_\theta(0) \simeq 42$ $\sigma_r(0) \simeq 42$
IP (d = 810 mm)	Surface	$\sigma_\theta(r_e) \simeq 22$
IP (d = 810 mm)	Center	$\sigma_\theta(0) \simeq 52$ $\sigma_r(0) \simeq 52$

The blades create also a radial stress at the surface which is taken into account using [14]:

$$\sigma_r(r_e) = p_e = \frac{\rho_b\omega^2 r_b(V_b N + 2\pi r_s t_s h_b)}{h_b 2\pi r_e}$$

Where ρ_b, V_b, N and h_b are, respectively, the density of the material, the volume, the number and the width of the blades, r_b is the distance of the center of gravity of the blades from the rotation axis; r_s and t_s are, respectively, the radius and thickness of the shroud.

Table 2. Blades Mass and Geometry Data

	HP	IP
Density $[kg/m^3]$	7800	7800
Distance between Blade's Centroid and Rotor Axis $[m]$	0.384	0.458
Blade Mass $[kg]$	0.131	0.305
Blade Number	103	103
Blade Width $[m]$	0.03	0.03
Shroud Radius $[m]$	0.402	0.511
Shroud Thickness $[mm]$	3	3

The shroud is an outer ring that surrounds each array of blades, having the task of stiffening them against vibrations. It is also the surface on which the seals work for reducing the escape of steam..
The blade's shape was approximated to a parallelepiped. The data taken into account for the blade centrifugal effect are reported in the Table 2.

