LIFETIME ASSESSMENT OF HYDROPOWER UNITS

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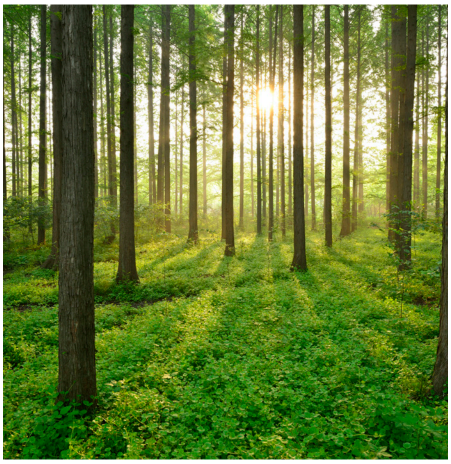

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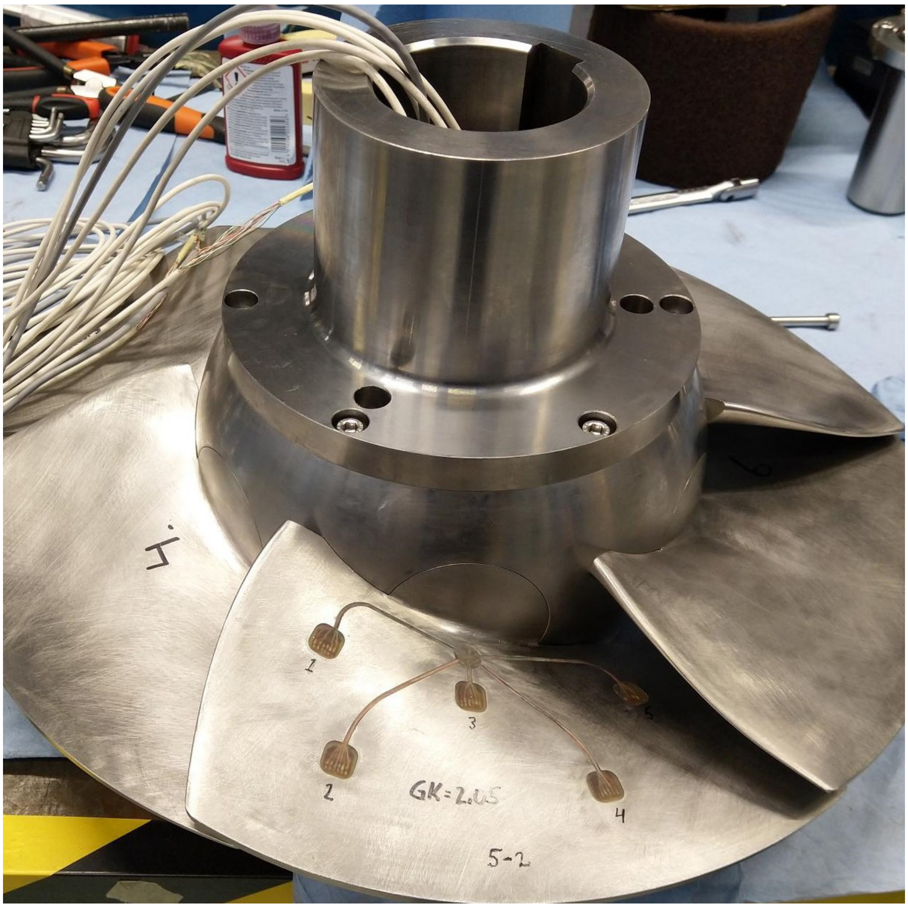

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Synthesis of research to enable lifetime assessment of hydropower units

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Foreword

Lifetime evaluation is crucial for ensuring the high availability and reliability of machines, aiming to achieve their expected service life by addressing specific damage mechanisms. Hydropower units are unique machines with varying sizes, geometries, and mechanical properties, making it challenging to define general turbine loads and their distribution throughout the machine.

The research and findings detailed in this report are the collective efforts of senior researchers from various esteemed institutions, including Jan-Olov Aidanpää, Michel Cervantes, and Kim Berglund from Luleå University of Technology, Håkan Nilsson from Chalmers University, and Carl-Maikel Högström and Rolf Gustavsson from Vattenfall. Their collaborative work synthesizes the current state of the art and presents insights from their own experiences within SVC research projects, laying the groundwork for future research aimed at developing a robust methodology for the lifetime assessment of hydropower unit

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Summary

Lifetime evaluation is an essential analysis to secure high availability and reliability of any machine, to reach its expected service life associated with a particular damage mechanism. Hydropower units are normally unique machines of different size, geometry and mechanical properties, which makes it even more difficult and challenging to define general turbine loads and how these loads are distributed through the machine.

One example is the blade setting mechanism for Kaplan machines where the load distribution between individual links, lever arms and bearings etc. are unknown. The load distribution may depend on variations in external and internal loads, bearing friction, manufacturing tolerances and how the runner is assembled. In addition, the rotors are normally vertical and hence the bearing models becomes nonlinear.

In this report, the senior researchers of the SVC work package “Hydropower Technology” have summarized the state of the art together with own experience from SVC research projects. The report is limited to the hydropower unit and the research areas fluid dynamics, rotordynamics, solid mechanics and machine elements. Possible damage mechanisms and loads are listed together with different instabilities that may be applicable to hydropower units. To make the research efficient and to prioritize and direct its resources where it creates most value, it is important to define which are the critical components depending on damage mechanism.

One main result and conclusion is that measurement technology needs to be evaluated and developed, to get the forces that excite the unit during different operating conditions as well as developing condition monitoring systems and tools for diagnose and prognosis. Knowledge of the forces on the system, operation pattern and damage mechanism is essential to be able to determine the expected life of different components. The report also suggests needed research to get closer to a methodology for lifetime assessment of hydropower units. One problem is that lifetime assessment needs a holistic approach while research is on a detailed level. Therefore, if lifetime assessment becomes a prioritized area there is a need of a project leader that guides and concludes research results towards a common goal.

The highlighted text discusses the importance of lifetime evaluation for ensuring the high availability and reliability of hydropower units. It highlights the unique challenges posed by the different sizes, geometries, and mechanical properties of these units. The text also underscores the need for advanced measurement technology and condition monitoring systems to understand the loads and damage mechanisms affecting these systems. Furthermore, it suggests that future research should aim to develop a comprehensive methodology for lifetime assessment and emphasizes the necessity of having a project leader to guide and unify the research efforts towards a common goal.

Keywords

Hydropower, lifetime, assessment, solid mechanics, fluid mechanics, electricity and machine elements

Vattenkraft, livslängd, bedömning, hållfasthet, strömningsteknik, elektricitetslära och maskinelement

Sammanfattning

Livstidsbedömning är en viktig analys för att säkerställa hög tillgänglighet och tillförlitlighet för alla maskiner, för att nå dess förväntade livslängd förknippad med en viss skademekanism. Vattenkraftaggregat är normalt unika maskiner av olika storlek, geometri och mekaniska egenskaper, vilket gör det ännu svårare och utmanande att definiera generella turbinlaster och hur dessa laster fördelas genom maskinen.

Ett exempel är bladinställningsmekanismen för Kaplan-maskiner där lastfördelningen mellan enskilda länkar, hävarmar och lager mm är okänd. Lastfördelningen kan bero på variationer i yttre och inre belastningar, lagerfriktion, tillverkningstoleranser och hur löparen är monterad. Dessutom är rotorerna normalt vertikala och därför blir lagermodellerna olinjära.

I denna rapport har seniorforskarna inom SVC-arbetspaketet ”Turbiner och generatorer” sammanfattat det aktuella läget tillsammans med egna erfarenheter från SVC-forskningsprojekt. Rapporten är begränsad till vattenkraftsaggregat och forskningsområdena strömningslära, rotordynamik, hållfasthetslära och maskinelement. Eventuella skademekanismer och belastningar listas tillsammans med olika instabiliteter som kan vara tillämpliga på vattenkraftaggregat. För att göra forskningen effektiv och för att prioritera och rikta dess resurser dit den skapar mest värde, är det viktigt att definiera vilka som är de kritiska komponenterna beroende på skademekanism.

Ett huvudresultat och slutsats är att mätteknik behöver utvärderas och utvecklas, för att få fram de krafter som exciterar enheten under olika driftsförhållanden samt utveckla tillståndsövervakningssystem och verktyg för diagnos och prognos. Kunskap om krafterna på systemet, driftmönster och skademekanism är väsentlig för att kunna bestämma den förväntade livslängden för olika komponenter. Rapporten föreslår också nödvändig forskning för att komma närmare en metodik för livstidsbedömning av vattenkraftenheter. Ett problem är att livstidsbedömning behöver ett helhetsgrepp medan forskning är på detaljnivå. Om livstidsbedömning blir ett prioriterat område finns det därför behov av en projektledare som vägleder och sammanfattar forskningsresultat mot ett gemensamt mål.

