THE INSTIGATING FACTORS BEHIND THE OCCURRENCE OF VIBRATION IN STEAM TURBINES - A REVIEW ANALYSIS

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The present study focuses on elucidating the fundamental reasons underlying the emergence of vibrations in steam turbines. During operation, vibrations are observed not only in the components of the machinery that undergo cyclical motion but also in those components connected to the equipment. Therefore, vibration monitoring holds great importance in identifying malfunctions in the functional operations of turbomachinery, enabling timely detection and prevention of potential accidents. Using the steam turbine unit as an example, it is noteworthy that the rotor primarily undergoes oscillatory motion, where it is essential to recognize that vibrations also manifest in bearings, housings, turbine foundations, pipelines and surrounding components. The thorough examination of vibration should encompass not only turbine rotors but also the entire turbine assembly, including the generator and all associated equipment. It is essential to conduct a comprehensive evaluation of the overall system to ensure optimal functionality. Academic research papers typically do not often assess the specific number of working hours and conditions which are leading to rotor damage, also in that sense, not determining if damage is a result of wear and tear during prolonged undesired operation. Instead, the emphasis is commonly placed on analyzing elevated levels of vibrations and investigating the associated occurrence of cracks. This paper aims to provide a comprehensive summary of the main causes of vibrations through a unified perspective on the various conclusions available, regarding the diverse causes behind these common and complex vibration occurrences.

Key words: Steam turbine vibration, vibration causes, vibration emergence

1. Introduction

A considerable number of steam turbines utilized in power generation are currently operating beyond their intended life service, surpassing a duration of 30 years in operation. Crucial components within this context are the rotating turbine elements, particularly the rotors that consist of discs and blades. These structural elements endure elevated levels of thermal and mechanical stresses, making them prone to reduced durability and a shorter service life [1]. In that sense, efficient machinery diagnosis is essential to avert catastrophic accidents and vibration monitoring serves as a pivotal tool in diagnostic systems. To create a monitoring system that can detect cracks early, it's essential to understand the unique vibration characteristics of a cracked rotor [2]. Numerous non-destructive testing techniques have been developed to proactively prevent failures. In some cases, detecting cracks, especially those in the material of the rotor, may be challenging for ultrasonic testing. Therefore, vibration monitoring serves as a valuable additional approach to identify such structural anomalies. This literature review will explicate prevalent factors causing vibrations.

Within the academic community, a multitude of research endeavors have been undertaken to elucidate and enhance our comprehension of this complex occurrence. In order to present a systematized overview of the causes, they are classified into nine sections, i.e. Vibrations caused by unbalance, Vibrations caused by shaft (coupling) misalignment, Vibrations caused in bearings, Resonance, Vibrations caused by steam flow irregularities, Vibrations caused by turbine loose parts, Vibrations caused by damaged turbine part, Vibrations caused by temperature influence and Vibrations caused by steam turbine generator and electric grid irregularities.

2. Vibrations Caused by Unbalance

The predominant source of heightened vibrational levels in rotors emanates from the condition of unbalance. The emergence of an elevated magnitude of vibratory motion is directly associated with the amplitude of vibration, reflecting the degree of unbalance present in the rotor system [3]. The main cause of this problem is an uneven distribution of mass around the rotational axis, which poses a substantial risk and leads to noticeable noise [4].

In a rotor dynamical system, dynamic nonlinearity can lead to excessive vibration caused by consequential events such as rotor-stator rubs and disturbances in journal bearings film. Over time, the intensified vibration resulting from imbalance can bring about two significant long-term effects: engagement to operating speed and interfering through critical speed transition. The crux of the issue in a machine, where a substantial proportion of the rotor's mass becomes dislodged from the rotor axis, lies in the challenge of maintaining critical speed without incurring a significant degree of damage. Reducing imbalance is crucial to minimize the mentioned consequences [5].

While rotors are generally balanced at the factory, additional on-site balancing might be required due to errors that can occur during rotor train assembly. The main goal of rotor balancing is to alleviate excitation forces caused by residual mass unbalance, particularly evident at critical speeds and established operating speeds of the machinery [6]. In the past, numerous additional factors have contributed to the aggravation of rotor vibration issues. For instance, insufficiently damping and dry friction consequences in the equipment, unstable rotation in electrical machinery due to varying side-to-side stiffness of the shaft and specific speed intervals have exhibited oscillations caused by non-linear influences of the foundation. Following a thorough analysis of the problem, standard techniques can be applied to mitigate or eliminate undesirable vibrations [7].

As the rotational velocity of the rotor increases, even a slight imbalance can have adverse effects, potentially leading to the failure of the entire rotor system. It is highly undesirable for rotors to possess residual imbalances surpassing a predefined critical speed threshold. Such imbalances tend to generate excessively high imbalance forces, consequently causing significant vibrations in the system. While in operation, it is possible that a rotor may experience a loss of balance, leading to an increase in clearance between the rotor and stator which may eventually exceed the specified limit [8]. This interaction between the rotor and stator results in changes in non-stationary stiffness of the rotor, while the introduction of frictional forces at contact boundary gives rise to a complex whirling motion [9].

Various factors can induce whirling motion; however, an imbalance appears to be a principal contributing factor. When the rotor experiences lateral displacement due to heightened forces or when its lateral orbital motion surpasses acceptable clearances, the undesirable contact between the rotor and stator appears, known as rotor-to-stator rub [10].

It is widely acknowledged that unbalance is often found to be distributed across the shaft, whereby the rotor may exhibit specific instances of unbalance at discrete points at discs as well as along the length of the shaft. Accordingly, it is imperative to conduct regular inspections of an in-service rotor and subject it to periodic balancing procedures as necessary. The mentioned factors emphasize the vital significance of incorporating operational rotor balancing [8].

The intensity of oscillation can become notably significant when the excitation frequency approaches one of the eigenfrequencies of the machinery framework [11]. The occurrence of rotor unbalance can lead to noteworthy torsional vibrations that require careful consideration. These vibrations occur due to the imposition of unidirectional lateral forces on the rotor, caused by tangential forces exerted by the surrounding fluid. Consequently, a gyroscopic effect is generated, which further contributes to the development of the rotor vibrations, synchronous and nonsynchronous, forward and backward perturbation [10].

Agreeing with [8], it must be noted that the classification of the imbalance is not meticulously specified, where numerous explained variations pertain to exist in the concept of imbalance. The specific location where vibrations act as a signal for the emergence of instability is still a subject of debate. In a lot of cases, it remains unclear whether the instability arose, but it can be assumed that vibrations appear due to inadequate field balancing practices, bad installation, incident circumstances, wear and tear or material fatigue over time.

