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DIPLOMA THESIS

EOH - Equivalent operating hours of steam turbines

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1. Description of turbine parts and their influence on the EOH.
2. EOH calculation methodology.
3. Case study of application of the EOH.
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V Plzni dne 2. června 2017. ....

Declaration of authorship

By this I submit the diploma thesis written in the end of my studies at Faculty of Mechanical Engineering of University of West Bohemia to be revisited and defended.
I proclaim that I have made this thesis on my own, with the use of literature and other sources listed inside the thesis.

In Pilsen, 2nd June 2017. ....

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STRUČNÝ POPIS (MAX 10 ŘÁDEK) **ZAMĚŘENÍ, TÉMA, CÍL POZNATKY A PŘÍNOSY**	Diplomová práce je věnována problematice ekvivalentních operačních hodin (EOH), návrhu jejich stanovení a možnému využití zákazníkem v praxi. V práci jsou také popsány pomocné moduly a systémy, které jsou nějakým způsobem spojeny se stanovením EOH a kontrolou životnosti parní turbíny. Dále je v práci uveden popis jednotlivých součástí parní turbíny, které se svým opotřebením nejvyšší měrou podílejí na spotřebě EOH, spolu s uvedením jejich nejčastějších poruch.
KLÍČOVÁ SLOVA **ZPRAVIDLA JEDNOSLOVNÉ POJMY, KTERÉ VYSTIHUJÍ PODSTATU PRÁCE**	EOH, servis, údržba, životnost turbíny, ekonomika provozu, parní turbína, nízkocyklová únava, operační hodiny, TCS, plánování údržby

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BRIEF DESCRIPTION **TOPIC, GOAL, RESULTS AND CONTRIBUTIONS**	This diploma thesis is focused on the issue of equivalent operating hours (EOH), proposal of its determination and possible use for client practice. This work describes supporting modules and systems, which are in some way determining EOH and control of steam turbine life-time. Furthermore, a description of individual parts of a steam turbine is a part of this work, whose wear in its maximum span influences EOH consumption. Most frequent failures of steam turbine components are also mentioned.
KEY WORDS	EOH, service and maintenance, turbine lifetime, economy of oparion, steam turbine, low-cycle fatigue, operating hours, TCS, maintenance planning

Contents

Nomenclature

Quantity	Unit	Name
A	$[-]$	Availability of a turboset
c	$\left[\frac{\mathrm{J}}{\mathrm{kg \cdot K}}\right]$	Specific heat capacity
c_j	[hours]	Equivalent operating hours consumption
dN	$\left[\frac{\mathrm{MW}}{\mathrm{min}}\right]$	Average loading rate
E	[Pa]	Young's modulus
EOH	[hours]	Number of equivalent operating hours
EOH_Y	$\left[\frac{\mathrm{hours}}{\mathrm{year}}\right]$	Number of equivalent operating hours per one year
F	$\left[\frac{\mathrm{W}}{\mathrm{m^2}}\right]$	Temperature flux
l_j	$[-]$	Lifetime consumption factor
k_c	$[-]$	Stress concentration factor
K	$\left[\frac{\mathrm{W}}{\mathrm{mK}}\right]$	Thermal conductivity
N_j	$[-]$	Number of start-ups
$N_{j,red}$	$[-]$	Reduced number of start-ups
N_p	[MW]	Initial load after synchronization
OH	[hours]	Number of operating hours
OH_Y	[hours]	Number of operating hours per one year
PAO	[MW]	Presently available power output
PH	[hours]	Number of period hours
PO	[MW]	Present power output
r	[m]	Radial coordinate
R	[m]	Rotor diameter
t	[s]	Time
$t_{start-up}$	[min]	Start-up time
t_1	[min]	Waiting time at a warming speed
T	[°C]	Temperature
$\triangle T$	[°C]	Temperature difference
$\triangle T_{max}$	[°C]	Maximal temperature difference
$\overline{T}$	[°C]	Rotor mean integral temperature
T_0	[°C]	Initial temperature
T_{min}	[°C]	Minimal start-up temperature temperature
T_S	[°C]	Rotor surface temperature
$\vec{u}$	[m]	Displacement
UOH	[hours]	Number of unplanned forced outage hours
Y	[years]	Number of years
∇	$[-]$	Laplace operator
α	$\left[\mathrm{K}^{-1}\right]$	Coefficient of thermal expansion
κ	$\left[\frac{\mathrm{m^2}}{\mathrm{s}}\right]$	Thermal diffusivity
μ	[Pa]	Lamé constant
ν	$[-]$	Poisson's ratio
ξ	[m]	Radial coordinate
ρ	$\left[\frac{\mathrm{kg}}{\mathrm{m^3}}\right]$	Density

σ_z	[Pa]	Axial stress
σ_φ	[Pa]	Hoop stress
Φ	$[\mathrm{Pa}\cdot\mathrm{m}]$	Thermoelastic displacement potential
χ_j	$[-]$	Lifetime allocation factor

Abbreviations

CHP	Combined heat and power
CSP	Concentrated solar thermal power
DCS	Distributed control system
EOH	Equivalent operating hours
EU	European union
FATT	Fracture appearance transition temperature
GO	General overhaul
HP	High pressure
IP	Intermediate pressure
LCF	Low cycle fatigue
LEC	Lifetime expenditure counter
LP	Low pressure
LSB	Last stage blade
LTSA	Long term service agreement
NDT	Non-destructive testing
OEM	Original equipment manufacturer
PAC	Preliminary acceptance certificate
PGIM	Power generation information manager
TCS	Turbine control system
TSM	Thermal stress module
VPN	Virtual private network

Introduction

In today's world the demand for electrical energy is still higher and higher. It is given by its increasing consumption in many different fields and by the fact that there still is not a possibility to store electrical energy effectively. There is also a need and an effort to electrify almost every settled place in the world. This is connected with building of new power plants especially combined cycle power plants and power plants which use renewable sources. From an environmental and economical point of view all these power plants should be operated with the highest possible efficiency and reliability. Both these requirements are important and advantageous for increasing of quality of environment as well as increasing profit for the contractor of electrical energy.

If we look at the reliability of a power plant one of the most crucial parts is the steam turbine whose operational availability is a key parameter for production of electric power. The availability of a turboset is closely linked with the operation and the maintenance plan. There are often fixed numbers of a steam turbine's operating hours which require specific types of maintenance. Unfortunately this type of maintenance planning is in no way bound to manner of its operation. This can cause early execution of agreed inspection, in a better case, or a steam turbine failure, in a worse case scenario. Both these cases are not good considering operation costs.

The approach of equivalent operating hours (EOH) should ensure the connection between the philosophy of turbine operation and its maintenance planning. The EOH is designed individually for every steam turbine because the design is based on a lifetime consumption of critical steam turbine components which differs for various steam turbine applications. For example for concentrated solar thermal power (CSP) plants the critical component is high pressure (HP) rotor which is enormously strained by daily steam turbine start-ups and shut-downs. The last stage blade (LSB) is a critical component for combined heat and power (CHP) application which may act as peaking power plants. In that case the LSB is threatened by frequent changes of operation, e.g. erosion at spray or operation in ventilation at partial loads. In contrast to CSP and CHP units there are output changes which have far greater impact in EOH determination by fossil units. These units participate more on the ancillary services of an electrical grid. Nowadays there are many different types of ancillary services, not only primary and secondary frequency regulations, which require smaller or larger output changes. All above mentioned aspects have to be dealt with for a proper EOH estimation. As has been said before, the EOH is used for an improvement of maintenance planning and also serve the steam turbine operator as a tool for quite simple verification of a steam turbine's lifetime consumption.

The reader will be acquainted with a historical evolution of the EOH in the first part of this thesis. Then steam turbine auxiliary modules and systems, which are somehow coupled with the EOH determination, will be reported. The third part will be focused on a description of turbine parts which mostly influence the EOH. There will also be shown its percentage distribution. After that, the theory of the EOH calculation will be approximated and the method of calculation will be proposed. The fifth part will be devoted to a case study of the EOH application with a comparison of three variants of lifetime consumption. There is also a clarification of the main maintenance inspections in the next chapter. Finally, there are recommendations and remarks of usage describing EOH calculation methodology for steam turbines in practice.

1 Equivalent operating hours

The application of EOH for steam turbines is known from past several years. This approach is based on operational experience of turbines. That is why it was firstly used by gas turbines. Gas turbines are more or less standardized therefore there were a lot of records and information from their operation. These collected records were deeply analyzed and a database of failures and critical components was created. Such information is very valuable for each OEM supplier with regards to the feedback to their machines. Such feedback may then be used for the determination of EOH to foresee future potential problems.

The required operational hours of the steam turbine are converted into EOH. The computational process is described in a separate chapter in this thesis. The advantage of using the equivalent operating hours approach is that the steam turbine operator can track the steam turbine lifetime in real time. Because if the steam turbine's lifetime is determined on 8000 operating hours for 25 years it does not mean a fixed service life. Thanks to EOH a steam turbine operator can easily predict if the lifetime of his steam turbine will be shortened, extended or abided. Actually the EOH introduces a sort of simplification of the lifetime expenditure counter (LEC) which will be described in more detail in next parts of this thesis.

As it was said the estimation of the number of the EOH is built on the operational records and a steam turbine manufacturer's experience. This is one of the reasons why a vast majority of new steam turbines is equipped by a very advanced remote monitoring system. The main advantage of these systems is that all operational data are sent in real time to the headquarters of the steam turbine manufacturer where they are evaluated and stored.

With reflection of [1], [5], [7], [11], for a better understanding and orientation in a historical evolution of the EOH.

1.1 Historical background

From the historical point of view of EOH evolution the operation of steam turbines could be divided into three main phases.

The first one is characterized by no coherent cooperation between maintenance departments and steam turbine operators. This phase could be dated from the 50's to 90's of the last century. The cooperation between service departments and steam turbine operators was carried out by an ad hoc manner. There were no maintenance plans or planned inspectional operations which could prevent long time overhauls. Sudden steam turbine trips may have been one of the main causes of shortening service lifetime of steam turbines. This also meant a lower operational reliability of whole turbosets.

At the beginning of the 21st century the second phase started mainly by gas turbines due to their extensive standardization. The beginning of a closer cooperation between steam turbine operators and maintenance departments has led to maintenance planning. Maintenance plan was based on data from LEC which is described in the next part herebelow. The steam turbine service lifetime for LEC was, and still is counted from the critical operating hours of the steam turbine HP rotor. The operational data are collected by the remote monitoring system. The operational data collection is very important for the application of EOH method. This method is based on the sophisticated estimation of the lifetime of individual steam turbine components which work together and influence each other.

The last phase just began not long ago by some steam turbine manufacturers and in my opinion it should begin in a few next years for the rest of steam turbine manufacturers, if they want to be able to compete in bids and sign new contracts. The cooperation between service departments and steam turbine operators should be fully connected. It means there will be a detailed service and inspectional plan based on steam turbine manufacturer's operational experience. These experience were gained especially during the second phase of EOH evolution. On the basis of real time monitoring modules and systems a steam turbine operator should be able to manage the lifetime consumption of the steam turbine. Hand in hand with this he can significantly influence the maintenance plan. The time between particular overhauls can be shortened or extended.

2 Steam turbine auxiliary modules and systems

Every steam turbine is a very complicated power device whose correct service would not be possible without series of auxiliary systems. These turbine auxiliaries are e.g. lube systems, overspeed trip devices, gland seals system, etc. but these systems do not directly influence the service lifetime of the steam turbine. Auxiliary modules and systems which are important and have an impact to steam turbine service lifetime are briefly described bellow. The explanation is founded on [13].

2.1 Thermal stress module

Every steam turbine of a significant steam turbine manufacturer is featured by its own turbine control system (TCS). As a part of the TCS there is an online thermal stress evaluation the thermal stress module (TSM). Current thermal stress is provided by the TSM determined in critical points of the steam turbine.

Critical points are locations with the highest wall thickness because there is the highest thermal difference, thus the highest thermal stress is occurring in these points. Therefore, the component which is the most prone to this kind of damage is the rotor. As can be seen in the Fig. 1 the radii at the inlets into the internal seal as well as the first stage labyrinth seal are one of those critical locations. The points of maximum stress concentration and the largest fluctuation of steam temperature are right there. For these reasons the turbine rotor is assessed as a determinative component which shall be further described hereunder for the low cycle mechanism.

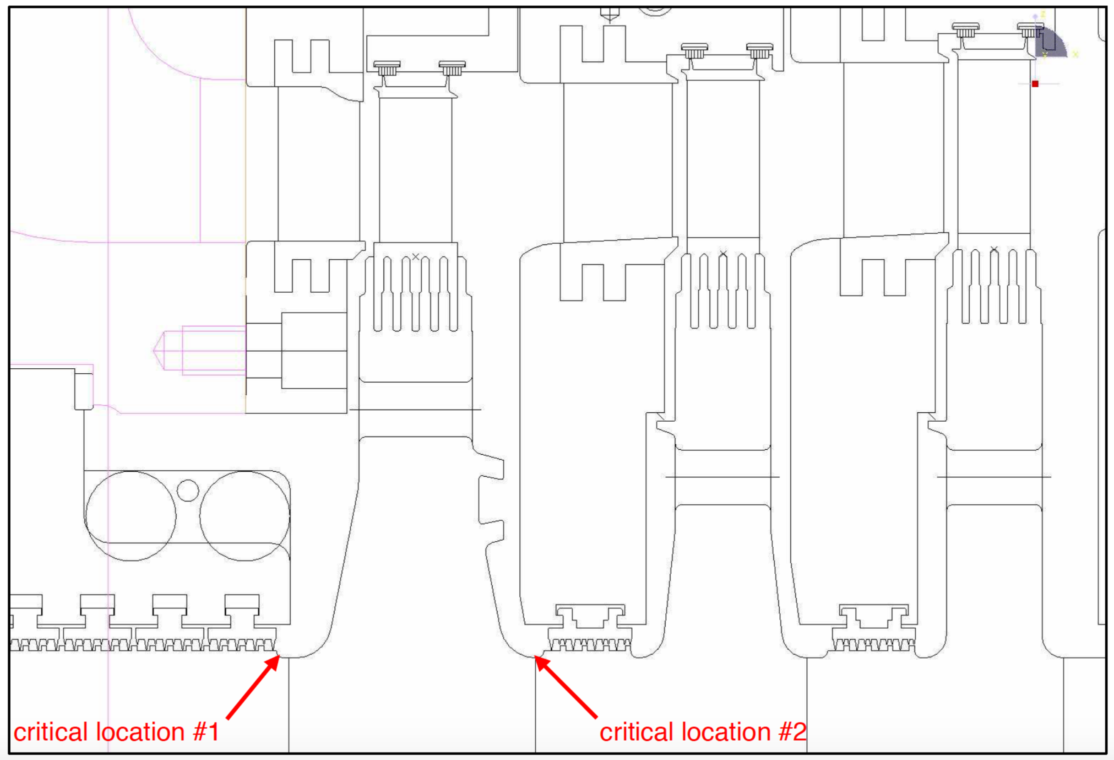

Figure 1: Critical points of turbine rotor

Thermal stress monitoring module contains two modules of thermal stress evaluation — for HP and for IP turbine rotors. The measured input values are automatically verified. The simulating model is used for evaluation, providing the difference between surface temperature and mean integral temperature of turbine rotor which fully represents the thermal stress.