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1 Introduction

Lifetime assessment together with condition monitoring are essential tools for obtaining high reliability and availability. Tools for lifetime evaluation depend on the research subject and in this document, we have focused on the hydropower rotor and generator and the research subjects: electricity, fluid mechanics, solid mechanics (rotor dynamics) and machine elements.

There is also a need to distinguish between different types of phenomena that can cause the failure. In this report, we have divided failure into two groups. Degradation due to operating conditions and age e.g. fatigue, wear and ageing of materials. The second group is failure due to sudden events, which include nonlinear phenomena, parametric resonance and instabilities. The aim is to use classical deterministic methods to predict life instead of probabilistic methods which are frequently used in availability and reliability assessment [1]. Deterministic methods are more suitable for hydropower units where each unit is unique while probabilistic methods are useful for mass produced products. To fully use deterministic methods, all excitation forces need to be known why a main part of the report describes forces under different operating conditions.

In publications on hydropower failures, the most frequent problems are related to the turbines. Some examples are fatigue in Francis turbines, cavitation and erosion. Many other common failures are not reported, and severe failures are mainly reported in newspapers. Therefore, there is a lack of information about common failures since there are normally legal issues which make them confidential. The aim of this report is to list possible failure mechanisms and excitation forces which can be useful in damage investigations and new areas of research needed.

In rotor dynamics the vibration levels can be solved if the loading condition is known. These vibration levels can then be transformed into stresses in critical components for fatigue or fracture analysis. Today both linear and non-linear rotor models for vertical hydropower rotors are developed and the main problem is to know the excitation forces and how they depend on different operating conditions. Methods for fatigue and fracture analyses are well defined in solid mechanic textbooks [2]. Besides fatigue, rotor dynamics also aims to avoid catastrophic events such as resonance or instabilities which can cause severe damage to the system.

2 Relation to Lifetime Assessment in Each Research Area

2.1 ROTOR DYNAMIC

The main purpose of rotor dynamic is to design a robust machine. In the design phase this is done by avoiding natural frequencies and low modal damping (damping ratio). To make a good analysis it is of importance that excitation frequencies are known and that the model is good enough. One main problem in hydropower dynamics is that rotors are normally vertically oriented. That results in unknown bearing loads and therefore difficulties to make a correct model.

If the forces are known (fluid and magnetic) the same models can be used to evaluate stress and strain on critical components. This can be used for analysing low and high-cycle fatigue or fracture mechanics, if the geometry and material properties are known. Other simulations can be used to evaluate e.g. nonlinear effects, instabilities, turbine contact, short circuit or parametric excitation. In such cases high loads can result in plasticity or direct failure.

Experimentally, forces on turbine and generator are the most interesting ones in order to evaluate the risk for failure. Therefore, a measuring technique has been developed at Vattenfall to find these forces. By using bearing models, the bearing forces can be found by measuring the shaft displacement. By adding some extra strain gauges on the shaft, the forces can be evaluated by use of FEM model [3]. From such measurements the cost for different operating conditions can be evaluated. The lifetime evaluation will be divided into two regions namely catastrophic failures and degradation due to excessive loads cycles. It is not possible to decide which category a failure mechanism belong, since it depends on the location if the failure becomes severe.

2.2 MACHINE ELEMENTS

Machine element research is important to ensure the efficiency, reliability, and longevity of hydropower plants. This research focuses on understanding and improving the performance of critical components such as bearings, lubricants, and lubrication systems.

One key area of research is tribology, which examines the principles of friction, lubrication, and wear. Advances in tribology can lead to the development of environmentally friendly lubricants and more wear resistant bearings, reducing maintenance costs and downtime. Additionally, there is a focus to develop methods to predict the service life of critical components, ensuring that hydropower plants can continue to operate efficiently and safely.

The increased use of other types of renewable energy sources like wind and solar increases the strain on hydropower to regulate and control the power output. Consequently, hydropower machines need to have the ability to operate more under varying operating conditions and starts and stops may become more frequent. Kaplan runner hub bearings and guide vane bearings are especially exposed to these issues.

2.3 FLUID MECHANICS

Fluid mechanics is used to design the flow passages of hydraulic turbines, generally composed of a spiral casing, distributor with a certain number of stay vanes and guide vanes, runner and draft tube. The design should grant a high hydraulic efficiency at the best efficiency point (BEP) while minimizing harmful flow phenomena such as excessive cavitation, flow separation and vortex breakdown away from the BEP region. The detrimental flow phenomena can substantially decrease the turbine life at certain operating conditions and tend thus to be avoided, i.e., they limit the operational range and thus flexibility. The different failure modes which may be induced by flow phenomena are described later.

The wide introduction of intermittent renewable energies has moved the function of hydraulic turbines from being mainly an energy provider to a frequency regulator through power injection. These market constraints induce a need for a wider operating range of the turbines. The design is thus moving toward this new paradigm by developing technologies enabling a mitigation of these harmful flow phenomena or consequences away from BEP. Therefore, flow structure interactions are becoming a necessity at the design stage for an efficient turbine design, i.e., a turbine with high hydraulic efficiency, flexibility and life expectancy.

Model testing is used to accurately quantify the operational capabilities of hydraulic turbines while CFD may be used at the design stage and help to elucidate some operational issues. Detailed full-scale experiments are sparse and challenging to perform. All these techniques enable quantifying loads on the runner of interest for rotor dynamic studies. The degree of details and quality of the data correlates to the cost. Methodologies for fluid-structure interactions using numerical and experimental approaches exist but need further refinements for an accurate quantification of the operating conditions on the structures to accurately quantify the cost associated to the operation of a turbine. Some research is being done to connect the flow-induced forces to rotor-dynamical models, to investigate the effects on the entire rotating system.

2.4 INDUSTRIAL R&D

Industrial research is essential to gain access to measurements in models and prototypes. Experiments at prototype scale are however expensive to perform, why experiments in model scale are more justified economically to e.g. validate numerical models and study loads. Research at prototype scale is however essential, and sometime also necessary to better study scale effects and load spectra of both periodic and stochastic nature in real structures.

Industrial partners are also essential to apply research results to new requirements when hydropower units are upgraded. An example is the “Nordic Generator Technical Requirements (NGTR)” which implements research results to develop better machines.

3 Damage Mechanisms

Damage mechanisms can be divided into two areas where the first is reduction of remining life due to unfavorable operating conditions and the second is sudden catastrophic failure. Several failure modes can occur in both areas, why we don't distinguish where each mode belong.

Below, the most common failure modes are listed restricted to the hydropower unit. Also, we have not included the turbine regulation although it is a source of failure.

3.1 WEAR

Wear is a process that occurs when two surfaces are in contact and move relative to each other, leading to the gradual removal of material. There are various types of wear processes, including abrasive, adhesive, erosive, fretting, tribo-chemical, and corrosive wear. The extent and type of wear depend on numerous factors such as contact pressure, sliding speeds, type of motion, material hardness, surface roughness, material compatibility, lubricant, and temperature. Wear can affect many different components in hydropower turbines, such as bearings, seals, hydraulic pumps, cylinders and valves. The expected service life of these components is often very long; for example, Kaplan hub bearings are typically designed to last around 40 years. However, unexpected or premature failure can result in significant costs due to the substantial downtime required for their replacement.

3.2 DEGRADATION AND FAILURE OF KAPLAN RUNNER HUB BEARINGS

Simultaneously as the usage pattern of hydropower is changing, environmental demands continue to increase. Therefore, greaseless self-lubricating bearings have been introduced to replace the grease and oil-lubricated bronze bearings previously used in the Kaplan runner hub and guide vanes [4, 5]. The self-lubricated bearings can be used in both water and air, and the environmentally harmful oil and grease can thus be avoided.

Each blade in a Kaplan turbine is supported by three bearings: two journal bearings and one thrust bearing. Additional journal bearings are present in the mechanical linkage that converts the translational motion of the hydraulic servo into the rotational motion of the blade. The design pressure for the journal bearings is typically around 40 MPa, but local pressures at the bearing edges can reach approximately 90 MPa. The actual pressure levels in the bearings depend on several factors, including the load, bearing clearances, and the materials used. The operating conditions of guide vane bearings are similar to that of the hub bearings, although contact pressures across the bearing are generally lower and more uniform.

System failures can result from changes in friction and excessive wear. Over time, increased friction can lead to the inability to move the blades. Friction levels also affect the loads on the hub, with higher friction accelerating fatigue and reducing service life. Variations in friction behavior, such as a high static friction coefficient compared to the dynamic friction coefficient, can cause stick-slip and significant machine vibrations. Additionally, excessive wear can lead to seal failure and/or the blade tip touching the chamber.

