Refurbishment of twin Francis turbines – maximizing the annual production

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Abstract. Before the invention of the Kaplan turbine, Francis turbines were also built as twin Francis turbines. An example for such a single-regulated horizontal axis twin Francis turbine is the hydro power plant Meitingen – 3 units with an inlet chamber and a single-sided shaft led through. This plant, which is located on a diversion channel, went in operation in 1922 with a nominal power of 11.6 MW for the three machine sets providing an annual production of 72.6 GWh. Due to cracks and cavitation damages, one of the units should be replaced, yet, after a comprehensive study the decision for a major mechanical and electrical refurbishment was taken. The geometry was scanned on-site and a CAD model was generated. The RANS calculation for the whole power plant provided information on the overall performance, the losses referring to each component and the flow situation in each of the turbines. An optimization of the runner design was carried out and an increase of 14.8% of the annual production could be realized. All of the units were brought into operation after the refurbishment and the performance level was checked successfully by means of an on-site efficiency measurement campaign according to IEC 60041.

1. Introduction

Even after having been operated for more nearly 100 years, hydropower turbines are still able to transfer the energy stored in water into mechanical energy, which is then converted into electricity in the electrical part of a machine set. In general, such systems show, how sustainable the use of hydropower is, as other technologies used for power generation have significantly shorter life times. However, these systems also have a potential that should be explored – especially in times of low electricity prices – to increase the efficiency of the energy conversion and the output. Regardless of the age of a plant, small as well as large plants have to face this issue. An increase in efficiency enables plant operators to secure their investments in the long run and also to improve their earnings position. To encourage such investments, there is a corresponding incentive system in many European countries.

Due to the increased exploitation of other regenerative energy sources, the demand situation has changed and therefore hydropower has to meet new requirements. When refurbishing an existing plant, specific measures have to be taken to meet the new framework conditions and to modernize the relevant plants. Mostly plants are brought up-to-date by exchanging or converting parts, or even constructing new components. In addition to plant efficiency and system services, such as regulating energy and stabilizing frequency, however, refurbishments also focus on noise emissions, vibrations and oscillations, cavitation safety and the operation range of power plants. Especially, as nowadays
plants are operated at load points unthinkable a few years ago – be it operation at lowest part load or with frequent load changes. The refurbishment of existing plants is therefore essential in order to increase plant efficiency and annual production as well as annual earnings. These three factors are (mostly) associated to an investment, whereas today, the corresponding investment decision can be backed up much better as, based on the constant advancements in the field of numerical flow simulation, there is an increasing number of empirical values determined by means of CFD. The better and more detailed the database at the beginning of a plant optimization, the runner more targeted and reliable the decision to rebuild or refurbish. In some cases, new investments have been made after just a few years of operation. In addition to improved hydraulic turbine efficiency, better operations management as well as sophisticated plant automation can result in higher plant efficiency. For decades already, Numerical Computational Fluid Dynamics (CFD) has been a constant companion of hydraulic machines. Thanks to further developments of turbulence models (e.g. length-scaling models such as the SAS turbulence model [1]) and the use of parametric models (e.g. automated optimization with evolutionary algorithms [2]), the flexible simplified computational grid design and the many evaluation options (e.g. the histogram method for the cavitation evaluation [3] [4]), CFD is now able to produce a sufficiently accurate quality statement on the performance of a hydraulic machine.

2. The Meitingen hydro power plant

Prior to the invention of the Kaplan turbine, Francis turbines were also built in twin, triple or double twin versions per machine set [5]. An example for such a machine set is the power plant Meitingen (Germany) with three identical Twin Francis turbines in open flume (vertical shaft) design [6] with a one-sided shaft feed through. The power plant (Figure 1) was built in the years 1918 to 1922 and has a maximum power of $P = 4.4$ MW per machine set with a runner diameter of $D = 2.41$ m. One machine set has a discharge flow rate of $Q = 42$ m³/s, whereby the very high specific speed per impeller is calculated to be $n_q = 114.2$ rpm. The power plant site is located at a diversion on the Lech channel, but no channel lock was provided there, as it was the case with the upstream power plants Gersthofen and Langweid [7]. A summary of the system data can be found in Table 1, where for the calculation of the specific speed the rotational speed in rpm is used.