TSM constitutes two limit levels — allowable thermal stress limits (called green limits — H1) and maximum thermal stress limits (called red limits — H3). These two levels are based on mean integral temperature of the rotor and responds to the actual operational stage. If the owner of the steam turbine wants to reach required turbine lifetime it is necessary to operate the steam turbine in green limits. Turbine trip occurs when the maximum thermal stress limits are exceeded.

The progression of the thermal stress in time together with its limit values is fully available for the operator. If the allowed thermal stress limits are overstepped, information about unplanned decreasing of residual turbine service life is given to the operator. This overstepping is also registered by the TCS. Limits of turbine operation are controlled by the TCS not only exceeding of their limit values. The turbine fully utilizes the thermal stress limit value is controlled by the TCS from the start-up to load changes as well. These functions can supply maximal operational flexibility and concurrently secure the planed lifetime expenditure.

2.2 Lifetime expenditure counter

The best effort of a steam turbine operator should be to maximize its service life and operate it efficiently. Earlier there were analyses to determine the residual life of turbine components towards projected lifetime which were established on recording of the past operation. Nevertheless this kind of evaluation of residual life wasn't that effective. A much more effective and useful tool for management of component residual life is a system which can evaluate residual life in real time and it is based on continuous data collection.

The LEC is a specialized equipment for online monitoring of low cycle fatigue (LCF) damage. It is another part or module of the TCS which has its mechanism based on outputs from the TSM. As the TSM all time evaluates the current value of thermal stress in critical points, LEC all time provides information about the rest of turbine's service life based on the whole turbine history of operation from the first start-up to the current condition.

The principal input variables requisite for the LEC to assess LCF are the thermal stress and the mean integral temperature of rotor. LCF is perceived as the dominant way of material damage. The algorithm of LCF evaluation is an online version of rain-flow method which is based on recording of thermal stress extremes. From these extremes thermal stress cycles are evaluated and each cycle causes certain damage with respect to responding material temperature. Damages caused by individual thermal cycles are summarized according to Corten-Dolan damage accumulation model. Since the fatigue damage caused by each cycle of given amplitude is calculated in the design stage and entered into the life expenditure counter as a parameter, the residual life of the rotor can be determined and displayed to the plant's operating personnel at any time. It is then possible to evaluate e.g. life expenditure caused by one start-up / shut-down cycle, or the rate of lifetime consumption throughout one year.

2.3 Tip-timing measurement system

Tip timing is a standalone diagnostic system, where several sensors are placed along the casing and detect the time of arrival of blades. The diagram of measurement and needed input data to measurement system can be seen in the Fig. 2. Blade vibrations are checked by tip-timing measurement systems usually on the last stage blade of the turbine. The last stage blade is the longest blade in the whole turbine therefore its vibrations are the biggest. Identification of these vibrations is prime because of their dependence to back pressure in the condenser.

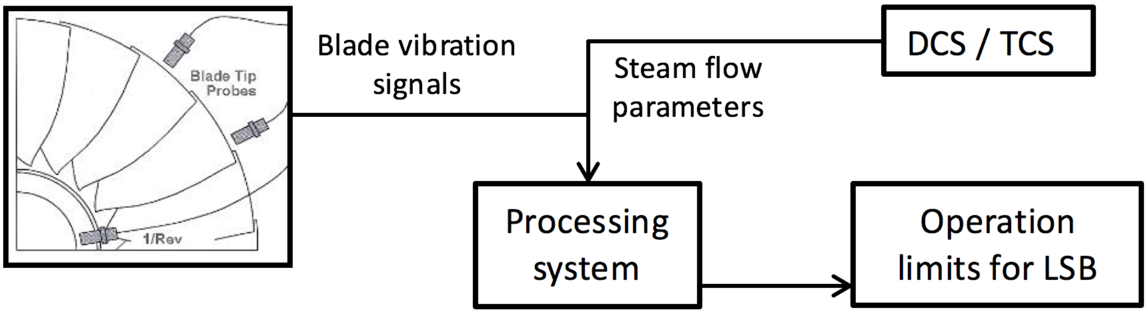

Figure 2: Diagram of tip - timing measurement system

The system is used for real time analysis of blade vibrations, long-time monitoring of blade properties (static lean, micro-crack detection) and monitoring of shortened life span due to temporary high amplitude vibrations. The system detects times of arrival of each blade. In comparison to the others it is possible to identify the properties of the vibration (amplitude, frequency etc.). It is also valuable to check blade vibrations during start-up and shut-down. In these cases, several synchronous resonance frequencies need to be crossed.

2.4 Lifetime consumption

As said above, the lifetime of the steam turbine is evaluated and monitored by the LEC throughout turbine operation. The full lifetime is understood as 100% of the LEC or as a planned number of operating hours. Counting of the life time expenditure from initial 100% to 0% at the moment of life consumption is transformed to the initial planned operating hours and gradual subtraction of EOH where the real operation of turbine is respected.

The residual lifetime is gauged separately for each critical point. It means e.g. for a double casing steam turbine with one high pressure part and one combined intermediate pressure / low pressure part there are two critical points: high pressure turbine rotor and intermediate pressure turbine rotor. Each of these critical parts are fitted out with specialized evaluating equipment (e.g. TSM, LEC) because each of them can become decisive under different operational conditions. The record from LEC is shown in the Fig. 3.

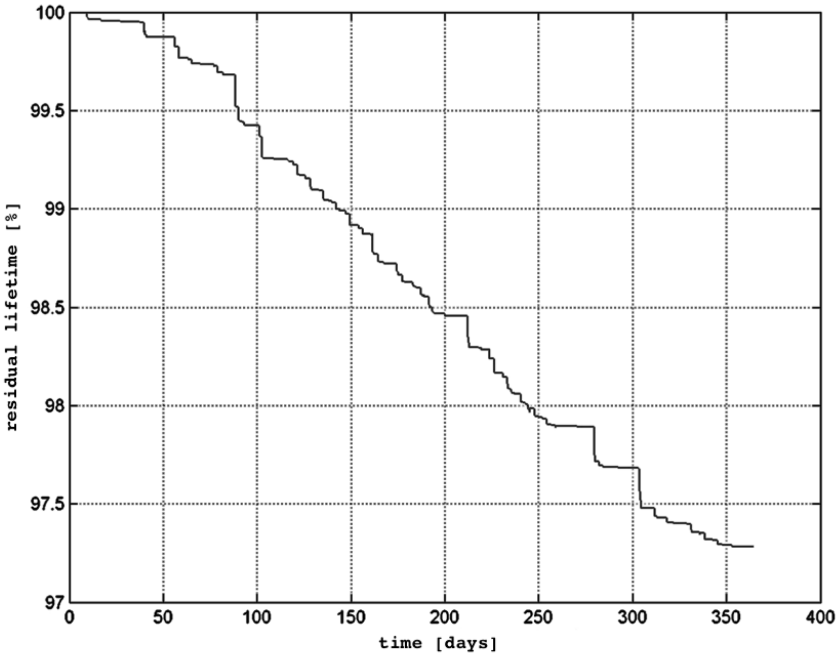

Figure 3: Lifetime consumption

2.5 Remote monitoring system

Remote monitoring system represents a certain kind of diagnostic centre where operational data are hoarded up, evaluated and stored. The main purpose and goal of this centre is an online and maintenance support to costumer during and after the warranty (it is specified in every implemented project by customer). The operational data collection is very important for the right calculation of EOH because this calculation or design is based on an operational experience of individual components and their wear in time.

For receiving and storage of monitored turbine operational data, the power generation information manager (PGIM) archiving station including software is supplied to provide data collection from relevant control systems through communication drivers. The archiving station is equipped with a monitor, keyboard and mouse and this assembly is located inside a separate cabinet.

Operational data is collected by PGIM server on-site from relevant control systems eventually from remote storage of data, and provided to the special diagnostic centre, located by a steam turbine manufacturer headquarters, by remote connection via Internet, VPN Internet access.

PGIM includes a process data server to store:

- signal descriptions,
- current process data (real-time data),
- historical process data (long-term data),
- messages (events).

The following information is stored for all process data:

- the time of acquisition,
- the physical value,
- detailed status information (e.g. measured value disturbed).

How the remote monitoring/diagnostic center can be realised is indicated in the Fig. 4.

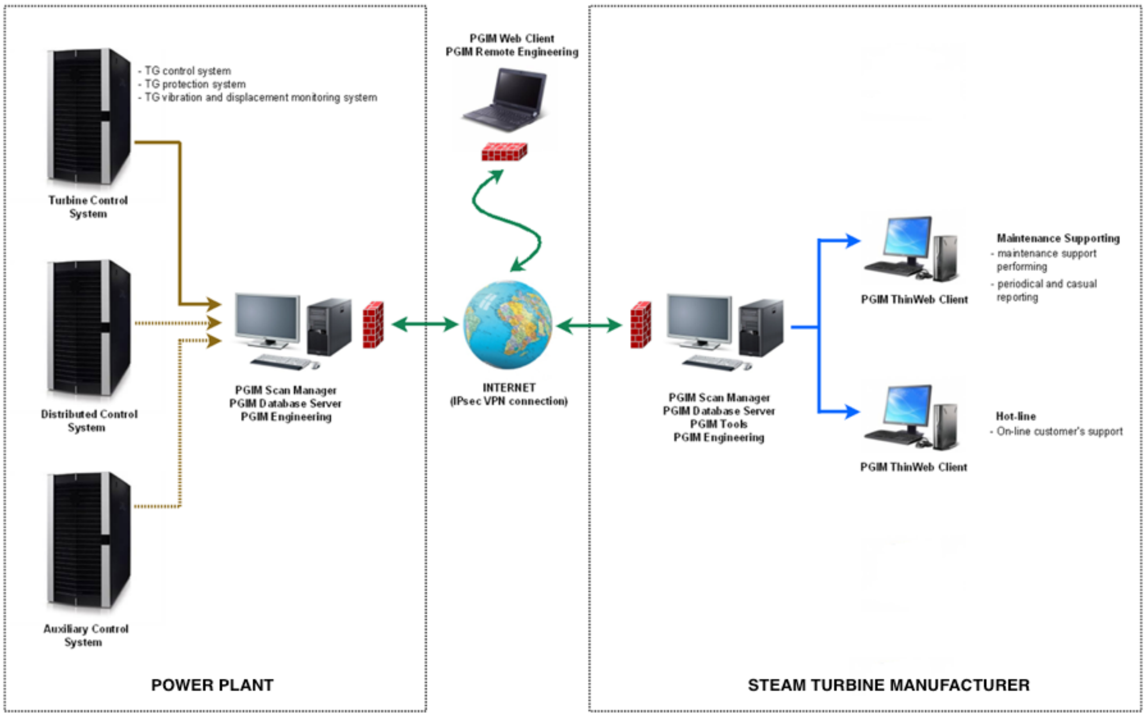

Figure 4: Remote monitoring/diagnostic center - block diagram

3 Description of turbine parts

Following pages will be dedicated to a brief description of individual turbine components which mostly influence the lifetime of the steam turbine. It also means their wear mostly consumes the EOH. The description is mainly made according to [3], [6], [8], [10], [12].

The distribution of turbine components, which are the most risky in view of turbine failure, is demonstrated in the Fig. 5. Chart details are proved in [7].

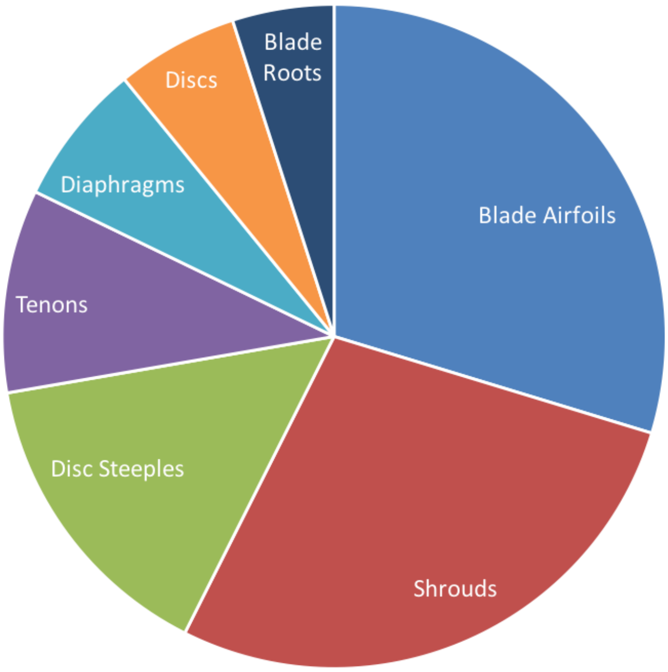

Figure 5: The most risky turbine components

3.1 HP and IP rotor

As said above the HP and IP rotor are the most critical components due to the possible LCF damage. In case of combined HP and IP part of turbine there is one common HP/IP rotor, in case of separated HP and IP parts of turbine there are two rotors which are coupled together. This is basically given if we are speaking about non-reheat or reheat steam turbine.

3.1.1 Auxiliary equipment associated with rotor

The rotor is typically supported by two journal bearings which help to posture the rotor in the correct radial position. The journal bearings also give support that withstands the reaction from shaft rotation. Typical position of thrust bearings is between HP and IP or first LP sections. Their function is an interception of axial thrust and holding the position of the turbine rotor axially relative to the stationary parts. Furthermore rotors are equipped with a turning gear which provides them turning during shut-downs and start-ups. Turning gear ensures the protection from thermal bending or wrapping of rotor caused by uneven cooling or warming.