Accurate prediction of service life relies on reliable data regarding how the friction and wear of the bearing material vary under different operating conditions. The friction and wear properties depend on the bearing and shaft materials, counter surface roughness and lay, and any lubricants used.

Practical design considerations include the tendency of polymer composite materials to creep and swell. Creep can affect interference fits, causing the bearing to loosen over time. Gluing can be used to secure the bearing in the housing, but it is crucial to ensure an even distribution of glue to guarantee that the entire bearing is rigidly attached. A summary of some of the important operating conditions affecting the degradation and eventual failure of a self-lubricating bearing can be seen in Table 1.

Table 1. Failure modes and degradation mechanisms of self-lubricating bearings used in Kaplan turbines.

Failure modes

Bearing seizure/Increased friction – No blade movement	Excessive wear can cause seal failure or blade tip touching chamber
Bearing material separation	Bearing material softening/melting
Bearing detached from housing	Changes in friction characteristics (stick-slip)

Degradation mechanisms

Fatigue	Fretting	Stress relaxation	Hysteresis	Creep
Abrasive wear	Adhesive wear	Thermal ageing	Hydrolysis	Corrosive wear

Operating conditions

Environment (*air, water, lubricant, abrasive particles*)	Temperature	Frequency and amplitude of reciprocating movement
Vibrations	Load level	Load oscillation (frequency and amplitude)

A test equipment and method to evaluate the friction and wear behavior of self-lubricating polymer composite bearing materials under various operating conditions representative of hydropower applications was developed in [6]. The test method is useful to screen the friction and wear performance of materials and can establish how the wear rate depends on the turbine's operating conditions. The impact of various operating condition parameters on friction and wear has been investigated for selected commercially available self-lubricating bearing materials used in hydropower [6-10]. A reduction in friction coefficient with an increase in contact pressure was observed [7]. The optimal counter surface lay and roughness to minimize wear and friction depends on the specific bearing material used and can vary from best to worst case scenario [8]. Among the three different surface roughness tested, two of the bearing materials showed the best wear performance for the mid-range surface roughness, indicating that both too smooth and too rough counter surface can decrease the service life of the bearing material used [8]. The stroke length was shown to affect both friction and wear levels with shorter stroke length being more beneficial, e.g. the friction coefficient was approximately twice as high for the longest stroke length compared to the shortest [9]. Based on the wear data acquired for a specific bearing and counter surface combination, changes in bearing clearance or axial bearing wear can be calculated if the blade load, frequency and amplitude of motion, and geometrical dimensions are known.

The challenge of obtaining reliable wear data under varying operating conditions has made it attractive to find alternative methods for tracking the wear of the Kaplan hub linkage, specifically online methods. Peyrano et al. used two proximity probes to measure the gap between blade tip and discharge ring and were able to identify excessive bearing clearance due to wear in individual linkages [11].

Although extensive research has been conducted to understand the service life of polymer composite bearings used in hydropower, there is still limited understanding of how load transients affect the service life. A sufficiently high load level could cause rapid failure of the bearing material. Additionally, the fatigue behavior of polymer composite bearings due to load variations during start-stop cycles and frequency regulation remains an area of concern. Furthermore, many of the commercially available materials today include PFAS-chemicals where polytetrafluoroethylene, PTFE, is one such solid lubricant component which is used to achieve favorable friction characteristics and avoid phenomena such as stick-slip. A coming PFAS ban can affect the use of conventional self-lubricating polymer composite bearing materials and lead to a need for more research and development of new bearing materials.

3.3 LOW CYCLE FATIGUE (LCF)

Many of the solid mechanic problems are due to high vibrations and a common reason is that excitations exist close to the natural frequencies. Therefore, the phenomenon of natural frequencies is explained in Appendix I. If the stress level is high, low cycle fatigue can occur if the material will plasticize during each cycle (less than 10000 cycles). This should not happen in a well-designed machine. Normal methods for evaluating LCF are “Neubers rule” or “Coffin-Mansons relation” [12].

3.4 HIGH CYCLE FATIGUE (HCF)

High cycle fatigue is common in all rotating machines and is included in basic courses in solid mechanics [13, 14]. By making tests at alternating and pulsating tests in S-N diagrams data can be obtained to perform analysis on safety factor to avoid high cycle fatigue. For most common materials data can be obtained from handbooks. The data is analysed in Haigh-diagrams and data is reduced according to size, surface roughness, stress concentration factor and environmental conditions.

If the load cycles are changing over time methods for partial damage analysis is necessary to use. The most common method is Palmgren-Miner's rule. In this method loads are grouped in load blocks and for each load block the consumed life is estimated. The consumed life is estimated by adding the consumed life from each load block. In this case the same S-N diagram is used for evaluating the consumed life for each load block.

There are some special considerations for ductile iron, malleable iron and cast steel [15] as well as for welded structures [16-17]. In such cases ordinary dimensioning rules cannot be used.

3.5 FRETTING

Fretting normally refers to wear and corrosion due to oscillatory motion of loaded surfaces in contact. Surface asperities are broken and causes debris to form in the contact area. Fretting corrosion is the term for oxidation of the debris in the contact area. The surface roughness increases together with micro pits which reduces the fatigue strange of the material. Fretting can cause micro cracks causing a high stress concentration factor failure of the component. In steel the damage can be identified as red iron oxide dust, but it is not rust since no water is needed. The evaluation of estimated life is still a research area and not a well-defined area today. Typical areas of fretting are shrink fits, bearing seats, bolted parts, splines and dovetail connections.

3.6 FRACTURE MECHANICS

When the material has an initial crack, the remaining life can be predicted if data is provided for the specific material. In the simplest case a linear theory is used (K_{ic})but normally the method of J-integral in a FEM analysis is necessary (J_{ic}). The constants K_{ic} and J_{ic} are used to determine stresses for crack growth. The linear K_{ic} can be find for some materials, but the nonlinear J_{ic} must normally be determined experimentally by specialists in the area [12].

3.7 BUCKLING

Buckling is the effect when a structure loses stability and suddenly deform or collapse. The most common case is the Euler critical load when a beam suddenly loses stability. Other common cases are buckling of plates. Typical evaluations are linear and non-linear stability analysis where nonlinear is in many cases the most reliable one. A typical error is to use linear theory for slender structures to estimate stiffness. In tension such structures are stiff but under compression there is almost no stiffness, and the structure will buckle. In hydropower the stiffness of support structures can be overestimated when badly designed beams are used, or thin plates are analysed with linear theory. One such example is the beams in support structures from the 80's. To reduce cost, unsymmetric beams were used according to the figure below.

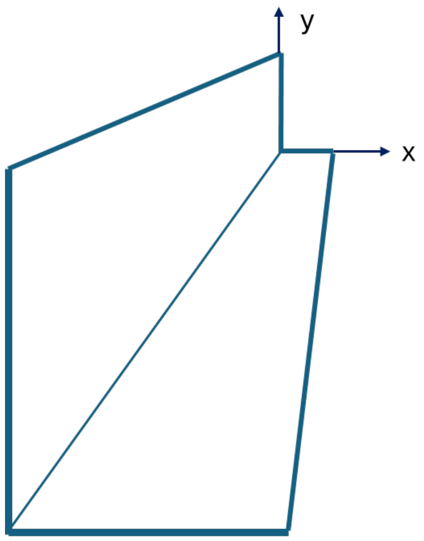

Figure 5.7.1 Figure of an unsymmetric beam.

Due to the unsymmetric geometry the beam will be subjected to skew bending where the deformation occurs in both x and y when a load is applied in one of the directions. Also, the long flange will start to deform when a load is applied, and buckling will happen. This could be avoided by thicker/shorter flanges and symmetric cross section together with nonlinear analysis.

3.8 DYNAMIC INSTABILITIES

In rotating systems several instabilities have been described in literature and in here, the most possible ones are described. Several of these phenomena have not been reported or investigated further in hydropower applications. Especially for vertical machines there is a lack of research in this area why even new phenomena can exist which has not yet been studied. Below some possible instabilities are described.

3.9 PARAMETRIC RESONANCE

When a system has time dependent parameters (e. g. stiffness or damping) an unexpected resonance or instability can occur. Parametric excitations produce whirling that can be synchronous [18], sub-synchronous [19] or super-synchronous [20], depending on how the parameters are varied. Some known cases from literature are

- Shaft stiffness asymmetry [18],
- Intermittent rotor/ stator rub [19],
- Excessive ball bearing clearance [19],
- Rotor mass asymmetry [21],
- Pulsating torque [22].

In hydropower rotors parametric excitation can occur due to unsymmetric properties of stationary or rotating components. Some possible sources are

- time dependent bearing properties [23],
- unsymmetric stiffness in support structures,
- misalignment of combi bearings [24].