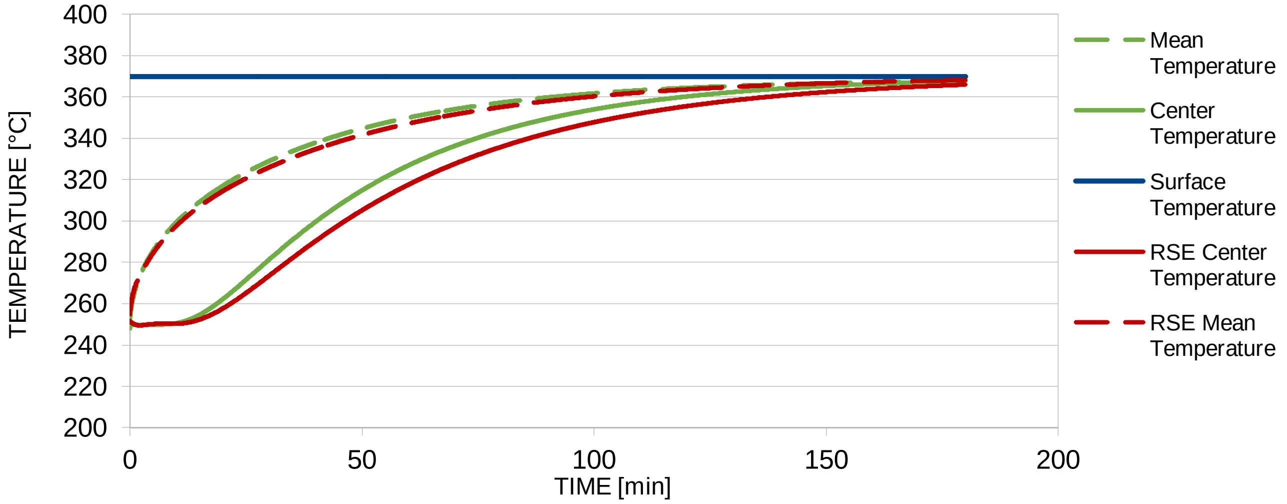

Figure 2. Temperature Comparison for a step change in the surface temperature

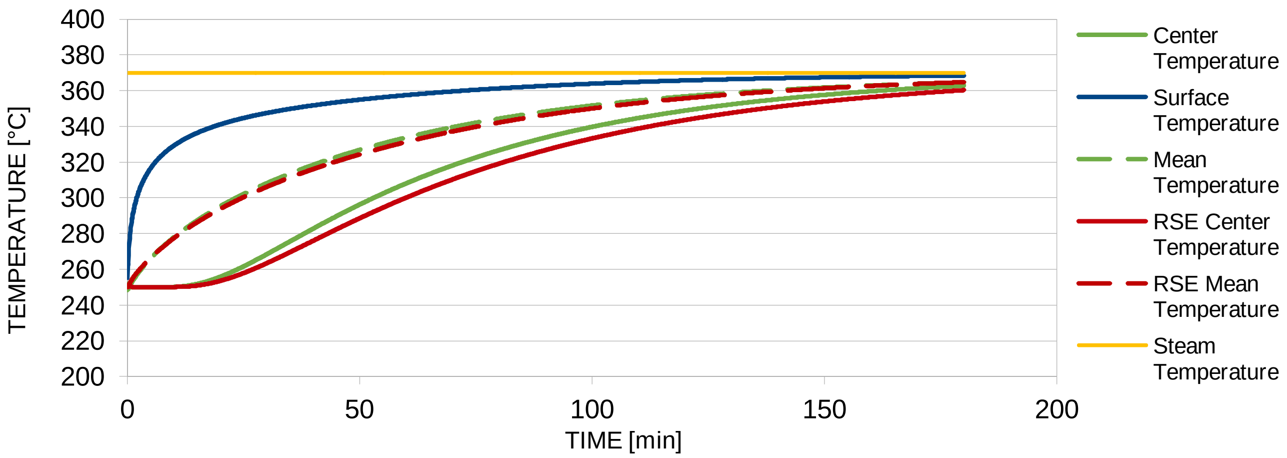

Figure 3. Temperature Comparison for a step change in the steam temperature

Stresses due to the pressure are calculated by the following formulas [15]:

$$\sigma_r = -\frac{{r_e}^2}{{r_e}^2 - {r_i}^2}\left(1 - \frac{{r_i}^2}{r^2}\right)p$$

$$\sigma_\theta = -\frac{{r_e}^2}{{r_e}^2 - {r_i}^2}\left(1 + \frac{{r_i}^2}{r^2}\right)p$$

The maximum pressure acting on the rotor is 126 bar and generates compressive stress at the surface of the rotor (-12 MPa), even if the steam enters the machine the first time with pressure higher than 63 bar resulting in a -6 MPa stress.
It can be easily noted as the thermal stresses are a lot higher than the stresses produced by both pressure and centrifugal forces, weighing for about the 95% of the overall stress.

3.1 Stress Concentration Factor

The effect of stress concentration has been taken into account at the blades' points of insertion on the rotor surface by introducing an appropriate theoretical stress concentration factor in elastic range K_t [16] which involves the radius of the groove of the first stage of blades. For the Tirreno Power turbine rotor it was assumed K_t = 2.8 for both the HP and IP sections.

4 FATIGUE LIFE EVALUATION

The assessment of the life consumption associated with start-up has been carried out through to the low-cycle fatigue theory. Manson Coffin curves have been used, according to the type and operating conditions of the alloy rotor.

4.1 Influence of Plastic Strain on the Stress Concentration Factor

In low-cycle fatigue (LCF), assessments are performed using strains. To calculate the actual value in the notches, where the plastic state is reached, it has been introduced a strain concentration factor K_ε by means of the Neuber's approach [17]. The linear elastic model for the evaluation of the strain provides a lower result than the real one.

After the calculation of the nominal stress σ_n, the value of K_ε is obtained by intersecting the Neuber's hyperbola, having equation $\sigma\varepsilon = K_\sigma K_\varepsilon \sigma_n \varepsilon_n = K_t^2 \sigma_n \varepsilon_n$ and the cyclic stress strain curve of the material.

The result of the evaluation has been represented in Figure 4 for the specific material in the red dashed curve, as a function of the normalized nominal strain range. The trend has been compared with the results proposed for similar materials [5].

