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Performance tests and evaluation of hydropower plant with double rotating hydropower screw system

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Professional paper

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Performance tests and evaluation of hydropower plant with double rotating hydropower screw system

The newly developed Archimedean double screw hydropower concept combines hydroelectric power generation and bi-directional fish passage in a single device. Efficiency tests were conducted at the first commercial facility to verify the design data. The downstream migration possibilities have previously been proven. However, this study demonstrates the operational capability for upstream migration. The hydraulic efficiency of this type of power plant is also determined. This is the first time that the hydraulic performance of this type of plant has been demonstrated.

Key words:
low-head, hydropower screw, fish-lift, hydraulic efficiency, fish migration

Stručni rad

Petr Lichtneger, Christine Sindelar, Helmut Habersack, Bernhard Zeiringer, Christian Lechner, Gernot Mayer, Nino Struska, Walter Albrecht

Ispitivanje učinkovitosti i ocjena hidroelektrane s dvojnim rotirajućim hidroenergetskim pužnim sustavom

Novorazvijeni koncept Arhimedovog dvostrukog hidroenergetskog pužnog sustava u jednom uređaju objedinjuje proizvodnju električne energije i dvosmjerni sustav za prolazak riba. Na prvom komercijalnom projektu provedena su ispitivanja učinkovitosti kako bi se provjerili podaci dobiveni tijekom projektiranja. Otprije je poznata mogućnost nizvodnog propuštanja. Međutim, u okviru ovog istraživanja dokazana je i stvarna mogućnost uzvodnog propuštanja. Također je utvrđena i hidroenergetska učinkovitost postrojenja ove vrste. Prvi je puta dokazana hidraulička učinkovitost ovog uređaja.

Ključne riječi:
niski tlak, hidroenergetski puž, uređaj za podizanje riba, hidraulička učinkovitost, propuštanje riba

Fachbericht

Petr Lichtneger, Christine Sindelar, Helmut Habersack, Bernhard Zeiringer, Christian Lechner, Gernot Mayer, Nino Struska, Walter Albrecht

Untersuchung der Effizienz und Bewertung des Wasserkraftwerks mit doppelt-rotierendem Wasserkraftschneckensystem

Das neuentwickelte Konzept des Archimedischen doppelten Wasserkraftschneckensystems in einer Anlage vereint die Erzeugung von elektrischer Energie und das Zweiwegesystem für den Durchgang von Fischen. Beim ersten kommerziellen Projekt wurde eine Untersuchung der Effizienz durchgeführt, um die während der Planung erhaltenen Daten zu überprüfen. Von früher ist die Möglichkeit des Durchlasses stromabwärts bekannt. Im Rahmen dieser Untersuchung wurde jedoch auch die tatsächliche Möglichkeit des Durchlasses stromaufwärts nachgewiesen. Darüber hinaus wurde auch eine hydroenergetische Effizienz solcher Anlagen festgestellt.

Schlüsselwörter:
niedriger Druck, hydroenergetische Schnecke, Fish-Lift, hydraulische Effizienz, Durchlass der Fische

1. Introduction

The hydropower screw turbine is a technology based on reversal of the old Archimedean pump. As of recently, the technology is increasingly being implemented for small low-head hydropower installations. The Archimedes screw turbine was patented in 1991 by Karl-August Radlik [3] and the first tests were conducted under the guidance of Karel Brada at the Technical University in Prague from 1995 to 1997. Since that time, hundreds of installations have been implemented worldwide [5-7]. With regard to low-head conditions, the screw turbine presents several advantages such as good efficiency, robustness, and low investment and maintenance costs. Furthermore, the hydropower screw with low rotation frequency, and therefore with insignificant shear forces or pressure changes, can enable fish population to migrate the watercourse downstream with a very low (or even no) risk of injury [8, 9].

The double screw concept newly developed by the Austrian company Hydroconnect [10] combines hydroelectric power generation and bi-directional fish passage in a single device. This solution fulfils very well principal requirements of the EU Water Framework Directive [11] on the passability of transverse structures in rivers. The prototype was installed at the site of the head office of Hydroconnect and its first commercial installation is situated on the Sulm River at the confluence of the Sulm and Mur rivers near Retznei, Austria. The project was commissioned in 2015 [12]. The installed double rotating screw was designed with the gross head of 5.5 m and the flow rate of 380 l/s for potamal fish migration. The predominant fish is the pike measuring 90 cm in length. The Retznei hydropower plant consists of a weir with the so called fish-belly flap gate, the primary run-off hydropower plant with a Kaplan turbine, and a new "fish-ladder" hydropower plant with the Hydroconnect screw, see Fig. 1 [13]. The fish migration monitoring [1] and hydraulic efficiency tests [2] had to be carried out to verify the design data. While the downstream fish migration has been proven before, the fish behaviour at the study site demonstrates operational capability for upstream migration. Most fish species of various age were able to migrate upstream. Regarding hydraulic performance, the hydraulic efficiency was determined for the first time at this type of power plant in normal operation. The double rotating screw technology and the hydropower conditions will mainly be presented in this paper.