Other sources are possible in the hydropower application and the area need further investigation.

3.10 LABYRINTH SEALS

In labyrinth seals destabilizing forces can occur especially in Francis turbines. A few studies have been presented [25] but more studies are necessary before we can handle these effects.

3.11 MISALIGNED ROTORS

Vibrations due to misalignment is usually characterised by 2x running speed component and high axial vibrations [26]. Studies has not been performed on vertical rotors why it is not known if it is a problem for hydropower rotors.

3.12 THE MORTON EFFECT

The Morton effect [27] is a synchronous instability that is possible especially for vertical rotors. The phenomenon (also called Newkirk effect) is caused by thermal bend of the rotor. If the motion is synchronous, the same surface will be in contact with the bearing along the orbit and the rotor will be unevenly heated which causes a thermal bend. So far, no such effect has been reported for vertical rotors, but it is a possible source for vibration if the motion is synchronous. The typical behavior for the Morton effect is a sudden increase of vibration when one side of the rotor is heated. As the heating increases, the vibration increases until the point where the temperature starts to become even over the cross section. The vibrations will then suddenly disappear, and the rotor runs smoothly for a while until the process starts again.

3.13 ALFORD FORCES

When a bladed turbine is subjected to an eccentricity, cross coupling forces will appear due to the variation of blade tip clearance between the blades [28, 29]. This phenomenon has been observed in steam turbines and compressors but has not been studied for hydropower rotors. However, due to the low viscosity of water it is likely that this effect will be negligible, but it would be easy to verify by fluid dynamic simulations. For Francis turbines this force would probably be higher.

3.14 HOLLOW ROTOR PARTIALLY FILLED WITH LIQUID (STEWARTSON INSTABILITY)

When a hollow rotor is partially filled with a liquid the rotation can cause an instability. This could occur in the runner cone if it is subjected to leakage. The instability occurs above the critical speed and therefore it is probably not an issue for hydropower rotors. On the other hand, the major critical speed will decrease which could be a problem (subharmonic resonance). Today, no such problem has been reported why the risk should be low.

3.15 CAVITATION

Cavitation is a phenomenon that occurs when the static pressure in the water locally reaches the vapour pressure, leading to the formation of small vapor-filled cavities or "cavitation bubbles". When subjected to higher pressure, the cavitation bubbles collapse (implode) and generate pressure shock waves that can cause damage, i.e. material erosion, if it occurs close to a surface of the machine. The local static pressure is reduced where the local flow velocity increases, as well at the core of vortices. The quality of the water is also an important parameter. The inception of cavitation is greatly enhanced with the number of nuclei (in most cases gas bubbles) present in the water. Cavitation erosion of turbine runners results in production losses since the hydraulic efficiency decreases, and further in cost related to repair and down-time. Since cavitation involves formation of gas bubbles, the compressibility and sound propagation speeds of the water/vapour mixture changes. This influences the frequencies and amplitudes emitted from flow-induced pulsations. Cavitation can be avoided by submerging the turbine and thus increasing the hydrostatic pressure. A local pressure reduction may then no longer reach below the vapour pressure. However, submerging the turbine must be done when building the power plant, and it increases the construction costs significantly. The hydropower plants in Sweden were originally built for base load close to the best efficiency condition, and they were submerged accordingly to avoid cavitation. New operating procedures, due to frequency regulation etc., are therefore limited due to cavitation inception. A possibility when refurbishing turbines is to apply new materials or coatings that can better withstand the cavitation erosion. An important question in industry is how much cavitation can be accepted, and what is the cost due to degradation when running the turbines in cavitating conditions.

3.16 EROSION

Erosion is related to particles that are being transported with the water, impinging on the surfaces of the machine. This leads to removal of material, which changes the shape of the surfaces and may eventually even create holes through the material. The rate of erosion depends on the amount of particles, the properties of the particles, how the particles approach the surface, and the material properties of the surface. [30]. The change of the surfaces may, in addition to changing the properties of the machine, induce additional risk of cavitation that may accelerate the erosive process. Erosion is an enormous problem in regions with high particle/sediment content in the water, such as many locations in India, Nepal and

China. A remedy to avoid extreme particle content, due to e.g. sudden landslides, is to monitor the particle content and shut down the power plant until the levels are reduced. On-going research is investigating different kinds of materials and coatings to reduce the erosion.

4 Loads

Most of the damaging mechanisms causing fatigue in hydraulic turbines have their origin in loads, and in particular varying loads with frequencies that coincide with any natural frequency in the system. The machine components can be designed to withstand static loads, and they can be designed to avoid natural frequencies that are known.

However, when operating the turbines at off-design or varying conditions there may at least occasionally be large load variations at frequencies close to some natural frequency in the system. The mechanical designs of the machines can be well-balanced to avoid forces originating in imbalance from the design itself. The varying loads thus have their origin in the fluid flow and in the electromagnetic interaction in the generator, which is discussed below.

4.1 FLUID FLOW LOADS

The fluid flow induces a varying load about a static mean load on different parts of the turbines. The static mean load is in most cases not a major problem. The machine components, including well-lubricated bearings, can be designed to withstand them for a very long time. However, when regulating the turbines (changing the operating conditions), also the static mean load will have an effect on the wear on blade bearings, bushings and the mechanisms used for the change in blade angles. The varying load is caused by several fluid flow excitations [31]. The induced forces are usually harmful when operating the machine away from BEP. They can create excitations of the blades, rotor, support structure or in some cases all of them together. The main frequencies appearing in hydraulic turbines are described below together with an attempt to distinguish if they cause blade or rotor vibrations.

4.1.1 Spiral Casing Frequency

The main purpose of the spiral casing is to distribute the flow evenly at the inlet to the guide vanes. Poorly designed spiral casings may induce a tangential variation in the flow at the inlet to the guide vanes, with a sudden jump at the spiral casing tongue. This gives a static horizontal force on the entire runner, superimposed with pulsations as the runner blades pass the tongue. The frequency that is excited on each runner blade is directly related to the runner rotational speed, as

$$f_{sc} = \frac{n}{60}$$

where *n* is the rotational speed of the runner (rpm). The entire runner will sense a frequency that is multiplied by the number of runner blades. These loads and frequencies directly affect the blades but are also transmitted into the runner blade regulation mechanism and also to the rotor system. The shape of the spiral casing, in combination with the development of the boundary layers, also induces secondary flow that somewhat influences the flow entering the guide vanes. However, these effects are greatly diminished as the flow undergoes a strong acceleration through the guide vane passage. The loads and frequencies originating from the spiral casing and acting on the runner and the runner blades could potentially also affect the spiral casing, but that is not something that is being reported in the literature.

4.1.2 Guide Vane Frequency

The flow leaving the guide vanes should ideally be distributed axi-symmetrically to the runner. The boundary layers at the guide vane surfaces however induce wakes that interrupt the axi-symmetry in the flow. There are also horse-shoe vortices formed as the boundary layer of the spiral casing interacts with the leading edges of the guide vanes, and there are leakage flow vortices formed due to the clearance between the guide vanes and the upper and lower surfaces. The wakes are low velocity regions that are experienced by the rotating runner blade leading edges as high-pressure regions compared to the rest of the flow leaving the distributor with higher velocity. Also, the direction of the flow after the guide vanes is affected locally in the wakes and by the vortices formed by the guide vanes, causing a varying angle of incidence of the flow reaching the runner blade leading edges and thus a varying load on the runner blades. Therefore, pressure or strain measurements performed on a runner blade particularly reveal a frequency related to the number of guide vanes. The frequency is given by

$$f_{gv} = \frac{n\, Z_{gv}}{60}$$

where n is the rotating speed of the runner (rpm) and Z_{gv} is the number of guide vanes. The horseshoe vortices and the vortices formed in the guide vane clearances may additionally induce a flow-induced frequency as the vortices start to break down, but the magnitude of the corresponding loads on the runner blades is much lower. To avoid resonance, the ratio ${}^{Z_{gv}}\!/_{Z_b}$ should be an uneven or irrational number, where Z_b is the number of runner blades. The unsteady loads may cause runner blade or rotor vibrations if not appropriately designed. The varying pressure distributions on the runner blades, due to the interaction with the inappropriate flow properties leaving the guide vanes, also affects the inception and development of cavitation, in the case that the static pressure locally falls below the vaporization pressure.

The number of stay vanes present ahead of the guide vanes is usually equal to the number of guide vanes or half of that number. The purpose of the stay vanes is only to keep the integrity of the spiral casing, which would otherwise "explode" due to the high pressure. They are thus designed to interrupt the flow as little as possible at BEP and are thus in most cases aligned with the guide vanes at BEP. This, together with the distance to the runner and the disturbance of the guide vanes yields that the stay vane wakes are not significantly experienced by the runner.