While a rotor undergoes balancing measures following its manufacture procedures, a rotor connected through rigid flange coupling (like those in a steam turbine) may still display irregular vibrations due to imbalanced conditions when the entire rotor system is assembled. This phenomenon can be attributed to two main factors. Firstly, interconnecting the rotors exacerbates errors in rotor geometry. Secondly, the operational context of the rotor within the field is distinct from encountered working conditions during post-manufacture testing. The act of connecting the rotor induces variations in its geometry along centerline additionally influenced by the turbine's operating environment.

The aforesaid environment includes the fluid's operating parameters and the foundation's bearings connection. Before balancing activities, it is imperative to mount all top halves of the turbine to guarantee that the balancing procedures are executed under equivalent conditions to those encountered during normal operation of the unit. Otherwise, an incorrect balancing procedure may ensue residual imbalance.

3. Vibrations Caused by Shaft (Coupling) Misalignment

Misalignment is widely recognized as a prevalent concern in rotor systems subsequent to unbalance [12]. The occurrence of noise and vibration is a common consequence to misalignment of: shafts, couplings, bearings, or gears as underlined in [13]. Turbine manufacturing procedures often result in the presence of residual misalignment, so the emergence of overall misalignment is present during utilization. When misalignment occurs between drive shafts, it results in the generation of reaction forces and moments, known as misalignment forces, at the coupling location. These forces have a substantial impact on the vibration characteristics of the rotors [12]. The misalignment forces vary for different types of couplings, as presented in [14].

Incorrect maintenance practices and significant disparities between hot dynamic alignment and cold static alignment contribute to abnormal machine misalignments, facilitating crack propagation [15]. Misalignment of shafts also has the potential to generate high economic costs for the industry due to the resulting downtime and consequent financial losses incurred from unscheduled shutdowns. During the shutdowns it is often necessary to realign shafts and, on rare occasions, open casing coverings. The presence of misaligned shafts is a predominant factor that accounts for over 50% of the forced outages in turbomachinery is the presence of misaligned shafts, according to scholarly sources [6].

Over time, machinery is prone to wear, fatigue, and deformation, that result in increased gaps between components, misalignment of shafts, formation of cracks and imbalance in rotors. These effects contribute to an elevated degree of oscillation, leading to additional dynamic loads on the bearings, as is pointed out in [16]. Common causes of heightened vibration in machinery include improper alignment, bent or eccentric shafts, uneven weight distribution in components, faulty bearings, malfunctioning gearboxes, impellers with flawed blades and dislodged mechanical parts [17].

Rotor misalignment can also lead to cyclic bending stress in the coupling, resulting in a whirling motion due to the cyclic tension occur in the rotor. The rotation of the rotor induces vibration and imposes additional stress on the bearings, posing a notable risk to the attachment of the coupling flange. Therefore, achieving an optimal precision in the rotor alignment is imperative [18]. Misalignment may be attributed to several factors such as: inconsistencies in the dimensions governing the centricity of the rotor with respect to the stator, uneven loading of the rotor in a single direction, and eccentric thermal deformation of the stator during service as is stated in [19].

Regarding shaft alignment, accuracy should be based on the positions of the shaft itself, rather than the coupling or the bending limits of its mechanical components. When a single bend point is present and the shafts are not in proper alignment, or when there is a concurrent occurrence of misalignment and angularity, the coupling will transfer exceedingly elevated radial forces into the bearings. The presence of an excess of two bending points can give rise to considerable unregulated motion between two linked shafts, which frequently culminates in exceedingly elevated vibration magnitudes within the turbine [20].

The specific phenomenon referred to as "unbalanced magnetic pull"[1] generates vibrations and noise emissions that accelerate bearing deterioration and may lead to contact between the rotor and stator [21]. This problem is also analyzed in [22] utilizing a model that calculates the magnetic pull in generators. Beginning with the precise determination of the clearance distribution, the model proceeds to define both the constant and dynamic forces resulting from the asymmetric clearance distribution via a Maxwell stress calculation.

Shaft alignment, which is commonly known as coupling alignment, can be accomplished through the utilization of conventional tools such as gages, calipers, dial indicators and straightedges for aligning the rotor. Advanced and refined techniques presently incorporate optics and/or laser systems [6]. During the commencement of turbine operation, the shaft endures elevated levels of stress

[1] "Unbalanced magnetic pull" is explained in more detail in chapter 10 of the paper.

and dynamic pressure as a consequence of its misalignment. Repeated startups over the weeks increase chance of detrimental effects i.e., damage as it is concluded in [23].

It should be noted that alignment of shafts in the past had practical applications involving the employment of both conventional equipment and laser systems in tandem. Neither of which was exclusively utilized, regardless of the type or power of the steam turbine. The alignment process is afforded considerable attention, particularly when fastening of measuring equipment have to be provided as well as conducting rotor run-out checks, precluding bearing rotation and axial rotor movement. It is also important to ensure appropriate competence and training of the measuring and positioning personnel during measurements.

At the onset of turbine operation, due to the aforementioned factors and occurrences, the process of turbine warming up has to endure for a period ranging between one to three hours or even more. The duration of the warming up period is contingent upon the type and capacity of the turbine as well as the guidelines stipulated by the manufacturer. The desire to commence operation of the unit after overhaul is typically immense, and therefore, there is often recourse to abbreviate this time interval. Unfortunately, this approach inevitably produces deleterious outcomes for the turbine in the form of heightened levels of vibration, noise, wear and tear, and even the occurrence of sparkling.

As a rule, the system is subjected to a cleaning process prior to the commencement of the turbine work. Despite concerted efforts by employees to prevent it, the presence of leftover parts of the overhaul, including screws, nuts, and other contaminants within the system is not unprecedented occurrence. Those appearances are also associated with flaws in turbine assembly and alignment. All these phenomena can lead to heightened vibrational activity, as well as increased wear and damage to the turbine's components, including bearings. Consequently, it is imperative to avoid abridging the warm-up phase to ensure comprehensive attention, devoted to the examination of all facets of the system that manifest operational deficiencies to detect misalignment errors as soon as possible.

4. Vibrations Caused in Bearings

The operation of bearings is determined by the principle of hydrodynamic lubrication, which enables an adequate amount of oil that must persistently occupy the clearance area separating the bearing and the shaft. The development of the oil-film wedge and the subsequent initiation of hydrodynamic lubrication is dependent upon several factors, namely, the bearing design, relative velocity, oil viscosity and applied load [24].

