Tampereen yliopisto

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BALANCING FACILITY OF POWER PLANT TURBO GENERATOR ROTORS

Economic effects of balancing facility dependability

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Master of Science Thesis
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ABSTRACT

Roni Juutinen: Balancing Facility of Power Plant Turbo Generator Rotors. Economic effects of balancing facility dependability
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An existing balancing facility has been in operation for decades and it needs to be modernized. High level of dependability is required to enable proficient project delivery. Valuating dependability is one of the ways to evaluate the need for modernization of software and components.

The main objective of this thesis is to investigate how the dependability of the balancing facility could be increased and what is the economic effect of these changes. In the beginning of the project, the objective was to do benchmarking of other balancing facilities and balancing software, but this was not possible due to current travel restrictions by the Covid-19 pandemic. However, the updating requirements for the software are discussed, and other recommendations for improving dependability are given.

The balancing facility consists of two balancing machines. The smaller one is used for rotors weighting under 30 tons and the larger one is for rotors up to 110 tons. The balancing software was previously updated in 2008. Evaluation of the dependability is done by inspecting spare part availability for the machines, condition monitoring and critical requirements of the software. Investigations are done by interviewing and literature survey.

According to the research, the dependability of the balancing facility can be increased by preferring preventive maintenance to corrective maintenance. To move towards preventive maintenance, it is recommended to start measuring vibrations of rotating machinery periodically. Another action to increase dependability is to provide for a backup computer for the balancing machine to decrease the downtime significantly, if the currently used computers would malfunction. Finally, an internal standard and documentation for the balancing procedure would summarize the important requirements, limits, and work phases in one document. Therefore, it is recommended to compile an internal standard for balancing procedures.

In conclusion, this thesis analyses the different components for the modernization of the balancing facility. There is no need for updating the balancing software based on the current needs. Benchmarking of other balancing facilities is highly recommended after the traveling restrictions of the Covid-19 pandemic are discharged.

Keywords: Rotor balancing, Balancing facility, Economic effect of dependability

TIIVISTELMÄ

Roni Juutinen: Voimalaitoksen turbogeneraattorien roottorin tasapainotuslaitos. Käyttövarmuuden taloudelliset vaikutukset tasapainotuslaitoksessa.
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Modernisointi on ajankohtaista vuosikymmeniä käytössä olleelle tasapainotuslaitokselle. Projektien toimituksen sujuvuuden takaamiseksi on tärkeää, että tasapainotuslaitoksen käyttövarmuus on mahdollisimman korkea. Käyttövarmuuden arviointi on yksi tapa tarkastella ohjelmistojen ja komponenttien päivitystarvetta.

Tämän työn päätavoitteena on tutkia, kuinka tasapainotuslaitoksen käyttövarmuutta voidaan kasvattaa ja kuinka nämä muutokset vaikuttaisivat laitoksen talouteen. Projektin alussa tavoitteena oli mitata muiden tasapainotuslaitosten ja saatavilla olevien tasapainotusohjelmistojen suorituskykyä, mutta tämä ei ollut mahdollista Covid-19 -pandemian aiheuttamien matkustusrajoitusten vuoksi. Työssä kuitenkin tarkastellaan tasapainotusohjelmiston päivittämisen vaatimuksia ja annetaan suosituksia käyttövarmuuden kasvattamiseksi.

Tasapainotuslaitos koostuu kahdesta tasapainotuskoneesta. Pienempää käytetään enintään 30 tonnia ja suurempaa enintään 110 tonnia painaville roottoreille. Tasapainotusohjelmisto on edellisen kerran päivitetty vuonna 2008. Käyttövarmuuden arviointi perustuu tasapainotuslaitteiden varaosien saatavuuden, kunnonvalvonnan sekä tasapainotusohjelmiston kriittisten vaatimusten tarkasteluun. Tutkimus tehdään haastattelujen ja kirjallisuuskatsauksen avulla.

Selvityksen perusteella käyttövarmuuden kasvattamiseksi tulisi pyrkiä ennakoivaan kunnossapitoon korjaavan kunnossapidon sijaan. Ennakoivan kunnossapidon saavuttamiseksi on suositeltavaa aloittaa pyörivien koneiden värähtelytasojen mittaus säännöllisesti. Toisaalta käyttövarmuutta voidaan kasvattaa hankkimalla tasapainotuskoneille varatietokone, jonka olemassaolo lyhentää merkittävästi häiriöaikaa, jos toinen nykyisistä tietokoneista rikkoutuisi. Tämän lisäksi sisäinen standardi ja dokumentaatio tasapainotusprosessista tiivistäisi tärkeät vaatimukset, raja-arvot ja tasapainotusprosessista yhteen asiakirjaan. On siis suositeltavaa koostaa sisäinen standardi tasapainotusprosessista.

Tämä työ analysoi tasapainotuslaitoksen modernisointiin vaikuttavia tekijöitä. Nykyisillä vaatimuksilla ei ole tarpeen päivittää tasapainotusohjelmistoa. On kuitenkin erittäin suositeltavaa arvioida ja vertailla tasapainotusohjelmistojen suorituskykyä, kun Covid-19 -pandemian aiheuttamat matkustusrajoitukset purkautuvat.

Avainsanat: roottorin tasapainotus, tasapainotuslaitos, käyttövarmuuden taloudelliset vaikutukset

PREFACE

This thesis concludes my Master of Science studies at Tampere University in Energy engineering. Thesis is done for Fortum eNext Turbine and Generator Services. I wish to thank my employer for giving me an opportunity and support to do the thesis alongside the work.

First, I want to thank my supervisor DSc Henrik Tolvanen for smooth collaboration and support during the work. My supervisor at work MSc. Jari Tenhunen gave me detailed and thoughtful comments during the project, thank you for those. Also, thank you to other colleagues for giving great support. Even if most of last year was spent at home it has been quite a busy year and I want to thank friends and family for giving support and helping to relax.

Special thanks to my wife for amazing support.

Lastly, for all the others who were not mentioned:
To whom it may concern, ***thank you***!

Tampere, 29.3.2021

Roni Juutinen

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LIST OF SYMBOLS AND ABBREVIATIONS

1X	1 times rotor speed, synchronous
CBM	condition-based monitoring
DOF	degree of freedom
HP turbine	high pressure turbine
LP turbine	low pressure turbine
MP turbine	medium pressure turbine
RMS	root mean square
$A0$	initial vibration
$A1$	correction plane 1 vibration
a	acceleration
A	vibration amplitude
α	influence coefficient matrix
c	viscous damping ratio
C	response
ΔA_1	effective vector
d	displacement
$\dot{d}$	first derivative of displacement
$\ddot{d}$	second derivative of displacement
e_{per}	permissible residual specific unbalance
f	frequency
F	force
F_c	damping force
f_d	frequency of the dampened vibration
F_k	spring force
f_n	natural frequency
F_t	force at the highest operating speed
G	balance quality grade
H_c	correction mass distribution ratio
k	spring constant
m_1	mass for location 1
m_2	mass for location 2
m	mass
m_b	balancing weight
m_t	trial weight mass
m_r	rotor mass
m_c	number of correction planes
n	number of measurement points
Ω	angular velocity of service speed
O	original run
rpm	rotation speed
r_t	radius of trial weight
θ	phase angle
t	time
T	run with the trial weight
U_{per}	permissible residual unbalance
v	velocity
$\dot{v}$	first derivative of velocity
W_c	matrix of correction masses
W_{ci}	correction mass
W_{tm}	trial weight mass phase

x	displacement of spring
$\dot{x}$	first derivative of displacement
$\ddot{x}$	second derivative of displacement
$\angle$	angle

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1. INTRODUCTION

Electricity consumption has increased continuously over the years, and in 2018 worldwide electricity generation was around 26.6 petawatt-hours, from which a great majority was produced with turbine generators [16]. Turbine generators are converting energy of the fluid into rotating energy and further into electricity. For example, in steam turbine machines water is heated into steam which expands through turbine releasing a part of its energy. This energy is converted into rotating energy.

Turbine generators operate at high rotating speeds, which in this thesis means operating speed of 1500 rpm or higher. In the worst case the failure for such a large and high-speed rotating machine can have significant consequences for the turbine generator operation as well as personnel safety. Therefore, preventive maintenance is well suited for turbine generators to avoid unexpected failures.

Unbalance is one of the common issues with rotating machinery. In the rotating machines unbalance means that the mass center differs from the geometric center of the rotating shaft. Every rotating machine has unbalance at some level since for example, the parts have manufacturing tolerances, and the materials are never completely homogenous. [2] When balancing is required the objective is to minimize the unbalance and at least fulfill the requirements defined in the applicable standard [9]. Balancing is typically performed at the balancing facility during the major overhaul of the turbine generator at least if the overhaul includes work that can change the unbalance distribution of the rotor. Limits for permissible unbalance are set in the standard. [9] Successful balancing can be verified by measuring the vibration levels of the rotor.

The balancing is only one small part of the major overhaul and the schedules are usually rather tight. The balancing is one of the last work phases before the rotor is installed back to the generator and thus, it is hard to catch up the schedule after balancing if there are delays. Since, the ability to perform as and when required is critical for the balancing machine. This ability is called dependability. [6]

Improvement of balancing facility dependability consists of analyzing the components of the balancing machinery, processes, and methods to perform balancing, and actions and requirements for fault situations. Prevention of the negative impact of faults improves dependability, but also, the improvement of daily practices makes the process more efficient.

This thesis investigates how the dependability of an existing balancing facility could be improved. The investigated facility has been in operation for several decades and is now being modernized. The focus is on analyzing the balancing software and dependability of the mechanical components. Moreover, the goal is to inspect the need for software update and methods to decrease unplanned outages of the balancing machinery.

The research methods in this thesis combine interviews of the internal staff and software suppliers with literature analysis of current standards and best practices. If the design phase of the facility is excluded, this is the first time when the dependability of this balancing facility is observed. Therefore, the investigation is limited to a general level and it gives recommendations for concrete actions and further inspections.

The research questions in this thesis are:

1. What are the advantages of preventive maintenance of generator rotors to the owner of the powerplant?
2. What kind of effects does the unbalance cause to normal operation of the turbine generator and when is balancing required?
3. How can dependability of the balancing facility be increased?
4. How improving the reliability of the balancing facility would impact the economy of the balancing projects?

Chapter 2 covers the balancing of the rotating machinery in power plants. The advantages of preventive maintenance are introduced in Section 2.1 where different types of maintenance are presented. The second research question is covered in Chapter 3 where vibration behavior of turbine generator is discussed. Chapter 4 introduces the status of the balancing facility and the methods for investigations are defined. The methods to increase dependability are presented in Section 4.3. In Chapter 5, the results of the investigation are presented and the findings for increasing dependability are presented in Section 5.3. The economic effects of dependability are analyzed in Section 5.4. Finally, the conclusions and recommendations are summarized in Chapter 6.

2. BALANCING OF ROTATING MACHINERY IN POWER PLANTS

This chapter introduces heat power plants and describes in more detail the meaning and justification of balancing, and the balancing process. As stated in the introduction chapter, a large amount of electricity is produced with turbine generators. In Figure 1, electricity and heat generation chains are visualized.