![Figure 1. Meridional section of the hydro power plant.](image-url)
Table 1. Nominal data.

|                                 | Year of construction 1918-1922 | Speed 187.5 rpm | Runner nominal diameter 2.41 m | Suction head +6.6 m | Head 12-15 m |
|---------------------------------|---------------------------------|-----------------|--------------------------------|---------------------|--------------|
| Max. flow rate                  | 42 m³/s                         | 21 m³/s         |                                |                     |              |
| Max. shaft power                | 4400 kW                         | 2200 kW         |                                |                     |              |
| Specific speed \((Q_{\text{BEP}}) n_q\) | 161.6 rpm                       | 114.2 rpm       |                                |                     |              |
| Specific speed \((P_{\text{Max}}) n_s\) | 519                             | 367             |                                |                     |              |

Figure 2. Analysis of flow rate at site location.

A data analysis showed that the largest proportion of the annual production is generated at overload operation. Figure 3 depicts the production of the machines in a segmented form. The basis for this analysis is the channel flow rate available from 1960 up to now, which is shown in Figure 2. The maximum flow is given by the diversion structure of the Lech channel. For the last 55 years illustrated, the average cumulative flow rate amounts to \(Q_{\text{averaged}} = 93.4 \text{ m}^3/\text{s}\). In order to estimate the potential for the efficiency increase, an efficiency curve measured by Voith in 1988 was used as a basis. The flow duration curve for the River Lech is available in detail on the website of the "Hydrological Service of Bavaria" and was evaluated for the years 2001 to 2014, as reliable production data are also available for these years. In this case the maximum amount determined by the diversion channel was set to be \(Q_{\text{Metingen,max}} = 121 \text{ m}^3/\text{s}\) (the water regime at the channel itself is well-known and the values of two other power plant sites could be taken as references), and the maximum flow rate of one machine is limited to \(Q_{\text{Unit,max}} = 41 \text{ m}^3/\text{s}\) (according to the adjustment of the guide vane locking in the system). The minimum amount of water is \(Q_{\text{Unit,min}} = 29 \text{ m}^3/\text{s}\), and since 1961 it has only been achieved once. It can be seen, that the highest annual productions were achieved in the area of the highest flow rate generated by one machine \((Q_{\text{Turbine}} = 39-41 \text{ m}^3/\text{s})\). The general assumption is though that the machine sets (in case of more than one machine set being operated) are subject to equal load. Thus, the one-machine operation resulted in 3.5 days per year, for a two-unit operation 137.9 days per year and for a three-unit operation 223.8 days per year.

Figure 3. Annual production.
The total generation added up to 72.88 GWh, whereby the self-consumption of 0.25 GWh still had to be subtracted. This annual production is in excellent agreement with the production data, i.e. the actual annual production over the period 1994-2014 (values measured by the central control room) of 72.5 GWh. The generator efficiency used was calculated on the basis of an acceptance protocol of the last generator renovation from the year 1997, in the course of which individual losses were measured. The losses were then split into constant (stator purely resistive losses, load-dependent losses and resistive losses of exciters and brushes) as well as variable losses (frictional losses, iron losses and load-independent additional losses), and thus the efficiency is calculated for the rated power factor of $\cos \varphi = 0.84$ for the operation range.

3. Numerical simulation
The numerical calculation of this plant was carried out in two stages. In order to get an understanding of the different factors influencing turbine performance and to get results faster, a simplified single-channel model was used. For this purpose, the boundary conditions were set at the entrance of the guide vane, with only one guide vane passage and one blade passage, each modelled with periodic interfaces. In a second step, an overall machine model was calculated and the boundary conditions were placed at the real inlet. Thus both, guide vane and runner, became 360-degree models. Finally, an additional outblock had been installed, so that the flow situation at the end of the two draft tubes (at the end of the junction) was not limited by a boundary condition. The model (Figure 4) of the overall machine included 21.13 million nodes for the mid-size grids, 11 million nodes for the coarse grid, and 45 million nodes for the fine grid. The $y^+$-values at the walls for runner and guide vane are below $y^+<20$.