3.1.2 Typical rotor design

There are schematically shown three general types of rotor construction in the Fig. 6. The most ordinary constructions for HP and IP turbine rotors are solid or welded construction. Build-up design with an integrally forged shaft onto which discs are shrunk and keyed is mostly used for LP rotors due to their large size. An advantage of using a welded construction is the combination of two or more different materials. This combination can provide optimization of both mechanical features (e.g. heat resistance, transition temperature) and cost (saving of more expensive material).

3.1.3 Rotor damage mechanisms

Among the most frequent damage mechanism belong LCF (thermal and mechanical), creep, creep-fatigue interactions, fatigue from rotating weight ("self-bending"), and embrittlement. The main locations of interest in the HP and IP rotors are the accumulation of damage at the bore and in the other locations of stress concentration resulting from cyclic loading and elevated temperature operation. Bores, radii, keyways, locking slots, heat and seal grooves, ventilation holes, and blade attachments are the common stress concentrators. Maximum creep resistance was given to earlier developed forged rotors. Production was focused on heat treating of the rotor. The material was austenitized at 1010 °C. This high creep strength was accompanied by poor creep ductility and poor toughness, since as the creep-rupture strength increases, so does the fracture appearance transition temperature (FATT). Through changes in steel-making practices, heat treatment, increased steel purity, and alloy content toughness was increased more recently. Improving of creep ductility has been prioritized against maximizing creep-rupture strength. However, for many operators there is still a problem with creep cracking at the blade attachment.

Over the years the scope of operational conditions of rotors has changed. An unceasing turbine operation at temperatures up to 565 °C, where the creep damage was the primary damage mechanism, was shifted into cyclic operational conditions. It means a power load cycling and full start-up/shut-down cycles, this kind of service is typical e.g. for steam turbines for solar applications. With this change other potential damage mechanisms like LCF and creep-fatigue were initiated. Start-up and shut-down induce thermal stresses that become a maximum at temperatures substantially lower than those that occur at steady state. Fracture toughness becomes the dominant property in the resistance of the steel to brittle fracture.

Bending of a rotor can be caused by several reasons. The most frequent reason is presence of water in a flow section of a steam turbine. Water can induce a local rotor cooling or it can cause an oscillation of a rotor which can lead to the touch between rotor and stator part, this may cause a temperature hike. Intake of water can be caused through a steam extraction or by dysfunction of a drainage. The next reason

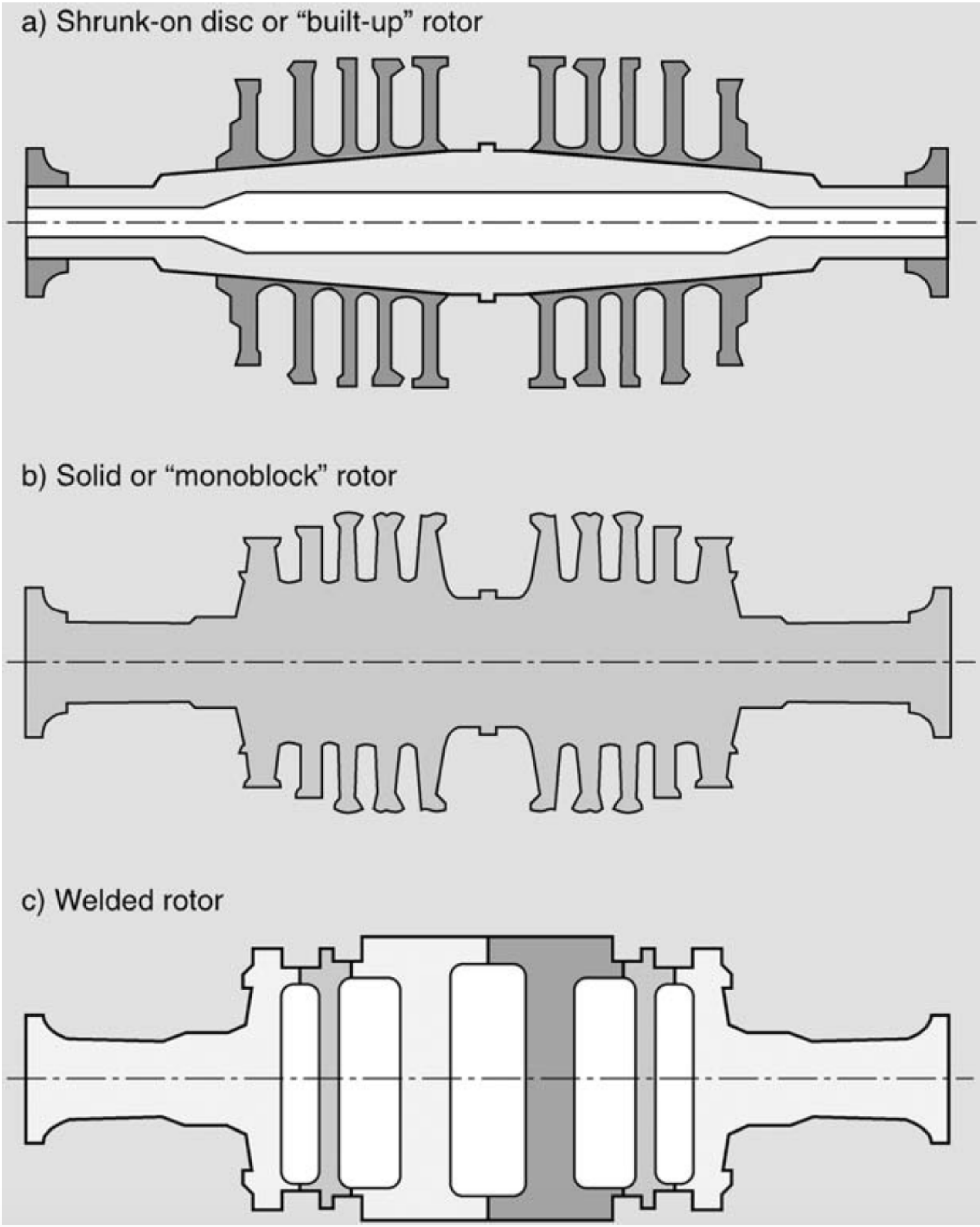

Figure 6: Typical steam turbine rotor constructions [8]

of bending can be too hot steam during turbine loading.

3.2 Turbine seals

Several types of seals are used to reduce leakage losses in single areas of flow path to each other or from inside of casings to the atmosphere. One of the largest reasons of performance reduction in HP turbines is exactly seal leakage. But there are significant losses caused by poor sealing in IP and LP parts of turbines as well. The most common damage of turbine seals is generated by the rotor when its vibration or bending results in the contact between a turbine seal and a rotor.

3.2.1 Interstage seals

They are mostly executed in a labyrinth design. The main aim of interstage seals is to prevent leakage around the rotating and stationary stage. A diaphragm seal is a typically used term in impulse stages against undershroud seals (seals individual stages among each other) and overshroud seals (prevents steam leakage over the rotating blade) that are used in reaction stages.

3.2.2 End seals

Minimization of leakage at the end of each shroud is realised by several section of end seals or packing glands, they are solely produced in the labyrinth design. Front HP packing glands are used to prevent leakage of input steam out of turbine casing. Rear LP packing glands are determined to prevent air leakage into the LP part of turbine and condenser.

3.3 HP and IP turbine blades

A turbine stage is composed of stationary blades (nozzles) and rotating blades (buckets). The enthalpy of input steam is transformed into rotational energy as it flows through a turbine stage. Pressure and thermal energy (potential energy) of the steam is converted into kinetic energy in stationary blades. The steam flow is directed onto rotating blades by stationary blades. Kinetic energy is converted into impulse and reaction forces by rotating blades. The rotation of the turbine rotor is achieved by impulse and reaction forces, caused by pressure drop. Features and structure of rotating and stationery blades are described bellow.

3.3.1 Types of airfoils

Blade airfoils are divided into three classes: constant area airfoils, tapered airfoils, and tapered twisted airfoils. The constant area airfoils is an impulse blade. It is usually applied in short blades in the HP part of steam turbine. Reduction of centrifugal stress on longer blades is provided by using of the tapered airfoil. The tapered twisted airfoil is basically used as a reaction blade. It is applied when both a reduction of centrifugal stress and a change in blade angles, from hub to tip for thermodynamic efficiency, are required.

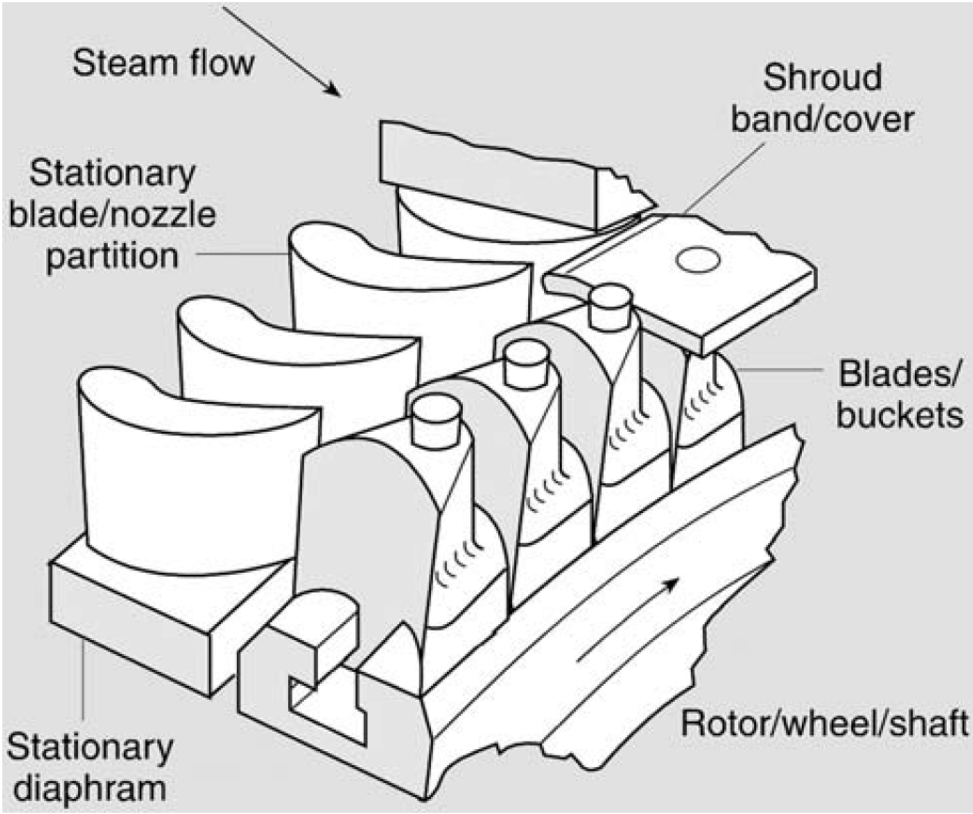

Figure 7: Schematic arrangement of HP turbine stage [8]

3.3.2 Rotating blades — buckets

Because of low volumetric steam flow in the HP part of turbine, the HP blades are shorter and narrower than elsewhere. Basic features of the HP turbine stage are indicated in Fig. 7 for better notion. The airfoil of rotating HP blade is usually straight but still more often leaned and bowed blade airfoils are used due to advantage of use three dimensional aspect of design.

The way of attachment of blades is fully depended on the manufacturer of steam turbine. The choice of type of attachment could be determined by number of factors, e.g. ease of replacement, wake frequency, high dynamic stress, type of inlet steam entry, etc. The most common types of root attachments are:

- fir tree root,
- inverted fir tree root,
- single - T root,
- double - T root,
- straddle - T root,
- pinned root,

- multi finger pinned root.

A special attention is put to design of the first (control) stage because of high dynamic stress. The high dynamic stress is caused by operation with partial arc admission. Other design factors like a choice of leading edge configuration or blade groupings are selected to decrease the production of vibratory stresses. The IP blades are very similar in design to HP blades. Due to higher volumetric flow they are longer and wider, they are also more twisted and most recently bowing and leaning to account for greater radial variation in the flow.

3.3.3 Stationary blades — nozzles

The stationary blades or nozzles can be divided into two design categories:

- wheel and diaphragm construction,
- drum — rotor construction.

A wheel diaphragm construction is typically used for impulse stages. This construction is consisted from nozzles or stationary blades, a ring in which the blades are located in the casing, and a web by which the gap between the shaft and the blade wheel is filled. Diaphragms in the HP and IP parts of turbine are mostly of welded design.

The drum — rotor construction is usually used for reaction stages. In reaction stages, stationary blades or nozzles are manufactured in a manner similar to that for rotating blades with a root attachment and in some cases a sealing shroud. The blades are fixed by the root attachment on a blade carrier. A blade carrier is placed in the outer casing of steam turbine.

Nozzles and diaphragms are bended in the plane perpendicular to the turbine axis by pressure differentials. The biggest value of these pressure differentials is in the HP turbine, even though the shorter blade length limits the bending stresses that develop. In the Fig. 7 the relationship between nozzles and buckets in a turbine stage is shown.

3.3.4 Blades damage

Generally there are two main sources of a blade damage: chemical purity of steam and mechanical impurities in steam.

Chemical purity of steam is important because it could cause salting of flow path section which could cause narrowing of flow canals. Salinization of flow section causes increasing of axial force which leads to change of pressures within the flow section. Another type of damage is settling of sediments on the flow surface of blades. It induces degradation of harshness of blade surface which leads to decreasing of its efficiency. One of the next possible damage is denting corrosion which is caused by an aggressive solution of sediments. Denting corrosion can initiate a crack which can lead to a blade tearing. This has not got as much fatal impacts to HP and IP turbine parts as on LP parts, because the last LP blades are much bigger and longer than HP and IP blades and their releasing can seriously damage the turbine casing.

Mechanical impurities in steam lead to mechanical damage of leading edges. Decreasing of turbine efficiency is then caused by change of flow field. Mechanical impurities can also initiate blade cracks or they can lead to touch of rotor and stator part in the area of over-shroud seal.

3.4 Shrouds

Even though some stages of steam turbines are assembled using shroudless, free standing blades, mostly condensing, LP stages, the majority of blade rows are equipped with shrouds or some kind of auxiliary dampers. There are many types of shrouds but the main purpose is the same. They are all designed to reduce service stress levels and vibratory response of the blade especially by improving structural stiffening, dampening, or a combination of both. Moreover, shrouds are often a significant component in the stage aerodynamics, forming the outside — diameter surface of the flow path and helping to minimize over blade tip leakage and secondary flows. Consequently, they are critical parts for reliable steam turbine operation. Some of the most common forms of shrouds will be introduced in the following rows.