Misalignment in journal bearings also has a significant impact on the static and dynamic characteristics of rotating machinery. Initially, when misalignment occurs, load concentration at the edges of the journal and sleeve axis increases, followed by partial lubrication, increased bearing temperature, as well as rotor instability. Over time, these conditions can cause faster bearings wear, vibrations, and even failure [25]. These phenomena were investigated in [22] with the conclusion that the vibration induced by steam excitation passing though blade rows and steam glands as well as low damping in bearing oil film are almost certainly the cause of this type of work disorder [26].

Extensive theoretical and empirical investigations have elucidated the impact of misalignment on the functionality of journal bearings, revealing spatially disproportionate provision of oil pressure and temperature [25]. Due to misalignment, the temperature field could exhibit considerable asymmetry in the axial direction, where temperature rises significantly caused by surface contact while the hydrodynamic pressure is being inclined to one side only. Partial (mixed) lubrication occurs readily when the bearing journal is misaligned. The empirical findings suggest that the frictional force is increased by the alignment angle, indicating the onset of partial lubrication of the bearing surface. Moreover, the incongruity between the misalignment and the previously established symmetrical distribution, coupled with the alteration in oil film thickness, has resulted in a noticeable influence on the journal bearing's elastic deformation, as articulated in [27].

A small gap between seals of a rotating and stationary components can lead to adverse effects. This is resulting in intensified swirling motion, causing a rapid increase in vibration amplitudes that ultimately surpass permissible levels. In that case the inherent unstable vibrations stemming from the initial natural frequency of the shaft can be damped either by the oil film in bearings, or by other external damping mechanisms. Inadequate damping requires adjustments in the geometry of the seal centerline, as suggested in recent literature [26]. The whirling motion may assume conical, semi-conical, or parallel configurations, as posited by the literature source [28]. The heightened excitation force brought by the steam flow diminishes the seals dynamic stability. The turbine may exhibit unstable vibrations owing to the interplay between the rotating shaft and the surrounding fluid forces, regarding the sealing mechanism [29].

Analysis presented in [19] point out that the clearances, looseness in the bearing or dead band in the vibration signal, can result in a limit or distortion to the peak values of the transmitted vibration wave. Such phenomenon is observed as vibrations at frequencies outside the typical design parameters of the components connected to the external static structure of the machine. In that case, non-uniform stiffness in a stationary rotor support element (while not a primary determinant of system response), can cause significant distortion in the simple harmonic oscillations of a linear system, especially when considering peak clipping related to these irregularities.

Phenomena that should also be paid attention to are oil whip and oil whirl and their difference is well explained in [30]. Oil whirl is an unstable free vibration in a fluid-film bearing caused by inadequate unit loading, resulting in circular dynamic motion of the shaft centerline in the direction of rotation. This typically happens when the oil flow velocity within the bearing is between 40 to 49% of the shaft speed. Oil whip occurs when the whirl frequency aligns with a shaft resonant frequency and becomes locked to it. It's important to note that oil whirl and whip can occur whenever there is fluid between two cylindrical surfaces. Also, in the case of a high-speed or lightweight rotor, the degree of journal eccentricity is generally enough small where instability component of the oil film force increases, causing oil whip phenomena [31].

However, implementing this corrective action can be challenging, especially for large-capacity steam turbines. Therefore, the common practice is to use stabilizing bearings to ensure continuous rotor stabilization in steam turbines [24]. Additionally, an inadequate machine foundation could result in vibration issues for the machine. The analysis of the dynamic response of a rotor-bearing-foundation system has highlighted the crucial significance of examining the primary direction of motion for each individual component. Moreover, the conditions of a specific component can profoundly influence the overarching behavior of the entire system [32].

The vibration of a rotor depends on several factors, including its geometry, the type of support it utilizes and the forces acting as sources of excitation. Such factors play crucial role in determining the degrees of vibration exhibited by a rotor. The rotor's oscillatory motion induces vibration in its support structure, and conversely, the resonance of a rotor is affected to a varying degree by its foundation. If there are concerns about the foundation of the steam turbine foundation, it is reasonable to conduct tests to identify whether heightened vibrations are a result of malfunctions or structural damage in the foundation. The expeditious assessment of conditions in concrete structures is often implemented through the utilization of measuring tapes made from glass. These tapes are strategically situated at locations suspected of undergoing any anomalous movement or where cracks and fissures in the material have emerged. In the event of a measuring tape breakage, it can be inferred that increased vibrations are affecting the bearings at designated locations. Subsequently, such issues may result in adverse consequences in the turbine.

5. Resonance

An explanation how each object possesses innate vibration frequencies, also known as resonant frequencies (eigenfrequencies), which are determined by its distinctive properties of mass, stiffness, and damping is presented and discussed in [18]. Despite the infinite nature of natural frequencies in theory, the field of machinery vibration calculation and measurement is limited to consideration of only the initial few frequencies, as indicated by research in [11]. Although it is hard to accept the conclusion where damping is presented as a property in [18], since the scholarly literature [33] provides a thorough explanation regarding rotordynamics but also the methods of rotor damping, which includes defining damping phenomenon just as an effect.

In the context of rotating machinery, the resonant frequency, also referred to as the natural frequency, denoting a pivotal point of unit speed known as the "critical speed". While a resonant frequency is frequently described as a single frequency, the reality is that it often encompasses a range or band of frequencies. The width of this frequency band is contingent on the level of damping. An effectively damped resonant frequency is characterized by lower amplitude and a broader frequency range, according to [34]. Resonant frequency commonly pertains to a distinct frequency in question. Fundamentally, resonance arises when a natural frequency inherent to the system coincides with one of the excitation frequencies of the system [35]. When the frequency of the forced vibration coincides with the natural frequency of the system, the amplitude increases significantly, much more than expected given by unbalance effect [18].

During operation, the natural frequencies in rotating machinery may exhibit a growth phenomenon in presence of fluid pressures, such as in turbine wheel bearings, seals, and blades, gives rise to an inherent trait. The phenomenon of rotordynamic instability, also referred to as self-excited instability [11]. The fatigue strength of blades is subject to degradation in resonance scenarios due to greater displacement amplitudes that exceed those typically encountered in design conditions. At the point of resonance, the blades commence to undergo fracture, and accordingly this phenomenon propagates expeditiously until complete failure ensues. In academic discussions, vigilant monitoring of turbine vibrations over extended periods is essential to ensure the intended operational lifespan of the blades. The presence of a crack has the potential to alter existing modes and vibration amplitudes, resulting in increased resonance and the possibility of entering unforeseen resonance zones [36].