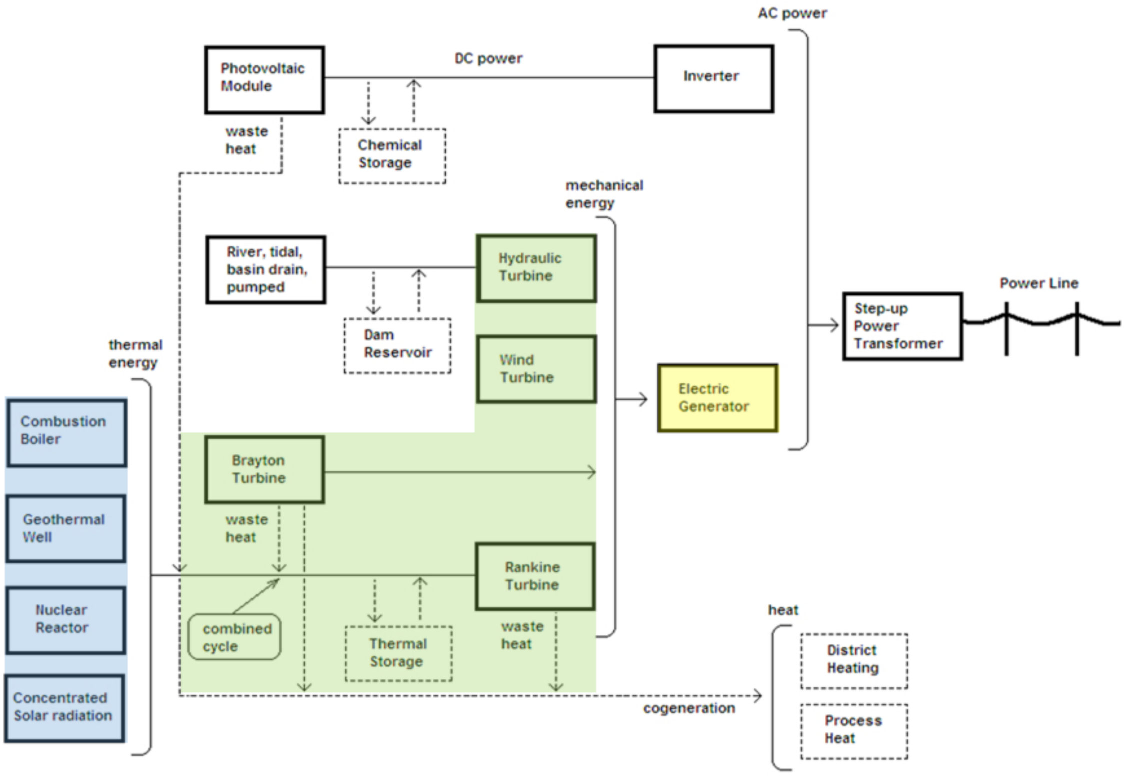

Figure 1: Heat and electricity production chain. Modified from [14]

Figure 1 shows that there are multiple ways of producing electricity and heat. In the left there are thermal energy sources highlighted with a blue background. Different turbine types are on the green background. Turbines convert energy of the fluid into kinetic energy of turbine rotor, and they are coupled with generators (yellow background). The generator further converts kinetic energy into electricity. Electricity is supplied to customers through power lines. This thesis focuses on turbo generators which are used with either Brayton or Rankine turbine. An example schematic of a power plant using Rankine turbine and turbo generator is presented in Figure 2.

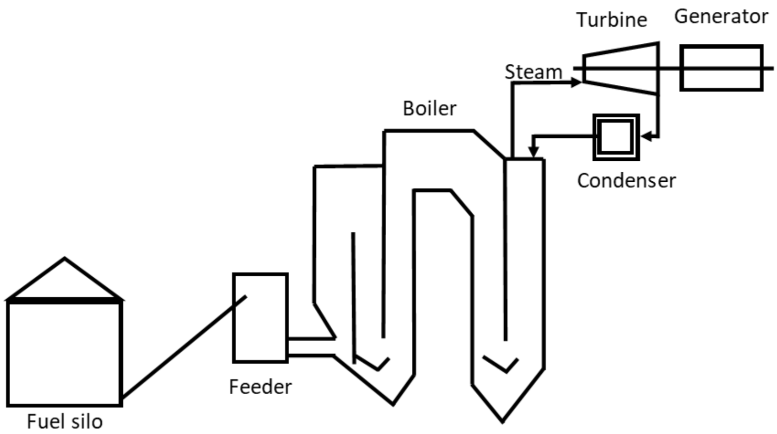

Figure 2: Example of a power plant scheme using Rankine turbine. Modified from [21]

Figure 2 shows the major components of a power plant. Fuel is transported into the fuel silo from where it is fed to the boiler through a feeder. In the boiler fuel is burned which generates large amount of heat. Closed water circuit has a large amount of tubing inside the boiler and when water flows through, it transforms into steam. Next, the steam goes through the turbine where it releases some of its energy into rotating energy of the turbine rotor when it expands through the turbine blading. Steam is then condensed back into water in the condenser from where it is ready to go back into the boiler. Turbine rotor is connected into the generator rotor through a coupling or a gearbox. The generator transforms kinetic energy into electricity.

First, the next sections present the basic principles of maintenance to help the reader to understand how balancing of a generator rotor is linked to the operation of the turbine generator. It is important to understand how the overhauls are planned. Next, the theory of vibration is presented and how vibration condition monitoring is performed.

2.1 Types of maintenance

It is a common assumption that maintenance means solely repairing faulty machines, but in modern understanding this is an incomplete description. Maintenance is upkeeping the machines' ability to perform as they are meant. Neglecting maintenance leads to malfunctions which usually end up costing more than a well-planned maintenance. [4]

Maintenance can be categorized in multiple ways. For example, in standard SFS-EN 13306 [7] maintenance is categorized as presented in Figure 3.

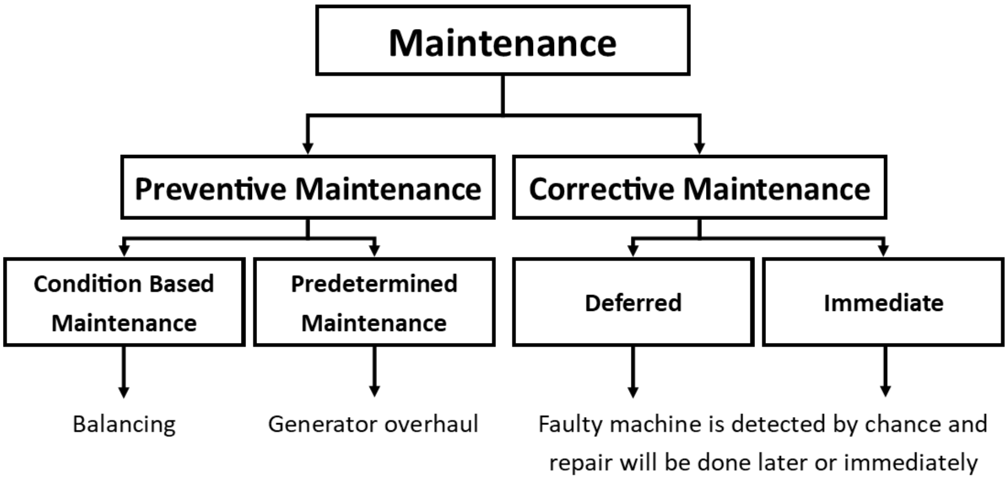

Figure 3. Types of maintenance, modified from SFS-EN 13306 [7]

On the bottom row of the Figure 3, examples of each maintenance group are presented. Preventive maintenance represents operations made before a fault is detected and corrective maintenance means that the fault is already detected before the action is made. These types of maintenance are described more in the following sections.

2.1.1 Preventive Maintenance

In the past, preventive maintenance has been the most used maintenance philosophy in maintaining the critical parts of power plants. In this case, a definition for the critical parts is that reliability is required from a machine and unplanned interruptions are not allowed. During the overhaul, the turbine generator itself and related machinery are all shut down together, and therefore the maintenance can be performed without extra loss of production. [5]

Planned maintenance is a combination of predictive maintenance and condition-based maintenance. Planning maintenance requires knowledge from reliability of the machine to estimate how long it can run between overhauls without problems. It is important to evaluate, for example, if there have been events affecting the condition of the machine or if condition monitoring has been giving signs that overhaul should be rescheduled. [5]

Before maintenance starts, all the repairs and inspections are planned. When an overhaul is planned it is important to take the machine history into account and use the knowledge from similar machines to estimate the required actions. Since it is not possible to notice all problems when the turbine generator is running it is quite common that inspections made during the overhaul will lead to additional works. [5]

Condition-Based Maintenance

As shortly mentioned in the previous section, condition-based maintenance (CBM) is heavily linked into preventive maintenance. CBM means that operations are made directly according to the condition of the machine. This means that regular condition monitoring is necessary to be aware of the condition of the machine.

The monitoring can be done by using real-time monitoring systems as well as periodic testing routines. For example, voltage, current, vibration, and temperature of core/windings can be measured continuously during the operation of a large generator. Online monitoring increases the initial capital investment required for the instrumentation, but the investment will provide more reliable and less expensive operation. [5]

For example, balancing is part of condition-based maintenance. The unbalance can be monitored with the vibration measurements, and if required, the balancing will be performed to reduce the unbalance of the machine. High unbalance causes cyclic stresses to the components that might cause fatigue failure.

Predetermined Maintenance

Predetermined maintenance is also part of preventive maintenance together with the CBM. The predetermined maintenance is not relying on the actual condition of the machine but rather using knowledge of failure mechanisms as well as mean time to failure statistics. This information is used to calculate the schedule for maintenances. [5]

Problem in predetermined in maintenance is that the actual condition of the machine is not considered. This might lead to unnecessary maintenance as well as failures which could have been avoided if the condition of the machine would have been considered. However, if the manufacturer knows the failure mechanisms well and the machine is new, it is possible to plan the required maintenances rather accurately.

2.1.2 Corrective maintenance

Corrective maintenance means that the machines are fixed only if they break. This type of maintenance is suitable for small and non-critical machinery, especially if the repair is more costly than the replacement of the device. Corrective maintenance can be immediate, which means a repair is done right after the broken equipment is noticed or the repairment is referred into a suitable maintenance time. For larger or critical machines this could be a costly option, since waiting for the spare parts extends downtime and decreases productivity.

In some cases, the operating costs must be cut down and maintenance is neglected. In short term that can decrease costs of maintenance but in long term maintenance backlog increases and the machines are run towards a breakdown. [5]

2.1.3 Maintenance of turbine generator

Turbine generators are critical machines, and their maintenance is carefully planned. Planned maintenance is a combination of predictive maintenance and CBM. For new turbine generators, the time intervals between overhauls are normally relied on the experience of the manufacturer for the specific machine. After some years in operation, the maintenance should move towards CBM instead of predetermined maintenance. It is also very important to coordinate the turbine and generator maintenance plans. Turbine generator overhauls can be divided into major, minor, and annual overhauls based on their scope, which is presented in Figure 4. [3, 5]

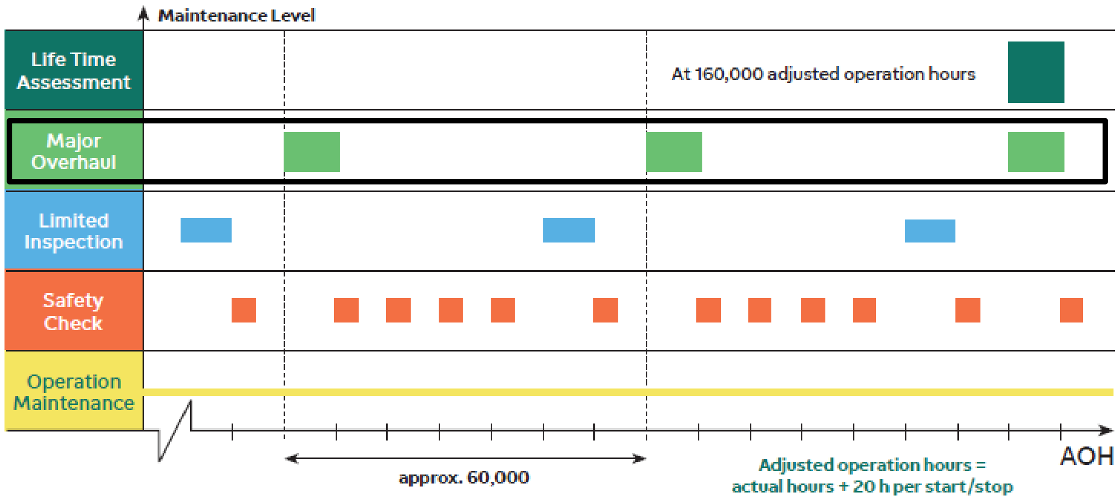

Figure 4: Example maintenance schedule of a generator, modified from [3]

A major overhaul is highlighted in the figure because rotor balancing is usually a part of the major overhaul and it is necessary if the overhaul requires such work at the rotor which can change the unbalance distribution. The major overhaul is the biggest and most important of the overhauls, where the machine is completely disassembled allowing comprehensive inspections and reconditioning. The information gathered from visual inspections and measurements are analyzed to identify possible faults. The major overhaul consists for example removing the bearings, winding covers and rotor. The components are visually inspected, and diagnostic testing is performed for the critical components. Balancing of the generator rotor in a balancing facility is part of a major overhaul. [3]

Minor overhauls are the most effective when they are performed 1 or 2 years before a major overhaul. This helps planning the major overhaul, since it gives information about the condition of the machine. In minor overhauls, the machine is only partly disassembled and the key-difference between major and minor overhauls is that the rotor is not removed during a minor overhaul. Visual inspections are carried out for accessible parts and diagnostic testing is performed but not as comprehensively as in the major overhaul.

