

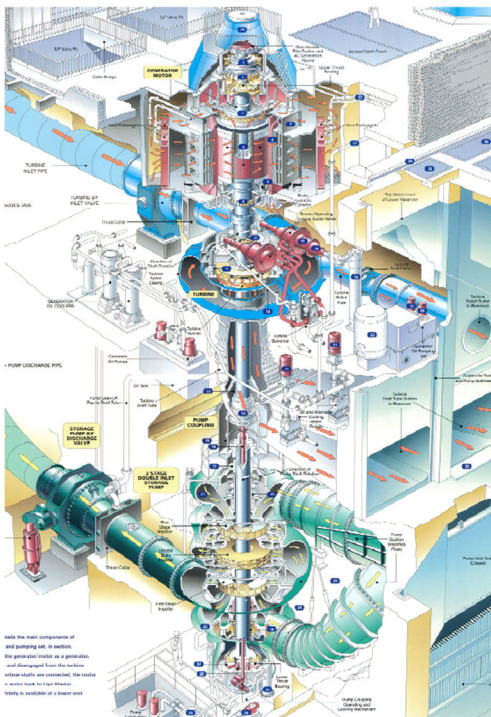
REHABILITATION OF FFESTINIOG PUMPED STORAGE POWER PLANT

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Abstract: Ffestiniog Power Station with its 4 ternary units and total output of 360MW can supply the entire power needs of North Wales for several hours. After more than 50 years of operation, a major overhaul of the units is executed. This paper focuses on some of the challenges faced during the rehabilitation works.

1 Introduction

Commissioned in 1963, Ffestiniog Power Station was the UK's first major pumped storage power facility. Ffestiniog's four generating units achieve a combined output of 360MW of electricity - enough to supply the entire power needs of North Wales for several hours. Each of the ternary units consists of a Motor-Generator, a Francis type turbine, an engaging gear type coupling and a double-flow double-stage storage pump as shown in Figure 1. After more than 50 years of operation, a major overhaul of the units is executed. The overhaul has been planned in 2 stages, starting with units 1 and 2, followed by units 3 and 4, currently under execution.



Main parameters of Ffestiniog units:

Motor-Generator:

- Rated voltage: 16 kV
- Rated power: 107,2 MVA

Vertical Francis units:

- Design net head: 279 m
- Synchronous speed: 428 rpm
- Nominal discharge: 35 m³/s
- Nominal power: 92 MW

2-stage pumps:

- Nominal discharge: 22 m³/s
- Nominal power: 69,8 MW

Fig 1: Cross-section of Ffestiniog ternary units

The rehabilitation work consisted of supplying new turbines with overhaul of embedded parts, new pump and turbine spherical valves, new turbine governors, new generators, replacement of rotating exciters by static ones, new pressure relief valves (PRV), new

couplings between turbines and pumps, new blow down systems as well as extensive balance of plant electrical and automation scope. In addition, replacement and overhaul of some bearings including brackets as well as overhaul of the pump housings was also part of the overall scope.

During the negotiation phase an extensive investigation of the transient behaviour of the units was carried out, and it could be shown that the pressure limits for both the spiral case and the draft tube can be respected without application of the pressure relieve valve (PRV) for all transients. New generator rotor design does not limit the overspeed during unit transients either. Though, it has been decided to keep the existing configuration and to provide new PRVs. This decision had an impact on the mechanical design of the turbine distributor, and it turned out to be also source of several challenges during the commissioning phase.

Further challenges were related to changed concept for the shaft line dynamics. Despite of the increased bending critical speed the shaft line system showed high bearing vibrations during load rejection, requiring modification of the bearing supports.

During recommissioning, it was discovered that significantly higher amplitudes of shaft vibration occur in partial-load operation at the lower turbine guide bearing (LTGB). While the ISO standard cannot be applied for several reasons, the question remained to be answered if repeated operation at partial load would put the mechanical safety at risk under the present conditions.

This paper also addresses investigations related to grid connection topics, specifically tuning of the AVR & PSS modules during the commissioning phase.

2 PRV Control

The plant layout has been designed from the very beginning using Pressure Relieve Valves connected to the Turbine spiral cases. This allows faster control of the turbine while remaining within defined pressure limits for the waterways. During the design phase several options were investigated for the interface between the PRV and distributor control. At the end hydraulic connection between one of the distributor servomotors and the PRV servomotors was chosen.