The occurrence of a phase shift and an amplitude peak response are commonly recognized attributes observed as a rotor approaches its critical velocity. In an isolated system devoid of external inputs, the phase shift would result in a 180° shift, and the critical speed would manifest itself at the midpoint of the phase shift. In the majority of scenarios, the phase shift exhibited by turbine-generator trains is commonly observed to be below 180°. There are several factors that contribute to this phenomenon, namely: the cross effects experienced by interconnected rotors, the damping of the bearing oil-film and the flexibility of the rotor. The aforementioned parameters possess the potential to exert an influence on the amplitude peaks [6].

With respect to system instability, the occurrence of sudden, significant alterations in power generation, manifested in the form of fluctuations, represents an undesirable outcome within a given power network. During transitional phases, rotor torsional resonance may be triggered, leading to excitation of the rotor blades and consequent imposition of significant mechanical pressures [37]. The phenomenon of torsional resonance manifests when the armature winding of a generator interacts with the primary magnetic field, which is produced by the turbine generator, due to sub-synchronous currents, as is pointed out in [38]. The interaction and rotation of magnetic fields bring about torque pulsations on the generator rotor, where the slip frequency serves as a crucial factor in the aforementioned phenomenon. Subsynchronous oscillation refers to a state within an electric power system experiences substantial energy exchanges with a turbine generator, occurring at one or more natural frequencies of the combined system which are notably lower than the synchronous frequency [39].

In general, the aforementioned torque fluctuations do not induce detrimental effects unless their frequency aligns with or approximates one of the torsional resonant frequencies of the turbine-generator shaft mechanism [38]. According to [40], the presence of a torsional resonance is observed in the turbogenerator system when it is subjected to an unbalanced load at the generator terminals, resulting in a torsional oscillation with a frequency approximately twice the grid frequency.

Concluding this section, based on the cited literature it can be emphasized that the phenomenon of resonance in global systems is a complex subject that involves intricate components. It is important to highlight that the generator's resistance during operation has a dampening effect on vibrations. This phenomenon is particularly evident during the initial start of the turbine after an overhaul when residual imbalances are prominent, especially during the first synchronization of the generator. While the most common scenario involves the damping of vibrations, there are instances where vibrations significantly increase due to residual imbalances.

However, in the scientific community is a widely accepted consensus and established definition that associates the phenomenon of damping with material properties or characteristics. This specific definition is considered inadequate. Damping refers to the transmission and dissipation of energy, and it can be explained as the system's ability to reduce its dynamic response through energy dissipation. The influence of damping on the natural frequency of a system's oscillation is minimal, and the inclusion of material damping in calculations for determining bending and torsional eigenfrequencies is often unnecessary. In instances whereby particular operational conditions such as specific turbine working regimes require analysis, the computation of torsional eigenfrequencies may necessitate the application of a damping material featuring a coefficient of 0.1%.

In order to determine the potential electrical failure modes of the generator, it is necessary to possess relevant electrical data that encompasses reactances, impedances, and damping coefficients. However, it has been observed that these factors do not have an impact on the torsional or bending eigenfrequencies. They have a significant influence on the material torsional stress level. However, damping plays a significant role in attenuating the amplitude of forced oscillations. The turbine rotor typically adheres to a configuration wherein it is positioned on sliding bearings. These bearings perform the dual role of providing support to the rotor and mitigating its vibrational tendencies.

6. Vibrations Caused by Steam Flow Irregularities

To analyze the impact of steam flow on rotor vibration, it is important to account for the effect of steam flow on the turbine blades. Turbine blades are subjected to various external factors, which include surface pressure arising from the steam flow, as well as centrifugal force arising from rotation. The applied loads possess the potential to induce a multitude of interactions among various components of the turbine, encompassing but not limited to the discs, vanes, blades/blade roots and even the seals located within the turbine structure [41]. Flow instability in turbine machines is a prevalent phenomenon that is often attributed to flow irregularities resulting from a diverse set of factors, which may include turbulence, inlet distortion, set-up effects, as well as blade-tip vortices [29].

The presence of non-uniform steam flow may result in the generation of forces within the resonance field at the blade system. When the frequency of the harmonic force is in close proximity or identical to the natural frequency of the blade system, the phenomenon of resonant vibration transpires, consequently resulting in an increased vibration of the blades [41]. Nevertheless, the fluctuating flow field may lead to pressure variations in the blade tip seal, resulting in temporal changes in the steam-induced vibration force of the blade tip clearance, as presented in [42].

The primary concern arising from the vibration of turbine blades in turbomachinery is linked to the fluctuation of the working medium distribution along the blade, undergoing changes throughout a single blade revolution. This array of circumstances is attributed to the angular dependence of the blade load, leading to the tangential loading of the rotor blades by the operational medium, causing disturbances in the flow. Non-uniform loading of blades during rotation may result from partial admission, where impacts of the active medium on rotor blades occur in single or multiple sections at the circumferential orientation of the turbine.

This gives rise to conditions with sectors that remain unloaded or partially loaded. The inherent frequencies of each rotor blade are fundamentally dependent on the angular velocity determined by the centrifugal force and stiffening factors [7]. Additionally, the introduction of vibration induced by the non-uniformity of the steam flow engenders fluctuating stresses which combined with tensile stresses, leading to corrosion cracking on the blade surface. The cumulative impact of various factors, along with the formation of fatigue paths, contributes to the initiation of blade cracking, as it is underlined in [43].

The phenomenon known as steam whirl manifests itself as a self-sustaining oscillation of the primary mode of vibration within the rotor during conditions of heightened operational demand.
Furthermore, despite the implementation of tilting-pad bearings as a preventive measure against oil whip, the phenomenon of steam whirl can still manifest. The production of a steam whirl is initiated by the forces of excitation that are generated as a result of the motion of the blades and seals. When

there is an increase in turbine load, it is common for steam whirl to manifest in rotors of high-pressure steam turbine systems.

Distinguishing the impact of steam whirl on a real rotor from other effects is challenging. This difficulty arises because of the unpredictable excitation effects on the rotor caused by partial admission operation in a high-pressure turbine, leading to fluctuations in bearing loads. These effects become more noticeable when the inherent frequency of the first mode is lower than the rotor speed. The resulting vibration of the rotor system's first mode due to flow instability, rubbing, and steam whirl is termed subsynchronous vibration [24].

The efficacy of the operational phase may also be adversely affected due to the formation of corrosive deposits on blade surfaces, indicating a reduction of several percentage points. The primary underlying cause of turbine blade degradation is attributed to the acceleration of stress-induced disintegration of surface quality. This assertion is notably applicable to process turbines, geothermal steam turbines, and large-scale utility steam turbines [7]. The aforementioned inconsistencies are evidenced through the occurrence of gusts i.e. vortical waves within the blade. The occurrence of such gusts is linked to various unfavorable consequences when the frequency of aerodynamic excitation closely matches the natural frequency of the machine blade in this case. Generation of destabilizing forces on the blade's surface can result in noise and forced vibration, as exemplified in [29].