The annual overhauls are also referred as safety checks which describes its coverage well. Only the inspection hatches are opened, accessible parts are visually inspected, and diagnostics are performed. Big changes in the condition of the machine can be detected in annual overhauls. [3]

2.2 Condition monitoring

Condition monitoring is an important part of the CBM. The objective of condition monitoring is to improve and maintain the efficiency of the equipment. This is done by acquiring data from the machine behavior, for example, temperatures and vibrations. These parameters are used to define the condition of the machine and plan the overhauls according to the condition. Having data from machines' condition makes it easier to plan the revisions. Overall, the goal is to save costs by optimizing maintenance actions using the information about the condition of the machine.

In industry, it is typical that the most critical machines are under a protection system. Typical systems are continuously measuring multiple parameters and they are set to give warnings to the operators if a parameter exceeds the warning limit and the system can even stop the machine automatically if the measured values exceed the limits. Warnings and trip limits should be set independently for each machine based on the design, standards, and their characteristic behavior. Vibration is one of the parameters that gives information about the condition of the machine.

Vibration measurements

Vibration measurements are part of condition monitoring and most of the modern powerplants have permanently installed vibration systems in critical machines. Most of the turbine generators have a vibration monitoring system but the coverage varies heavily.

Typically, these systems only show overall level of vibration making more detailed analysis impossible without additional measurements. To get a better understanding about the vibration behavior, additional measurements should be done regularly. Analyzing vibration data is useful and it is possible to notice bearing faults even before they are visible in overall vibration levels. For important machines, for example, turbine generators, it is recommended to measure the vibrations in coast down before overhauls and in run up after the overhaul. Coast down measurements give data of the current condition of the machine and this information might reveal problems which should be taken care of during the overhaul. [2]

Vibration measurements are not limited to just turbine generators, also other critical machines are typically measured to get information about their condition. Vibration analysis can reveal various problems. Examples of these problems are listed below [17]:

- Unbalance
- Looseness
- Bearing failures
- Resonances and natural frequencies
- Electrical motor faults

Vibrations can reveal common problems in rotating machines which helps to notice issues before they cause downtime. If the vibrations are measured frequently, it is possible to notice problems in very early stage and their development can be monitored. For example, measurements might show if the interval of bearing lubrication is too long and needs to be adjusted. Optimizing lubrication intervals takes time but it results in longer lifetime for the bearings and reduces costs by reducing unnecessary bearing faults. Furthermore, condition monitoring helps to plan upcoming maintenances. [17]

2.3 Balancing of rotating machines

Balancing means that the center of gravity is moved closer to the center of rotation to reduce vibrations caused by the unbalance. Mathematical models for balancing assume that the vibrations are behaving coherently when adding or removing balancing weights. This means that the amplitude of the vibrations caused by the unbalance is proportional to the level of unbalance, and a phase shift of the vibration is corresponding to the angular position change of the unbalance. There are situations where incoherent behavior appears and balancing of the rotor requires more balancing runs, since the rotor is not behaving like the mathematical model assumes. [8]

In the following Sections, the different types of unbalance are explained and then a simple example of how balancing is performed is shown to give better understanding of the balancing procedure. Also, the influence coefficient method and modal balancing are presented.

Unbalance

Unbalance is a common reason for vibration problems. Unbalance occurs when the center of the gravity and the center of rotation are not coincident. This causes a force to the system which can be noticed as vibration. If the dynamic stiffness of the system is known, it is possible to calculate the vibration caused by unbalance by using Equation 1. [2]

$$Vibration = \frac{Force}{Dynamic\ Stiffness} \tag{1}$$

Unbalance is attached to the rotor which means that the force caused by the unbalance is synchronous with rotating frequency (1X). In a totally linear system, a force which appears regularly once per revolution would cause vibrations at 1X and nothing else, but stiffness of the rotor system is not linear. For example, stiffness of the fluid-film bearings strongly increases at higher eccentricity ratios and a rotor-to-stator rub can increase stiffness of the rotor system. These phenomena are causing nonlinearity and they can cause harmonics of 1X vibration. More details about vibration theory are presented in Chapter 3. [2]

The simplest type of unbalance is static unbalance, which means that there is an unbalance equivalent to a heavy spot at a single point. In theory, static unbalance will show up even when the rotor is not rotating, and the heavy spot will turn facing downwards when the rotor is on very low friction bearings. However, generator rotors have so little unbalance compared to their weight that the static unbalance cannot be determined from a stationary rotor. An example schematic of static unbalance is shown in Figure 5. [8]

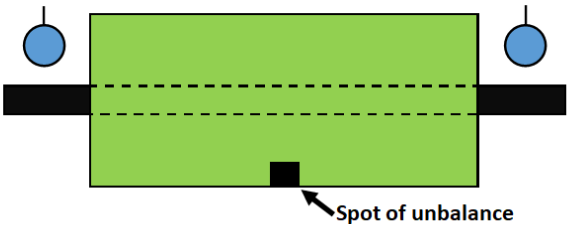

Figure 5: Example schematic of static unbalance.

In Figure 5, the shaft is presented as a black shaft, the rotor is presented as a green box and unbalance is presented with small black bow in bottom of the rotor. Blue circles with black indicators represent phase, thus in this case both ends of the machine have the same phase. [8]

All unbalance is not notable when the rotor is stationary, but it can cause problems during rotation. Two equal unbalance spots on the opposite ends of the rotor with 180° phase difference are countering each other. This is called couple unbalance and an example is shown in Figure 6. [8]

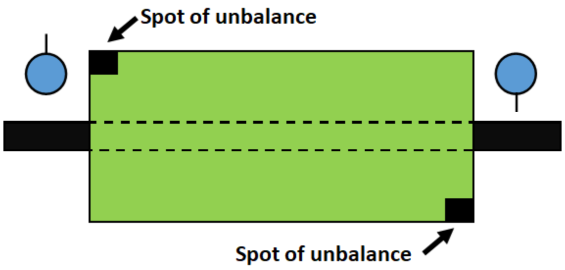

Figure 6: Example schematic of couple unbalance.

In practice, pure couple unbalance is rarely found but understanding the phenomenon is helpful in diagnosing dynamic unbalance. Dynamic unbalance is a combination of static and couple unbalance. An example of dynamic unbalance is shown in Figure 7.

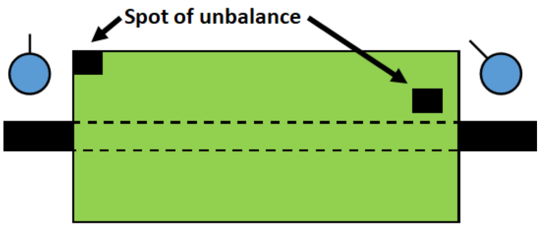

Figure 7: Example schematic of dynamic unbalance.

In dynamic unbalance the mass centerline does not coincide with the shaft's geometric centerline.

Balance grading

It is not possible or even feasible to manufacture rotors without any residual unbalance and therefore, it is important to have a grading system for the unbalance. In ISO 21940-11 [9], the balance quality is determined with balance quality grade G. The standard defines G as follows: "The balance quality grade G is designated according to the magnitude of the product $e_{per}\Omega$, expressed in mm/s". The standard includes a table where guidance for recommended balance quality grades is given for different types of machinery. For example, for gas turbines and steam turbines the guided balance quality grade is G 2,5 which means that the allowed magnitude for $e_{per}\Omega$ is 2,5 mm/s.

$$e_{per} = \frac{G}{\Omega}, \quad (2)$$

where G is balance quality grade and Ω is the angular velocity of the service speed in rad/s. Next, the permissible residual unbalance U_{per} can be calculated as shown in Equation 3.

$$U_{per} = e_{per}m, \quad (3)$$

where m is the mass [kg] of the rotor. Balance quality grade (G) is a worldwide used grading system which gives classifications of balance quality for typical machinery types. The standard states the guidelines for balancing grades for different types of machines. For example, for steam turbines the recommended balancing quality is G 2.5. [9]

2.4 Balancing methods

Single-plane balancing

Rotors that do not fill the unbalance tolerances need correction, which means either adding or removing weight. The weights are installed on correction planes which in rotors are typically a machined groove or threaded holes where the weights can be installed in the required positions and then locked into place. The amount of the correction planes depends on the design of the rotor. Single-plane rotors have rigid behavior in their operating speed range.

Single-plane balancing means that the rotor has only one plane for inserting balancing weights. The first step in balancing is to run the rotor and do vibration and phase measurements. More about vibration measurements is presented in Chapter 3. These measurement results are typically referred as the original condition of the machine. Balancing is usually done at the normal operating speed of the machine. [12, 13]

Balancing requires that we must discover the weight we need to add and the location where it must be added. A general guideline to choose the trial weight is to produce a force equal to 10 % of the rotor weight. This guideline is suitable for flexible rotors, but for rigid rotors the mass should be 2–3 times larger. To calculate the mass of the trial weight (m_t), we must define the force (F_t) caused by a mass at the highest anticipated operating speed: [12]

$$F_t = m_t r_t \Omega^2 \tag{4}$$

$$F_t = 0.1 m_r g \tag{5}$$

Where m_t means mass of the trial weight (kg), r_t means the radius of trial weight (m), Ω means rotating speed (rad/s). In Equation 5 m_r means rotor mass (kg). This gives the mass of trial weight [12]:

$$m_t = \frac{0.1 m_r g}{r_t \Omega^2} \tag{6}$$

The trial weight should cause at least 20 % change in the amplitude or 20° change in the phase angle. If the change is smaller, a heavier trial weight should be used, or the position of the trial weight should be changed. [18]

When selecting the position for the trial weight, it is important to know the resonance speed of the machine since it will have an impact on 1X phase reading as shown in Table 1. [2]

Table 1: Phase of the vibration relative to the heavy spot. When measuring displacement runout compensation must be used to filter out surface imperfections. [2]

	Low speed	Resonance speed	High speed
Displacement	0°	90° Lag	180° Lag
Velocity	90° Lead	0°	90° Lag
Acceleration	180° Lead	90° Lead	0°

With this information and vibration measurements from the machine at the original state, we can estimate the location of the heavy spot and install the trial weight to the opposite side. Figure 8 presents an example bode plot which shows if the machine is operating at low, resonance or high speed.

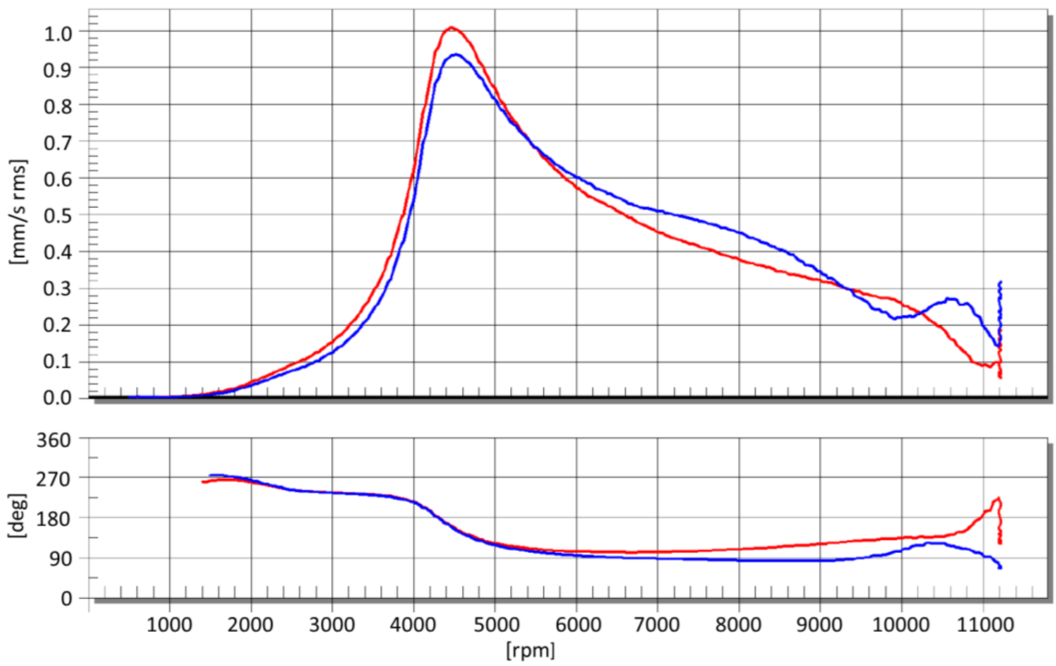

Figure 8: Example bode diagram for an arbitrary measurement.