3.4.1 Riveted shrouds

The riveted shroud is still the most commonly used shroud design in service today, although other designs are applied in newly manufactured steam turbines. By this type of shroud a separate strip of material is fastened to the tips of each blade airfoils by one or more peened tenons. In the Fig. 8 the riveted shroud made from a series of shroud segments is demonstrated.

Several blades are covered and tied together by one segment of shroud, discrete packets of blades are created. The number and length of packets are determined by the stage designer to adjust the vibratory behaviour of the bladed disk and to evade potentially dangerous specific mode shapes and frequencies of vibration for the bladed disk assembly. The damage of this type of shroud is caused by the tenon, when the shroud is completely or partially released.

3.4.2 Integral shrouds

In contrast to riveted shrouds, where discrete packets of blades are attached by peened tenons after assembling in the turbine disk, integral shrouds are designed as integral parts of the single blades. As displayed in the Fig. 9 the complete shrouded wheel is shaped from the segments of the shroud which are part of each blade tip.

3.4.3 Z-lock shrouds

Z-lock shroud is a special kind of integral shroud, which is equipped with interlocking shroud segments. These segments are placed against one another in service to ensure contact at the abutment surfaces. Z-lock design is almost every time utilized for twisted blade airfoils because they are internally activated by the twisting of the airfoil with increasing speed. Untwisting of airfoil is caused by increasing speed and centrifugal

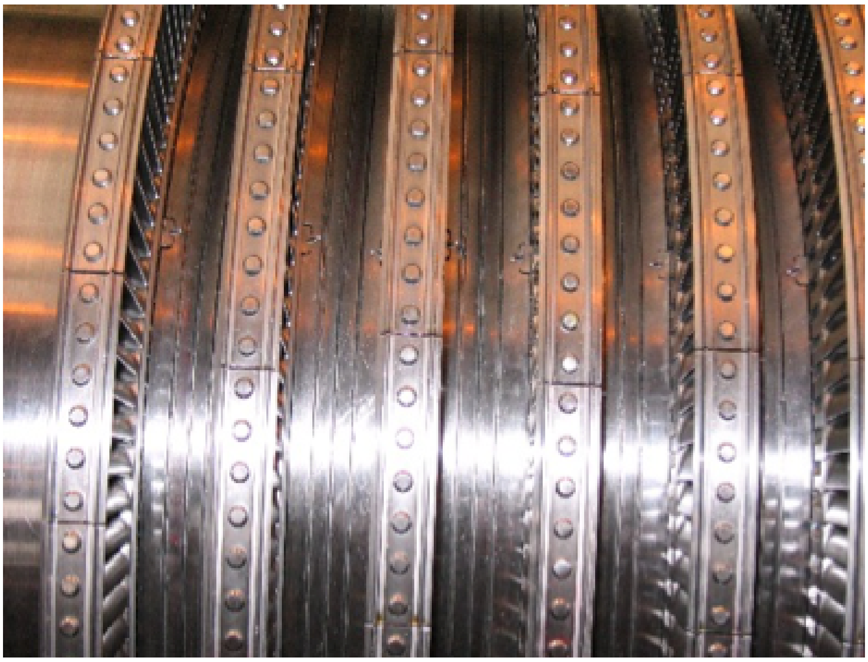

Figure 8: Riveted shroud design

Figure 9: Integral shroud design

load. How the airfoil is untwisted under centrifugal load, the Z-lock abutment surfaces are forced into contact, providing stiffening and damping for the blade disk. The Z-lock surfaces are oriented to provide positive contact at the abutment surfaces at speed. Regular and careful control of Z-lock positioning and preload are necessary to ensure effective and correct function of the shroud.

The blade with Z-lock shroud is almost always manufactured with axial fir tree root design. This kind of root design is used to prevent relative twist between nearby blades which could cause variations in contact force, defeating the purpose of the interlocking shrouds.

The prevention of wear and fretting corrosion on the abutment surfaces are often assured by using some type of surface finishing method. There can be a relative move (rubbing) between abutment surfaces during the service that can lead to unacceptable wear of Z-lock contact points. This wear is reduced by application of variety of coatings. One of the surface treatment is e.g. depositing of a Stellite 6 welded overlay.

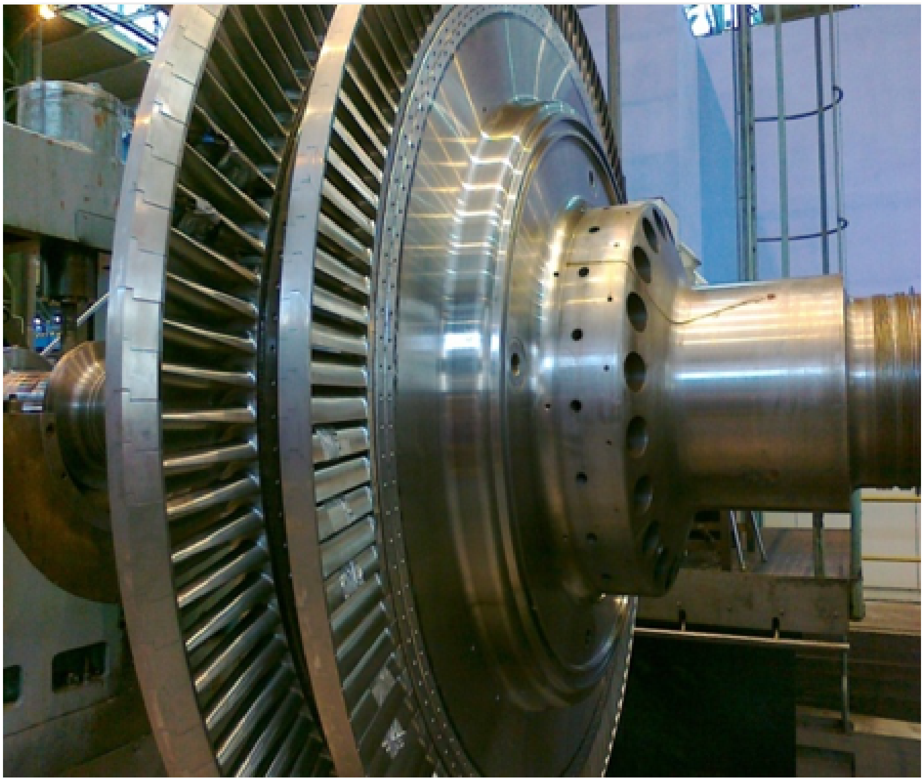

Figure 10: Z - lock shroud design

4 Theory of calculation of EOH

On the few next pages the reader should be familiarised with a theory of the EOH calculation. There are some calculations which have to be done as basis for the calculation of the EOH. The results from this calculations represent inputs to the EOH calculation. The steps which have to be done before EOH computation are:

- allocate rotor life among hot, warm and cold start-ups,
- calculate permissible thermal stress amplitude using life allocation and LCF data,
- convert the permissible stress into permissible temperature difference across the section of the rotor,
- determine minimum time for hot, warm and cold start-up for which the permissible temperature difference is not exceeded.

The customer should give to the steam turbine manufacturer a conception of how the steam turbine will be operated. This means if it will serve as a peak unit or if it will operate with a constant load. These pieces of information are very important for EOH calculation as well. Following paragraphs have been written with a reflection of [13].

4.1 Lifetime allocation

The lifetime allocation among the individual types of start-ups is more or less arbitrary and depends on operational priorities given by the customer. The reduced number of start-ups / shut-downs cycles $N_{j,red}$ which is different for each type of the start-up is calculated by the equation.

$$N_{j,red} = \frac{N_j}{\chi_j} \text{ [-]} \tag{1}$$

N_j is the number of the start-ups, χ_j represents a portion of lifetime allocated towards cycles of specific start-up type and the index j denotes the start-up type (cold, warm, hot).

Type of start-up, j	**Standstill duration**	**Number of start-ups, N_j**
Cold	$48 \div \infty$ hours	125
Warm	$8 \div 48$ hours	1000
Hot	$0 \div 8$ hours	9125

Table 1: Start-up definitions

The best way how to distinguish the type of turbine start-up is by the rotor temperature. But how it can be seen in practice, this option is not used because there would be numerous minor different start-up diagrams which would be more confusing than useful. Therefore three main types of turbine start-ups are classified by most of a steam turbine manufacturers. Classification of turbine start-ups and number of the each start-up for an exemplary steam turbine project is shown in the Tab. 1.

$N_{j,red}$ is the theoretical number of start-ups of the given type j that would alone lead to complete expenditure of the component lifetime. In this case it is the lifetime of the HP rotor. The enumeration is based on the assumption that each thermal cycle of given magnitude causes the same range of damage regardless of where it occurs in the load history. This assumption is known as the Miner linear damage accumulation rule. For a thoughtful reader this rule can be found in [9].

Type of start-up, j	**Number of start-ups,** N_j	**Lifetime allocation,** χ_j	$N_{j,red}$
Cold	125	0.025	5000
Warm	1000	0.250	4000
Hot	9125	0.625	14600
Total		0.900	

Table 2: Lifetime allocation

4.2 Rotor temperature difference

The type of the steam turbine start-up is determined by the temperature difference $\triangle T$ between the rotor surface temperature T_S and the rotor mean integral temperature $\overline{T}$. The temperature difference is given by following formula.

$$\triangle T = T_S - \overline{T} = T_S - \frac{2}{R^2} \cdot \int_0^R rT(r)dr \ [°\mathrm{C}] \tag{2}$$

R[m] is a rotor diameter, r[m] is a radial coordinate in the examined location and T[°C] is a temperature in examined location which is dependent on the radial coordinate.

The permissible temperature difference for individual type of start-up for specific rotor surface temperature is given by this calculation. This value has to be checked because of permissible temperature stress. Mean integral temperature of the HP rotor is a decisive condition for the type of the steam turbine start-up. For each of the steam turbine start-up the different start-up requirements are stated. The steam turbine start-up conditions are:

- start-up time - $t_{start-up}$ [min],
- minimal steam temperature for a start-up - T_{min} [°C],
- waiting time at a warming speed - t_1 [min],
- initial load after synchronization - N_p [MW],
- average loading rate - dN [MW/min].

For the better illustration look at the Appendix A where the model start-up diagram is showed. Dissimilarities in steam turbine loading among individual types of start-ups are denoted in the Appendix B.

4.3 Stress on the rotor surface

The determination of the rotor temperature difference is a necessary parameter for the calculation of the permissible stress on the rotor surface. If the steam turbine rotor is approximated like a solid cylinder of infinite length which is heated or cooled down on the surface. The calculation of stress on the rotor surface is realized by using the well known formula.

$$\sigma_z = \sigma_\varphi = -\frac{E\alpha\triangle T}{1-\nu} \text{ [Pa]} \tag{3}$$

σ_z[Pa] represents an axial stress which is in this case the same like a hoop stress σ_φ[Pa]. The verification that $\sigma_z = \sigma_\varphi$ is demonstrated in the Appendix C. E[Pa] is an Young's modulus (or elastic modulus) specific for given material of rotor, α[K^{-1}] is a coefficient of thermal expansion, $\triangle T$[°C] is a rotor temperature difference known from a previous chapter and ν[-] is a Poisson's ratio (the ratio of the proportional decrease in a lateral measurement to the proportional increase in length in a sample of material that is elastically stretched).

On the rotor surface the locations with a sudden change in geometry are the biggest stress concentrators, these locations were closer described in a chapter about thermal stress module. For this reason the equation 3 for the calculation of the permissible stress has to be multiplied by the stress concentration factor k_c[-].

$$\sigma_z = -k_c \cdot \frac{E\alpha\triangle T}{1-\nu} \text{ [Pa]} \tag{4}$$

The stress concentration factor takes values in the range 1.6 ÷ 2.0 in labyrinth seals. The steam turbine manufacturer assumes $k_c = 2$ in all his computations.

The permissible temperature difference of particular start-ups for symmetric load cycles is stated in the Tab. 3. The individual temperature differences for each type of the turbine start-up and for each rotor temperature are based on the equations (4) and (2), the Tab. 2 and a specific rotor material properties.

$\triangle T_j$	**Rotor surface temperature, T_S**			
	200°C	**400°C**	**500°C**	**600°C**
$\triangle T_{cold}$	60.1	48.3	48.3	27.4
$\triangle T_{warm}$	60.1	48.3	48.3	27.4
$\triangle T_{hot}$	-	43.0	32.6	23.7

Table 3: Permissible temperature difference $\triangle T_j$ for symmetric load cycles

After an inclusion of results from all previous calculations and an implementation of a mathematical simulation the specialist in a rotor stress is able to distribute a lifetime consumption of the HP rotor among individual types of start-ups and output changes. This detail is a part of the start-up diagram and it is various for each steam turbine rotor. For the exemplary rotor lifetime consumption distribution among start-ups and output changes see the Fig. 11.

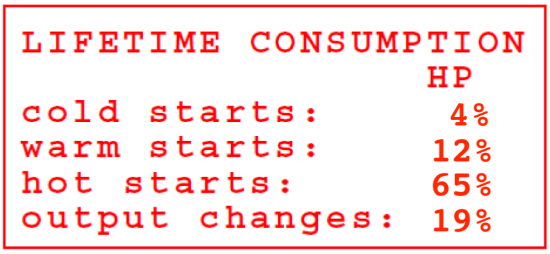

Figure 11: Rotor lifetime distribution

The whole HP rotor lifetime is perceived like a 100% which are then divided into individual types of a start-up and an output changes. With rising instability of a grid in a given state the influence of output changes is increased too. For the next calculation of the EOH, the lifetime consumption factor l_j[-] is introduced. The relation between distributed lifetime and the lifetime consumption factor is indicated in the Tab. 4. The rotor lifetime consumption distribution is one of the most significant statement for the EOH calculation.

Type of start-up, j	**Lifetime consumption**	**Lifetime consumption factor,** l_j
Cold	4 %	0.04
Warm	12 %	0.12
Hot	65 %	0.65
Output changes	19 %	0.19
Total	100 %	1

Table 4: Lifetime consumption factor

4.4 Calculation of the EOH

4.4.1 Inputs from the customer

Besides the rotor lifetime distribution, the customer's requirements about an operation of a steam turbine are very underlying for a proper estimation of a number of EOH. The steam turbine service life is usually designed by a steam turbine manufacturer for 25 years. The customer should be able to estimate how many operating hours the steam turbine will work per a year. Thereafter the overall operating hours (OH) of the steam turbine are very easily determined by the equation 5. Where Y is a number of years and OH_Y is a number of operating hours per one year.

$$OH = OH_Y \cdot Y \text{ [hours]} \tag{5}$$

The number of operating hours per year is influenced by application where the steam turbine will be served in. There are big differences in a steam turbine operation if it is used for example in a solar power plant, in a CHP application or like an industrial steam turbine.