The loads acting on the runner blades affect the pressure field, which due to its elliptic behavior in incompressible flow propagates in all directions. It means that also the guide vanes (and to a lesser extent the stay vanes) are also affected by the passing of the runner blades. Both the static load and the varying load on the guide vanes may have an effect on the guide vane regulating mechanisms and the wear in bearings and bushings, although this is not something that is commonly discussed in the literature.

4.1.3 Runner Blade Rotating Frequency

Similarly to the guide vanes, the flow leaving the runner blades generates wakes behind the runner blades. Also horseshoe vortices are formed due to the interaction between the blade and the boundary layers at the hub and tip. Moreover, Kaplan turbines have clearance flow both at the hub and tip of the blades, causing vortices that travel downstream. The main frequency experienced in a point in the draft tube due to these effects is the blade passing frequency

$$f_b = \frac{n\, Z_b}{60}$$

where Z_b is the number of blades. As for the guide vanes, the vortices may undergo breakdown that causes additional frequencies that are typically of lower amplitude.

The vortices cause a reduced static pressure in their cores, which in combination with the low static pressure at the suction side of the blades may cause cavitation. This cavitation happens close to blade, hub and shroud surfaces, which may cause cavitation erosion.

4.1.4 Von Karman Vortices

A body situated in a free stream creates vortices in its wake above a certain Reynolds number, known as Von Karman vortices. They are found behind the trailing edge of stay vanes, guide vanes and runner blades in turbines at all operating conditions. Their frequency is given by

$$f_k = St\frac{U}{L}$$

where Sh is the Strouhal number with values [0.12-0.30]. U is the mean flow velocity and L is the characteristic length, e.g., the hydrofoil thickness.

Von Karman vortices may excite the object generating them if the shedding frequency of the vortices matches the natural frequency of the body considered and the damping is low, i.e., resonance. This happens with stay vanes, guide vanes and runner blades. Such a problem is circumvented by having an oblique trailing edge enabling the simultaneous detachment of vortices which damped themselves and thus the pressure amplitude source of the resonance.

4.1.5 Vortex Rope

At the best efficiency point (BEP), the guide vane and runner blade angles are in the optimum configuration, allowing a smooth transfer of the fluid flow energy to the blades. As the flow rate decreases to part load (PL), the angle of the guide vanes decreases leading to an excessive rotation of the fluid at the runner leading edge. The runner cannot transform all this fluid rotation into torque, and thus some remaining swirl enters the draft tube. Above a certain threshold of angular momentum to axial momentum (swirl number), the columnar vortex in the draft tube breaks down, leading to the formation of a so-called rotating vortex rope. This happens for single-regulated turbines of type Francis and propeller but is not a problem of the doubly regulated turbine of type Kaplan which have adjustable runner blades. Kaplan turbines with failed blade mechanisms often have their blades welded at a particular angle, transforming them to propeller turbines.

The rotating vortex rope has a frequency in the range

$$f_v = [0.2 - 0.4]f_r$$

where the runner rotational frequency is given by

$$f_r = \frac{n}{60}$$

This is just a typical range for hydropower rotors, but there is actually no connection between the runner rotational frequency and the vortex rope frequency. The same vortex rope frequency can be predicted numerically without a runner if the swirl number and the axi-symmetric velocity profiles are kept the same. The rotating vortex rope exhibits both axial and circumferential motions. The axial motion propagates through the entire rotor system and can be a source of excitation for resonance of a component such as the generator. The circumferential motion induces radial/horizontal forces on the rotor, which is transferred through the rotor system to the bearings, the support structures and the generator. The runner blades are indeed also affected by these pressure fluctuations, possibly causing fatigue.

The rotating vortex rope has a reduced static pressure at its core and may be cavitating, which changes its properties and frequency. A cavitating vortex rope is in most cases not a problem with respect to cavitation erosion, since it happens far from surfaces. However, in some cases it may come close to the draft tube walls and cause cavitation erosion.

4.1.6 Inter-Blade Vortices

With a further decrease of the flow rate to deep part load of Francis turbines, the smaller opening angle of the guide vanes leads to an elevated incidence angle between the leading edge of the runner blades and the incoming flow [32]. The flow in the blade channel is then twisted, deviating from its optimal configuration, leading to inter-blade vortices. These inter-blade vortices lead to instabilities and large pressure pulsations resulting in early fatigue of the runner and lower efficiency. The inter-blade vortices usually cavitate with a time dependent vapor volume. The collapse of the vapor column occurs often near the trailing edge of the runner blade suction sides. The occurrence of the phenomenon depends on the head, i.e., it appears at different relative flow rate function of the head [33]. The frequencies associated with such phenomenon vary but present in general a wide range extending up to 20 times the runner frequency with large amplitudes.

4.1.7 Hydraulic Fluctuations Caused by Transient Flow in Penstock.

When a turbine load is suddenly changed, such as a rapid closure of the guide vanes, pressure waves may form in the penstock, referred to as water hammer. These fluctuations have a frequency given by

$$f_{wh} = \frac{a\,m}{2\,L}$$

where a is the sound speed of water, m is the number of resonance order (m=1,2…n) and L is the length of the penstock. This may cause sudden failure of the upstream pressure conduits. At the downstream side of the guide vanes there is a risk that a large vapor pocket is formed, and when the water comes back it may hit the machine with a force that may damage the machine.

Slower but repeated variations in operation at a frequency that coincides with the natural frequency of the free surface system may cause a standing wave in the system, referred to as surge. These happen at very low frequencies, far from any natural frequency of the machine and the construction, but it can for instance lead to flooding of the machine hall.

4.1.8 Added Properties.

One important aspect of the presence of the water at a vibrating surface is that it changes the properties of the vibrating system. This is referred to as added mass and added damping. In addition to vibrating the structure, also the water touching the structure needs to move or deflect. Both the vibrating mass of the water and the motion of the water affect the properties of the vibrating system. The added properties correspond to an additional inertia force (added mass) and viscous force (added damping) needed to maintain the same displacement of an oscillating object submerged in a fluid (water) compared vacuum (or air). The effect of added mass depends on the object's shape and is directional, generally increasing in a confined fluid domain [37], [38], [39]. Therefore, added mass will increase, for example, for an oscillating turbine runner when it is located in a waterway compared to added mass in open water. The effect is further different if the water is standing still or moving, and the presence of nearby surfaces may also play a role. Added mass can therefore potentially change the dynamical system to move closer to, or further away from, problematic frequencies. In rotordynamic applications it is therefore important to include added mass of the runner in order to better estimate the natural frequencies of the shaft train [36]. For the runner itself it is also important to include the effect of added properties in modal and response analysis to avoid dangerous resonance frequencies which can rapidly cause cracks and failures to secure a long service life of the runner.

Energiforsk

4.2 ELECTROMAGNETIC LOADS

The magnetic field in generators and exciter results in displacement dependent forces called magnetic pull. Commonly for small displacements (<10% eccentricity), this magnetic pull force is described by a magnetic stiffness. In SVC the tangential force has been studied in a research collaboration between UU and LTU. From numerical simulations at UU, it has been found that a tangential force exists, and it depends on the whirling frequency of the rotor. The tangential force is generated by the damping wires and will be zero at synchronous whirl. The existence of tangential force has also been shown experimentally [34] at UU. In figure 6.2.1 the radial and tangential force is shown for the experimental generator Svante at UU.

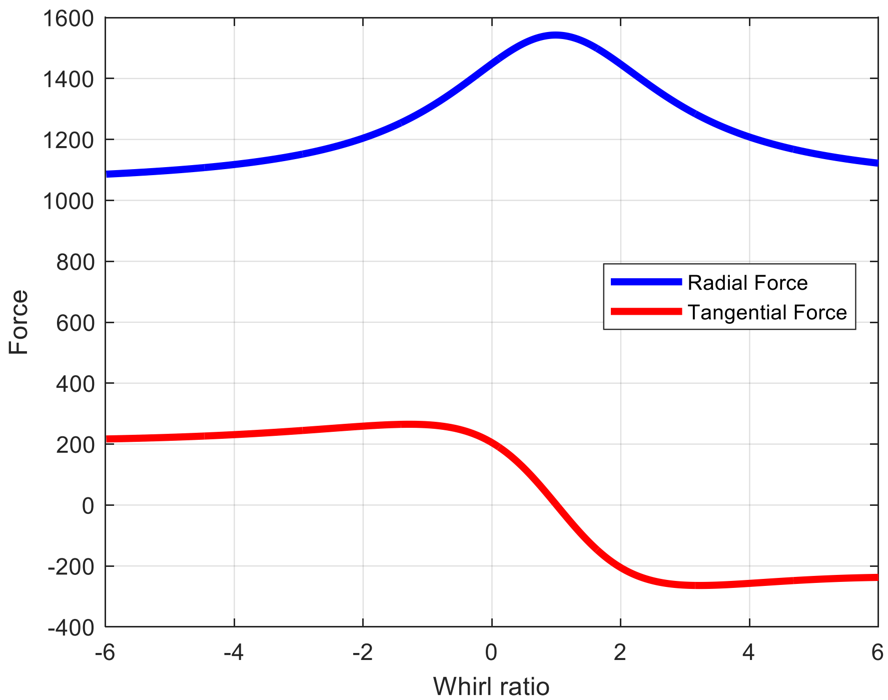

Figure 6.2.1. Whirling dependent forces from Svante at UU.