Figure 8 presents the rotating speed at horizontal axis (rpm), and amplitude and phase in vertical axis (mm/s and °, respectively). In this example, the operating speed is well above the critical speed and the phase change is clearly notable when the critical speed is passed. The heavy spot seems to be at 180° and the trial weight should be installed at the opposite side at the 0° mark.

Single plane balancing

Next, an example of how balancing is done using a graphical method is presented. First, a polar plot should be sketched, and an external reference direction should be marked. Then, the degree marks should be pointed out according to the position of the tachometer. In Figure 9, an example of the polar plot after the first and second run is presented.

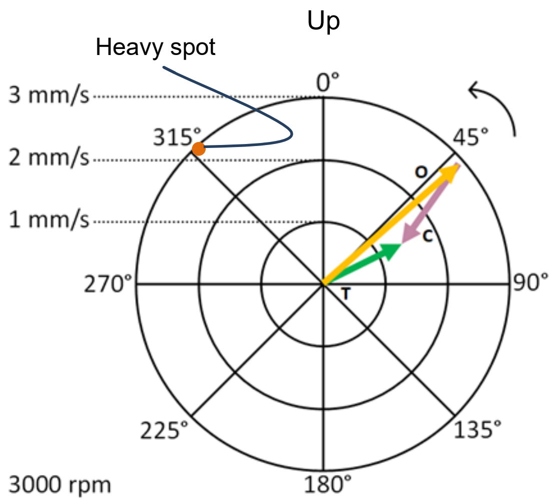

Figure 9: A polar plot with 1X filtered original vibration vector (O), vector of the trial weight (T), and response due to the trial weight (C).

In Figure 9, the phase angles are presented with 45° gaps and the amplitude is presented with circles with 1 mm/s gaps. Estimated location of the heavy spot is marked with an orange dot located at 318°. 1X filtering means that everything but the vibration at the rotating speed is filtered out. This is done since the unbalance is causing vibration at the rotating frequency. If the measurements are made with displacement sensors it is also important to do slow roll compensating. Slow roll compensating means that the shaft is measured at slow speed allowing to study the topography of the measuring area and then measured imperfections in the surface are removed from the signal. This way scratches or other small imperfections of the rotor surface do not affect compensated measurement results. In this occasion slow speed means rotating speed where significant unbalance forces do not exist. For example, normal speed range for slow roll compensating is about 200-400 rpm for the shaft lines that have normal operating speed of 3000 rpm. Table 2 presents the values used in the example.

Table 2: Values used in the example.

Original run (O)	2.9 mm/s ∠ 48°
Rotor mass (m_r)	2000 kg
Radius of trial weight (r_t)	350 mm
Mass of the trial weight (m_t)	57 g ∠ 138°
Run with the trial weight (T)	1.4 mm/s ∠ 64°
Response (C)	1.6 mm/s ∠ 214°

In Table 2 the required values to calculate the mass of the trial weight are presented and Equation 15 is used to calculate mass of the trial weight. In this example, the rotor is running well under the resonance speed and thus it is considered as a rigid rotor. According to Table 2, the phase is 90° lead since the measuring point is under the resonance speed and the measurements are presented as velocity. This means the heavy spot is around 318° and the trial weight will be placed to the opposite side at 138°. Also, the vibrations at the trial run and the system response to the trial weight are presented in Table 2. The response C is calculated using vector mathematics in Equation 7.

$$C = T - O \tag{7}$$

The next step is to calculate the balance weight. We assume that the system is linear and therefore, if m_t is doubled then the amplitude of C will also be doubled. Also, if m_t is rotated, then C will rotate, respectively.

$$m_b = \frac{-O}{C} m_t \tag{8}$$

$$\angle m_b = \angle O + 180° - \angle C + \angle m_t \tag{9}$$

where m_b is the required balancing weight when the trial weight is removed and $\angle m_b$ is its location.

The phases presented above are required for the balancing process. After the last step, the machine is run again, and the residual unbalance is calculated from the vibration measurement results. If the residual unbalance is higher than what is allowed according to the applied standard, then a new balance weight mass and a position is calculated to balance the machine more precisely. This will be repeated until acceptable vibration levels are achieved.

The limitation of the graphical method is that it is only effective for rotors with one balancing plane. For rotors with multiple balancing planes there are more efficient methods available, for example, the influence coefficient method.

Multiplane balancing

Modern turbomachinery rotors have usually at least two balancing planes and the rotors can run above the second or even higher vibration mode. Vibration mode means the vibration deflection shape of the rotor system. When the machine has multiple vibration modes under the operating speed, the balancing is usually more complex since multiple planes must be balanced simultaneously.

Rotors have axial, radial and torsional vibration modes but only radial vibration modes should occur during rotor balancing. Axial and torsion modes cannot be compensated with balancing and they should be observed already in design phase of the generator rotor. In the generator rotor configurations, most of the rotor mass is between the bearing centers. An example of modes is shown in Figure 10.

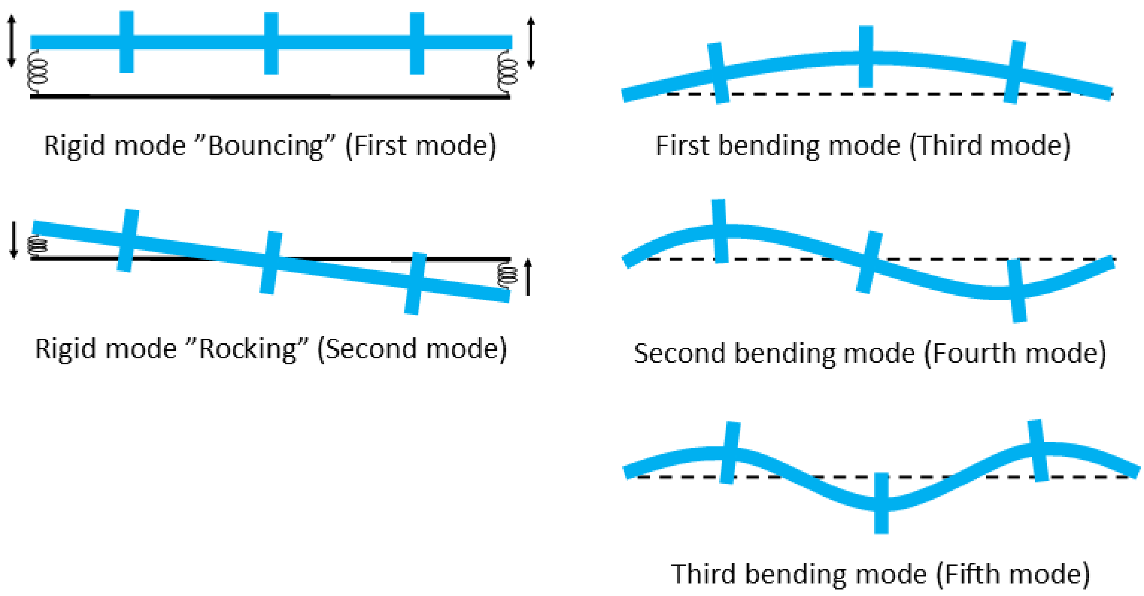

Figure 10: Vibration modes of a rotor. Modified from [11]

Large generator rotors have fluid film bearings on them and thus, they will act as rigid body at very lowest modes and then have a transition to flexible rotor behavior at higher modes. The points where minimum vibration occurs are called nodes and respectively, the points with the highest amplitude are called anti-nodes. [2]

Regarding the vibration modes it is important to remember that the first two modes are rigid modes and the bending modes come after these. This is important to keep in mind to avoid misunderstandings between the first vibration mode and the first bending mode.

If the rotor is flexible it means that it is operating above the first bending mode (third vibration mode) and it requires more than two balancing planes to balance it precisely.

For example, the third vibration mode is practically impossible to balance if the only balancing planes are at both ends of the shaft but there is no balancing plane in the middle. This is practically impossible, because the planes are at the node points, or very close to them, and thus adding a mass in the end of rotor has little or no effect in the third mode. Examples of how balancing weights affect to different bending modes are presented in Figure 11.

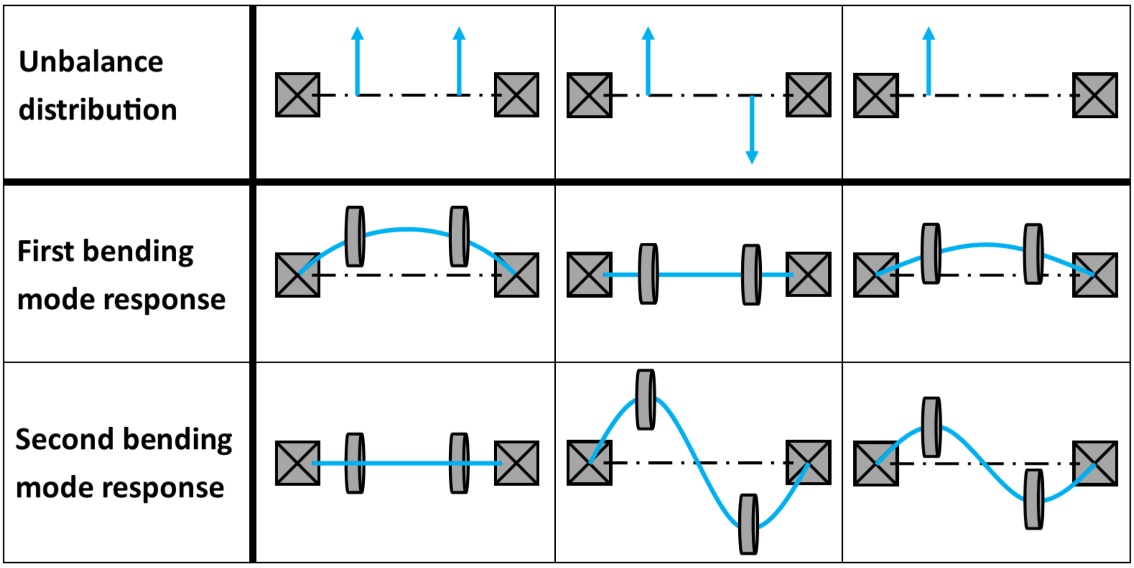

Figure 11: Unbalance distribution and mode excitation. Modified from [2]

In Figure 11, the first row shows with arrows how the unbalance is distributed in the shaft. The second row shows how the first bending mode responds to the unbalance distribution presented above. Further, the third row shows how the second bending mode responds. The axial position of the unbalance also effects to the mode response. For example, if there is unbalance at a single spot in the middle of the shaft it will excite the first bending mode but not the second mode, since the unbalance is located at the node of the second mode. This enables balancing only one specific mode without affecting the other vibration mode. Mode excitation must be considered during the design phase since the amount of required balancing planes depends on the existing vibration modes.

The flexible rotors should be high-speed balanced to correct each mode existing in operating speed range. Balancing should include a series of balancing speeds which are selected so that there is at least a balancing speed close to each resonance speed. After each mode is corrected, the remaining higher modes are corrected at the highest balancing speed. [10]

Balancing with Influence Coefficient Method

An influence vector is a complex number describing the changes of the 1X vibration of a machine when a balance weight is added. It is based on the same theory as the example shown above but it enables processing data for multiple planes. Commercial balancing systems use the Influence Coefficient Method since it is a powerful tool for multiplane balancing. [2] This Section explains how balancing is done using influence coefficients.