4.4.2 Estimation of the total number of the EOH

The first step for an estimation of the total number of the EOH is a determining EOH for each type of the turbine start-up. Once the number of individual start-ups (N_j) is known from the customer and the lifetime consumption factor (l_j) is designated by the rotor stress specialist, the EOH for individual type of start-up (c_j) could be calculated using following formula.

$$c_j = OH \cdot \frac{l_j}{N_j} \text{ [hours]} \tag{6}$$

As it will be proved in the next part of this thesis, previous calculations give the steam turbine operator a very decent notion how much the consumption of EOH, which in fact represents the steam turbine service life consumption, is influenced by a various types of start-ups.

The next step is a calculation of a total number of the EOH with use of all achieved results and information so far. For this part the equation (7) is used by the steam turbine manufacturer.

$$EOH = OH + \sum_{j=1}^{3} c_j \cdot N_j \text{ [hours]} \tag{7}$$

The total number of the EOH is always higher than the number of operating hours. The number of the EOH per year (EOH_Y) also occurs in practice.

$$EOH_Y = \frac{EOH}{Y} \text{ [hours/year]} \tag{8}$$

Further, the maintenance plan is defined by the steam turbine producer in a cooperation with his service departments. The maintenance plan is based on a calculated number of the EOH per year and on a steam turbine producer's experience.

4.5 Guarantee of an availability of a turboset

An expending of the EOH is also related with availability or reliability of a turboset. Still more often the customer demands guarantee of availability in a contractual documentation.

The availability guarantee is applied to an operation of the delivered equipment by the steam turbine manufacturer in terms of a guarantee period. There will be considered only those forced outages of a turboset within an overall availability evaluation that will be caused by delivered equipment failures.

The availability of a turboset A is qualified by the following formula.

$$A = \left(1 - \frac{UOH}{PH}\right) \cdot 100 \text{ [\%]} \tag{9}$$

Period hours PH [hours] is the number of hours when a turboset was in an evaluated period, minus time for annual maintenance — not more than 2 x 336 hours for two years. Unplanned forced outage hours UOH [hours] is the number of hours when the turbine generator or its accessories are not operational due to failure which the supplier (the steam turbine manufacturer) is responsible for. UOH are defined according to the formula below.

$$UOH = \sum \left(1 - \frac{PO}{PAO}\right) \cdot t \text{ [hours]} \tag{10}$$

A present power output PO [MW] is considered like a power output presently measured at the generator terminals. And a presently available power output PAO [MW] is defined as a corrected power output preset by an operator and achievable under actual circumstances of the power plant. t [hours] represents unplanned forced outage hours (full or partial outage) when the turbine generator or its accessories are not operational due to failure which the supplier (the steam turbine manufacturer) is responsible for.

For an unplanned forced outages (UOH) are counted only unplanned reductions of the power output below 90% of presently available power output (PAO) and longer than 1 hour, which the supplier is responsible for.

The availability of the delivered equipment is a general rule determined by the steam turbine manufacturer higher than or equal to 96%.

$$A \geq 96\%$$

The evaluated period for an availability guarantee is delimited from 3 months after a preliminary acceptance certificate (PAC) to the end of a warranty period.

5 Exemplary EOH calculation

For the better explanation and understanding of problematics of the EOH an exemplary EOH calculation will be shown. A model estimation of the EOH will be implemented for the solar power station using a CSP technology. For solar power stations projects the EOH approach is very often used and useful due to their everyday start-up and shut-down.

5.1 Necessary inputs

A required number of start-ups and operating hours per a year given by the costumer:

$$N_{cold} = 125, N_{warm} = 1000, N_{hot} = 9125, OH_Y = 8000.$$

An individual start-up times and distributed HP rotor lifetime consumption are displayed in the Tab. 5. These details are provided by the rotor stress specialist.

Type of start-up, j	**Lifetime consumption**	**Lifetime consumption factor,** l_j	**Real time duration,** t_j
Cold	4 %	0.04	142 min
Warm	12 %	0.12	85 min
Hot	65 %	0.65	13 min
Output changes	19 %	0.19	

Table 5: EOH inputs

5.2 EOH calculation

The first step in the EOH calculation is an assessment of a total operating hours.

$$OH = OH_Y \cdot Y = 8000 \cdot 25 = 200000 \text{ hours} \tag{11}$$

For the next part of the EOH calculation the EOH for an individual type of start-up have to be computed.

$$c_{cold} = OH \cdot \frac{l_{cold}}{N_{cold}} = 200000 \cdot \frac{0.04}{125} = 64 \text{ hours} \tag{12}$$

$$c_{warm} = OH \cdot \frac{l_{warm}}{N_{warm}} = 200000 \cdot \frac{0.12}{1000} = 24 \text{ hours} \tag{13}$$

$$c_{hot} = OH \cdot \frac{l_{hot}}{N_{hot}} = 200000 \cdot \frac{0.65}{9125} = 14.2 \text{ hours} \tag{14}$$

Time differences between the EOH and operating hours of individual start-ups are for a clearer view introduced in the Tab. 6. The total number of the EOH is given by the formula (7).

$$EOH = OH + [(c_{cold} \cdot N_{cold}) + (c_{warm} \cdot N_{warm}) + (c_{hot} \cdot N_{hot})] \text{ [hours]} \tag{15}$$

Type of start-up, j	**Real time duration, t_j**	**EOH consumption, c_j**	**Time difference**
Cold	142 min	64 hours	61.63 hours
Warm	85 min	24 hours	22.58 hours
Hot	13 min	14.2 hours	13.98 hours

Table 6: Comparison of OH and EOH

$$EOH = 200000 + [(64 \cdot 125) + (24 \cdot 1000) + (14.2 \cdot 9125)] = 361575 \text{ hours} \tag{16}$$

After that it is possible to determine the number of the EOH per year (EOH_Y).

$$EOH_Y = \frac{EOH}{Y} = \frac{361575}{25} = 14463 \text{ [hours/year]} \tag{17}$$

The maintenance plan is then suggested on the base of: a steam turbine manufacturer's experience, the number of the EOH per year and the predicted number of start-ups per year. The suggested maintenance plan is shown in the Fig. 12.

Year of operation	Inspection type	EOH per 1 year	Sum of EOH	OH per 1 year	Sum of OH
1		14463,0	14463,0	8000	8000
2	Minor	14463,0	28926,0	8000	16000
3	Minor	14463,0	43389,0	8000	24000
4	Intermediate	14463,0	57852,0	8000	32000
5		14463,0	72315,0	8000	40000
6	Minor	14463,0	86778,0	8000	48000
7	Minor	14463,0	101241,0	8000	56000
8		14463,0	115704,0	8000	64000
9	GO	14463,0	130167,0	8000	72000
10		14463,0	144630,0	8000	80000
11	Minor	14463,0	159093,0	8000	88000
12	Minor	14463,0	173556,0	8000	96000
13	Intermediate	14463,0	188019,0	8000	104000
14		14463,0	202482,0	8000	112000
15	Minor	14463,0	216945,0	8000	120000
16	Minor	14463,0	231408,0	8000	128000
17		14463,0	245871,0	8000	136000
18	GO	14463,0	260334,0	8000	144000
19		14463,0	274797,0	8000	152000
20	Minor	14463,0	289260,0	8000	160000
21	Minor	14463,0	303723,0	8000	168000
22	Intermediate	14463,0	318186,0	8000	176000
23		14463,0	332649,0	8000	184000
24	Minor	14463,0	347112,0	8000	192000
25	Minor	14463,0	361575,0	8000	200000

Figure 12: Suggested maintenance plan

5.3 Possible changes of maintenance plan

The maintenance plan in the Fig. 12 is valid if the number of all types of start-ups will be more or less abided. What will happen if e.g. number of hot start-ups is significantly reduced or increased and numbers of other start-ups stay the same, it is visible from modified maintenance plans in the Fig. 13 and the Fig. 14.

Blue fields show where the general overhaul (GO) should be carried out and red fields indicate how planned terms of GO are moved depending on the number of hot start-ups.

The comparison of individual steam turbine's inspections from time and economical point of view is indicated in the Tab. 7.

In the Tab. 8 there is a quantification of total maintenance prices for all three variants which were mentioned above.

Changes among individual maintenance plans are clearly visible from the Fig. 15.

Year of operation	Inspection type	EOH per 1 year	Sum of EOH	OH per 1 year	Sum of OH
1		11836,0	11836,0	8000	8000
2	Minor	11836,0	23672,0	8000	16000
3	Minor	11836,0	35508,0	8000	24000
4		11836,0	47344,0	8000	32000
5	Intermediate	11836,0	59180,0	8000	40000
6		11836,0	71016,0	8000	48000
7		11836,0	82852,0	8000	56000
8	Minor	11836,0	94688,0	8000	64000
9	Minor	11836,0	106524,0	8000	72000
10		11836,0	118360,0	8000	80000
11	GO	11836,0	130196,0	8000	88000
12		11836,0	142032,0	8000	96000
13		11836,0	153868,0	8000	104000
14	Minor	11836,0	165704,0	8000	112000
15	Intermediate	11836,0	177540,0	8000	120000
16		11836,0	189376,0	8000	128000
17		11836,0	201212,0	8000	136000
18	Minor	11836,0	213048,0	8000	144000
19	Minor	11836,0	224884,0	8000	152000
20		11836,0	236720,0	8000	160000
21	GO	11836,0	248556,0	8000	168000
22		11836,0	260392,0	8000	176000
23	Minor	11836,0	272228,0	8000	184000
24	Intermediate	11836,0	284064,0	8000	192000
25		11836,0	295900,0	8000	200000

Figure 13: Modified maintenance plan for 50% reduction of hot start-ups

Year of operation	Inspection type	EOH per 1 year	Sum of EOH	OH per 1 year	Sum of OH
1		17061,6	17061,6	8000	8000
2	Minor	17061,6	34123,2	8000	16000
3	Minor	17061,6	51184,8	8000	24000
4	Intermediate	17061,6	68246,4	8000	32000
5		17061,6	85308,0	8000	40000
6	Minor	17061,6	102369,6	8000	48000
7	GO	17061,6	119431,2	8000	56000
8		17061,6	136492,8	8000	64000
9	Minor	17061,6	153554,4	8000	72000
10	Minor	17061,6	170616,0	8000	80000
11	Intermediate	17061,6	187677,6	8000	88000
12		17061,6	204739,2	8000	96000
13	Minor	17061,6	221800,8	8000	104000
14	Minor	17061,6	238862,4	8000	112000
15	GO	17061,6	255924,0	8000	120000
16		17061,6	272985,6	8000	128000
17	Minor	17061,6	290047,2	8000	136000
18	Minor	17061,6	307108,8	8000	144000
19	Intermediate	17061,6	324170,4	8000	152000
20		17061,6	341232,0	8000	160000
21	Minor	17061,6	358293,6	8000	168000
22	GO	17061,6	375355,2	8000	176000
23		17061,6	392416,8	8000	184000
24	Minor	17061,6	409478,4	8000	192000
25	Minor	17061,6	426540,0	8000	200000

Figure 14: Modified maintenance plan for 50% increase of hot start-ups

Inspection type	**Outage time**	**Inspection price**
Minor	2 — 7 days	15.000 EUR
Intermediate	14 — 20 days	200.000 EUR
GO	40 — 60 days	1 mil. EUR

Table 7: Comparison of individual steam turbine's inspections

Maintenance plans variants	**Total maintenance price**
50 % of N_{hot}	2.72 mil. EUR
100 % of N_{hot}	2.78 mil. EUR
150 % of N_{hot}	3.78 mil. EUR

Table 8: Quantification of total maintenance prices

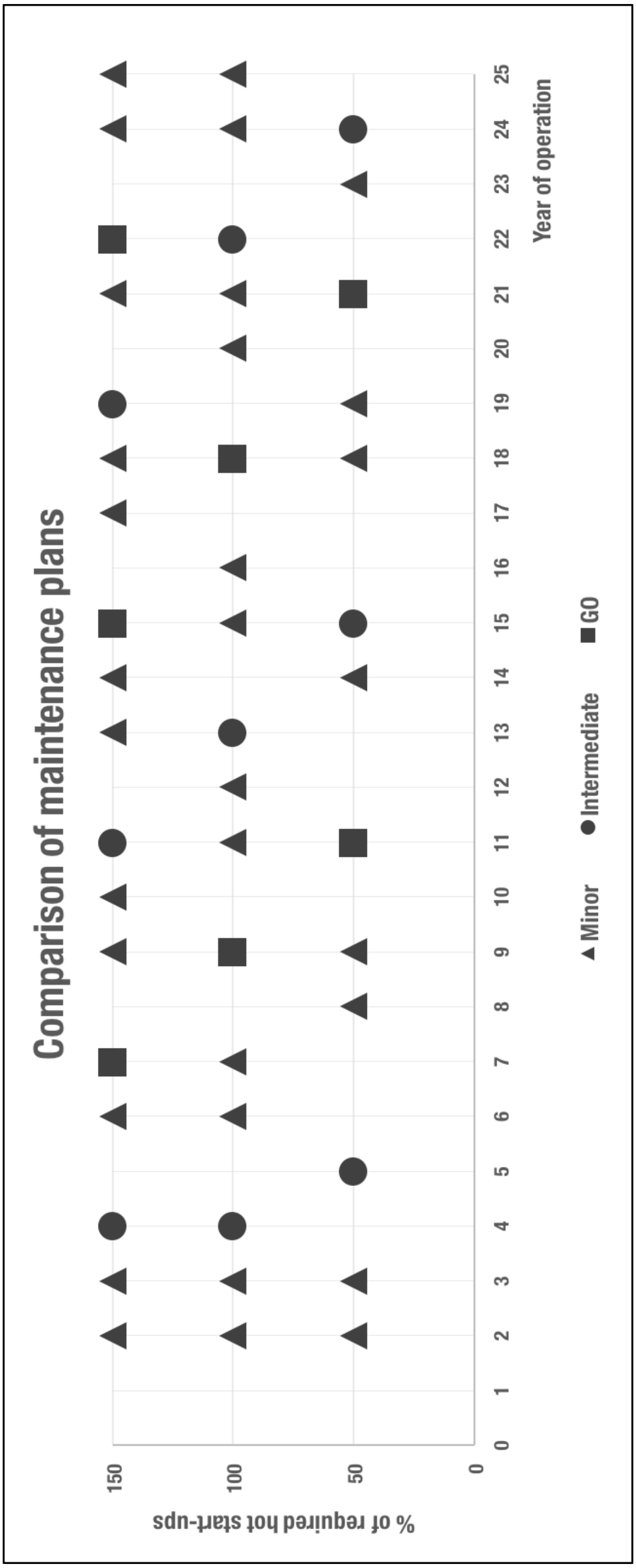

Figure 15: Comparison of maintenance plans

6 Service and maintenance programme

Because every steam turbine manufacturer knows best his own steam turbine, he should dedicate to his customers and would like to be a partner for the entire lifetime of the steam turbine. Therefore steam turbine manufacturers offer various service products according to the specific customer's needs and local requirements. These products most often are:

- maintenance programme,
- plant performance audits,
- remote operations support,
- advanced vibration diagnostics and predictive systems,
- residual life assessment.