Hydropower generators are however more complicated to simulate, and we do not have any detailed load models today. One machine has been studied for a specific case, but no whirling dependent load model was presented [35]. It would be of importance to numerically simulate and by measurements validate these forces on a real machine.

5 Load Measurements

Analysis of force measurement from generator and turbine is generally a difficult task. The reason is that in a dynamic system the forces are affected by inertia why a static approach can give wrong results.

However, in hydropower applications the operating speed is normally significantly below the critical speed (typically about 30%). In such case the quasistatic approach is likely to give a reliable result for low frequency excitation. In here two approaches will be presented namely a static and a dynamic approach.

5.1 STATIC APPROACH

In [3] a static approach was presented where the loads are predicted by use of stiffness matrix. Bearing displacements were measured ant the forces calculated by use of bearing models [23]. In addition, an extra measurement of bending moment was used to get necessary input to solve the excitation forces and moments. The hypothesis is that interesting frequency are close to or below the operating frequency. To study the hypothesis the same model as in [44] is used. Applying a stationary force of 10000 N at the generator and runner results in the following response curves for the runner and the generator. The excitation frequency is varied from 0-40 rad/s and the operating speed is 12 rad/s.

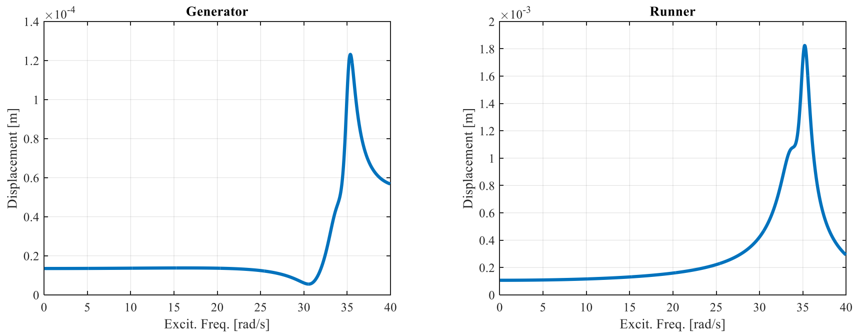

Figure 7.1.1. Displacement of generator and runner when subjected to stationary force (10000N) in generator and runner at different excitation frequencies.

The figure indicates that a static model gives realistic results on the forces for frequencies below the operating speed. In the measuring method the displacements from the bearings will be used to predict the forces. Therefore, the displacement at the bearings is studied at different excitation frequencies. Relative displacement and frequencies are used where the amplitude is scaled with the displacement at zero excitation frequency (the static case). The excitation frequency is scaled with the operating speed.

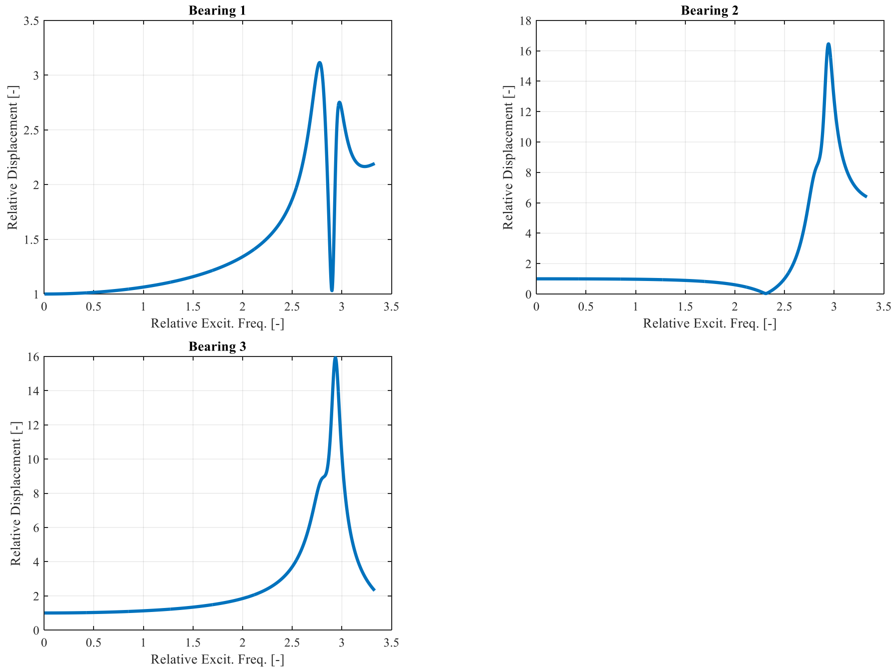

Figure 7.1.2. Displacement at bearings when subjected to stationary force (10000N) in generator and runner at different excitation frequencies.

At excitation frequency equal to the operating speed, the error in bearing 1-3 is 6.4, -2.7, 12.8 %. The figure also indicates that frequency 2x the operating speed can result in 100% error. Therefore, the method of predicting force by quasistatic approach is useful for excitation frequencies 1.5x operating speed (maximum error 34%). The method needs deeper evaluation to handle higher frequencies and is sensitive to the system properties.

5.2 DYNAMIC APPROACH

If the forces are stationary, the forces could be solved by analyzing the particular solution in complex form. The equation of motion can be written as

$$(K - \omega^2 M + i\omega C)\{x\}e^{i(\omega t+\theta)} = \{f\}e^{i(\omega t)}$$

Where a particular solution similar to the excitation is assumed. On matrix form, the equation of motion can be written as

$$[\alpha]\{x\} = \{f\}$$

By rearranging the equations, the problem can be written as

$$\begin{bmatrix} \alpha_{ll} & \alpha_{lm} & \alpha_{ln} \\ \alpha_{ml} & \alpha_{mm} & \alpha_{mn} \\ \alpha_{nl} & \alpha_{nm} & \alpha_{nn} \end{bmatrix} \begin{Bmatrix} xk_l \\ xu1_m \\ xu2_n \end{Bmatrix} = \begin{Bmatrix} fk1_l \\ fk2_m \\ fu_n \end{Bmatrix}$$

where,

$fk1_l$ = l known forces,	xk_l = l known displacements
$fk2_m$= m known zero forces,	$xu1_m$= m unknown displacements
fu_n = n unknown forces	$xu2_n$ = n unknown displacements

Using Fourier transform the unknown forces can be solved for each frequency. The method has been suggested during this project and needs to be validated by experiments before we know if it is a possible method for hydropower rotors. One has also to evaluate if inertance or mobility form (of the equation of motion) is more effective than receptance for the analysis. It is also a need to evaluate if any dynamic method is possible for analyzing transient loads.

5.3 LOAD MEASUREMENT ON TURBINE BLADES

To estimate the fatigue service life or the costs for degradation of a turbine runner, it is necessary to determine the loads which runners are exposed to at different operating conditions such as start/stop, part load operation (PL) etc. Prototype measurements (onboard measurements) by means of strain gauge sensors glued to the blade surface and further to route the signal cables trough a rotating shaft to be connected to a telemetry and a data acquisition system is usually complicated and expensive. Examples of the procedure on how to measure and assess the relative partial fatigue damage using the Miner´s rule and operational dependent costs for a Francis turbine is presented by Monette at al. [40] and Huang at al. [43]. To calculate the absolute fatigue service life or remaining service life based on a historical operation pattern, requires however access to the real geometry which usually is not the case for hydropower utilities and owners.

Besides prototype measurements, the general approach is to estimate the fatigue service life by means of numerical simulations or physical model test or from a combination of both. For numerical simulations the flow and pressure field are calculated using Computational Fluid Dynamics, CFD. The calculated pressure field is mapped to a Finite Element Model, FEM to calculate the stresses and locations of the hot spots in the runner followed by fatigue damage calculations. The assumption here regarding fluid and structural interaction (FSI), is that the structural response is small and will not affect the flow field. The uncertainties associated with used methodology to determine dynamic stresses of periodic and stochastic nature using numerical methods or experiments are however not yet fully investigated, why prototype measurements still are needed for validation. Some of the uncertainties are related to

- Deviations in geometry between real geometry and for simulations often used CAD drawings or 3D models why laser scanned geometries are preferred.
- Simulation parameters such as mesh size and quality, choice of turbulence and cavitation models, simulation time to gather enough statistical data to evaluate the partial damage etc. which is in turn is associated with computational costs. For example, simulation of speed no load conditions (SNL) will probably require longer simulation time or equivalent several runner rotations compared with simulation at best efficiency point (BEP) to resolve a more chaotic flow field in the case of SNL.