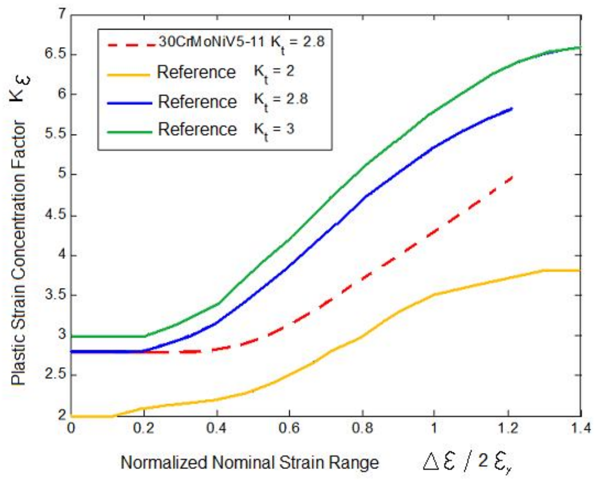

Figure 4. Strain Concentration Factor

4.2 Multiaxial Loading Conditions

Once the type of fatigue cycle is evaluated, the mean and alternate components have been identified for the three kind of stresses acting on the rotor. These have been combined into equivalent stresses, according to the von Mises yield criterion, obtaining the values σ_{idm} and σ_{ida}. This criterion can be applied since the rotor is subjected to centrifugal and thermal loads that act always along the principal directions.

4.3 Influence of the Mean Stress

Two approximate methods have been used to take into account the presence of the mean component of the fatigue cycle. They have been used to define an equivalent stress which allows using the Manson Coffin's curves.

The first one exploits the nominal results obtained in the elastic range, amplified by the concentration coefficient of deformation earlier discussed. The equivalent stress is simply calculated as half the difference between the maximum and the minimum value of the fatiguing cycle.

The second method is taken from high cycle fatigue where the mean stress has a linear influence on the fatigue life. The following equation is used, where s_s is a safety factor (value = 1.2) introduced to consider the uncertainty on the strength characteristics of the material, obtained by [18], since the thermal treatment of the rotor is not known.

$$\tilde{\sigma}_{ida} = \frac{\sigma_{ida}}{1 - \frac{\sigma_{idm} s_s}{\sigma_r}}$$

In both cases, the amplitude of strain is obtained as

$$\Delta\varepsilon = \frac{2 \cdot \tilde{\sigma}_{ida}}{E}$$

Since the approximate estimates are affected by uncertainty, between the two obtained results it has been always used the more conservative one.

4.4 Low Cycle Fatigue Curves

For appraising the maximum number of allowable cycles, related to a given deformation amplitude , the Manson Coffin's curves in the Figure 5 have been used [5].

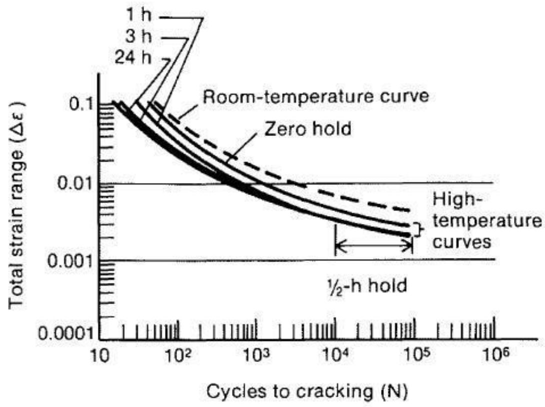

Figure 5. Low Cycle Fatigue Curves for Cr-Mo-V Rotor Steels at Approximately 540°C [5]

The curves also consider the effects of hold time, i.e. the possible residual plastic strain remaining after intense maneuvers.

For the sake of safety, only the curves referred to the maximum temperature reached during the maneuvers have been considered. Such approximation is conservative, since the maximum strain is not necessarily reached at the maximum temperature.

In rotors subjected to thermal stress, hold times last about 3 hours (i.e. the time of the thermal transient); various authors [8][18][19][20], however, suggest to use the curves with 24 hours "hold time" only for the high strain region (number of cycles less than 500).

This choice has been applied for the entire operating range, acting in a conservative manner.

5 CYCLIC LIFE EXPENDITURE CURVES CALCULATION

5.1 Description

The strain intensity due to thermal stress depends on the magnitude and gradient of the temperature variation, the heat exchange coefficient, the mass of the rotor and its thermal properties, as well as the presence of notches that can concentrate stress/strain.

The CLE curves give an indication on the consumption of life associated to a given operation.

They also indicate the region to be avoided during start-ups, or the conditions which not to operate in.