Above mentioned service products are not included in supplier's basic scope of supply and services and from this reason it is up to customers who will take care of their steam turbine. Individual types of maintenance in following subheads are interpreted in accordance with [10], [13].

6.1 Maintenance programme

A typical maintenance programme concerning the type of maintenance to be undertaken, the scope of maintenance works and the schedule of inspections are described and explained in this chapter. The owner's operational personnel operate the equipment and provide routine daily maintenance of the equipment in accordance with the supplier's operating manual both during and after the expiry of the warranty period. The following activities shall be carried out during the lifetime of a steam turbine:

- routine maintenance,
- minor inspection,
- intermediate inspection,
- major inspection/general overhaul.

Equipment availability and reliability are among the most important factors when a customer decides between various equipment suppliers. Steam turbine manufacturers are well aware of this fact and they have frequently prepared a compact solution - Long Term Service Agreement (LTSA). The aim of LTSA is to, among others, establish true partnerships to the benefit of both parties.

A whole service care about delivered equipment is covered in the LTSA. The LTSA guarantee the customer fixed prices of services for the whole duration of the LTSA, escalated only by the EU raw material and labour indexes. The LTSA is focused on decreasing customer's risks with respect to reaching high availability, decreasing the number of starts-ups etc., by help of professional maintenance support.

6.1.1 Routine maintenance

The focus of a routine maintenance is to eliminate minor damages detected by operators or equipment monitoring systems during operation of the equipment. Regular checking and inspection of the equipment and performance of routine maintenance in accordance with the steam turbine manufacturer's operating manual forms where the basis of preventative maintenance are described in more detail. This generally represents only checking and inspection of the system, possible removal of minor failures during operation or during short-term shut-downs of the system forced by other reasons.

Intervals of activities are either forced by the operating situation (e.g. signalling of filter clogging, results of the chemical analysis of samples) and/or they are prescribed in the operating part of the instructions. Most terms of testing periodicity are prescribed by the table of secured quantities that is a part of the operating instructions of the steam turbine. Routine maintenance is habitually performed by the customer's staff.

6.1.2 Minor inspection

Minor inspection of the steam turbine is a preventive inspection during which the turbine casings, bearing pedestals and generator remain closed.

The typical scope of operations within minor inspection is as follows:

- diagnostic and adjustments as may be necessary of the control system,
- complete functional check of the protection system of the turbine including linearity of control valves,
- boroscopic inspection if possible (casing temperature must be under 65°C),
- overall visual check of the turboset,
- maintenance which is impossible to execute on a running machine,
- repair of imperfections which may have been found during operation or subsequent inspections,
- diagnostic measuring during the operation.

The minor inspection is typically scheduled as 2 — 7 days outage of the steam turbine, generator and auxiliary equipment. Individual operations are carried out under steam turbine manufacturer supervision, by local maintenance personnel and associated services provided by the customer.

6.1.3 Intermediate inspection

An intermediate inspection of the steam turbine is a regular inspection, the scope of operations is surely wider than during a minor inspection. Besides operations from previous chapter the scope of work is extended by:

- decoupling and alignment of rotors,
- boroscopic inspection,
- diagnostic measurements on generator,
- adjustment of lube and control oil system,
- check of vibration monitoring,
- bearing visual inspection .

Throughout these activities, the steam turbine and generator remain closed.

The minor inspection is typically scheduled as at least 14 days outage of the steam turbine, generator and auxiliary equipment. An individual operations are carried out under steam turbine manufacturer supervision, by local maintenance personnel and associated services provided by the customer.

6.1.4 Major inspection

Major Inspection is a package of complex maintenance activities, which are considered necessary for returning the equipment close to its original state.

The main target of general overhaul (major inspection) is to ensure trouble-free operation of the steam turbine and auxiliary equipment during the next maintenance cycle. It is a thorough inspection of the actual state of all equipment and accessories — especially internals. Worn out or damaged parts are exchanged or repaired. During this overhaul spare parts are replenished at the stock.

The typical scope of service activity within the GO includes besides the scope of intermediate inspection also:

- opening the steam turbine casings,
- execution of maintenance tasks which take longer than one week,
- cleaning and repair of turbine steam flow path to ensure turbine efficiency and reliability, cleaning and flushing of lubrication oil/hydraulic oil systems, cleaning of condenser/heaters, repair of control armatures, resetting of clearances and the mechanical items within the control system,
- repair of imperfections which may have been found during operation or subsequent inspection,
- revision of and repair exciter and generator,
- check of radial and axial tolerances at flow through part.

During a GO all the equipment and accessories should be opened to permit visual inspection of the internals, Non-Destructive Testing (NDT) inspection where it is necessary and large-scale diagnostic measurements.

General overhaul is to be applied in a period $6 \div 9$ years according to operating and technical conditions and the EOH. Just the consumption of the EOH is a very practical tool, thanks which a steam turbine operator can estimate when he will have to do a GO. The scope of the first GO may have a different form than the second GO because of the same reason (the steam turbine after $6 \div 9$ years of operation requires normally different care and scope of spare parts from the steam turbine that has been in operation twice as long).

A GO is typically scheduled as a minimum of 40 and up to 60 days outage of the steam turbine, generator and auxiliary equipment. Individual maintenance operations are carried out under steam turbine manufacturer supervision, by local maintenance personnel and associated services provided by the customer. A representative of the steam turbine manufacturer is usually presented when the steam turbine is putting back to the operation.

The typical activities are individually described in more detail in the Appendix E. Exact scope of works, list of spare parts and overhaul duration should be discussed at least one year before a planned GO.

Conclusion

This submitted diploma thesis generally deals with introduction of the concept of EOH by steam turbines and with a suggestion of its determination and utilization in practice. Its aim is to offer this type of concept to further potential customers. For a better notion a case study of application of the EOH with different variants was presented.

Firstly, the reader was acquainted with the idea of the EOH and its historical evolution. The second part of this thesis was dedicated to steam turbine auxiliary modules and systems, which are somehow coupled with the EOH determination or an estimation of a steam turbine residual lifetime. The third part was focused on the description of individual turbine components which mostly influence the steam turbine's lifetime resulting in their wear consuming EOH. In the next part, the reader was familiarised with a theory and the proposal of the EOH calculation. A case study of EOH application with different variants in the fifth part was presented. The last part was focused on the clarification and sorting of steam turbine's inspections.

Every outage of a steam turbine poses certain expenditures to power plant operator. Moreover an operator can not sell electricity which is normally produced by serviced device. It, to a certain extent, decreases his revenue.

The concept of the EOH especially improves maintenance planning which is closely linked with economy of operation. But it does not primarily mean that EOH will reduce number of steam turbine inspections. EOH more likely determine the right time where the certain type of inspection should be accomplished depending on the manner of steam turbine operation. On the one hand it leads to the prevention of many early inspections which would be ineffective from an economical point of view. On the other hand it also should help to obviate steam turbine outages owing to failure of its certain component. Outages caused by a component failure usually last longer and they can have weighty consequences in terms of steam turbine damage. Because of the EOH with usage of operational data steam turbine operator can relatively easily verify if steam turbine lifetime is consumed in accordance with suggested maintenance plan or if it is consumed slower or faster and the maintenance plan then would have to be modified. A modification of the maintenance plan is accompanied by total maintenance expenditures as well.

Above mentioned situations as well as illustration of usage of the EOH in practice are demonstrated in the fifth part of this thesis. The EOH determination and the maintenance plan are suggested on the base of customer's conception of a steam turbine operation. For example if the customer will know that the number of hot start-ups will be significantly lower than the proposed number. From EOH calculation and resulting maintenance plan (Fig. 13), they are able to find out that first GO will be done two years later compared to original proposal and the period between GOs will be extended. Steam turbine operator is also able to calculate that if he operates a steam turbine this way throughout the service life he will save about 60.000 EUR (approximately 400.000 CZK), as stated in the Tab. 8. This savings are not so substantial as total maintenance plan price increase in case of a significantly rise of number of hot start-ups. Modified maintenance plan is shown in the Fig. 14. For the reason of more frequent start-ups turbine components are worn out quicker. Therefore the first GO is scheduled two years earlier than the original proposal. Periods among GOs are shortened and one GO has to be added. Total maintenance price is then enormously increased, opposite to original maintenance plan about 1 mil. EUR (approximately 26.5 mil. CZK).

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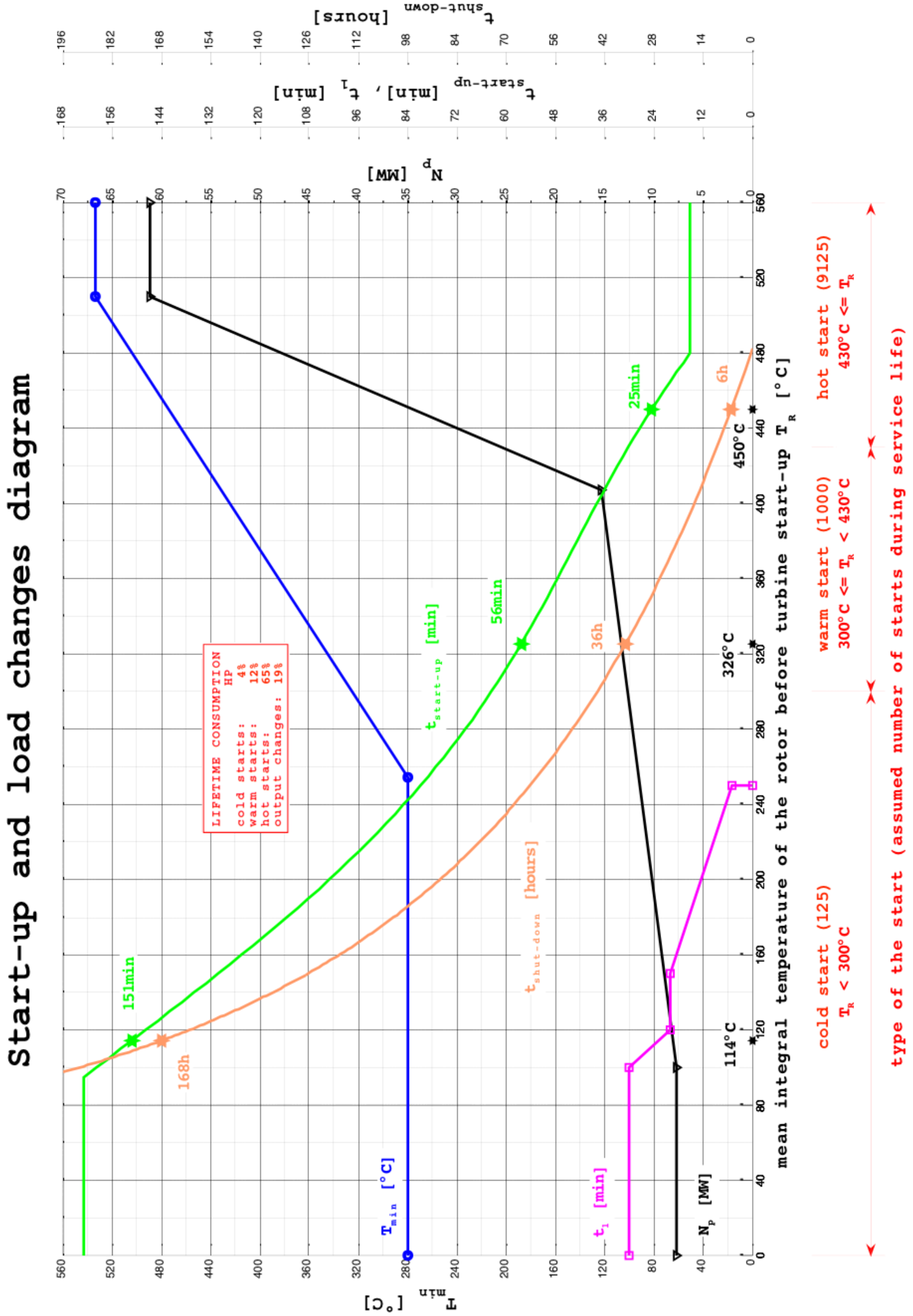
Appendix A
Start-up and load changes diagram


Start-up and load changes diagram
LIFETIME CONSUMPTION
HP
cold starts: 4%
warm starts: 12%
hot starts: 65%
output changes: 19%
151min
56min
25min
168h
36h
6h
114°C
326°C
450°C
mean integral temperature of the rotor before turbine start-up T_R [°C]
cold start (125)
T_R < 300°C
warm start (1000)
300°C <= T_R < 430°C
hot start (9125)
430°C <= T_R
type of the start (assumed number of starts during service life)
T_min [°C]
t_1 [min]
N_p [MW]
t_start-up [min]
t_shut-down [hours]
t_start-up [min], t_1 [min]

Appendix B
Start-up and load changes dissimilarities for individual types of start-ups