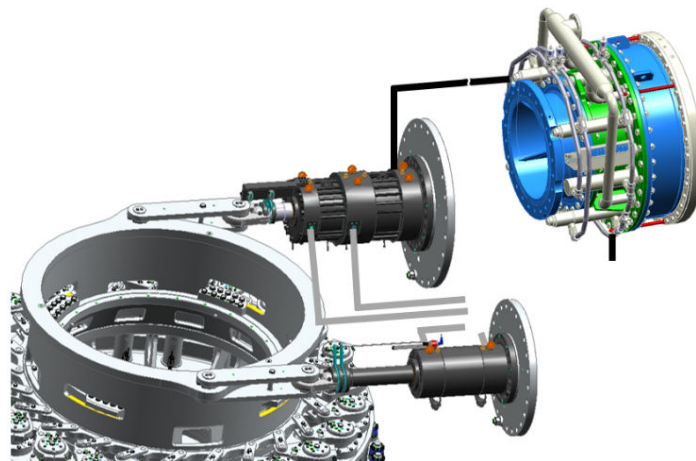


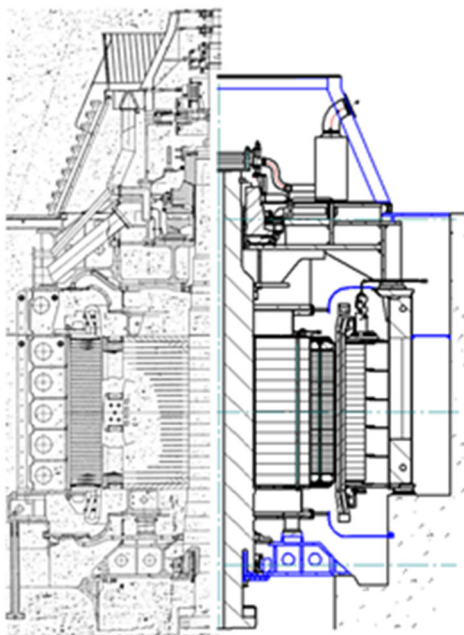
Fig 2: Hydraulic connection between the distributor and PRV servomotors

However, with this arrangement the regulating ring experiences a significant force asymmetry, resulting in additional loading of counter thrust segments of the regulating ring, which fixation had to be reinforced during the first months of operation.

Another challenge showed up to be the right setting of the timing of both distributor and the PRV, as the systems are not independent and adjusting the orifice of one system influences the timing of the other one as well. This tuning at site was very time consuming, especially seeing that the hydraulic control system is very sensitive and small orifice adjustment leads to significant timing change. Further on, large deviation of the timing between dry and wet tests could be observed.

3 Shaft and bearing vibrations during load rejection, critical bending speed

During the rehabilitation works the rotating exciters have been removed and replaced by static excitation systems. Additionally, several stiffening modifications have been implemented, such as stiffening of the rotor rim and of the upper bracket, which is supported by the stator frame. Also, the axis of the upper generator guide bearing has been lowered to decrease the span between the generator guide bearings. Fixation of the lower generator bracket has been improved as well. However, later analysis revealed that the new stator frame becomes the weak point of the overall stiffness chain for the upper guide bearing. See Fig. 3 for comparison of the original and new cross-section of the generator.



The original shaftline model consisted of the shaft elements and single (overall) stiffness elements for each bearing support – see Fig. 4. The maximum overspeed during transient operation derived from waterhammer calculation equals 696 rpm. Safety margin to the 1st critical bending speed was determined to be 15%.

During the commissioning tests of Load Rejections from the maximum power with deactivated Pressure Relieve Valve (PRV) it could be observed that the bearing vibrations increase with the rising speed to relatively high values. Partial improvement could be reached by additional balancing. However, detailed analysis of its 1st harmonic component showed that the rotor gets already close to the 1st Critical Bending Speed, not keeping the safety margin of 15%.

From the vibration data records the actual CBS was identified to be around 680 rpm instead of expected 800rpm. In the following the focus was set to understanding of the reason for this discrepancy and defining measures to reach the required safety margin for the CBS.

One of the important learnings was the fact that the stator dynamic behaviour must be included in the shaftline analysis when the upper bracket is fixed to the stator frame

and not directly supported by the concrete. Modal analysis has shown natural frequencies of the stator in the vicinity of the CBS – see Fig. 5.