- Stochastic flow phenomena which cannot be reproduced in numerical models or experiments for example resulting from interaction between prototype waterways and the turbine which may cause dynamic stochastic stresses that are essential to evaluate partial damage [43].
- Material properties and SN-data (Wöhler curves).
- Transformation of measured static and dynamic strain from model experiments to prototype conditions and in both cases establish a correlation between measuring locations and the hotpots (location of static and dynamic hotspots may differ).

5.4 RUNNER LOAD MEASUREMENT IN MODEL EXPERIMENTS

To experimentally measure the runner load using a model turbine requires that the test is performed at geometrical & hydrodynamic similitude to achieve similarity in hydraulic excitations of the runner. Moreover, for response analysis and fatigue damage assessment it also requires similitude between the hydraulic excitation frequencies, f_h and the runner structural modes i.e.,

$$\frac{f_{n,M}}{f_{h,M}} = \frac{f_{n,P}}{f_{h,P}}$$

where the subscript P = prototype, M = model and the variables f_n = natural frequency of the runner. Generally dominant frequencies (f_h) of periodic or quasi-periodic phenomena (e.g., rotor-stator interactions, part-load vortex) will scale directly with the rotational speed (n) of the turbine, i.e.,

$$f_{h,M} = \frac{n_M}{n_P} f_{h,P}$$

To achieve similitude between the hydraulic excitation frequencies, f_h and the runner structural modes, f_n, Tanaka et al. [41] proposed and performed measurements on a model pump-turbine under hydrodynamic similarity by using a scaled model runner made of the same material as the prototype runner. From dimensional analysis they linked the structural response of the prototype and model runner through the relationship:

$$f_{n,M} = \left(\frac{E_M}{E_P} \cdot \frac{\rho_P}{\rho_M}\right)^{1/2} \frac{D_P}{D_M} \cdot f_{n,P}$$

where, E = material young modulus, ρ = material density and D = runner reference diameter. Hence, the natural frequency of geometrically identical runners is function of the scale factor D_P/D_M and the material ratio $(E/\rho)^{1/2}$. If the same material is used for the model and prototype, $f_{n,M}= f_{n,P}$ the structural response ratio depends only on the scale factor (D_P/D_M). However, use of different material enables adaptation of the runner response to the test stand capacity.

For transient operating conditions such as start/stop, load variation and load rejections, it further requires similitude between the time period for moving guide vanes (and for Kaplan the runner blades) which according to O Kirschner et al. [42] can be expressed as a time factor equal to ratio between the rotational speed i.e.,

$$\frac{\Delta t_M}{\Delta t_P} = \frac{n_P}{n_M}$$

To achieve similarity during acceleration/deacceleration of the runner/rotor during start/stop and load rejection, the polar moment of inertia (J) in the model needs to be adapted. From dimension analysis using the Π therorem the inertia can be transposed from prototype to model through the expression

$$J_M = J_P \left(\frac{\rho_M}{\rho_P}\right) \cdot \left(\frac{D_M}{D_P}\right)^5$$

The inertia may however need to be adjusted to compensate for differences in friction and hydraulic head losses. The polar moment of inertia can be adjusted by means of a flywheel with weights that can be added or removed from the flywheel.

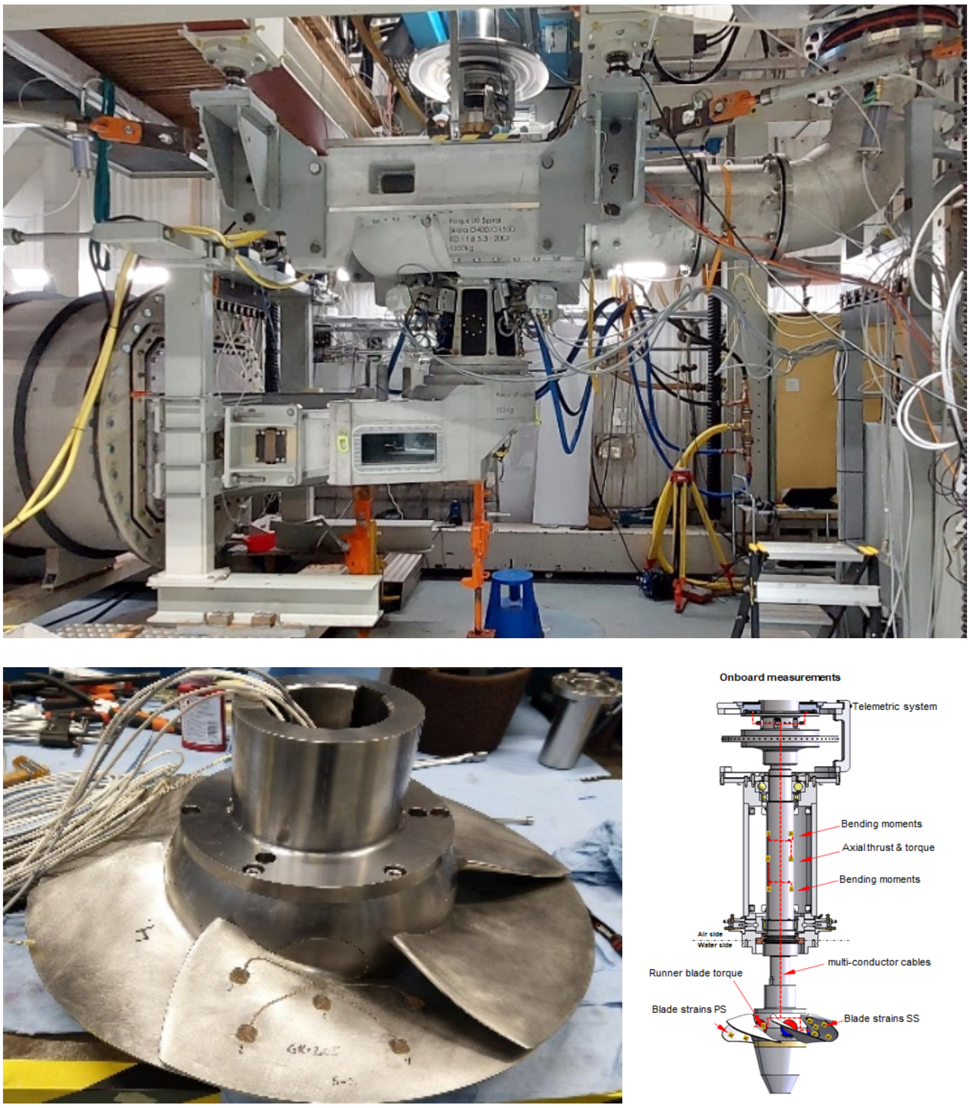

Figure 7.4.1. Vattenfall’s transient test stand in Älvkarleby, Sweden. Porjus U9 model turbine (D=400 mm) assembled with movable and instrumented runner blades for load measurements. The shaft train includes a flywheel with adjustable inertia and a servomotor for adjusting the blade angle and a telemetric system.

6 Need for New Research

The research at SVC turbine and generators is mainly focusing on creating new knowledge in this area.

Some examples of research milestones that would increase the knowledge for lifetime evaluation are

- Cost effective technique (dynamic or static) to measure turbine and generator loads at different operating conditions,
- Measurements on hydropower units to create load models,
- Better understanding of the reason for failure of Kaplan turbines,
- Understanding loads at different components in the Kaplan turbine during different operating conditions,
- Dynamic models on Francis turbines,
- The reason for fatigue in Francis turbines,
- Scaling of model forces and frequencies to prototype,
- Understanding of failure modes for different components,
- Validation of electromagnetic forces on a real hydropower generator.

Several of these projects are under development within SVC but more research is needed before we have data for lifetime evaluation. With a stronger focus on lifetime evaluation, we could coordinate our research to solve these problems and create methodology for evaluation of expected lifetime due to operating conditions.

7 Synthesis

Normally, it is difficult to collaborate between different research groups due to large differences in methods and terminology. However, within SVC the senior group in "Hydropower Technology" and industrial partners have worked together for over 20 years and is therefore well suited for developing methods for lifetime evaluation of hydropower units.

The problem of lifetime evaluation is that it is applied on a holistic level on a system while research is normally on detailed level. Several research projects are contributing to lifetime assessment but are not today guided towards a common goal. To further develop the field, it would help to have a leader guiding the research towards the goal.