The article number [44] provides an investigation into the interaction of rotor blade oscillation and rotation, as well as the external distribution of non-uniform pressure. The inter-blade phase angle of oscillations is influenced not only by the lag of unstable forces but also by the inherent frequencies of the blades. The non-uniform distribution of pressure results in a heightened presence of low frequency unsteady forces, which surpass the impact of high frequency excitations. The amplitude-frequency spectrum has been empirically revealed to encompass harmonics that exhibit frequencies which do not correspond to integer multiples of the rotation frequency.

The partial opening of a steam control valve can also engender substantial vibratory activity within the interconnected piping subsystem, predominantly in a middle-opening state, as is discussed in [45]. Yet, understanding the flow characteristics of a valve near flow through experimental means poses a challenging task. This is attributed to the intricate three-dimensional structure inherent in the flow near the valve, compounded by its supersonic conditions ($M>1$). Furthermore, an asymmetrical flow that is linked to the body of the valve (also known as the "valve-attached flow") becomes evident, the flow interaction from the opposing side at the mid-opening condition creates a local high-pressure zone in a specific location [45].

The region in question undergoes circumferential rotation, which results in the application of a cyclical lateral force on the valve structure, commonly referred to as "rotational pressure oscillations." The present inquiry is centered upon the correlation between the interaction of valve and fluid with the lift, denoting the gap between the valve and valve seat, as well as the presence of the spinning pressure fluctuation [45]. In the event of partial admission, random excitation forces primarily excite the first bending mode, and if the random excitation force is quite great, the rotor vibration becomes apparent.

As a result, most random rotor oscillations can be eliminated or reduced by removing or reducing the undesired random excitation force. Increasing the rotor damping is another useful countermeasure for reducing the rotor's reaction to random excitation forces. Since partial admission not only provides random rotor excitation forces, but also varies the bearing load, it causes changes in the dynamic characteristics (stiffness and damping) of the bearing. As a result, when designing high-pressure steam turbine rotors, the effect of partial admission operation on rotor dynamics must be considered, as is underlined in [24].

In concluding remarks for this section, it can be underline that the occurrence of supersonic flow leads to the generation of shock waves, culminating in fluctuations in pressure and impairment of steam flow at the turbine intake. The aforementioned analysis leads to the inference that perturbations in the flow of the control valve can trigger undesired outcomes like blade and rotor oscillations. Forced vibration can have detrimental effects when the frequency of aerodynamic stimulation aligns with the natural frequency of the equipment. Identifying and eliminating irregular patterns that are leading to significant induced force amplitudes is crucial, regarding the pervasive nature of certain non-uniformities.

The formation of deposits, while typically resolvable, remains a matter of concern that cannot be disregarded. The chemical water treatment plant and the overall condition of the plant, including the steam and condensate pipelines, heat exchangers, valves, among other factors, greatly influence the quality of steam. The introduction of impurities, i.e. inclusions into a system has the potential to exert a deleterious effect on both the quality of steam and the overall functioning of the plant. In instances where deposits on turbine blades exceed a thickness of 30μm or are deemed to pose a threat to steam flow stability despite being thinner, have to be eliminated. To mitigate the potential occurrence of heightened axial thrust and the onset of corrosion. The chemical composition of the deposits necessitates evaluation as a fundamental component of a comprehensive assessment of the deposits, since it could facilitate water conditioning procedures to establish a precise elucidation of the source of the deposits.

It is noteworthy to mention that the nozzle control governing also plays a part in inducing flow perturbations. The result of this phenomenon is the occurrence of an unsteady flow regime surrounding the entire circumference, leading to incomplete stage filling and boundary layer separation. It is also worth to mention the possibility of inserting instrumented probes into the flow section, considering the various research analyzes that can be implanted, including the monitoring of measurements during operation.

7. Vibrations caused by turbine loose parts

During the operation of a steam turbine, the various turbine parts are subject to an arduous operating environment. Various external stimuli such as vibrations, temperatures, and pressure forces can engender deleterious consequences, encompassing erosion, corrosion and pitting are ultimately leading to the loosening of steam turbine components [46].

The rigidity of the system over time experiences fluctuations, which may result in components becoming dislodged from their respective positions within the system. The presence of looseness in a machine pedestal joint or an oversized bearing become apparent only when the periodically acting vertically upward rotating unbalance force surpasses the gravitational force responsible for maintaining close contact between the joints [10]. The vibration data, predominantly characterized by a low-frequency component that exhibits temporal variations, can be correlated with the loosening of low-pressure rotor blades [43]. By scholarly source [47], in the maintenance process turbine blade inspection was conducted following standard procedures. The investigation determined that the abnormal vibration in the low-pressure rotor is primarily resulted by blade looseness.

The emergence of looseness can be attributed to wear and tear on the supporting mechanism, as well as the gradual loosening of fasteners over an extended duration [10]. Typically, rotor lateral vibration design criteria are established through practical application. They are determined not only by rotor resistance, but also by environmental factors, bolt failure and loosening, bearing life and probability of failure, and etc. [24].

The unevenness of stiffness that is produced by the aforementioned influences, causes irregular oscillations instead of the expected operational behavior and working harmonics [19]. Seal strips, if not properly secured, may be loosed, detach and travel through the steam path. It is crucial to inspect seal strips after caulking them into a stationary component to guarantee their stability against the steam forces that affect them during operation. This verification is essential to prevent potential issues caused by loose or detached seal strips in the steam path, as is mentioned in [48].

Looseness in nonrotating connections, such as bearing caps, bearing mounts, or base mounts can also lead to vibration issues. This is because these connections play a critical role in constraining the shaft to its rotational centerline. The dynamic characteristics resulting from the looseness of these components are likely to introduce a significant degree of dynamic nonlinearity into the vibratory system. For example, intermittent in-and-out hard contacts may occur as components vibrate through dead-band gaps created by the mechanical component looseness. The looseness in a rotor-mounted component, such as a thrust collar, spacer collar, impeller ring, or slinger disk, can also cause vibration problems. This kind of looseness is likely to induce mass unbalance on the rotor, potentially resulting in nonsynchronous vibration, although synchronous vibration is also possible. If the combined effects of looseness and other factors permit the rotor-loose component to spin at a speed different from the shaft speed, nonsynchronous vibration is likely to be present. Various factors, including prevailing friction conditions, clearances, and fluid or aerodynamic drag forces provide a wide range of possibilities for different types of steady or unsteady vibrations. A loose component might lock its rotational speed into one of the rotor–bearing system's subsynchronous orbital natural frequencies, disguising itself as a self-excited vibration type. In the case of a driven element like a turbine disk, resulting nonsynchronous vibration would occur at a frequency above the rotor spin frequency. Otherwise, any nonsynchronous vibration would likely be at a subsynchronous frequency [5].