After investigation of several alternatives like stiffening of the stator frame, it has been decided to implement additional supports of the upper bracket against the pit concrete walls, whereas the load limits of the existing structure had to be respected. With implementation of these supports the required CBS could be reached.

Using the adjusted shaftline model the system prior to the rehabilitation has been checked. The results indicate that the units prior to the modification featured 1st CBS slightly above 500rpm. However, as the generators were balanced at rated speed not far away from the 1st CBS the vibration level during load rejection events were still in a well acceptable range.

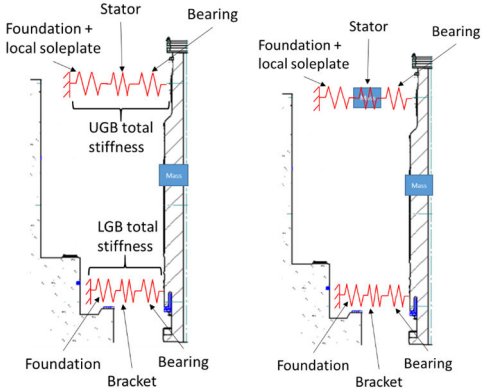


Fig 4: Shaftline models – old (left) and new (right)

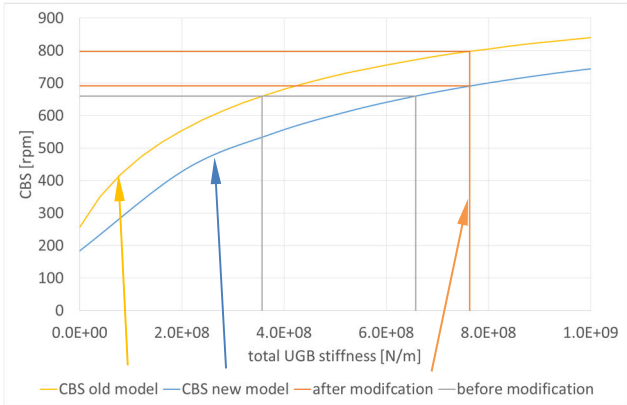


Fig 5: Calculated CBS with old and new models

4 Shaft vibrations during part load operation at lower turbine guide bearing

Because of the ternary type of the units there are two turbine guide bearings. The lower one is of a grease lubricated shell type and has been reused and refurbished during the rehabilitation works. Due to initial issues with bearing temperature at Unit 1 its clearance was slightly increased. Afterwards higher shaft vibration values especially during turbine part load operation were observed.

Fig 6: compares the amplitude of shaft vibrations before rehab (broken lines) and after rehabilitation (solid lines). Blue lines indicate direction x, red lines perpendicular direction y. Prior to rehabilitation, minimum amplitudes occur in the range between 65 - 70MW; very likely the condition of zero runner exit swirl was there as the units were originally designed with a 70MW output.

Amplitudes increase as the output (and discharge) differ from this condition. Test data show a maximum of vibration close to half of the zero-swirl discharge, vibration also increases in case of high load (negative swirl).

At rehabilitation, the design load has shifted to higher peak efficiency point and the 'calm' load is now close to 90MW, a change achieved via increased runner blade exit angle. Accordingly, the zone of critical swirl at part load also moved to higher output, about 45MW instead of 32MW. At the required minimum output of 55MW, the new runner layout therefore has higher residual swirl and higher draft tube pressure pulsations and vibration amplitudes. The vibrations of a Francis turbine in partial load are dominated by the vortex rope. When the turbine discharge Q is below its design value, the residual swirl downstream of the runner creates a spiral-shaped in the draft tube, rotating at 20-40% of runner speed in the sense of runner rotation. As a consequence, the pressure in the draft tube is distorted and neighbouring components, such as the runner or, in case of Ffestiniog, the shaft, experience a rotating radial force.