Typically, rotors with more than two balancing planes are flexible rotors. The system is measured at n measurement points and it has m_c correction planes for balancing and $n \geq m_c$. The initial vibration (A_0) is presented as $A_0 = amplitude \angle phase$ where the phase is measured at the rotational speed close to the critical speed. [8]

$$A_0 = [A_{10} \quad A_{20} \quad \cdots \quad A_{n0}]^T, \tag{10}$$

and then the vibration, with the trial weight $W_{t1} = mass\angle phase$ attached to the correction plane 1 is measured at the same rotational speed.

$$A_1 = [A_{11} \quad A_{21} \quad \cdots \quad A_{n1}]^T, \tag{11}$$

Next, the effect of the trial weight to the vibrations is calculated. This is called the effective vector ΔA_1.

$$\Delta A_1 = A_1 - A_0 = [A_{11} - A_{10} \quad A_{21} - A_{20} \quad \cdots \quad A_{n1} - A_{n0}]^T \tag{12}$$

The influence coefficient ($\alpha_{m_c n}$) is the effective vector per unit eccentricity. Influence co-efficients of the trial weight W_{t1} in correction plane 1 are:

$$\alpha_1 = [\alpha_{11} \quad \alpha_{21} \quad \cdots \quad \alpha_{m_c 1}]^T = \frac{\Delta A_1}{W_{t1}} = \frac{A_1 - A_0}{W_{t1}}, \tag{13}$$

After the influence coefficients are calculated for all planes it is possible to form the in-fluence coefficient matrix α:

$$\alpha = [\alpha_1 \quad \alpha_2 \quad \cdots \quad \alpha_{m_c}] \tag{14}$$

Balancing requires that a correction mass W_{ci} is attached to each correction plane to cancel the vibration.

$$-A = \alpha W_c, \tag{15}$$

where $W_c = \begin{bmatrix} W_{c1} & W_{c2} & \cdots & W_{cm_c} \end{bmatrix}^T$. If $n = m_c$ then correction masses can be solved from Equation 25:

$$W_c = -\alpha^{-1} A_0, \tag{16}$$

and if $n \geq m_c$ then the least square method can be used.

$$W_c = -(\alpha^T \alpha)^{-1} \alpha^T A_0, \tag{17}$$

and if $n \leq m_c$, then more sensors should be added if it is possible. If adding sensors is not possible and all planes are needed, then calculating correction mass is done by using ratio H_c. H_c describes how many times and how many degrees the trial mass has to be multiplied and advanced. Equation 25 is modified as follows: $\alpha \rightarrow \Lambda = [\Delta A_1 \quad \Delta A_2 \quad \cdots \quad \Delta A_{m_c}]$, $W_c \rightarrow H_c$ resulting:

$$-A = \Lambda H_c. \tag{18}$$

Then using equations 15 and 17, the correction masses W_c are determined as:

$$W_c = \begin{bmatrix} W_{t1} & 0 & 0 & 0 \\ 0 & W_{t2} & 0 & 0 \\ 0 & 0 & \cdots & 0 \\ 0 & 0 & 0 & W_{tm_c} \end{bmatrix} H_c \tag{19}$$

Equation 19 gives the correction masses and the phase for every correction plane. After installing the weights, the test run is done to confirm the balancing. [8]

Weight splitting

Balance weight installation locations depend on the rotor. In some machines, it is possible to locate the weight everywhere on the balancing plane and other machines can have evenly spaced holes for the weights. If the weights are installed at holes, then it might be necessary to split the weight if the calculated location is between holes. Weight splitting means that multiple weights can be installed, and their combined resultant is the same as the calculated balance solution. The masses and angles can be treated as complex numbers:

$$m_1 e^{j\theta_1} + m_2 e^{j\theta_2} = m e^{j\theta} \tag{20}$$

where indices 1 and 2 stand for available weight locations and variables without an index mean the calculated balance solution. Phase angles (θ_x) in this case are instrumentation convention phase lag angles. Equating real and imaginary parts gives:

$$\begin{bmatrix} cos\theta_1 & cos\theta_2 \\ sin\theta_1 & sin\theta_2 \end{bmatrix} \begin{pmatrix} m_1 \\ m_2 \end{pmatrix} = \begin{pmatrix} \mathrm{m}\cos\theta \\ msin\theta \end{pmatrix} \tag{21}$$

From where we can solve m_1 and m_2:

$$m_1 = m \left[\frac{\sin(\theta_2 - \theta)}{\sin(\theta_2 - \theta_1)}\right], \tag{22}$$

$$m_2 = m \left[\frac{\sin(\theta - \theta_1)}{\sin(\theta_2 - \theta_1)}\right], \tag{23}$$

which are the masses to be attached in the defined phase angles (θ_x). [2]

Some of the balancing software allow to define what are the possible locations for the balancing weights and the software automatically splits the weights accordingly. This makes the balancing process slightly faster and simpler. Automated weight splitting also decreases a risk for a human error.

3. VIBRATION OF TURBO GENERATOR

In this Chapter the vibration behavior of a turbo generator is introduced. First, the main parts of the turbine generator are presented. Then, typical vibration measurement methods in modern turbo generators are discussed.

3.1 Power chain of steam turbine

The basic principle is the same for all types of turbines, fluid containing potential and kinetic energy enter the turbine and when it passes by the turbine, a part of its energy transforms into rotation energy of the turbine rotor. High-capacity turbines use different turbine stages of steam turbines such as high-pressure turbine (HP turbine), mid-pressure turbine (MP turbine) and low-pressure turbine (LP turbine). The power chain of a steam turbine is presented in Figure 12.

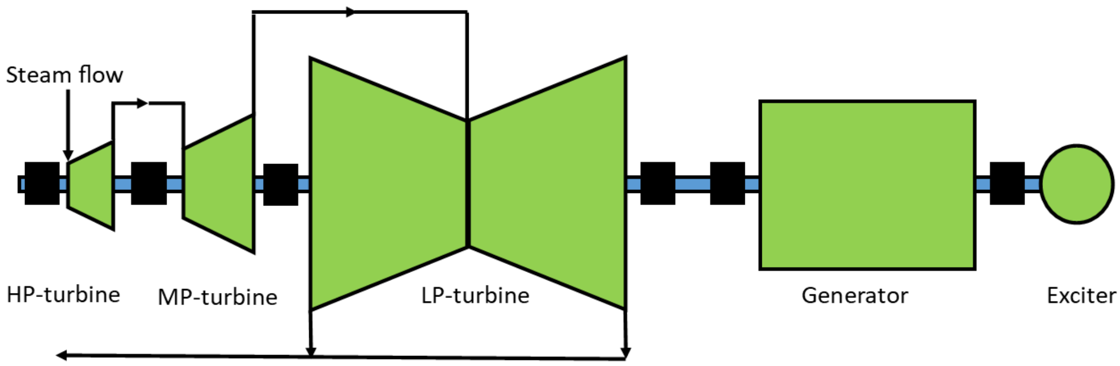

Figure 12: Power chain of steam turbine.

In Figure 12, the steam flow is presented with black arrows. First, steam flows into the HP turbine, next to the MP turbine and finally to a LP turbine. Black boxes on the shaft represent the bearings. The turbine shaft is connected to the generator shaft with either a clutch or a gearbox. Typically, gearboxes are only used in smaller machines.

Mechanical energy from the turbine is converted into electricity in a generator. Generators have two main components: a stator, and a rotor but both consist of multiple parts. The structure of the turbo generator and its typical failure modes are presented in Figure

13.

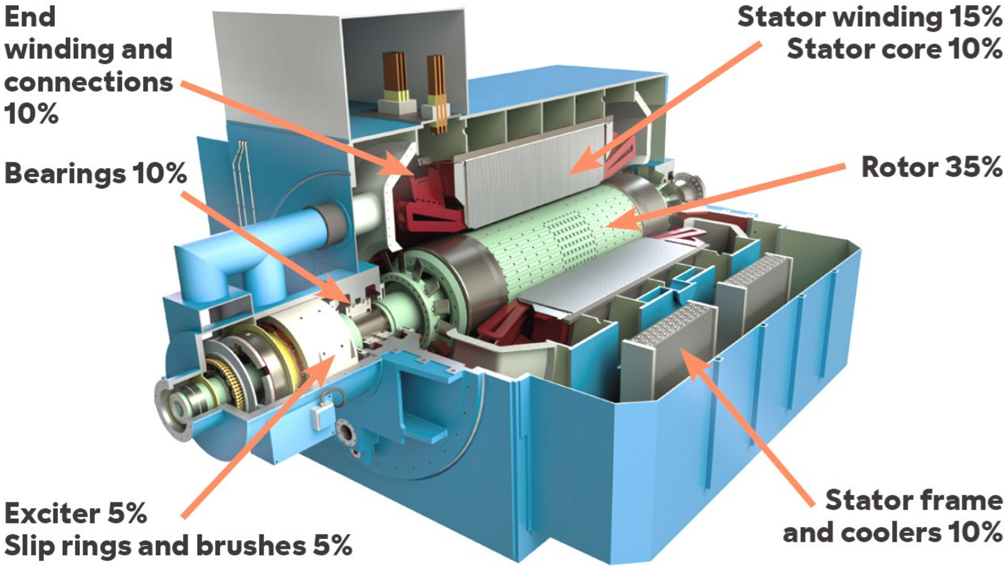

Figure 13: Structure of an air-cooled turbo generator. The distribution of the different failure modes is presented as percentages. [3]

The figure above shows that 35 % of possible failure modes have been identified in the rotor. This means that often problems in the generator are related to the rotor. When the rotor is repaired, balancing is also required to ensure the rotor is still in balance after the repair. [3]

Turbo generators are synchronous machines which means they operate at a very specific speed to produce a constant system frequency. In Europe, the grid frequency is 50 Hz and turbo generators are either two or four pole designs and thus, the running speed is either 3000 or 1500 rpm. High rotating speeds cause the rotors to encounter severe dynamic mechanical, electromagnetic and thermal loading, which explains why the highest probability of possible failure modes are identified to the rotor. [5]

The rotor is an electromagnet meaning that an electric current is needed to produce a magnetic field, and this done by an excitation system. The excitation system is typically installed as an extension of the generator shaft and it controls the applied voltage which gives control of the generator output. When the generator is operating, the rotor produces a rotating magnetic field generating three-phase alternating currents and voltages in the copper winding of the stator. The power is delivered to the system from the stator winding through the terminals. [5]

3.2 Bearings

In turbine generators both turbine and generator rotors are installed on fluid film bearings. Fluid-film bearing means that there are no rotating elements in the bearing but instead the load is supported with a thin layer of pressurized fluid. Compared to other types of bearings, fluid film bearings are much more resistant to wear since in normal operation there are no moving or contacting parts. [15]

Fluid-film bearings are the primary source of damping in the most rotor systems. They also have a specific behavior compared to the other bearing types since their spring constant changes when eccentricity or rotating speed is increased. Since the spring constant of the bearing affects the dynamic stiffness of the system, it also has an effect to the vibrations. [2]

3.3 Vibration

Vibration is a periodic motion of an object. The fundamental physics behind vibration is the Newton's second law, according to which the sum of the forces acting upon an object is equal to its mass multiplied by its acceleration. This is written in Equation 24.

$$F = ma, \tag{24}$$

where F is force, m is mass, and a is acceleration. Vibration means that oscillations occur around an equilibrium point. Vibration can be free or forced based on if there is an external force effecting the system or not. Simple vibration is vibration with single frequency and constant amplitude. This means that the measured time waveform is a pure sine wave where relationships between displacement, velocity and acceleration can simply be presented with mathematical equations: [2, 15]

$$d = Asin(\Omega t), \tag{25}$$

$$v = A\Omega sin(\Omega t + 90°) = \dot{d}, \tag{26}$$

$$a = A\Omega^2(\Omega t + 180°) = \dot{v} = \ddot{d}, \tag{27}$$

Where d means displacement, v means velocity, a means acceleration, A means vibration amplitude, Ω means angular frequency and t means time. The first derivative of displacement, in respect of time, is velocity and the second derivative is acceleration. Derivatives are represented with the dots $(\dot{d}, \dot{v}, \ddot{d})$.

Amplitude describes the magnitude of vibration and it can be expressed either in terms of the signal level or in engineering units. Amplitude can be measured as peak (pk), peak-to-peak (pk-pk) and root mean square (RMS) amplitude. Different methods are presented in Figure 14. [2]

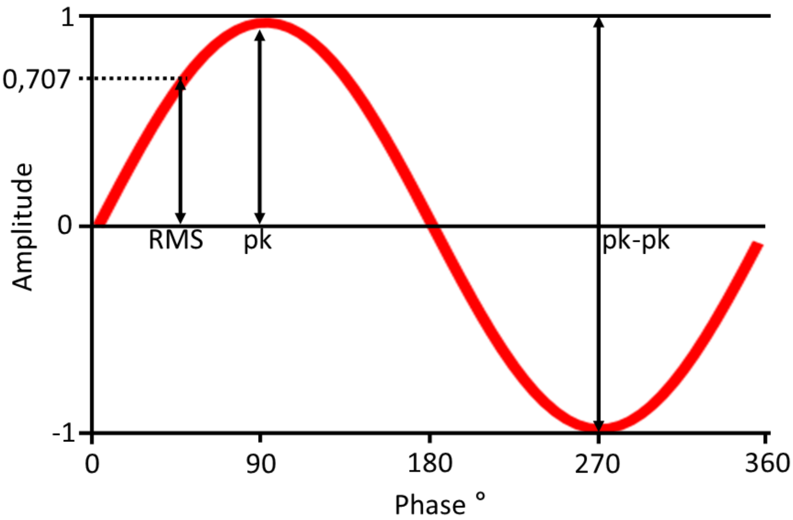

Figure 14: Amplitude measuring methods expressed with an example sine wave. Based on [2].