The case study pertaining to the 350MW cross-compound turbine-generator as explicated in reference [5] is deemed highly significant, considering the possibility of design errors that may still arise within turbine production. The journal bearings of the high steam turbine, experiencing a loosening phenomenon after a six-month period of operation, causing significant escalation in the critical-speed vibration peak magnitude of the HP turbine. In sense of loosening and rubs, a envious number of analysis and experiments are performed and described in [10]. The purpose of these experiments is to establish which pattern of vibrations represent steady-state processes, and what are the most sensitive system parameters affecting changes in these steady-state regimes.

Detection of loosed components through vibration monitoring can be certain type of challenge due to the presence of interference in vibration signals. These features may include abrupt changes in vibration signals, the occurrence of subharmonics, unstable phases within the signal, and fluctuations in amplitude. Even false readings may occur as a result of errors in vibration measuring devices. However, while loosening often occurs during operation, it can also result from inadequate bolt fastening, typically due to failure to follow proper procedures.

8. Vibrations caused by damaged turbine part

One of the most common reasons for premature structural failures in mechanical systems is the development and propagation of cracks. Rotating shafts, even when not exposed to regular operating conditions, are prone to the initiation and progression of cracks, especially in areas with significant stress concentrations. Cracks in shrink-fit connections can also arise due to fretting corrosion, particularly in environments with high moisture and corrosion levels. Despite the absence of any variation in the imposed external loads, the emergence of even a minor crack can result in the development of high stress and strain intensity factors at the crack tip, ultimately leading to its further propagation [49]. It is important to note that propagating microstructural short crack can continue to grow acceleratingly. Crack length, rate of growth, shape, orientation and their coalescence, these factors take precedence in influencing the failure process [50].

Transverse cracks are a prevalent issue in rotating shafts found in industrial machinery, and their occurrence and propagation are not to be underestimated. Nevertheless, in the majority of cases, such shafts are decommissioned prior to experiencing a catastrophic failure. Detecting cracks in rotating configurations presents a more significant challenge compared to stationary (non-rotating) equipment, mainly due to the inherent difficulties in monitoring a rotating shaft during operation. Therefore, it is crucial to identify quantifiable indicators that can serve as evidence of a crack within the crystal structure of a rotating shaft material, which is explained in [49].

The examination of bending vibrations provides a distinct approach for diagnosing rotor vibration that does not require the disassembly of the shaft. The application of longitudinal and torsional vibrations for diagnostic purposes in individual rotors necessitates the disassembly of the shaft, thereby exposing detectable fissures that are only visible in dismantled rotors. It's important to note that torsional vibration-based diagnostics demonstrate slightly higher sensitivity compared to their longitudinal vibration-based counterparts. However, the application of longitudinal and torsional oscillations for shaft vibration diagnosis during turbine operation has generally yielded unsatisfactory results. Holding other variables constant (e.g. crack location and step diameter), a small crack becomes noticeable under heightened loading in the breached section. The degree of variation in a fractured segment is in a direct and linear correlation to the intensity of the load applied to the fractured area but inversely proportional with the amplitude of longitudinal bending, and torsional vibrations of the affected region's diameter.-Therefore, in certain situations, it is easier to detect a crack in a moderately loaded step of the rotor with a relatively small diameter [51].

The location of a crack can greatly affect turbine vibration. Crack tensions and forces near the coupling or bearing are higher than those in the central section between bearings, which experiences increased bending moments. Open and closed cracks have natural ability to change shaft's stiffness, affecting static deflection and natural frequency. However, significant fissures may cause minimal changes in rigidity and vibration frequency, depending on the crack location. The magnitude of crack-induced forces depends on stimulating forces and dynamic amplification variables related to rotational velocity and fracture locations. These factors change as cracks propagate through the material [49].

Factors contributing to fatigue failure are influenced by temperature fluctuations and operating conditions. Complete failure of the shaft can lead to severe consequences, including reduced productivity, equipment damage, and potential risk to human safety. Any characteristic that hinders a component's suitability for its intended purpose is considered a flaw or defect in academic terms. To ensure safe and reliable operation, diligent maintenance and diagnostic technologies are essential, as is concluded in [52].

The occurrence of torsion appearances carries noteworthy consequences. Torsion vibrations in rotating shafts arise from the same coupling mechanism responsible for generating torsion deflection in a static context. As a result, the lateral oscillations display a proportional connection with the static bending moment in alignment with the crack, while the torsional stimulation is directly linked to the shear force at the exact location of the crack [53]. The occurrence of unintended natural frequency, resonance, and consequent blade vibration, or stress-induced blade cracks can transpire as a result of an aberration from the nominal design of blades during the turbine operational [24]. Fractures in the blade attachment region can lead to the rupture of the disc rim and release of the blade, causing cascading damage to the steam pathway, increased shaft vibration, impairment of bearings, and other potential effects. However, cracks on the internal bore surface may pose comparable, if not more significant risks, as they can propagate and fracture the entire disc, resulting in the destruction of the turbine [54].

The occurrence of blade cracking, leading to blade detachment, can cause a sudden increase in vibrations. This happens because the absence of blades on one side of the rotor creates an imbalance. In such cases, it's crucial to stop the turbine, rebalance it, and consider replacing individual blades. Reinstatement to operational status for the turbine without blade replacement, after inspections and the determination of its satisfactory condition of the complete rotor is a rare but plausible occurrence, attributed to the exigencies of the need for an electrical power supply.

9. Vibration Caused By Temperature Influence

In order to meet the high power needs, high-capacity steam turbines, especially in supercritical and ultra-supercritical power units, have been extensively incorporated into power plant systems. To meet demanding requirements, martensitic steels are widely used in various components that face loading conditions requiring creep resistance to withstand elevated temperatures and pressures in steam turbines. Due to the operational stresses, steam turbine rotors are prone to damage from long-term creep, in addition to more rapid damage during startup and shutdown phases. The duration of temperature dwell time exerts significant influence on the phenomenon of creep fatigue manifested by rotors that are continuously operative under conditions of peak loading. The understanding of analytical projections pertaining to the impact of dwell time on creep-fatigue phenomenon is a significant resource in the enhancement of steam turbine rotor design optimization, as is shown in the literature [55].