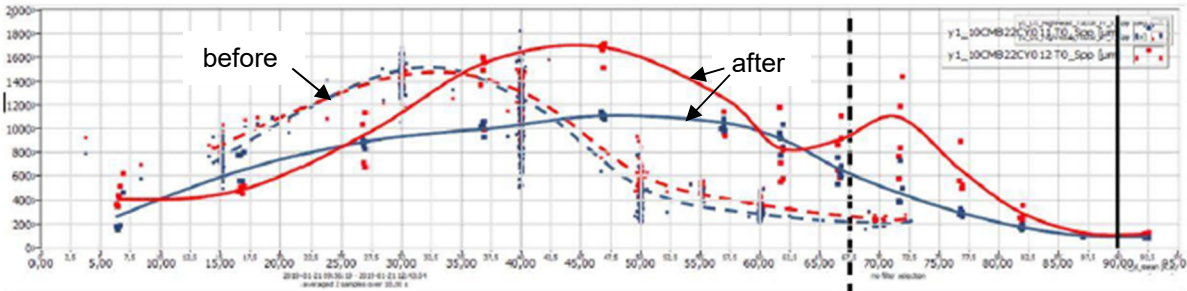


Fig 6: Shaft displacement (μm peak-peak) vs. output (MW)

Even though the international standards ISO 20816-5 showing the shaft vibration action limits, it also states that large shell type bearings with a diametrical clearance higher than 0.4 mm are not covered by the standard due to lack of statistical data. This means that there is no basis for the Ffestiniog lower turbine guide bearing having 1.0 mm clearance.

The most fundamental reason for not applying ISO 20816-5 however is that this standard is explicitly set up for oil-lubricated bearings. The statistical basis does not cover any grease-lubricated bearings like the lower turbine guide bearing having at Ffestiniog and are not applicable for part load operation either.

The vibration behaviour was analysed in detail to exclude any bearing malfunction. Even though the relative shaft vibrations reached up to 1600 μm pk-pk at part load with clear contribution from the vortex rope excitation, at full power the pk-pk value dropped below 200 μm . In addition, measurement of absolute bearing vibration is already installed. It is recommendable to check the casing vibration level with the Action limits in the ISO standard and from operating the units at part load, the level of absolute bearing vibration was found to be well below the lower Action limit of 0.9 mm/s rms thus confirming that the radial loads are not being transferred into the bearing housing or supporting structure.

5 Shaft vibrations @ lower generator guide bearing (LGB) during synchronous condenser mode of operation

After several months of operation of the rehabilitated unit 2 it started to trip during steady synchronous condenser mode of operation. The tripping signal came from relative shaft vibrations measured on the lower generator guide bearing. The phenomenon started on a sporadic basis, but the number of occurrences increased with the time. A reasonable increase of the limit values did not solve the issue.

Even though the absolute values of both the relative shaft and absolute bearing vibrations were on rather low level, it was decided not to increase the alarm and trip limits, and to investigate the root cause, as the measured vibration trend did not indicate any stabilization of the vibration behaviour – see Fig. 7. From the measured data analysis, a soft rubbing phenomenon in the vicinity of the bearing was suspected. This hypothesis could be confirmed with detailed infrared camera measurement. An asymmetrical temperature distribution caused by soft rubbing between shaft surface and the lower generator bearing air flow ring could be found soon – see Fig. 8. A visual inspection indicated that the gap between the shaft and the ring was asymmetrical (approx. 2-3 mm in one side, touching at the opposite side). Readjustment of this ring solved this issue, and this phenomenon has not been recorded since then.

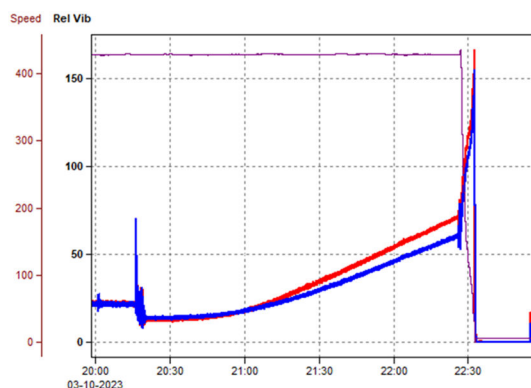


Fig. 7. Relative shaft vibrations at LGB

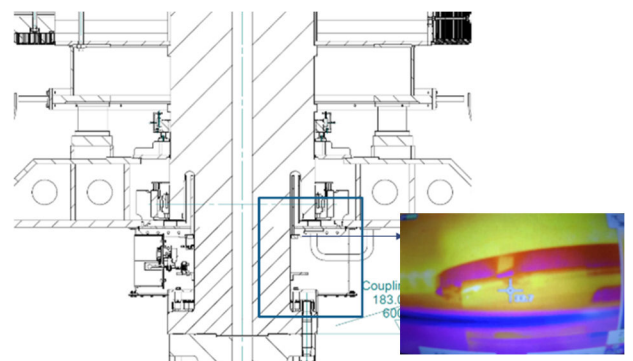


Fig. 8. Location of the soft rubbing

6 Tuning of AVR & PSS parameters

The UK National Grid (NG) grid code defines several requirements for generating units with respect to voltage control arrangement and performance.