In Figure 14, one cycle (360°) of sine wave is presented and different methods to measure amplitude are presented. In this example the RMS amplitude is 0.707, the peak amplitude is 1 and the peak-to-peak amplitude is 2.

Angular frequency (rad/s) means how fast an object is rotating. The relationship between angular velocity and frequency is represented in Equation 28. [2]

$$f = \frac{\Omega}{2\pi}, \tag{28}$$

where f is frequency and Ω means angular velocity.

Vibration of One-Degree-of-Freedom system

A simple example of vibration can be modelled as a one-degree-of-freedom (DOF) system with a damper. Described system is presented in Figure 15. [1]

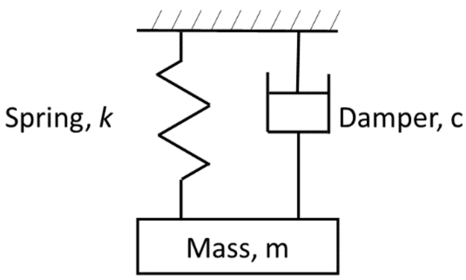

Figure 15. 1-DOF system

1-DOF means that the mass can move only on one axis. The spring force for linear and frictionless spring according to Hooke's law is presented in Equation 29.

$$F_k = kx, \tag{29}$$

Where F_k means the spring force, k means the spring constant and x is the displacement of the spring from its relaxed position.

The system in Figure 15 consists of a damper which can be modelled as a viscous damper. The damping force correlates with the viscous damping ratio and the speed of vibration. The force of viscous damping is presented in Equation 30.

$$F_C = cv = c\dot{x}, \tag{30}$$

where c is the viscous damping coefficient (Nsm^{-1}) and v is the velocity [m/s].

When Equations 1, 4 and 5 are combined and we expect the system to be isolated from all external forces, we can write a general equation for 1-DOF system as follows in Equation 31.

$$m\ddot{x} + c\dot{x} + kx = 0, \tag{31}$$

Every object has a natural frequency, which is the frequency in which the object will vibrate if it is displaced from it is relaxed position and no external forces are affecting it. For the undampened 1-DOF system the natural frequency is presented in Equation 32.

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{m}} \tag{32}$$

The frequency of the dampened free vibration is written in Equation 33.

$$f_d = f_n\sqrt{1 - \left(\frac{c}{2\sqrt{mk}}\right)^2} \tag{33}$$

These equations present the theoretical behavior of 1-DOF systems. [20] Real machines, however, do not behave as 1-DOF systems and they should be analyzed as multi-degree-of-freedom systems, where the model consists of multiple masses arranged according to the shape of an object.

Identification of the unbalance

Vibrations are measured with a vibration transducer which converts mechanical motion into an electric signal. Shaft vibrations are measured by measuring relative shaft displacement to the bearing pad with an eddy-current sensor. Bearing pedestal vibrations are absolute vibrations of the pedestal and they are measured with either velocity or acceleration sensors. Continuous monitoring of the vibrations can be used to detect changes which may indicate faults in an early stage. [2]

However, protection systems are usually only measuring the overall levels of vibrations which gives enough data to notice a change in the vibration behavior, but it is not possible to identify the cause of the change. To analyze it further, it is necessary to get more comprehensive data. Vibration analyzers that are meant for diagnosing vibration problems measure for example, spectrums, time series and shaft orbits. If a rising trend or clear change in vibration overall level is noticed it should be examined in more detail and find out the root cause.

Unbalance is one possible and the most common cause for vibrations. A typical indicator for unbalance is that the vibrations are increasing at 1X frequency in radial directions and typically, phase difference between horizontal and vertical measurement is 90°. Couple unbalance can also cause the highest vibration in the axial direction. It should also be considered how a change in operation conditions affects to the vibrations. For example, if vibrations are caused by mechanical unbalance the vibrations increase when the rotating speed increases and changing the generator load should not have an effect to 1X vibrations. [2]

Reasons for increased vibrations should be investigated to prevent issues. According to Bently et al. [2], possible problems that high 1X vibration can cause:

- Increased stresses due to bending of the rotor increase the risk for crack initiation and fatigue failure
- The bending of the rotor can damage the couplings
- If a component has natural frequency near the rotating speed, the increased excitation force can cause a fatigue failure
- Rotor can contact stationary parts in the machine which is called rub

The severity of unbalance depends on the amount of unbalance and the characteristics of the machine. Some designs have tighter tolerances and rub can occur more easily while in other machines, rub is not likely to happen due to unbalance. Also, machines have natural frequencies near the operating speed and therefore, higher excitation force may greatly increase the risk for a fatigue failure. If there is a sudden change in the unbalance condition, it probably is due to detached components in the rotor. In this situation, the machine should be inspected immediately, since a detached component at high speeds might cause severe damage to other components. [2]

4. MATERIALS AND METHODS

This Chapter introduces the balancing facility examined in this thesis. Next, the characteristics of the balancing machinery and balancing protocol are presented. Moreover, how to estimate the reliability of the balancing machine is defined. Figure 16 shows the components that are inspected in this thesis.

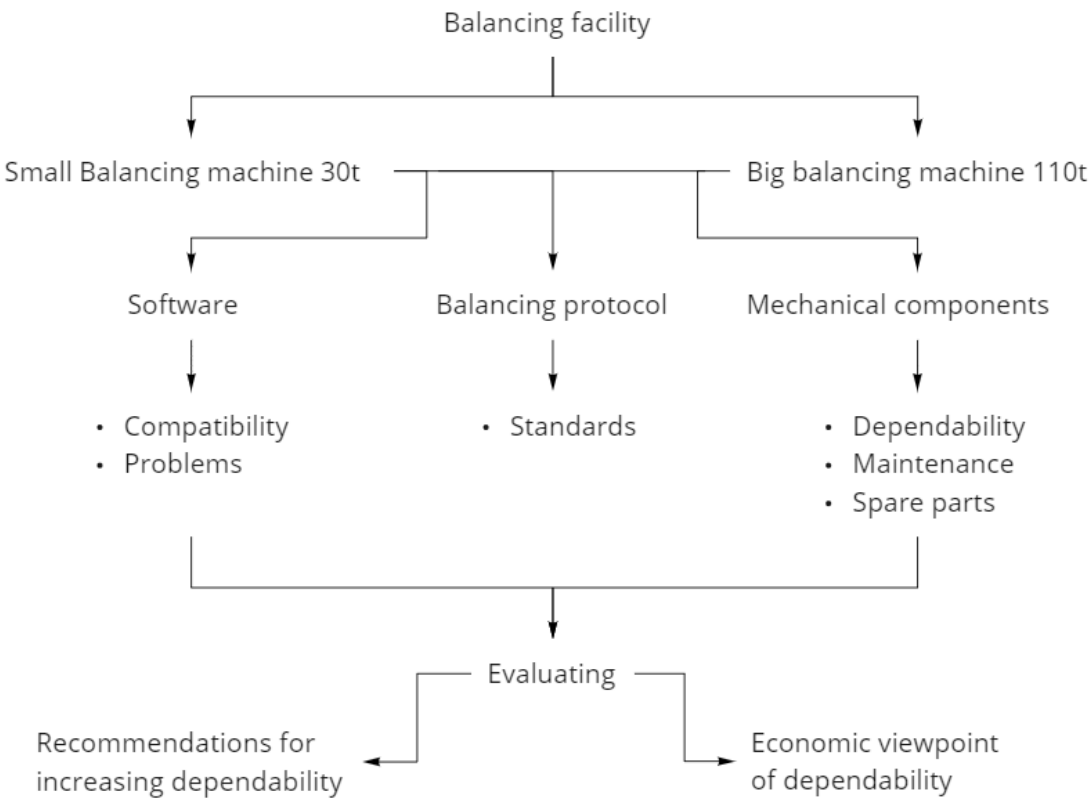

Figure 16: Inspected areas and overall evaluation procedure of the thesis.

Software, protocol and mechanical components of the balancing machines and their impact to dependability of balancing is inspected. In the following sections, these areas are presented in more detail. Increasing dependability of the balancing machine is investigated by interviewing the users of the balancing machine and the software manufacturer, and by analyzing the product specifications of the components.

4.1 Current balancing plant characteristics

The examined balancing facility includes two balancing machines as divided in Figure 16. The smaller one is for generator rotors weighting 2–30 tons and the bigger one for rotors weighting up to 110 tons. The balancing facility is built in the 1970-s, and some of its components require updating. Figure 17 presents the beginning of a balancing process: a rotor is lifted to the small balancing machine to be balanced for example, at the end of an overhaul.

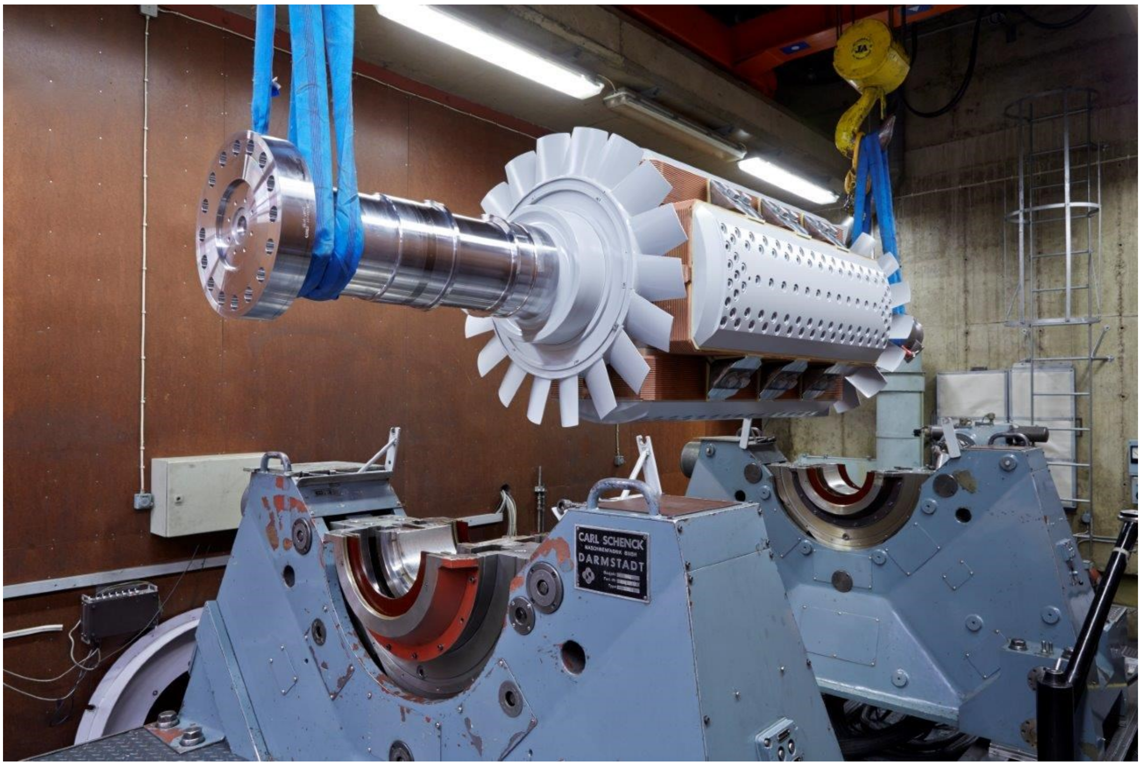

Figure 17: A rotor is lifted to the balancing machine. (photo credits: Fortum eNext, published with permission)

The balancing machine includes two movable bearing pedestals which allows rotors with different dimensions to be installed. Both pedestals have 3 acceleration sensors measuring the bearing pedestal vibrations and 2 displacement sensors. The acceleration sensors are installed perpendicular to each other measuring horizontal, vertical and axial directions. Directions are referred as x, y, and z axis where x axis means horizontal direction, y axis means vertical direction and z axis is the shaft direction. The current system can measure the overall vibration level, vibration at the rotating frequency (1X) and 1X phase. Usually, these measurements are enough to perform the balancing, but if there are problems for example, with the bearings a more comprehensive vibration measurements would be helpful to diagnose the root cause of the problems. Currently, all vibration signals have outputs in the control room which enables to connect an additional vibration measuring device. This enables to perform more comprehensive measurements: for example, shaft orbits, shaft centerline, and spectrums can be measured.