With the help of the commercial CFD code Ansys CFX V16.1 [8] the Navier-Stokes equations were solved. These Navier-Stokes equations describe the fluid motion in all three dimensions and were used with a Reynolds-averaged Navier-Stokes (RANS) formulation. RANS uses equations where the instantaneous variables are decomposed into mean and fluctuating values with the help of a Reynolds-decomposition, whereas these variables are time-averaged. Additionally, a MFR (multiple

![Figure 4. CFD Modell, (a) Model overview full setup, (B) detail of guide vanes and runners with surface mesh on draft tubes, (C) runner view from downstream including shaft feed through.](image-url)
frame of references) approach was used for the rotating domains (= runners). Menter’s [9] SST turbulence model with automatic wall functions was applied for the stationary calculations, and in order to achieve a satisfying convergence level all sensitive variables and imbalances were monitored.

For the evaluation of the hydraulic performance, the key figures as mentioned in the following are of interest. In general, the net head is the difference between the total pressure at the inlet of the spiral and the total pressure at the outlet of the draft tube. According to the IEC standard [10], the net head represents the difference between the static pressure at the inlet (inflow of spiral) and the static pressure at the outlet (end of draft tube) where the mean kinetic energy head is added to the inlet and outlet pressures (equation 1).

\[
H = \frac{P_{\text{total\-inlet}} - P_{\text{total\-outlet}}}{\rho \cdot g}
\]

\[
H_{\text{Loss-Runner}} = \frac{P_{\text{total\-Runner\-inlet}} - P_{\text{total\-Runner\-Outlet}}}{\rho \cdot g}
\]

\[
\eta_{\text{total}} = 1 - \frac{\Sigma H_{\text{Loss}}}{H} = 1 - \left( \frac{H_{\text{Loss-Runner}} + H_{\text{Loss-Guidevane}} + H_{\text{Loss-Intake}}}{H} \right)
\]

In order to analyse each component separately, a head loss analysis (see equation 2) was performed to calculate a cumulative distribution of the total unit. In this case, the total pressure difference between inlet and outlet of each component was set in comparison to the net head. For the runner, the shaft power was also taken into account and subtracted from the losses (equation 3).

![Figure 5. Existing runner pressure and velocity plot, cavitation damages.](image)

For a wide range of operation the recalculation of the turbine confirmed the measurement realized in 1988, whereas the numerical recalculation is above the measurement in the area of full load. A plant measurement from the year 2015 indicates that the calculated efficiencies are slightly below the measured ones. Figure 5 shows the location on the runner providing optimization potential \(Q_{\text{BEP}}\). An inhomogeneous distribution of the pressure generation, especially in the inlet area of the blades (a), can be seen. As a result of the unfavorable inflow velocity distribution in the shroud area, the stagnation point lies on the pressure side of the blade for the calculation of the optimum point (d).
There is a rather high deviation between the flow angle and the blade angle in the area of the outer diameter (95% blade height) (e). The low pressure zones on the suction side of the blade were numerically calculated (b) and confirmed during the runner scan, as cavitation zones (c) were found.

4. Runner optimization

Then, runner blade has been remodeled in a blade design tool and gradually optimized (manually) by changing the runner blade angles and profiles. Every major change was verified in a full-machine simulation. At the end of the optimization, there was a uniform pressure distribution (Figure 6 and Figure 7) on the pressure side of the blade. The suction-side low-pressure zones could be reduced and moved more into the middle of the blade. Thus, the stagnation point was exactly on the leading edge and lead to an equalization of the pressure distribution (Figure 6c).