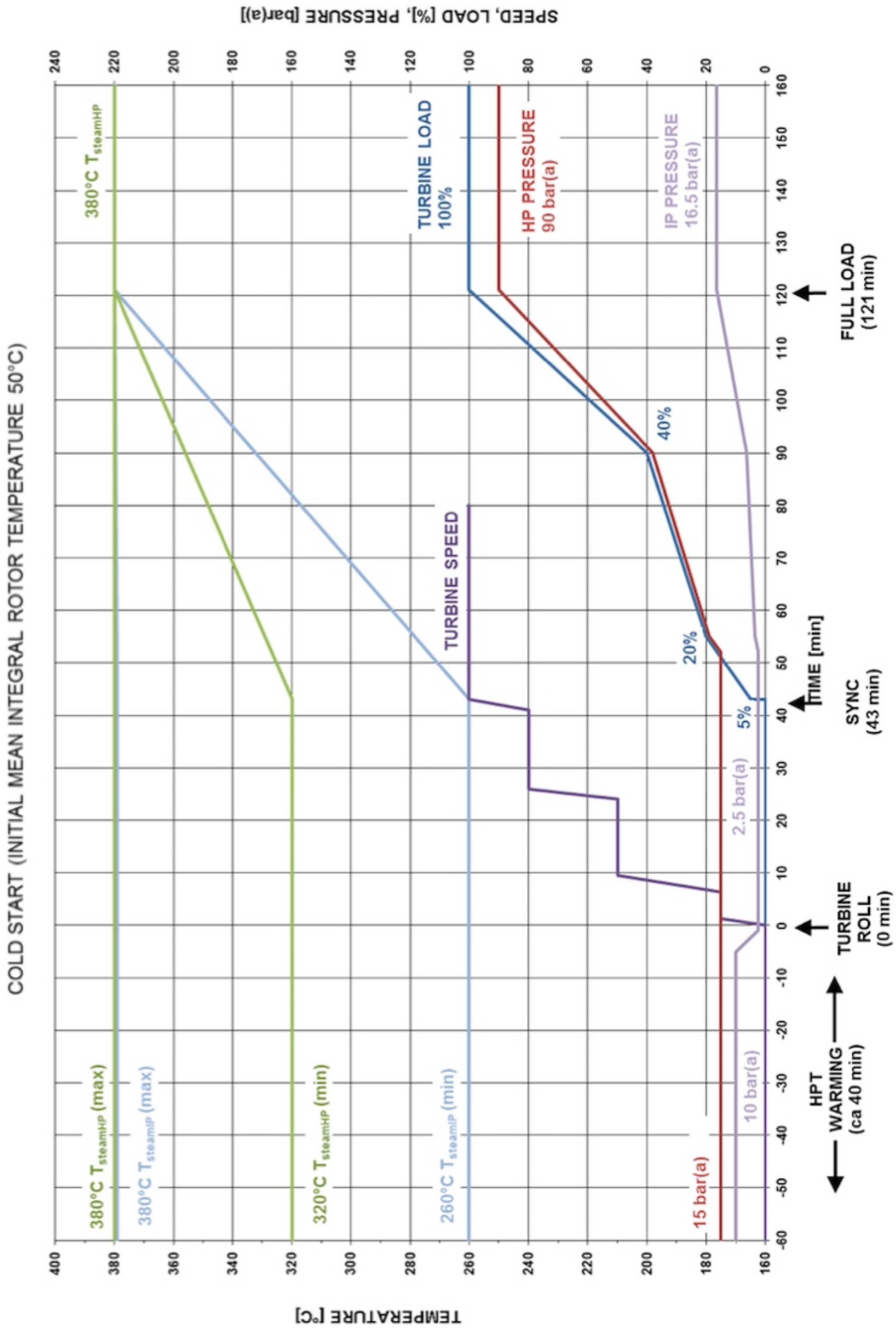


Figure 16: Cold start-up

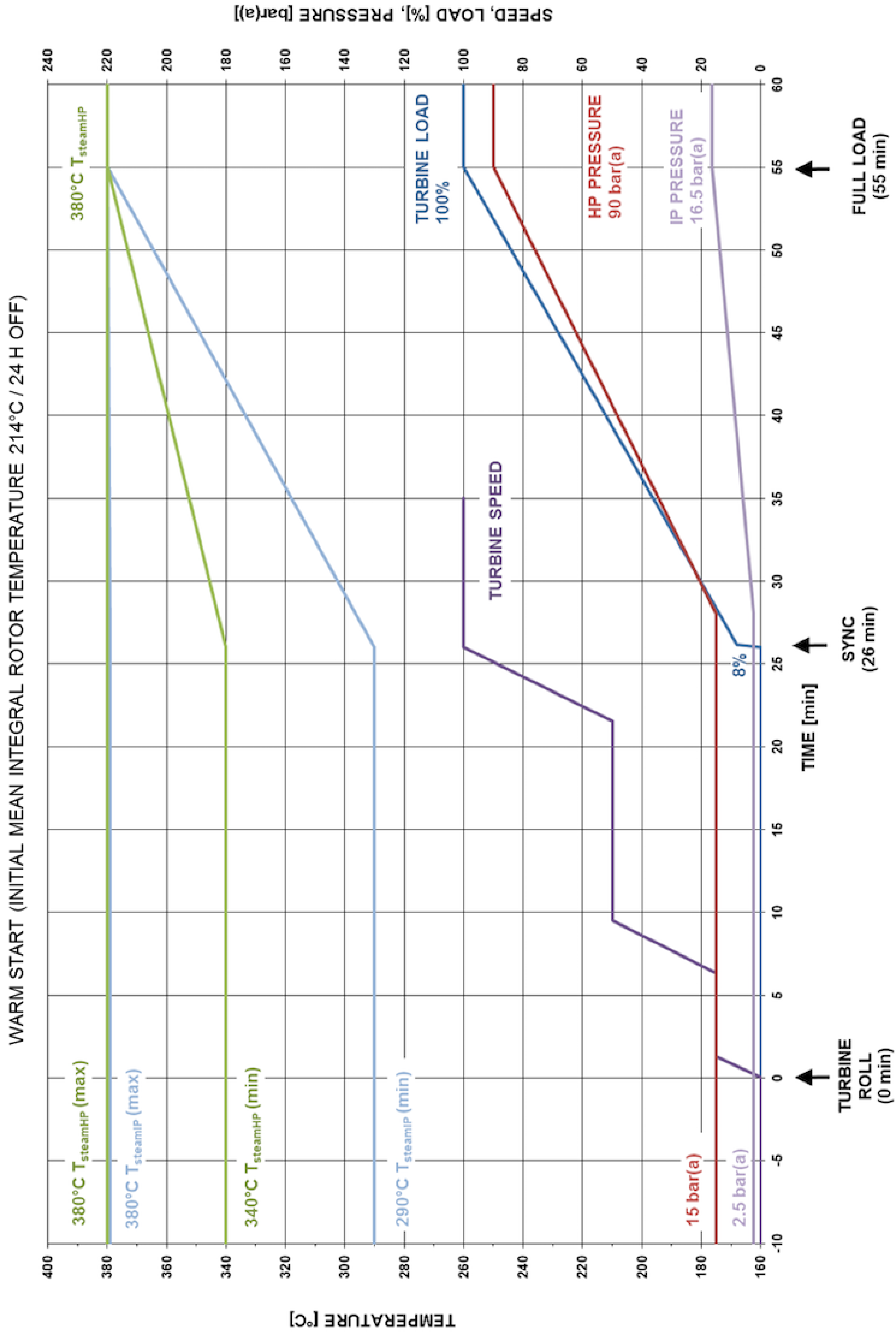


Figure 17: Warm start-up

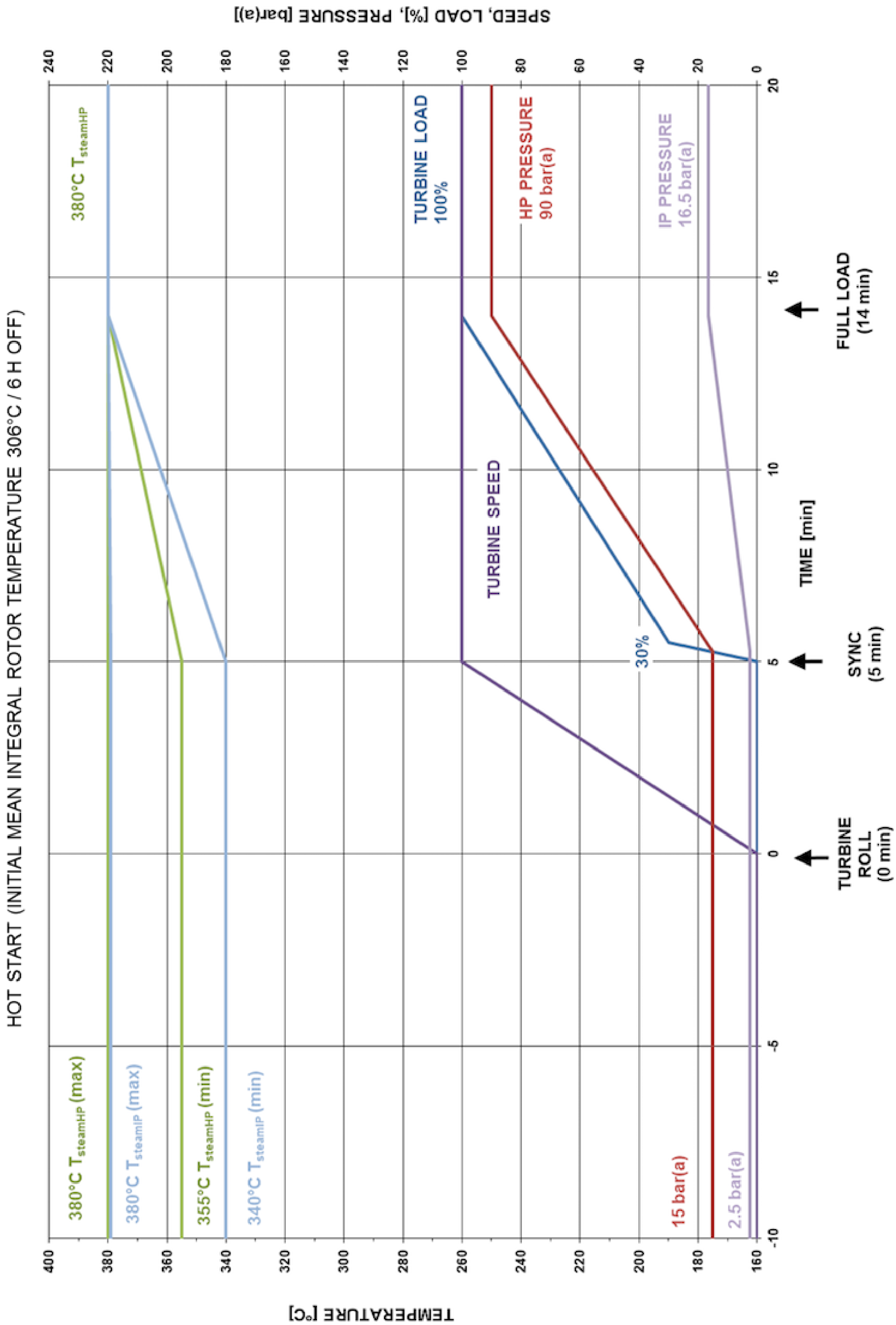


Figure 18: Hot start-up

Appendix C
Verification of the formula (3)

According to the monograph [2] chapter 20, a stress, caused by a temperature field, can be solved with reflections of a thermoelastic displacement potential Φ. Its gradient is known as

$$\nabla\Phi = 2\mu\vec{u}. \tag{18}$$

The Lamé constant is defined like

$$\mu = \frac{E}{2\cdot(1+\nu)}, \tag{19}$$

and $\vec{u}$ is a vector of a displacement.

The particular solution of the equation (18) is given by following formula.

$$\nabla^2\Phi = \frac{E\alpha T}{1-\nu} \tag{20}$$

T is a appropriate temperature field, the Laplace operator ∇^2 is formulated as

$$\nabla^2 = \frac{\partial}{\partial x^2} + \frac{\partial}{\partial y^2} + \frac{\partial}{\partial z^2}. \tag{21}$$

In case of axisymmetric problems, the Laplace operator is transformed into cylindrical polar coordinates and appropriate components of stress then have form:

$$\sigma_{rr} = \frac{\partial^2\Phi}{\partial r^2} - \nabla^2\Phi, \sigma_{\varphi\varphi} = \frac{1}{r}\cdot\frac{\partial\Phi}{\partial r} - \nabla^2\Phi, \sigma_{zz} = \frac{\partial^2\Phi}{\partial z^2} - \nabla^2\Phi,$$

$$\sigma_{r\varphi} = 0, \sigma_{\varphi z} = 0, \sigma_{zr} = \frac{\partial^2\Phi}{\partial z\partial r}.$$

In our case the temperature field T of a cylinder is dependent only on the coordinate r and not on coordinate z, the solution is simplified because

$$\frac{\partial\Phi}{\partial z} = 0. \tag{22}$$

The Laplace operator in cylindrical polar coordinates is then defined by coming equation.

$$\nabla^2 = \frac{1}{r}\frac{d}{dr}\left(r\frac{d}{dr}\right) \tag{23}$$

The equation (20) is converted into a new form.

$$\frac{1}{r}\frac{d}{dr}\left(r\frac{d\Phi}{dr}\right) = \frac{E\alpha T(r)}{1-\nu} \tag{24}$$

Then the particular integral is formulated as

$$\frac{d\Phi}{dr} = \frac{E\alpha}{1-\nu}\cdot\frac{1}{r}\int_0^r \xi T(\xi)d\xi. \tag{25}$$

The general solution is then obtained as

$$\sigma_{rr} = -\frac{E\alpha}{1-\nu} \cdot \frac{1}{r^2} \int_0^r \xi T(\xi) d\xi + \frac{E}{1+\nu} A, \tag{26}$$

$$\sigma_{\varphi\varphi} = -\frac{E\alpha}{1-\nu} \cdot \left(\frac{1}{r^2} \int_0^r \xi T(\xi) d\xi - T(r) \right) + \frac{E}{1+\nu} A, \tag{27}$$

$$\sigma_{zz} = -\frac{E\alpha}{1-\nu} T(r) + C, \tag{28}$$

$$\sigma_{r\varphi} = 0, \sigma_{\varphi z} = 0, \sigma_{zr} = 0. \tag{29}$$

The constant A is evaluated from the boundary condition $\sigma_{rr}(R) = 0$.

$$A = \frac{\alpha(1+\nu)}{1-\nu} \cdot \frac{1}{R^2} \int_0^R \xi T(\xi) d\xi = \frac{\alpha(1+\nu)}{1-\nu} \cdot \frac{\overline{T}}{2} \tag{30}$$

Thereafter we can write for the hoop stress on the surface

$$\begin{aligned} \sigma_{\varphi\varphi}(R) &= \frac{E\alpha}{1-\nu} \cdot \left(\frac{1}{2}\overline{T} - T(R) \right) + \frac{E}{1+\nu} \cdot \frac{\alpha(1+\nu)}{1-\nu} \cdot \frac{\overline{T}}{2} \\ &= \frac{E\alpha}{1-\nu} \cdot \left(\overline{T} - T(R) \right) = -\frac{E\alpha}{1-\nu} \triangle T(R). \end{aligned} \tag{31}$$

The formula (31) is not dependent on the appropriate process in temperature field in a cylinder but it is dependent only on its mean integral temperature and surface temperature.