The following list presents some of the major requirements:

- Steady State Voltage Control
- Transient Voltage Control (Ceiling voltage levels and reaction times)
- Power Oscillations Damping Control (Power System Stabilizer settings)
- Power System Stabiliser Tuning (PSS analysis in time and frequency domain)
- Fault Ride Through simulations for
 - o 3-phase voltage drop to 0% for 140 milliseconds
 - o 3-phase voltage drop to 50% for 450 milliseconds
 - o 3-phase voltage drop to 85% for 180 seconds

All these requirements have been checked and considered during the design work, summarized in a technical report and submitted through First Hydro to the grid operator for approval. As a part of the acceptance process some of the load cases had to be demonstrated at site, comparing the simulation results with the measured ones, including the impact of PSS on AVR and generator response in the frequency domain.

As one example the tuning of PSS parameters is presented here. The standard approach for PSS parameter selection is to increase damping of the generator natural frequency (at Ffestiniog units at app. 1,8Hz) and thus to ensure that the power plant does participate in the damping of potential power oscillations in the grid. However, in Ffestiniog project the optimization target function had to be extended for increased damping at lower frequencies (0,6 – 1,3Hz) to increase stability margins for inter-regional modes. In other words, the gain of the transfer function in frequency domain with PSS must be lower than w/o PSS activated, in the above-mentioned frequency range.

In addition, some discrepancies between measurement and simulation results were found, which led to several time-consuming iterations of analysis, report submission, approval by NG and verification at site. During these iterations several improvements of the calculation models were done, e.g. extension for the acquisition card behavior. During the optimization loops the HIL approach was used as well, leading to selection of AVR and PSS parameters fulfilling all requirements of the National Grid, see Fig. 9.

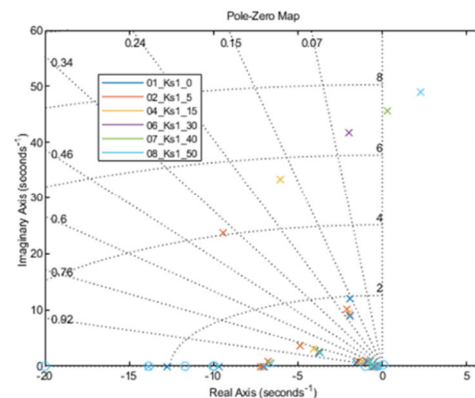
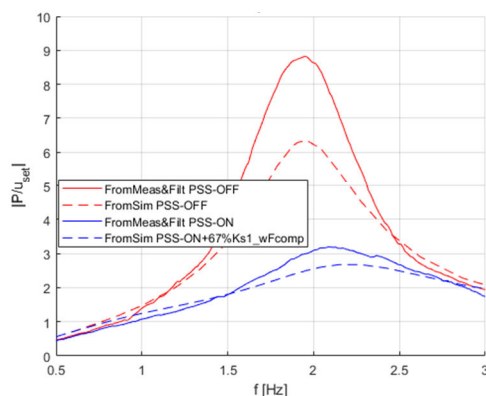


Fig. 9. Bode diagram for optimized PSS setting Fig. 10. Stability chart for different gain factors
As a part of the stability checks for the AVR / PSS parameter selection it must be demonstrated that the control remains stable even for 3 times higher proportional gain factor of the AVR – see Fig. 10.

7 Conclusion

This paper gives a few examples of challenges that aroused during the major rehabilitation work of the Ffestiniog pumped storage power plant units. Aside from the common issues arising at many power stations like vibration level or pressure pulsation topics, the following aspects were identified. For major rehabilitation works a correct assessment of the existing units is essential for correct evaluation of any modifications to be implemented. This point was underestimated regarding the critical bending speed topic in case of Ffestiniog. The other observation is related to growing effort necessary for complying with the grid connection regulations, both from the technical point of view and the approval process.

References

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- [2] The Grid Code, *Issue 6, Revision 3*, May 2021
- [3] Grid Connection Compliance Analysis Calculation Report, Internal report 2-10065127, December 2021

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