Figure 6. New runner, (a) pressure plot, (b) pressure plot (view from downstream), (c) Pressure at turbo surface at 95 % blade height.

Figure 7. Blade pressure.

Figure 8. Velocity components runner outlet.

The pressure drops present in the original runner could be improved to a better level, ultimately leading to a uniform distribution of the meridional velocity at the runner outlet. Figure 7 and Figure 8 shows the pressure and velocity distribution. With Figure 7, the pressure curve around the blade is shown at 75% blade height (height from hub to shroud). While with the original design, especially on the suction side, an extremely strong pressure drop at the inlet area occurred and then the blade pumps to almost 60% blade length, the pressure reduction was rather moderate during the re-design and a pumping zone was not present (horizontal solid black line of the optimized variant up to about 40% blade length). In Figure 9 the blade angle beta is shown versus the meridional length for both the original as well as the optimized hydraulic shape. Additionally, the blade thickness distribution is shown in grey. The original hydraulics were scanned and therefore the shape is not that smooth. The inlet angle for the optimized version is higher, where the meridional length is even higher for the new runner design. The blade itself has a slightly smaller thickest portion, where the trailing edge has the same dimension. Also shown in Figure 9 is a meridional view.
This pressure drop was pronounced for larger blade heights too (for example \(\text{span} = 0.95\), see also pressure plot in Figure 5d), which led to the cavitation damage already described above. In Figure 8, the velocity distribution at the blade outlet can be seen versus the blade span, with the meridional component showing a much flatter and more uniform distribution at the runner outlet for the optimized version. This is the case in particular from 60% blade height. The swirl component could also be improved. The remaining residual swirl could be lowered, although this increased pressure on the shroud as well as on the original runner. The optimized hydraulic design shows a significantly improved peak efficiency, which is illustrated in Figure 10 together with the existing one. Figure 10 additionally shows a streamline plot in the two draft tubes of the turbine halves for the overload operation point.

5. Economics
Originally, only the replacement of a double Francis runner was envisaged, as aging-out had been reached due to wear, cracks and cavitation damage. It has to be mentioned that, according to the German Act for the Development of Renewable Energies (Renewable Energy Act – EEG 2017), even plants with a maximum output of more than \(P = 5\) MW for the entire power plant can receive funding if modernization measures increase annual work by more than 10%. In the case of the HPP Meitingen and its \(P = 11.6\) MW expansion capacity, it would therefore be possible to sell the more electricity generated after modernization at 5.3 ct/kWh according to the EEG 2014 – instead of the EEX market price of less than 3 ct/kWh.
By changing merely the runner, the annual production can already be increased significantly, but a more radical approach has been chosen in order to profit as much as possible. The majority of twin Francis machines were designed with a coupled guide vane. Just a few machines have been designed with decoupled runner halves, since these have always been designed for such use. This approach was analyzed as a result of a decoupling of the guide vane of one unit. Analytic considerations showed that the modification of the guide vanes of all three units does not provide any significant benefit and would result in a delay for the overall project.

The technical convertibility of the single-regulated (decoupled) guide vane has been taken into account, and the higher axial forces were absorbed with a new, more efficient axial thrust bearing. In addition, a leverage design for the decoupled distributor had to be provided. Also, a new operation scenario was developed, see Figure 11. Depending on the flow, one turbine half of the machine with the decoupled guide vane diffuser was switched on and off. M1 is the decoupled unit, M1-1 represents the operation with only the decoupled runner half, and M1-2 indicates the generation with both runner halves, which then operates with the same performance as M2 or M3.

Now, the efficiency curve of the runner optimization has been used as the basis for further investigations, see Figure 12 for different scenarios. The black line symbolizes the basis; whilst the start and stop points of the individual machines are clearly visible (lower points with lowest efficiencies for each scenario). The black dashed line displays the use of the new runner, and the gray line represents the new runner and, in addition, the decoupled guide vanes of one unit. The turbine efficiency is a flow-rate-weighted average efficiency of the individual sets of machines. The increase in efficiency induced by the new runner design (gray vs. black line) as well as the additional effect of the decoupled guide vane for one Twin Francis turbine in two single turbines halves (black dashed vs. gray line) are clearly visible, whereas especially the lower flow rate range provides for significantly increased efficiency.