The manifestation of high-cycle fatigue in rotational configurations poses a noteworthy predicament, which may culminate in the malfunctioning of the system. Insufficient comprehension of the genuine characteristics of a rotor can result in its damage. An increase in temperature has been identified as a additional influence in the development of cracks during high-cycle fatigue, as evidenced in the literature [52]. The initiation and dissemination of fractures may be initiated by thermal stresses and thermal shocks, both of which give rise to significant local stress intensity factors [49]. In fact, there is a certain combination of high-cycle fatigue and low-cycle fatigue.

Because of the aforementioned factors, the designation "thermal sensitivity" was designated to denote an atypical response specific to flawed rotor systems, wherein vibrational fluctuations are modulated by thermally induced transients acting upon the rotor structure. The transition of the operating condition from full valve arc admission to partial valve arc results in a reduction of the first stage steam temperature and induces a cooling transient in the rotor. The converse scenario ensues when the operational state is transitioning from partial arc to full arc. The thermal sensitivity in question is considered to be a highly discernible and dependable indicator of a transverse fracture within a rotor [53].

The phenomenon of differential thermal expansion is observed during the beginning of unit startup where the rotors tend to exhibit a faster expansion rate as compared to the stationary components, specifically in the axial direction. While during a period of shutdown, there occurs an inverse trend. Insufficient clearance between the casing and rotor may lead to axial friction known as rubs that can occur within turbines. As has been aforementioned, disparate rates of thermal expansion between rotors and casings may result in the constrained axial clearance. Both axial and radial rubbing scenarios exert direct impact on the rotor, inevitably resulting in an escalation of the rotor's vibration levels. The phenomenon of a pure axial rub event is characterized by an amplified rotor displacement and/or an inordinate axial expansion in contrast to a pure radial rub event, which solely induces lateral deflection. The rubs can exhibit predominantly radial or predominantly axial manifestations contingent upon the orientation of the cylinder distortion as is underlined in [6].

The vibration known as bowed rotor vibration, commonly referred to as a "bent rotor" is a characteristic consequence of thermal bending. Inability to restart a high temperature rotor that has been stationary for an extended period arises from a noteworthy imbalance. The rotor undergoes bending due to a thermal gradient between the upper and lower portions induced by the accumulation of high-temperature gases on its top surface. In this conditions, rotor-stator friction appears which is also recognized as the Newkirk effect [24].

Viscous heating which causes a unidirectional temperature rise in the bearing journal, causing thermal expansion and subsequent bending of the rotor is called Morton effect. This results in distribution of unbalance with heating and bending escalation, ultimately leading to an unstable or self-amplifying response. A similar behavior associated with Morton effect is called Newkirk Effect, where the same process of heating, thermal expansion and shaft bending occurs, but the heat source is a labyrinth rub [56]. In the event that the temperature of the lubricating oil surpasses the predetermined threshold for safe operation, it leads to an increment in the hydraulic pressure fluid employed in mechanical hydraulic control systems, thereby elevating the displacement of the turbine shaft [57].

The probability of encountering radial friction escalates proportionately to the degree of relative misalignment from the initial design alignment especially when the housing becomes unintentionally misaligned with a rotor. The vibrational responses of the rotor and the structure can be attributed to the presence of radial and axial friction. They commonly manifest during the commencement or cessation of operations of the equipment, whereby the rotational speed of the shaft is increased or decreased [6]. To comprehensively analyze bend thermal instability of the rotor, it is crucial to consider the dynamic thermal nature of the problem. Relying solely on steady state is insufficient, synchronous journal orbits doesn't suffice to explain potential thermally unstable behavior. Anticipated modifications in orbit ellipticity, both in magnitude and orientation, occur on a slow timescale. A similar phenomenon is observed in situations involving frictional heating when the rotor contacts stationary components [58]. An issue concerning the increase in bearing temperature caused by wear is documented in [59], it has been posited that this phenomenon can negatively affect the efficient and stable operation of the steam turbine unit.

The degree of rotor deformation is subject to variation, contingent upon the particular operational conditions, that are determined by the intensity of the thermal influence. In principle, dynamic effects of thermal bending can be influenced by multiple parameters, specifically the proximity of the running velocity to a resonance frequency of the system, the degree of system damping, and the deflection of journal within the system. There is a probability of encountering instability in rotor dynamic systems that operate at high speeds in close proximity to modes where rotor thermal distortion gives rise to substantial imbalance forces, as per source [58]. In the context of low power steam turbines, it is noteworthy to acknowledge the values of thermal expansion that may be incorporated into the process of shaft alignment. The misalignment of the shaft train can have a significant impact on the failure not only of the shafts but also of the parts connected to a turbine gearbox, which are already influenced by thermal expansion [60].

Concluding this section, the manifestation of amplified vibration in the rotor can be also attributed to the detrimental impact on a particular section of either the complete rotor or blade. In the event of a partial or complete blade fracture, significant vibrations are promptly exhibited due to the absence of blades in a specific section of the turbine. This difference may lead to significant imbalances, potentially worsening damage if corrective actions are not taken and operations are stopped. In view of the apparent damage observed in the rotor and the manifestation of crack formation, it is imperative to further investigate this phenomenon. Irrespective of its location within a material, damage regardless of whether it is on the surface or inside the material itself has the propensity to initiate an imbalance.

Newkirk effect explanation by [24] is quite unusual, considering numerous papers written on this manner, giving the view of the effect during turbine operation, not in stationary conditions. Nevertheless, explanation by [56] is insufficient seeing that thermal source of Newkirk effect is made by contact of the any part of the rotor’s sleeve with the turbine bearing or any stationary components, not just labyrinths. In case of Morton effect heat is generated by shearing of the lubricant film without presence of rubbing where bending and rubbing can be manifested as secondarily consequential phenomenon. Thorough explanation about difference of those effects, can be found in [61].

In the presence of temperature, a localized crack on a rotor results in the expansion of the material in the affected region, ultimately leading to a change in the overall geometry of the rotor. As the crack propagates, the geometric configuration of the rotor will continue to alter, consequently leading to a state of imbalance. The phenomenon of vibrations is a complex physical process that involves the oscillation of an object or system about its equilibrium point. It is characterized by the periodic motion of particles within the medium that carries the vibration, and can be mathematically described by various wave equations and models.

The problem mentioned above does not exclusively originate from exploitation or escalated vibrations induced by other phenomena discussed within this manuscript. Impurities can lead to damage as well. The aforementioned refers to the inclusions that are present within the material constituting the rotor that were generated as a result of the manufacturing process. In cases where the elevation of vibratory activity proves persistent, it is recommended that the involved manufacturer of the equipment be contacted with a view to find a viable solution to the issue at hand. In the case of an accident, it is imperative for the involved parties to refrain from relocating the turbine equipment and components until a comprehensive inquiry is conducted, determining the precise cause of the incident. This cautious approach ensures a thorough understanding before any further actions are taken.