The pedestals have commutable bearings and correct bearings must be installed depending on the rotor. The most common bearing types are journal and tilting pad bearings. Both types require lubricating and jacking oil systems. The jacking oil system is used at lower rotating speeds to ensure that the oil film is strong enough to support the rotor. Lubricating oil must be pumped continuously to the bearings during the operation. [5]

After the rotor is installed on its bearings, it is connected to the rotating motor with a coupling. Since there is variation between rotors, different couplings are required. During the balancing, the rotor is rotated by an electrical motor connected through a gearbox.

4.1.1 Components of a balancing machine

The critical components require for balancing are an electrical motor, which is rotating the rotor, a gearbox, a lube oil pump, a jacking oil pump, and vibration sensors. Currently, none of these components are duplicated and thus, if one of these components is not working the balancing is interrupted until the component is repaired.

Currently, the condition of the components of the balancing machines are monitored periodically. The periodic checks include greasing of the bearings, inspection of the oil levels, and checking the condition of the carbon brushes. Operating hours of the machines are not monitored. There are basic spare parts in the stock, but no further analysis is made if there are other spare parts that should be held in the stock.

4.1.2 Balancing software

Both balancing machines are currently using a balancing software called ROBAL which is made by VIKON [19]. The software is built around the theories developed by Dr. Ing. J. Drechler and Lars Ove Larsson in the 1970s and these theories have since been used successfully for balancing of both rigid and flexible rotor shafts.

ROBAL supports up to 15 correction planes and 30 measurement points. The software is based on the influence coefficient method which was presented in Section 2.4.2. The influence matrix is based on the design of the rotor, and if the dynamic characteristics remain the same then the matrix is also unchanged. This allows the use of the old influence matrix when the next rotor balancing is required, thus making the process faster.

The current version from the software is from 2008 and it is operated with a Windows XP based computer. The user interface of the software is presented in the Figure 18.

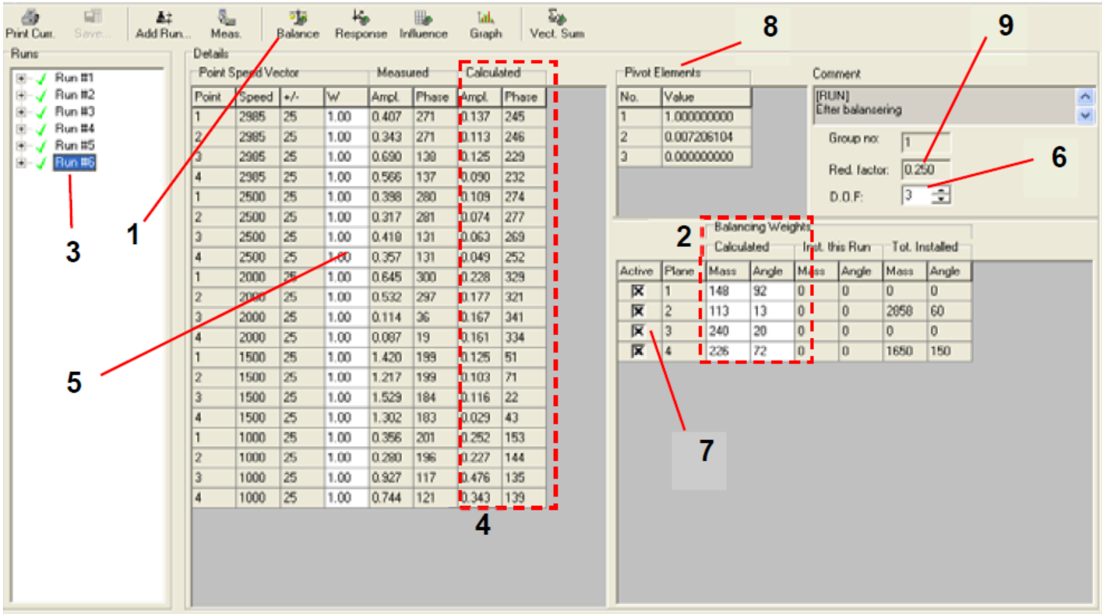

Figure 18. Basic user interface in ROBAL balancing software. [13]

User interface shows the runs in the left side of the screen as shown in Figure 18. Data of a specific run is shown when the run is selected (3). The green check mark next to the run shows that the data of the run is included when the balancing weight for the next run are calculated (2). The field *Calculated* (4) under *Details* shows the calculated response which should be achieved when the calculated balancing weight are installed. It is also possible to choose active balancing planes (7) which can be useful if some of the planes cannot be used. Changing weight factor *W* (5) can be used for example, to investigate unbalance in different speed ranges. Pivot elements (8) indicates how many DOFs can be solved. If the last pivot value is very small and there is not enough data to solve the equation, then DOF (6) should be reduced. If the weight factor (5), active balancing planes (7) or DOF (6) is changed, the calculation of the balancing weight must be initiated by clicking *Balance* (1). *Red. factor* (9) means reduction factor and it estimates the expected improvement in vibration levels.

4.2 Balancing projects

For a smaller rotor which have a simple design and only a couple of balancing planes the balancing takes around 3 days, whereas the balancing of big rotors can take more than a week. The reason for the differences in the balancing project times is that larger rotors have more balancing planes and therefore, they require more runs to be balanced.

The balancing project is a part of a major overhaul of the generator, but balancing can also be ordered without an overhaul. Figure 19 presents an example project schedule for a major overhaul of a generator.

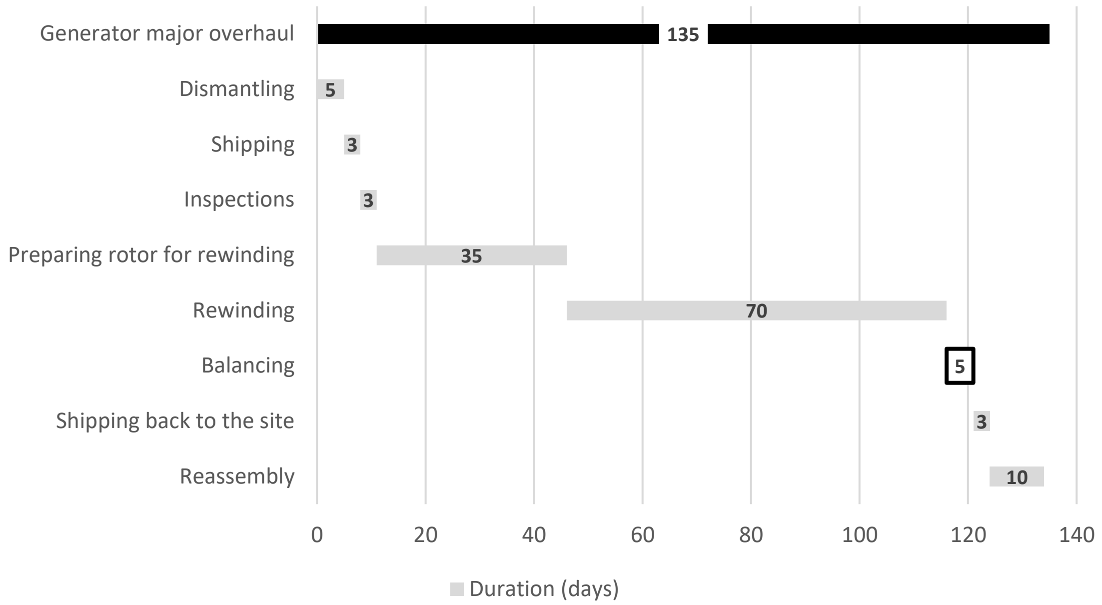

Figure 19: An example of generator major overhaul schedule.

In Figure 19, the work phases are named on the left side of the Gantt chart and the duration in days is marked in the bars. The durations are estimates based on experience and their purpose is to give insight into project phases. The black bar on the top is the total duration of the generator major overhaul project. The work phases are in chronological order. Balancing is marked with a black border. The chart shows that in the project that has a delivery time of 135 days the last 18 days are reserved for the balancing, shipping, and reassembling the generator.

4.3 Methods to evaluate dependability

The balancing project can be considered successful if the balancing quality grade exceeds the requirements and the balancing is done according to the schedule. To keep up with the schedule, the balancing machine must work as intended, thus the dependability of the balancing machine is important. Dependability means the ability to perform as and when required. The balancing machine operates only for short periods a few times per day which means the operating rate for mechanical components is quite low. [6]

Increasing the dependability of the critical components will also increase the dependability of the whole balancing machine. When evaluating the dependability of the components, condition monitoring and spare part availability are investigated. The goal is to investigate, if there are ways to enhance condition monitoring or if there are spare parts that should be ordered to the stock in advance. The investigation is done by interviewing users about the situation of the current condition monitoring and the spare parts.

Furthermore, other ways to enhance the condition monitoring with feasible costs are investigated. The condition monitoring makes it possible to notice faults before the machine is malfunctioning. Together with the condition monitoring spare part management can reduce downtimes of the balancing machines. Since there are not that many critical mechanical components and the operating rate of them is quite low, it is not feasible to maintain all the spare parts in the stock. However, it might be cost-efficient to have components that would have rather long delivery time in stock and order the spare parts required if a fault is detected during the condition monitoring.

The critical requirements for the software are the ones that make the balancing possible and reliable. The following requirements are considered as critical requirements:

1. The calculations in the software are reliable
2. Manufacturer has a valid support for the software
3. The software must be compatible with modern computers and operating systems

The backbone for a balancing software is reliable calculations. If the software calculates the unbalance incorrectly, it makes the software useless. Since the software has been used for several years the users have a lot of knowledge how the software works and about the problems affecting work phases.

The second important criterion is that the software must be actively supported by the manufacturer and this is checked from the manufacturer. The lack of support during a software malfunction might lead to a need of purchasing a new software. However, purchase of a software should be planned and controlled to avoid issues in the software change.

The third criterion is the compatibility with modern computers and operating systems. This is also be checked form the manufacturer. The reason for this criterion is that some of the older software do not work well in the newer operating systems leading into compatibility issues. It is important that the software can be used with a new computer without a problem to ensure easy change of the operating computer.

In this thesis, these critical requirements are inspected to evaluate the dependability of the balancing software. The inspections are made by interviewing the users of the software and consulting the manufacturer.

Dependability has an effect to the economy of the balancing facility. Shortly, if the balancing machine is not working, the balancing and the whole project is delayed. Estimating the costs on a component level is hard and not feasible, since delivery times, prices, and installation times for each component vary based on the availability of the component and work force. Therefore, the economic effects of dependability are inspected in terms of how much each day of delay costs. Also, effects for the imago are discussed since it is also related to the economy.

5. RESULTS AND DISCUSSION

The research questions of this thesis were:

1. What are the advantages of preventive maintenance of generator rotors to the owner of the powerplant?
2. What kind of effects does the unbalance cause to normal operation of the turbine generator and when is balancing required?
3. How can dependability of the balancing plant be increased?
4. How improving reliability would impact the economy of the balancing projects?

The answers for the first two questions are based mostly on literature and the answers for questions 3 and 4 are based on a combination of interviews and literature analysis. The results concerning these questions are presented in Sections 5.1–5.4.

5.1 Preventive maintenance

Preventive maintenance includes both CBM and predetermined maintenance. As presented in the Section 2.1.3, the new generators have preplanned maintenance schedules planned by the manufacturer. Manufacturers have good knowledge about the failure modes of the generators enabling to plan the maintenance schedule accordingly. However, with additional measurements the power plant owner receives more information about the machine which can be used to plan the overhauls precisely, and during the warranty period more thorough investigations can be performed to detect issues that would not necessarily be detected with the manufacturer's inspection schedule.

The advantage of preventive maintenance for critical machines in powerplants is to minimize downtime, which is very expensive due to production losses. It also helps to keep the overhaul costs lower since the overhauls can be planned and executed in suitable time periods. Condition monitoring also helps to prevent total failures of the machines.

5.2 Effects of unbalance in the turbo generator

The second research question was about the effects of unbalance in normal operation of turbo generators. As presented in Chapter 3, unbalance causes vibration at rotating frequency of the shaft. If the turbine generator has a vibration system installed, the unbalance can be noticed as an increase of the vibration levels. Without the vibration measurement system, it is hard to notice the changes in vibrations before the changes are severe and vibration magnitudes are very high. When a change in the vibrations is noticed, additional measurements are usually required to analyze the cause of the change.

Unbalance can occur either due to wear of components or even a part breaking away from the rotor. If the vibrations at the rotation speed increase suddenly, there is a reason to suspect that something has broken off. It is critical to investigate what has happened since the loose part can cause a massive damage to the machine if it is not prevented.