CLE curves are parametric ones obtained by connecting points and are function of:

- total range of temperature and its gradient
- stress concentration factor
- rotor diameter

For a given value of the diameter and the stress concentration factor, such curves allows to:

- knowing in advance the reduction of turbine's life connected to a given maneuver or gradient of temperature change
- making choices on the reduction of warm up and load times, and then on the increase of temperature gradients that the rotor will have to endure
- choosing whether to sacrifice some of the useful life of the rotor to gain greater operating flexibility

5.2 Realization

Depending on the type of start-up or performed maneuver, a temperature gradient (linear loading ramp) that simulates the one detected by the starting probes is imposed to the RSE. The algorithm ensures then to calculate the temperature on the axis and the mean temperature, thus allowing the evaluation of the stress components due to temperature, pressure and rotation.

The equivalent stress is then calculated through the Von Mises criterion, using an appropriate value of K_ε (which is greater than K_t for stresses inside the plastic range while is equal to K_t in the elastic range, as said above).

Amplitude of strain is then calculated with the methods previously cited, in order to use Manson-Coffin's curves for low cycle fatigue.

Once the number N of life cycles is obtained, its inverse value is multiplied by 100 thus defining the specific percentage life consumption for each maneuver. It can be represented by a point in a graph having the temperature change on the abscissa, the temperature gradient on the ordinate, fixing the stress concentration factor and the rotor's diameter.

Repeating the process, changing each time the total temperature variation and the slope of the loading ramp, different values of N associated with the maneuvers are obtained. They are reported in our graph to obtain a set of iso-consumption life curves. The CLE of the High pressure turbine sections (HP) are shown in Figure 9.

For example, it is possible to observe that life consumption equal to 0.01% may be caused by a temperature change of 100 °C performed at 250 °C/h. In other words, the rotor might tolerate up to 10000 cycles before cracks nucleation.

While increasing the temperature difference of the rotor, to keep constant the life consumption it is necessary to reduce the temperature gradient.

If this action is not made, the curve moves upward with a consequent increase of percentage life consumption.

These CLE curves can be looked up to predict the effects that a certain type of maneuver may cause to the rotor, allowing to choose in advance the best way to operate the unit start up.

To show how a number of cycles N has been obtained and then the identification of a point in the graphic $\Delta T - \Delta T/\Delta t$, a warm start up operation performed in Tirreno Power plant is shown (Table 3). It's characterized by an initial rotor temperature of 345 °C, reached 40 hours after the turbine stop (for example, during a weekend).

The data shown in the graphs below (Figures 6, 7, 8) have been directly recorded from the combined cycle's control unit and concern only the HP turbine.

The first graph (Figure 6) shows the trend of the powers generated by the gas turbine and by the steam turbine. It shows also the trend of the temperatures calculated by RSE.

In the second graph (Figure 7), the nominal stresses on the outer surface of the rotor have been calculated using the discussed approximate methods. They are deprived of the contribution of the stress concentration factor to highlight the stresses due to the thermal transient.

The stress components are expressed in the three principal directions, to show their mutual intensity and conserve their sign, in order to understand whether the stress is tensile or compressive.

Finally, in the third graph (Figure 8), it's possible to see the assessment of the equivalent stresses through the use of the Von Mises yield criterion, on the outer surface and on the axis of the rotor.

Observing the shut down it's possible to note that the beginning of the cooling occurs slowly, thus generating thermal stress not exceeding 40 MPa.

This allows stating that even considering as fatiguing cycle the stresses caused by start up and shut down, a fully-reversed loading condition is not got. Therefore it is possible to consider a zero-max-zero profile loading condition. Evaluating of the maximum stresses during start up and applying the approximate method, the calculation of the fatigue life has been performed for this maneuver.