10. Vibration caused by steam turbine generator and electricity grid

Turbine-generators utilized for supplying power to the electrical grid may exhibit sensitivity to the diverse torques exerted by the grid during routine operation. This phenomenon orders the requirement for turbine-generators to have the function capacity under prescribed steady-state oscillating torques. These torques may arise from the generator's interaction with the natural frequencies of certain electrical grids, such as sub-synchronous resonance (SSR) or from negative sequence current (NSC) torques, produced at twice the grid frequency due to unbalanced loads on the generator terminals. Furthermore, turbine-generators are frequently subjected to sudden transient torques stemming from various operational disturbances such as synchronization, line switching, turbine or generator trips, and electrical grid faults. These impulse loads can excite all torsional modes of the system. In that sense, grid has the capability to produce both steady-state and transient oscillating torques on the generator [62]. Torque enhancement in turbine generators associated to a series-compensated can cause excessive fatigue loss or unexpected shaft failure, thus it is posing a serious threat to shaft safety [63].

The electric grid may experience unbalanced load conditions, such as those observed during short circuit events or when subjected to highly unbalanced loads. This phenomenon will predominantly result in the stimulation of torsional shaft oscillations, posing a significant hazard when the eigenmodes align closely with the line frequency and double line-frequency of the electrical system. Radial and torsional vibrations can be stimulated by impact of additional network-induced phenomena resulting from non-synchronous loads. Those loads occur in proximity to significant electrical energy consumers, as well as grid resonance in systems characterized by long transmission lines. Also, the primary consequence of this phenomenon is the generation of radial shaft vibrations, particularly as the eccentricity of the rotor within the stator increases [64].

When the rotor displays eccentricity, a consequential imbalance of forces transpires, giving rise to the development of a radial force. The imposition of a radial force, commonly referred to as Unbalanced Magnetic Pull (UMP) has been found to induce vibrations and noise accelerating bearings deterioration. Potentially resulting frictional contact between the rotor and stator, leading to stator windings damage [65]. In case of research by [64], it was found that the vibrations were caused by the nonlinear behavior of the UMP. This excitation was enhanced under certain network conditions. After determination that vibration amplitude remains at an acceptable and stable level over time, no special short-term measures were taken to reduce the vibrations. A few months after the vibration increase, it was enough to change the network configuration to return the vibration amplitude to the original values. This action also indicates that the cause of the vibration is likely to be the condition of the network rather than deterioration of the electric components or their installation conditions.

The occurrence of a short-circuit fault in a generator or its inaccurate synchronization upon being brought on-line result in a brief surge of reactionary torque that is exerted from the generator end onto the turbine shaft, thereby inducing torsional vibrations in the shaft. The amplitude of the vibrations in such scenarios is contingent upon the characteristic parameters of the reactionary torque surge and the level of vibration damping present within the system. In such a state, turbine loading may exhibit diverse responses depending on the varying surge parameters, incurring fatigue damage primarily within the shafting material when subjected to a sudden force acting on the generator side. The degree of damage sustained is contingent upon the waveform of the reactionary torque surge. It is observed that the rectangular surge poses a greater threat to cyclic damage in the rotor material compared to the biharmonic surge. Furthermore, the magnitude of the surge directly correlates with the extent of fatigue damage, where a larger surge magnitude results in greater damage. Additionally, the level of vibration damping in the system plays a crucial role, higher damping levels are associated with reduced torsional vibration amplitudes and subsequently less cyclic damage in the material [66].

In the scholarly literature [66] has been noted that fatigue damage can accumulate in a turbine shaft over an extended period of operation, potentially leading to failure when the shaft reaches its limit state, as evidenced by the case of the Gallatin Power Plant. The current status of assessing fatigue damage in turbine components as a result of abnormal operating conditions of a turbine-driven generator set remains an unresolved issue. This phenomenon can be demonstrated through empirical data regarding the reactionary torque surge and torsional vibrations experienced in turbine shafting, typically acquired through vibration monitoring of a steam turbine during actual operational usage.

Fatigue damage in the general case is determined by three parameters: the number of failed connections, duration of the electromagnetic torque at the failed connections and the phase shift angle, where the intensity of torsional vibrations essentially depends on this angle as well as from the duration of electromagnetic torque. It is imperative to verify the measurement of angle whenever the turbine generator is connected to the electrical network, with the objective of maintaining its value as minimized as feasible considering that operational safety of a steam turbine is significantly influenced by the frequency of faulty connections and the specific angle [67].

This topic addresses the explanation of phenomena and categorizes occurrences into two distinct groups. The initial category would consist of deficiencies in the equipment interfacing with the generator terminals, as well as the operational parameters of the turbine-generator within the power plant referenced in this paper. And there would be the second category, consisting of faults or short circuits, scheduled or emergency switching of transmission lines, continuous fluctuations due to unbalanced phase currents, fluctuations due to the application of power electronics, etc.

Due to the aforementioned reasons, it is imperative to ascertain the precise underlying cause of the issue in order to avert the progression of the situation towards more unfavorable outcomes.

Conclusions:

In academic research papers, it is infrequent for an assessment to be made regarding the specific number of working hours, prior to the emergence of rotor damage or whether damage was due to prolonged wear and tear during excessive stress operation. Rather, a typical focus consists of the examination of heightened levels of vibrations and the ensuing inquiry into the associated crack occurrence. Prolonged testing aimed at evaluating the propagation of cracks or other forms of damage resulting from increased vibration may not be adequate, as vibration testing is solely intended for defect detection as a preemptive measure to avoid further escalation of the damage. In the event where damage has already transpired, it is imperative to engage in proactive measures through monitoring heightened levels of vibrations, ceasing the operation of the turbine and identifying the causative factors to avert a substantial malfunction.

The comprehensive analysis of vibration should not only be related to turbine rotors. Considering that, attention must be paid to the entirety of the turbine assemblage, including the generator and all belonging equipment. It is of crucial significance to conduct an extensive evaluation of the overall system to ensure optimal functionality.

The phenomenon of vibrations attributed to rubs as a distinct cause was not considered in a separate topic of the vibration appearances, as it is perceived as an outcome of the underlying causes expounded in this paper.

In light of access to simulation programs, it is possible to establish and assess the operating parameters and antecedent factors that lead to heightened vibrations in diverse phenomena. Through this appraisal, it can be offered meaningful contributions to the body of knowledge in a progressive manner enabling the proactive anticipation and mitigation potential issues that may arise during the operation of turbomachinery.

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