Over the time, excessive vibration can cause fatigue to the components leading to premature failure. High vibration can also cause rub, which means that the rotating part is contacting the stationary part. Altogether, too high unbalance can prohibit the operation of the turbo generator causing large production losses.

5.3 Dependability of the balancing plant

In Section 4.3, the methods for evaluating the dependability were split in details of the components and software. The results of this investigation are presented next.

Components

Evaluation of the dependability of the components was made by investigating how the condition monitoring is performed and how the spare part stock is administered. The condition monitoring of the mechanical components is currently following the periodical schedule, which is presented in Table 3.

Table 3: Periodical maintenance schedule and recommendations for further improvements.

Component	Procedure and interval
Electrical motors	Check of carbon brushes < 2 years Lubrication of bearings < 12-18 months
Balancing machine	Oil filters are changed when required and oil level is checked regularly Calibration < 3 years
Backup battery	Voltage check of batteries < 2 years Water level < 1 year
Rotating machinery	**Recommendation:** Start measuring vibrations periodically twice a year.

The maintenance schedule also includes procedures that are done only when needed. To further improve the maintenance schedule, it would be a good practice to write down notes from every fault in critical machines. Each log should include the date when the fault occurred, a short description of the fault, operating hours of the machine since the last maintenance, and since the last bearing relubrication. Keeping log of operating hours of each machine is also a good practice. [6]

The goal of the maintenance functions are overall equipment effectiveness and high dependability, but safety and cost effectiveness are also important objectives. The time required to write down the notes of every fault is quite minimal and in long term it helps to predict cyclic faults and therefor, the maintenance schedule can be updated accordingly. Accurate maintenance history helps to improve the dependability of the machines, but it can be increased even more by collecting more data from the machines to evaluate their condition.

One way to monitor the condition of the rotating machinery is to perform vibration measurements periodically. As presented in Section 2.2, various faults can be detected with vibration measurements. For example, unbalance, bearing failures, looseness, misalignment, and electrical motor faults are typical faults that can be analyzed from the data. Accurate information about machine condition would greatly help to prevent unplanned downtimes since it enables planning of the maintenance.

Since the operating rate of the balancing facility machines are quite low compared to normal operating rates in the industry, a suitable period for vibration measurements could be 6-12 months. Time required for the measurements, analysis and a short report would be around 1-2 workdays. If analysis indicates a fault at an early stage, it is recommended to monitor vibrations more frequently and start preparations for an overhaul.

Currently, the spare part stock is not actively administrated. There are spare parts in the stock but no research for required spare parts has been made. It is recommended to do an inventory and investigate what is the availability of the spare parts from the suppliers. Based on the investigation, spare parts should be added to the stock if needed. The inventory could be performed by the staff of the balancing facility, or later by visiting the facility when the current traveling restrictions have been eased.

Software

The methods to evaluate the dependability of the software are:

1. The calculations in the software are reliable.
2. Manufacturer has a valid support for the software.
3. The software must be compatible with modern computers and operating systems.

Unreliability of the software would be noticed in vibration levels during the balancing process. So far, if the results have not been logical compared to the software estimation, the reason has not been unreliable calculations but instead something mechanical has changed in the system. Users have reported that there are a few known bugs in the software which are not causing that much harm to usability of the software. For example, on some occasion the order of point speed vectors shown in Figure 18 can be wrong but the data in each row is correct. The performance of the software is followed in case the software would become unreliable.

There are also features that could be included in the software. For example, a tool for weight splitting or advanced vibration analysis is not included in the current version. Currently, weight splitting calculations are performed manually which is feasible, but causes a risk for a human error. This error risk could be avoided if the calculations would be done automatically.

The possibility for more advanced vibration analysis tool was investigated but currently, the only option is to get an external condition monitoring system. Since this tool is only required in special occasions, it is not feasible to purchase an additional system. However, there are signal outputs for vibration transducers which can be used for measuring the vibrations with an existing external analyzer device. When the balancing software needs to be updated for other reasons, then the inclusion of more advanced vibration analysis tools is an important matter to be considered.

The software manufacturer (VIKON) was contacted, and the ROBAL [19] software for rotor balancing is still available and actively supported. There is a newer version available but according to VIKON the only difference is that the new version is translated into another programming language. Therefore, there is no notable difference for the end users or to software compatibility. They also confirmed that the currently owned version also works in Windows 10 operating system.

Currently, there are two computers for balancing at the facility, one for each balancing machine. It is highly recommended to have at least one spare computer where the software is preinstalled and preferably tested, thus the computer can be replaced if an issue occurs. Also, this would be a good chance to verify that the software works in a modern computer and operating system. The possible downtime could be shortened significantly if everything is prepared ahead. Obtaining of the spare computer enables testing software compatibility during a quiet period. Fixing possible issues could be time consuming and requires testing which is why it is beneficial to do it beforehand.

According to user feedback at the balancing facility from the current balancing software ROBAL is performing well in its task. There are a few known bugs but their effect to the usability is quite minimal. Current software lacks weight splitting and vibration analysis which are features that could be included in the next software, but they alone are not the reason to update the software. The manufacturer confirmed that there is no need to update the software since there are no critical updates and the current version works in Windows 10 operating system. In conclusion, it is highly recommended to obtain a spare computer and to verify compatibility for the currently owned software version.

Internal standard for the balancing procedure

Currently, there is no written protocol which would include the work phases, safety requirements, standards which are fulfilled and what the balancing reports include. It is recommended to compile internal balancing procedure documentation to. This protocol should also be kept up to date.

This document could be helpful in multiple ways. Internally, colleagues could use it to get a basic understanding what balancing means and what it includes. It could also be used to explain the work phases of balancing to customers and justify recommendations for further actions. It also gives a professional expression since an official document shows well planned work phases in a justified manner.

The internal standard should include important topics regarding the balancing procedure. It should be described how the permissible residual unbalance is defined. For example, used standards should be named and important details shortly described. Also, the limits for different kind of rotors should be explained. This should include the number of affecting vibration modes and available balancing planes.

Next, the balancing procedure should be explained. Work phases could be described to give an overview of the process. It is important to mention if a work phase is done according to a specific standard to ensure the standards are then always utilized. Also, calculation of the residual unbalance should be described at the basic level.

The evaluation of the protocol ensures the working phases are optimized and reliable. Carefully planned work phases reduce the risk for errors during balancing. Errors could be caused by, for example, the lack of right tools, or a human error.

5.4 Economic effects of dependability

Economic effects of dependability were divided between the balancing facility and the balancing projects. This division was done because not all the costs can be considered as project costs. The effects are presented in the sections below.

Effects for the balancing facility

As stated earlier in Chapter 2, maintenance should decrease overall costs. This means that costs caused by maintenance should be lower than the benefits achieved by maintenance. Since the costs of delays in balancing are rather high, preventive maintenance is a good way to keep costs down. However, to keep preventive maintenance efficient, good knowledge of the condition of the machines is required. Therefore, periodical vibration measurements are strongly recommended to follow the condition of the balancing machines and to keep them working efficiently.

With condition monitoring and preventive maintenance, it is possible to decrease the number of sudden machine failures which helps to keep projects on schedule. Repairing unexpected equipment failures is more expensive because the maintenance is not planned, and the balancing work is interrupted. This ensures the maintenance costs are more predictable and not unexpected.

Any improvements of the balancing facility should be considered as investments which require financial planning and justification of the required changes. Investments are only made if they improve the economy of the facility. To justify the improvements, it is necessary to know the costs of delays caused by undependable machines.

Effects for the balancing projects

Issues during balancing can cause additional costs to the project by causing delays. The typical economic issues of the balancing facility are presented below:

1. Unexpected malfunctioning of the balancing machine causes a delay to whole overhaul project.
2. Delays will not give a professional impression to the customer causing harm for the imago.

The economic impact cost estimates are used to evaluate the costs of the balancing machine outages and how the increased dependability could decrease the overall costs. Since it is difficult to evaluate the costs in component level, the costs of delays are calculated in a daily level. Furthermore, the economic impact of upgrades needs to be inspected case-by-case.

The estimated basic properties of a major overhaul project are presented in Table 4.

Table 4: The basic properties of a major overhaul project.

Variable	Average value	Unit
Overall project duration	1 – 6	months
Balancing duration	2 – 5	days
Project value	200 000 – 800 000	€
Penalty from delay	2 000 – 8 000	€ / day
Lost production	E_p * t * Generator active power	€

In Table 4 E_p presents the price of electricity and t presents the time of delay in hours.

The duration of a major overhaul for the turbo generator is 1–6 months and it depends on the scope of the work. In a project with an extensive scope, the duration of the balancing is quite minimal compared to the total duration of the project. However, balancing is the last work phase before the rotor is sent back to the powerplant and the generator is assembled. Assembling takes around one week which means that even small delays are hard to catch up in such a short time.

As presented in Section 4.2, there is typically approximately two weeks of time remaining to have the generator reassembled after balancing is finished. The shipping takes around three days without actual possibilities to speed it up. During the reassembly, it is possible to work continuously in shifts which enables catching up the schedule. This requires that there would be workforce available for the project and working around the clock also increases personnel costs quite a bit.

The lost production and other costs caused by delays can be significant to the power plant owner, thus the contracts usually have a penalty for delays. For example, the penalty can be 1% of the total project value per day, capped at 10% after 10 days of delay. However, the penalties do not cover all the expenses caused by the delay and thus, delayed projects can leave a bad impression.

In this business area, the image is also important. It is important to show to the customers that we are professionals in overhaul projects and can deliver them as agreed. Increasing dependability of the balancing decreases the risk for issues during the project delivery. It is hard to prevent every issue, but it is possible to decrease risks by planning. However, when an issue occurs it is important to take care of it as well as possible.

5.5 Overall discussion

In this section the contribution of the study is presented. The initial goal of this thesis was to study, what balancing software would be the best one for our use. However, investigations revealed that currently there is no need for an update. If the requirements for the software change or new beneficial features become available, it is then recommended to re-evaluate the need for an update.

The subject of thesis was focused to the dependability of the balancing facility. This is the first time after the design phase of the facility that the dependability has been evaluated. The contribution of this thesis was to evaluate the general aspects of dependability in the balancing facility and give recommendations for actions and further studies.

Another goal was to benchmark our procedures and balancing equipment to other balancing facilities. This would require travelling in Europe which was not possible at the given schedule due to the COVID-19 travel restrictions. For the same reason it was not possible to travel to the balancing facility and investigations needed to be done remotely by contacting the local personnel. It is recommended to do the benchmarking after travelling is allowed again. This would give information if our procedure or the balancing machines could be improved.

6. CONCLUSION

The balancing of a turbo generator rotor is a part of a major overhaul which is the most comprehensive type of maintenance in power plants. Base load power plants have high utilization rates and therefore, downtimes are carefully planned, and any unexpected downtime would have major costs. In general, it is necessary to minimize downtime of the power plants to be efficient.

The objective of this thesis was to evaluate the dependability of the balancing machines and software, and to find ways to reduce unexpected downtime in the balancing facility. This study inspected the characteristics of the current balancing facility and the balancing process. Based on inspections, recommendations were given to improve the dependability of the process. A summary of the findings is presented below.

Currently, maintenance is combination of preventive and corrective maintenance. Basic checks are made according to the schedule and faults are repaired when they occur. It is recommended to start performing vibration measurements for rotating machinery as a part of periodic inspections. By monitoring the condition of the machines, it is possible to start moving more towards preventive maintenance.

The balancing software is currently installed on two computers which are designated to the balancing machines. There is no backup computer, and it is highly recommended to get one and install all required software in advance. Backup computer reduces unexpected downtime significantly and enables testing the software compatibility.

The current software does not have an option for advanced vibration analysis. Only the overall values as well as vibration amplitude and phase at rotating frequency are available in the software. For example, more advanced vibration analysis could help to detect problems with bearings. This alone is not feasible reason to get a new software and thus it is recommended to use existing handheld analyzers to measure additional vibrations from the signal outputs.

There is no internal standard which would define the work phases and details of the balancing procedure. It is recommended to start compiling the internal documentation. The standard should be updated regularly. Having a documentation of the procedure helps to reduce time spent for inquiries, since answers for basic questions are available.

In conclusion, this thesis investigated the need for updating the balancing software, but not enough justification was found to initiate the updating process. User feedback of the current software is positive and therefore, no acute changes are needed. However, benchmarking of the other balancing facilities in Europe was planned but not succeeded because of the traveling restrictions during the COVID-19 pandemic. Benchmarking and follow-up of the software development should be proceeded in the future to facilitate continuous improvement and learning.

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