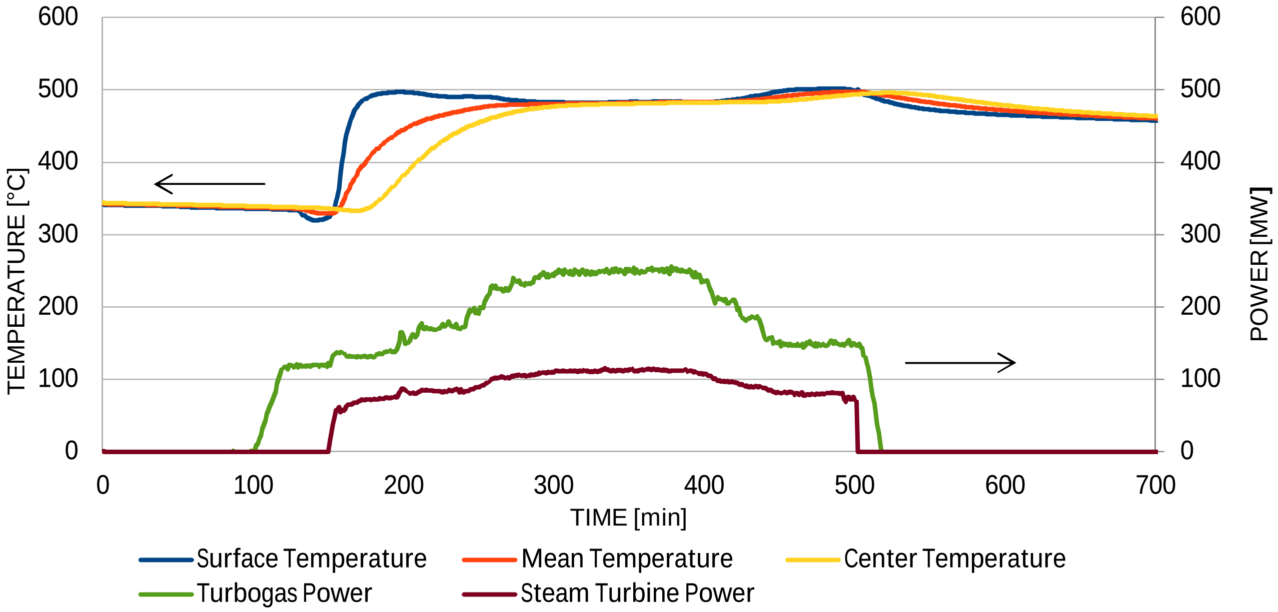

Figure 6. Temperatures and Powers

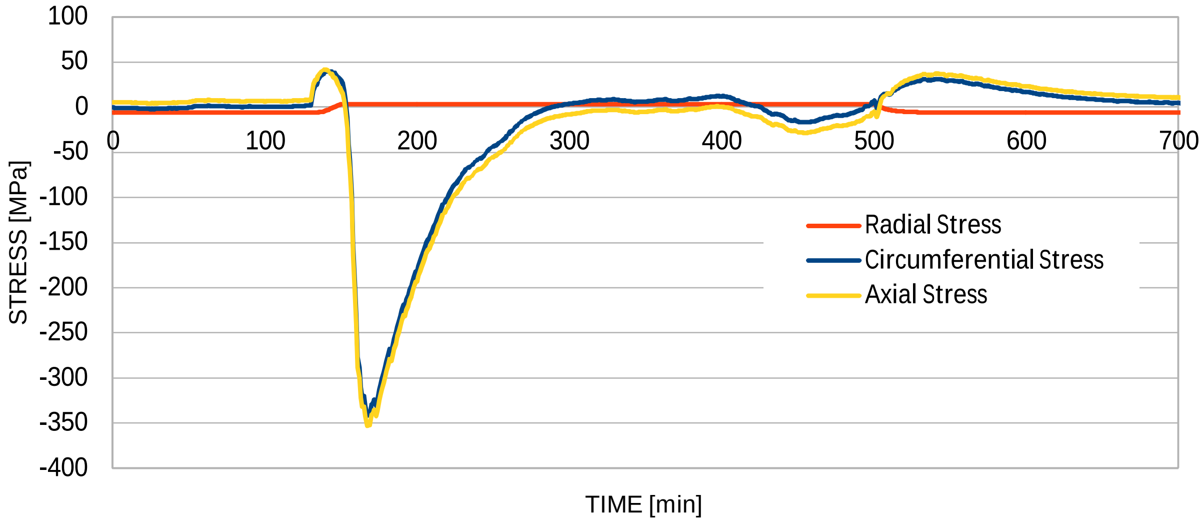

Figure 7. Surface - Radial, Circumferential and Axial Stresses

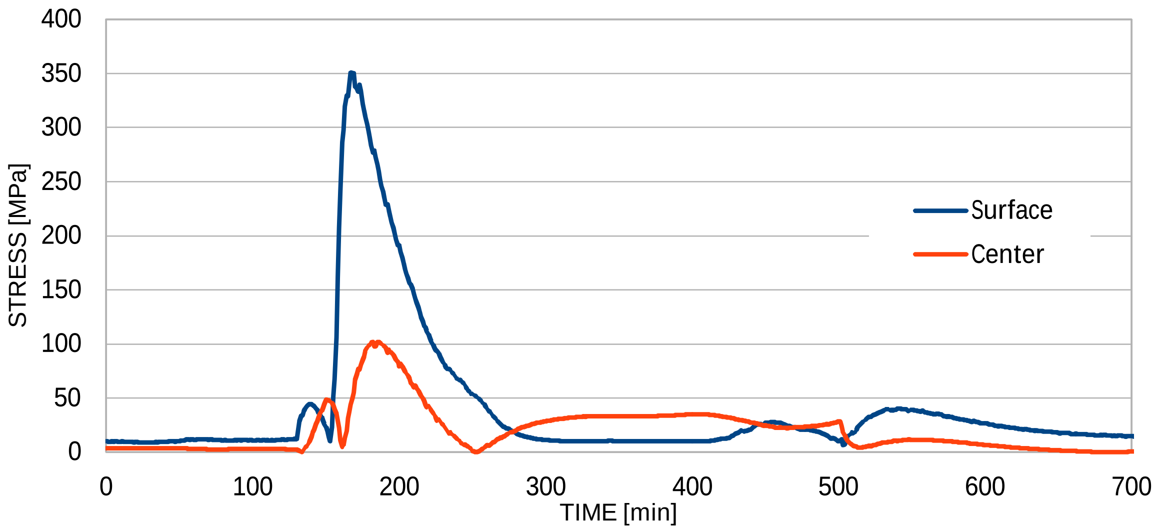

Figure 8. Von Mises Equivalent Stress

Table 3. Warm Start Up

Start Up	Downtime [h]	Initial Rotor Temperature [°C]	Mean Gradient [°C/h]	Temperature Variation [°C]	Maximum Number of Cycles	Specific Life Consumption [%]
Warm	40	345	255	170	3300	0.03