The development of simulation tools is today giving opportunities to solve more complex problems. Some examples of general developments that contribute to lifetime assessment are

- Simulations of turbine and generator loads and added properties
- Reduction of uncertainties to determine loads and risk of failures which in turn can motivate longer service life
- Monitoring systems that can detect anomalies, diagnose (what?) and prognosticate (when?)
- Bearing models
- Model experiments at VRD
- Prototype measurements
- Friction and wear properties of components
- Dynamic simulations of stresses due to vibrations
- Development of measuring techniques for finding turbine and generator loads

Critical developments would be to solve how model experiments and simulation could be scaled /transposed (particular dynamic quantities) to prototypes and the development of measuring techniques to find the loads on turbine and generator for different operating conditions. Then by simulation and experiments it is likely that we could build up a database on loads that could be used for lifetime evaluation. In addition, there is a need for a plan to implement results from each research project into a lifetime assessment work package.

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Participating companies and industry associations are: AFRY, Andritz Hydro, Boliden, Fortum Sverige, Holmen Energi, Jämtkraft, Karlstads Energi, LKAB, Mälarenergi, Norconsult, Aker Solutions, Skellefteå Kraft, Statkraft Sverige, Sweco Sverige, Tekniska verken i Linköping, Uniper, Umeå Energi, Vattenfall R&D, Vattenfall Vattenkraft, Vattenkraftens miljöfond, Voith Hydro, WSP Sverige and Zinkgruvan".

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Appendix I: Natural Frequencies

The natural frequency analysis of a vertical rotor is difficult to perform due to the unknown bearing loads. For a horizontal rotor, the bearing loads can be determined statically due to gravitation, while it for a vertical rotor the loads depend on fluid and electromagnetic forces. Today, one must rely on the experience from the generator and turbine supplier. But relatively simple measurements are possible and by performing measurements one can get better knowledge in future. To increase the safety margin, at least three loads (low, medium and high) should be used to evaluate the bearing properties and thereby the natural frequencies. In addition to natural frequency, it is important that the damping ratio is calculated since low damping will result in high vibrations and possible severe failures. The target should be to avoid excitation close to natural frequencies and increase damping to ensure a reliable design.

Example:
To describe the importance of natural frequency and damping ratio, a one degree of freedom model is shown below with mass M, stiffness K and damping C. The system is subjected to a harmonic force $F_0\cos(\omega t)$

In Fig. A.1 a harmonically forced SDOF system is shown when a damper is introduced. The target is to find if damping can be used to avoid severe failures when the driving frequency is close to the natural frequency.

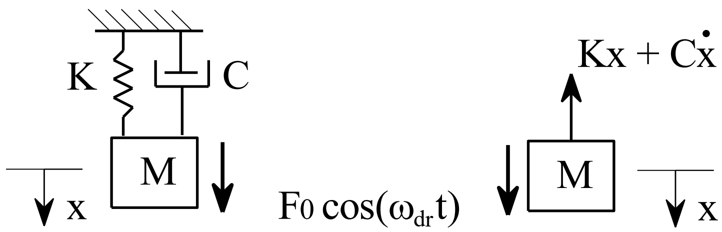

Figure A.1 Harmonically forced SDOF system with damper.

From the FBD the equation of motion can be derived by Newton's equations

$$M\ddot{x} + C\dot{x} + Kx = F_0\cos(\omega_{dr}t) \quad (1)$$

which can be simplified by dividing with the mass M

$$\ddot{x} + 2\zeta\omega\dot{x} + \omega^2 x = f_0\cos(\omega_{dr}t) \quad (2)$$

In the equation above the definition of damping ratio ($\zeta = {}^{C}/_{2M\omega}$) can be found. The undamped natural frequency is defined as ω and $\left(r = {}^{\omega_{dr}}/_{\omega}\right)$. The particular solution to this problem is for :

$$x_p = A_0\cos(\omega_{dr}t - \phi) \quad (3)$$

$$A_0 = \frac{1}{\sqrt{(1-r^2)^2+(2\zeta r)^2}}\frac{f_0}{\omega^2}; \tag{4}$$

$$\phi = \tan^{-1}\left(\frac{2\zeta r}{1-r^2}\right) \tag{5}$$

From this equation one can analyse how the amplitude depends on the forcing frequency and damping ratio. It is clear, that the denominator will not be zero if damping is introduced and therefore the amplitude will be limited. The design of the damping ratio is therefore crucial to avoid failure if the machine is subjected to driving frequencies close to the natural frequency. In Fig. A.2 the amplitude at resonance is shown for a damped system. One can see that in a resonance the amplitude will grow during a number of cycles before it reaches its maximum values and the maximum value depends on the damping ratio.

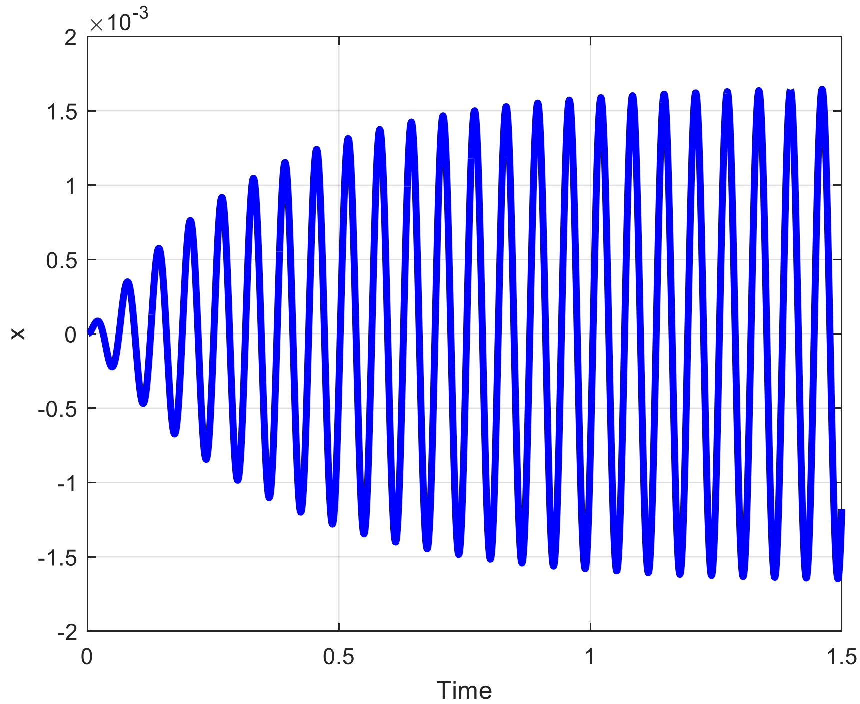

Figure A.2 Forced SDOF system at resonance.

In Fig. A.3 (left) the amplitude is shown for three different damping ratios and the corresponding phase at the upper right figure. The lower right figure shows the amplitude at resonance for different values of damping ratio.

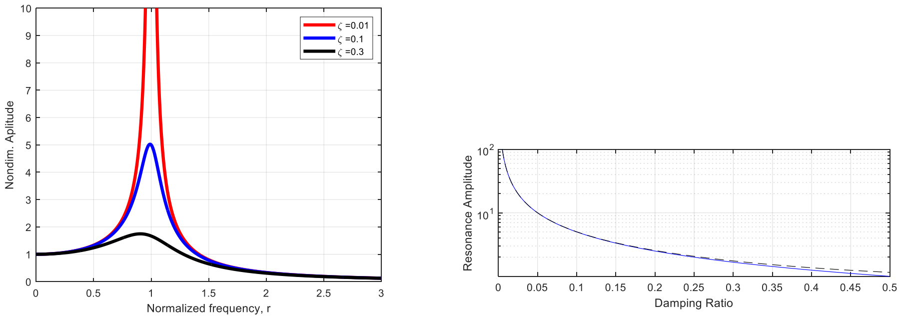

Figure n.3 Forced SDOF system according to Eq. nn. Left amplitude as fkn of r, and right figure amplitude at resonance for different damping ratios.

From the figure we can observe that the system starts at *r=0* at the static displacement for a constant load F_0 ($x_p=F_0/K$ at *r=0*). In this case the system is normalised so that that the static displacement is 1. At the natural frequency *r=1* the maximum amplitude is five times the static displacement for $\zeta = 0.1$ and 50 times for $\zeta = 0.01$. For higher frequencies the amplitude goes to towards zero. Typical damping values for metallic materials are 0.1-1%. It is therefore important to include damping elements in machine designs to avoid failure.

LIFETIME ASSESSMENT OF HYDROPOWER UNITS

The senior researchers in work package hydropower technology, have worked together with the industry for over two decades in developing experimental and simulation techniques to increase the reliability of hydropower units. However, there are still areas to develop before lifetime evaluation can be performed. This document is aiming to describe the state of the art and suggest fields to develop further before detailed lifetime evaluation can be performed.