The ends of cylinder are not fixed therefore the cylinder can relax in its axial direction. The boundary condition for the determination of the constant C is then

$$\frac{2}{R^2} \int_0^R \sigma_{zz} \cdot r dr = 0, \tag{32}$$

and according to (18)

$$0 = C - \frac{E\alpha}{1-\nu} \overline{T}. \tag{33}$$

That is why the formula (28) gives

$$\sigma_{zz}(R) = -\frac{E\alpha}{1-\nu} \cdot \left(T(R) - \overline{T} \right) = -\frac{E\alpha}{1-\nu} \triangle T(R). \tag{34}$$

As can be seen, right sides of (31) and (34) are equal, which means the hoop stress is the same like the axial stress on the cylinder surface.

Appendix D
Cooling of the infinite cylinder with a constant temperature flux at the surface

Analytical solutions temperature fields for problems with various boundary conditions are derived in the monograph [4]. In our case, the constant temperature flux F at the cylinder surface is considered. Its sign responds to cylinder cooling. The next consideration is a constant initial temperature T_0 in the time $t = 0$.

According to the paragraph 7.8. of the monograph [4], the corresponding radial temperature field $T(r,t)$ of the infinite cylinder with radius R can be described by the following formula

$$T(r,t) = T_0 - \frac{2F\kappa t}{K \cdot R} - \frac{F \cdot R}{K} \cdot \left[\frac{r^2}{2R^2} - \frac{1}{4} - 2 \sum_{n=1}^{\infty} e^{-\kappa \alpha_n^2 \frac{t}{R^2}} \frac{J_0\left(r\frac{\alpha_n}{R}\right)}{\alpha_n^2 J_0(\alpha_n)} \right], \tag{35}$$

where t is time, R is the cylinder diameter, K is the thermal conductivity coefficient, κ is the thermal diffusivity and α_n are positive roots of the equation

$$J_1(\alpha) = 0. \tag{36}$$

Here J_0 and J_1 denote the Bessel functions

$$J_v(r) = \sum_{n=0}^{\infty} \frac{(-1)^n \cdot \left(\frac{1}{2}r\right)^{v+2n}}{n!\Gamma \cdot (v+n+1)}. \tag{37}$$

The Bessel function J_v is the solution of the equation

$$\frac{d^2y}{dr^2} + \frac{1}{r}\frac{dy}{dr} + \left(1 - \frac{v^2}{r^2}\right) y = 0, \tag{38}$$

Γ denotes the Gamma function.

The integral

$$\int_0^R r J_0(\alpha r) dr = \frac{1}{\alpha} R J_1(\alpha R), \tag{39}$$

from [4] page 198, is used to the calculation of the mean integral temperature.

With taking into account of the equation (36) the previous integral is changed into

$$\int_0^R r J_0\left(r\frac{\alpha_n}{R}\right) dr = \frac{R^2}{\alpha_n} J_1(\alpha_n) = 0. \tag{40}$$

With use of formula (35) the mean integral temperature in a time t is then defined as

$$\overline{T}(t) = \left(T_0 - \frac{2F\kappa t}{KR} + \frac{1}{4}\frac{FR}{K}\right) \int_0^R r \frac{dr}{\frac{R^2}{2}} - \frac{FR}{K}\frac{1}{2R^2} \int_0^R r^3 \frac{dr}{\frac{R^2}{2}}, \tag{41}$$

$$\overline{T}(t) = T_0 - \frac{2F\kappa t}{KR} + \frac{1}{4}\frac{FR}{K} - \frac{1}{4}\frac{FR}{K} = T_0 - 2\frac{F\kappa t}{KR}. \tag{42}$$

The temperature difference is then

$$\begin{aligned} \triangle T(t) = T(R,t) - \overline{T}(t) &= \left(-\frac{1}{2} + \frac{1}{4}\right)\frac{FR}{K} + 2\frac{FR}{K}\sum_{n=1}^{\infty} e^{-\kappa\alpha_n^2 \frac{t}{R^2}} \cdot \frac{1}{\alpha_n^2} \\ &= \frac{FR}{K} \cdot \left(-\frac{1}{4} + 2\sum_{n=1}^{\infty} \frac{1}{\alpha_n^2} e^{-\kappa\alpha_n^2 \frac{t}{R^2}}\right). \end{aligned} \tag{43}$$

Thereafter the maximal achievable temperature difference is $|\triangle T|_{max} = \frac{1}{4}\frac{FR}{K}$. It is theoretically achievable in a time $t = \infty$.

The dependency of $\frac{\triangle T}{\frac{FR}{K}}$ on the dimensionless time $\kappa\frac{t}{R^2}$ is shown in following figures. The vertical axis is negative by both figures.

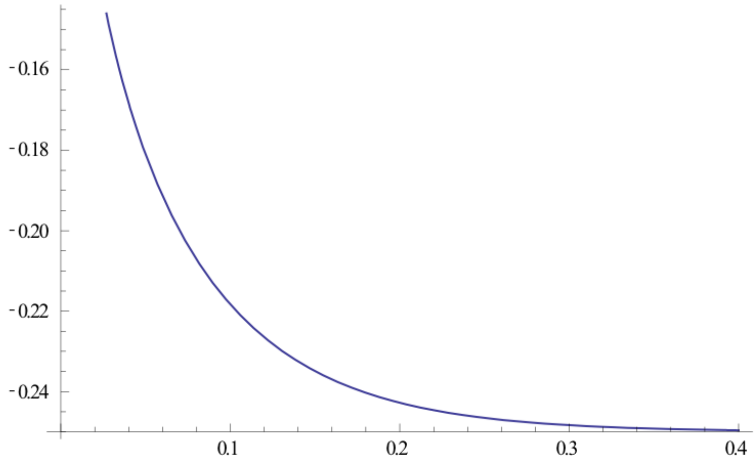


Figure 19: Dependency of $\frac{\triangle T}{\frac{FR}{K}}$ on a dimensionless time $\kappa\frac{t}{R^2}$

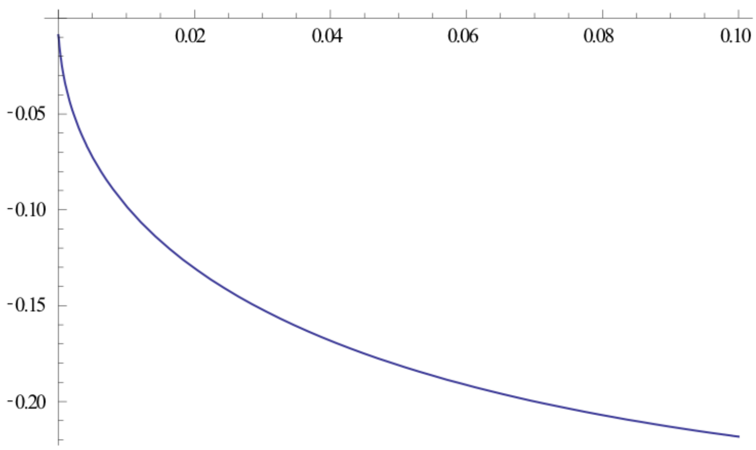


Figure 20: Dependency of $\frac{\triangle T}{\frac{FR}{K}}$ on a dimensionless time $\kappa\frac{t}{R^2}$

The previous figures are certain kind of explanation of the Tab. 3. Hereby it is shown that a rotor permissible temperature difference can be at various temperatures and after expiry of a certain time the same.

Appendix E
Typical scope of a maintenance operations

Typical scope of maintenance operations

X	In responsibility of the steam turbine manufacturer's supply
O	In responsibility of the customer

1. Turbine and steam fittings of the turbine

SYSTEM	ACTIVITY	Mi.I.	Int.I.	Ma.I.
Steam turbine casing	Dismantling of steam pipelines from valves into nozzle chambers, removal of field instrumentation, opening of the turbine casings, dismantling of internal casing, guide wheel carriers, guide wheels and the rotor, glands.			X
Steam turbine casing	Checking of inner parts - checking endoscope		X	
	Visual inspection for leaks of steam during operation	X	X	
Flow part	Measurement of clearances (radial and axial position rotor against stator)			X
Rotor	Inspection of the surface of bearing journals		X	X
	Run-out check of selected points of the rotor according to the basic according to assembly description			X
	Cleaning of rotor - sandblasting with a smooth abrasive (protect journals of rotor, coupling faces and axial disc of rotor)			O
	NDT check of chosen parts			X
	Checking of condition of sealing edges and balancing weights			X
	Check of condition of teeth - toothed ring for measurement of speed at the front of the rotor.		X	X
Moving blades of the rotor	Checking of condition of the leading and trailing edges, sealing parts, bandages, pins and coarseness of flow part channels			X
The outer casing	Checking of deformation of parting plane			X
	Checking of diameters of the casing (checking of ovality)			X
	Checking for cracks, NDT			X
	Visual checking of condition of internal parts and droplet of erosion			X
The internal casing	Checking of deformation of parting plane			X

The internal casing	Checking of diameters of the casing (checking of ovality)			X
	Checking for cracks, NDT			X
	Visual checking of condition of internal parts and droplet of erosion			X
	Checking of condition of the centre pins and condition of guide plates			X
Guide wheels	Cleaning - sandblasting with a smooth abrasive (protect joint faces and sealing faces)			O
	Checking of the flow part, alignment of guide wheels, correction			X
	Checking for cracks, NDT			X
	Checking of condition of the leading and trailing edges of guide blades and the surface quality of flow canals			X
	Checking of condition of suspensions, sealing elements of the shaft and over bandage seal.			X
Glands	Cleaning - sandblasting with a smooth abrasive (protect joint faces and sealing faces)			O
	Checking of flat springs			X
	Checking of condition of suspensions, sealing elements of the shaft and over bandage seal.			X
	Checking of the flow part, alignment of glands, correction			X
	Checking for cracks, NDT			X
Connecting bolts and nuts of the turbine casings	Checking of condition thread , checking of extension length			X
	Checking for cracks, NDT			X
Safety rupture disc	Checking of the condition of safety membranes		X	X
Thermocouples of casings	Checking of the condition of thermocouples (we recommend replacement during Major inspection)			X
Water cooling output of turbine casing	Check the piping line and water spray nozzle inside of turbine part.			X
	Checking of water filter (cleaning if necessary)		X	X

Turbine control and emergency stop valve chambers (steam part)	Opening of valve chambers, dismantling of internal parts		X	X
	Checking of the condition of thermocouples (we recommend replacement during Major inspection)			X
	Checking of the condition of surface of sealing planes of the chambers		X	X
	Checking for cracks, NDT		X	X
	Connecting bolts and nut -checking of condition thread		X	X
Steam strainer	Checking and cleaning of strainer in ESV		X	X

Emergency stop valves (steam part)	Visual inspection for leaks of steam during operation	X	X	X
	Checking of mobility		X	X
	Checking of setting of the position sensors		X	X
	Checking of tightness of the spindle gland of the steam part – visual check			
	Replacement of spindle gland		X	X
	Tightening of graphite glands according to instructions		X	X
	Checking of the condition of surface of sealing planes of the cone and diffusor		X	X
	Checking for cracks, NDT - (cone, diffusor)		X	X
	Tightness of cones – grinding or lapping of sealing areas (cone and diffusor)		X	X
Control valves (steam part)	Visual inspection for leaks of steam during operation	X	X	X
	Checking of mobility		X	X
	Checking of setting of the position sensors		X	X
	Checking of tightness of the spindle gland of the steam part – visual check			
	Replacement of spindle gland		X	X
	Tightening of graphite glands according to instructions		X	X
	Checking of the condition of surface of sealing planes of the cone and diffusor		X	X
	Checking for cracks, NDT - (cone, diffusor)		X	X
	Tightness of cones – grinding or lapping of sealing areas (cone and diffusor)		X	X
LP control flap	Visual inspection for leaks of steam during operation	X	X	X
	Mobility check		X	X
	Check of condition of the sealing surfaces of the flap - inner part			X
	NDT check for cracks			X
LP emergency stop flap	Visual inspection for leaks of steam during operation	X	X	X
	Mobility check		X	X
	Check of condition of the sealing surfaces of the flap - inner part			X
	NDT check for cracks			X
Servo drive of control valve	Disconnecting and connecting of HP oil system		X	X
	Disassembly and assembly actuators		X	X
Servo drive of emergency stop valve	Disconnecting and connecting of HP oil system		X	X
	Disassembly and assembly actuators		X	X
Servo drive of control flap	Disconnecting and connecting of HP oil system		X	X
	Disassembly and assembly actuators		X	X

Servo drive of emergency stop flap	Disconnecting and connecting of HP oil system		X	X
	Disassembly and assembly actuators		X	X

2. Turbine Control and Protection System (TCS/TPS)

SYSTEM	ACTIVITY	Mi.I.	Int.I.	Ma.I.
Speed sensors	Checking of mechanical fastening and cleanness of actual sensors and the geared ring at the tip of the turbine rotor (the sensors work in the electromagnetic principle and possible metal particles held by the magnetic field can cause a failure), check of clearances between the sensors and the geared ring		X	X
Connecting conductors and connectors	Check whether conductors are mechanically damaged or their connection.		X	X
Speed display	Checking of display	X	X	X
	Cleaning of dust, checking of connection cabling	X	X	X
TCS/TPS Switchgears	Inspection of the TCS/TPS cubicles, electrical operational parameters measurement, checking of component parts function, cleaning	X	X	X
TCS/TPS controller	Maintenance of TCS/TPS – replacement of worn (e.g. batteries) a/o defective parts with new EXTRA SPARE PARTS if needed from the Customer's stock		X	X
Communication	Inspection of the communication link within TCS/TPS and the communication link with supervisory Distributed Control System (DCS).	X	X	X
Operator Stations	Inspection of data transfer between TCS/TPS and operator stations (OS), OS serviceability checking.	X	X	X
TCS/TPS relationships with the Electrical Part Protection System	Inspection of TCS/TPS relationships with the Electrical Part Protection System.		X	X
TCS/TPS relationships with the Vibration and Displacement Monitoring System	Inspection of TCS/TPS relationships with the Vibration and Displacement Monitoring System.		X	X
Other inspection and service activities	Other inspection and service activities concerning TCS/TPS in accordance with OEM manual.	X	X	X
Protection loops	Inspection of selected TG protection loops.		X	X
Turbine start-up	Technical assistance during TG start-up.		X	X
Maintenance Report	TCS/TPS Maintenance Report elaboration and handover.	X	X	X