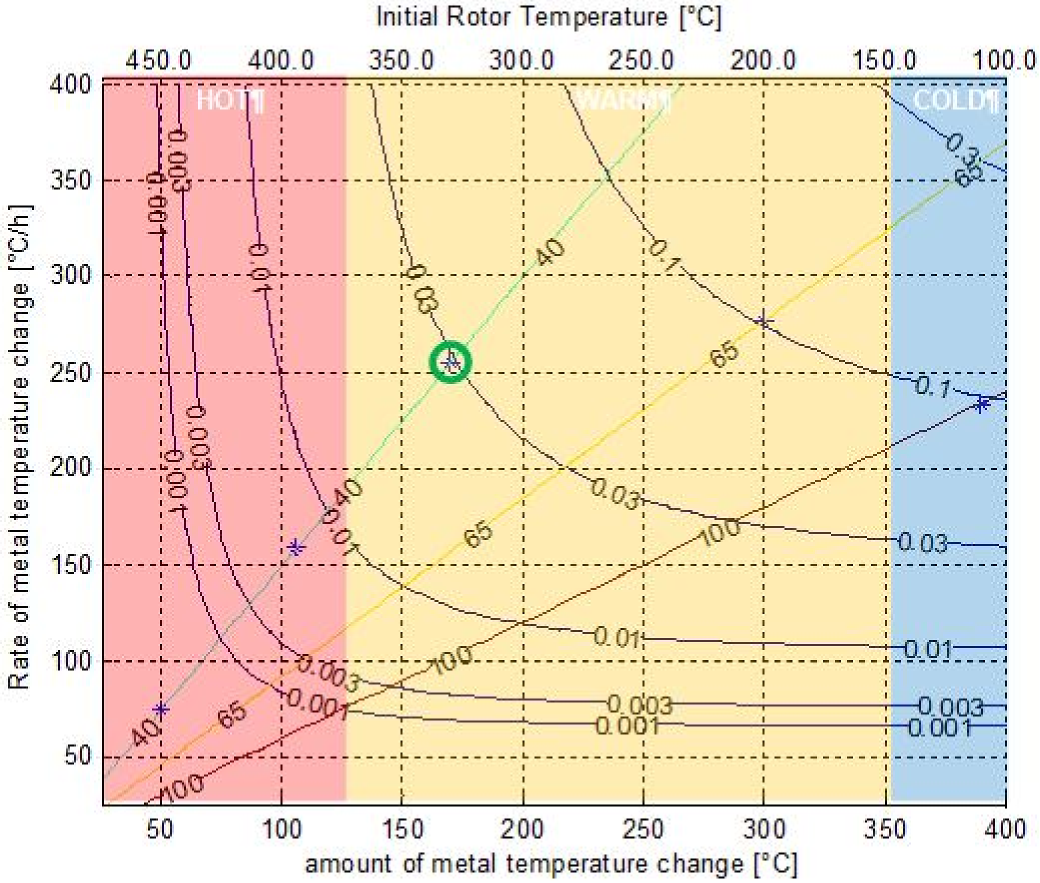

Figure 9. Cyclic Life Expenditure curves for the High Pressure part of the Rotor (blue curves); experimental start up data (blue stars): result about the Warm Start Up of Table 3 (green circle); minutes to full temperature (colored straight lines).

5.3 Yearly Life Consumption Estimation

Moreover, experimental data for several startups were used to evaluate the actual life expenditure over one operating year of the Combined Cycle. The generalized information were combined through the Miner-Robinson rule [5] taking into account also the creep effect due to high temperature normal operation (creep-rupture life under the nominal operating conditions is 300,000 h [5]). The Table 4 shows the different start ups' contributions to the life expenditure of the rotor in one year of average cycling operations: as effect of the low cycle fatigue a yearly consumption of 5.05% was calculated (as sum of the terms in the last column). The creep effect, considering an average of 3000 OH per year, adds 1% of yearly life expenditure (3000/300,000). The two cited contributions are simply summed up (6.05%) and the inverse result represents the number of years until defects nucleation (17 years).

Table 4. Average yearly start-ups and related life consumption

Start up	Standstill Time [h]	Initial Rotor Temp [°C]	Yearly Start up Number (n_i)	Life Expenditure/ Startup ($1/N_i$) [%]	Yearly Life Expenditure ($n_i \cdot 1/N_i$) [%]
Hot	<16	T >370	53	0.0050	0.265
Warm	16< t <108	370> T >150	123	0.030	3.69
Cold	t >108	T <150	11	0.100	1.1

6 CONCLUSION

The activity led to quantify the effects of start-ups and of the load variations on the steam turbine rotor of the Tirreno Power's combined cycle plant in Vado Ligure, Italy, in terms of life expenditure.

The impact of combined cycle startup on the steam turbine life was assessed on the basis of the actual operational data and a simple but rigorous approach, evaluating the Cycle Life Expenditure (CLE) Curves. The data were used to estimate the yearly average life expenditure. The algorithm was then implemented on the Power Plant Distributed Control System (DCS).

The construction of the CLE curves was the goal of the work allowing to predict the consumption of life associated with different operation modes.

Further development will be the exploitation of the designed methodology for optimizing the maneuvers that generate thermal transients.

Knowing in advance the damage caused to the rotor of the turbine, allows to manage the speed of start-ups with greater awareness. As mentioned, this grants the flexibility required by today's energy market: the goal is to shorten the time required for the operations, even at the expense of a higher -but expected- damage with the benefit of improving the competitiveness and efficiency in service delivery.

However, the efficiency might be improved reducing the power consumption and the time required for starting-up, for equal spent life. The better understanding of the effects of the maneuvers will also reduce the risk of damage or failure that may suddenly stop production, planning the maintenance in relation to the calculated cumulative damage, as well.

The work done can be improved with the use of a finite element numerical model for both the calculation of the thermal loads distribution and of the stresses in the rotor.
This would make it possible to take into account the specific geometry of the insertion region of the blades, giving more accurate results on the local stresses and deformation in those points. In addition, a more detailed characterization of the fatigue life to the particular study conditions will be the decisive step to achieve trustworthy final results.

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8 ACRONYMS

CCPP	Combined Cycle Power Plant
CLE	Cyclic Life Expenditure
DCS	Distributed Control System
EOH	Equivalent Operating Hours
GT/ST	Gas Turbine / Steam Turbine
HP/IP	High Pressure / Intermediate Pressure
OEM	Original Equipment Manufacturer
OH	Operating Hours
RSE	Rotor Stress Evaluator