In addition, the frequency of occurrence is displayed (gray dotted line). It shows the relative operating days put in reference to an average year. As sampling rate a flow rate of $Q = 0.1 \, \text{m}^3/\text{s}$ was used, and the operation days for each flow rate were added up and normalized. Thus, the annual productions as well as the operation mode are optimized by 14.8% to 83.37 GWh.

Since there was the risk that the EEG 2014, which was applicable at the beginning of the project planning, would expire by the end of 2016, a very ambitious schedule had to be developed. In addition to the fact that the original project was extended, commissioning would have to take place at the end of 2016 as of the notified amendment of the EEG in 2017. So, the reconstruction time originally planned was reduced by 50%.

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**Figure 11.** Operation plan of the units. **Figure 12.** Plant turbine efficiency and relative frequency.
6. Technical design aspects

The redesign of the adjustment mechanism to decouple the guide vanes for a unit had to meet particular technical challenges. The regulating ring of each turbine half was maintained and a synchronous operation of the two guide vanes in coupled operation had to be realized. Also, the non-operating guide vane had to be locked in decoupled operation. This was technically solved by means of proprietary locking cylinders (Figure 13a).

The existing axial collar thrust bearing had to be replaced by a more massive design, the turbine shaft itself had to be maintained and the force had to be introduced safely into the structure. The thrust bearing was completely redesigned for a maximum axial force of \( f_{ax} = 230 \text{ kN} \). The modification of the turbine shaft included the turn in for two axial clamping rings and the welding in the assembled state (Figure 13b). After a temporary stress relief heat treatment, the finishing was realized in installed state. For a better transmission of the force, two additional struts with special brackets were integrated into the draft tube sheet metal construction (red struts in Figure 13c). Normally, compressive forces are initiated. If, due to a malfunction or system failure of the guide vane, runner 1 is closed and runner 2 remains in operation, the opposite axial forces can be absorbed by these struts.

The runner half not in operation should provide only ventilation losses and should not wade in water. For this purpose, a reliable lowering of the water level in the non-active draft tube had to be ensured. Such lowering was reached by means of ventilation in the draft tube conus when switching from double (coupled) to single (decoupled) operation. The opening and closing of the new ventilation line is controlled with an electric ball valve.

7. Measurement

After the reconstruction of the first unit, an on-site efficiency measurement was carried out at the HPP Meitingen for comparison with the CFD calculation and the efficiency curve of the old runner. This was performed strictly according to IEC 60041 [10] with a current meter measurement for the flow rate. The measured unit 3 is the one without the decoupled guide vane. The efficiency curve is now shifted to full load, and therefore the efficiency at part load is lower than the existing one. This disadvantage is negligible as with the decoupled unit (see Figure 14) this operation range could be avoided.

The measured unit 3 achieves a turbine efficiency of 89.9% assuming a generator efficiency of 97.0%. At maximum output of the machine (with 90% opening of the guide vane), with a flow rate of \( Q = 42.34 \text{ m}^3/\text{s} \) and a net head of \( H = 12.79 \text{ m} \), an active power of \( P = 4554 \text{ kW} \) can be achieved. The efficiency curve is shown in Figure 14, together with the efficiency curve of the former runner and the CFD calculation curve. The deviations between measurement and simulation are minor.
8. Conclusion
The HPP Meitingen example shows that even with existing hydropower plants, remarkable ideas and cross-tested solutions can be realized to develop significant potential for the generation of more renewable hydroelectric power. In this case the annual power production increased by 14.79%, which corresponds to an additional production of 10.74 GWh. Furthermore, the HPP Meitingen will receive higher reimbursement according to EEG 2014 for the additional electricity generated over a period of 20 years. The analysis of the power production for the year 2017 confirm the estimated benefit.

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