2. Fish-lift concept with double rotating screw

The hydropower screw from Hydroconnect connects the screw turbine and the screw pump technologies together, and they operate simultaneously. The cylindrical rotor consists of an inside tube screw aligned as a pump and a shorter outside tube screw aligned as a turbine, see Figure 2 left. Thus, the whole rotor is a gap free cylinder that is mounted using one axial bearing near the generator, and a double flat belt bearing holding the radial forces. A rubber seal is implemented at the turbine water inlet. The inner pipe works as an integrated fish-lift lifting up the fish inside a "water pocket" of the inner screw. If the fish reaches the upper end it falls out of the pipe into the head water, or into a trough which routes the fish further away from the turbine inlet. The entrance of fish into the fish-lift tube from the tail water is shown in Figure 2 right. Although there is only one screw blade in the inner pipe, the screw is doubled in the lower section. In this way the surplus of the water taken through the doubled screw returns back downstream of the pipe, thus ensuring an additional axial attractive flow for the fish.

Figure 1. View of Retznei Barrage with the original hydropower plant (photo on the left) and the new hydropower plant with the tube-type double rotating screw on the left bank side (photo on the right)

Figure 2. View of demonstration model of the double rotating screw with the fish lift inside (on the left) and a snapshot of fish entrance from the tail water (on the right) [10]

3. Technical specification

The Retznei hydropower plant site was enhanced with a new installation with the double rotating screw on the left bank side in order to restore the river passability for fish as well as macrozoobenthos. Nevertheless, technical and economic issues such as the efficiency, maintenance effort, and amortization rates, play a significant role and must be considered. The main geometry and operational parameters of the new Retznei power plant are given in Table 1.

Table 1. Main parameters of the new hydropower plant

Parameter	Value	Unit
Inner tube diameter	1220	mm
Outer tube diameter	1800	mm
Nominal gross head	5,5	m
Inclination	30	°
Pumping screw blades no.	1 2 (at entrance)	-
Turbine screw blades no.	3	-
Pitch ratio of pumping screw	1,08	-
Pitch ratio of turbine screw	1,10	-
Speed range	0 – 20	rpm
Discharge range	0 – 550	l/s
Nominal electric power	15	kW
Net-head during absolute efficiency measurements	5,13 – 5,16	m

The total plant efficiency of the turbine can be calculated as follows:

$$\eta = \frac{P_{gen}}{\rho g H \left(Q_t - Q_p\right)} \quad (1)$$

In Eq. (1), Q_t is the turbine water discharge, Q_p is the pump water discharge, the term ρgH is the net specific hydraulic energy, and P_{gen} is the generated electrical power. The pump water discharge circulates in the system and can be re-used for energy production, thus partly compensating the energy spent for pumping. The denominator therefore substitutes the available hydraulic power P_h. Applying the torque T and rotation speed n in revolutions per minute, the mechanical power P_m can be calculated from:

$$P_m = \frac{2\pi n T}{60} \quad (2)$$

Specifically, the hydraulic and mechanical efficiency, η_h and η_m, can be calculated as follows:

$$\eta_h = \frac{P_m}{P_h} \quad (3)$$

$$\eta_m = \frac{P_{gen}}{P_m} \quad (4)$$

The turbine is operated at variable speeds, and so the discharge can be controlled. The speed-controlled generator from Siemens (type 1PH8133) and a gear-box from Siemens Flender (type H3HH506) is used at the Retznei Power Plant. A slice gate is implemented at the beginning of the intake channel so that the inlet water can be closed.

4. Field tests to determine hydraulic performance

The primary goal of the tests was to determine hydraulic efficiency (η) of the power plant with double screw. The head water level (z_0) upstream of the inlet gate, the head water level (z_1) at the intake behind the inlet gate, and the tail water level (z_2), were measured with levelled acoustic sensors. The speed (n) was measured with an inductive switch sensor using magnets fastened onto the rotor. The electricity power (P_{gen}) was read out from the generator signal output pins. The opening (a) of the inlet gate was gauged manually. Furthermore, the noise level in the power house (L_{p1}) and at the outlet (L_{p2}), the water turbidity at the intake (TR1) and outlet (TR1), the air and water temperatures (T_A, T_W), and the ambient pressure (p_{amb}) were measured at numerous operating points. A complementary torque measurement was also carried out using strain gauges glued on the cantilever and the tie bar holding the generator. An overview of all measured variables is given in Figure 3. Additional details can be found in [2].

4.1. Discharge

The total discharge (Q_{tot}) was hydrometrically measured behind the inlet gate, which was fully open. The absolute integrative

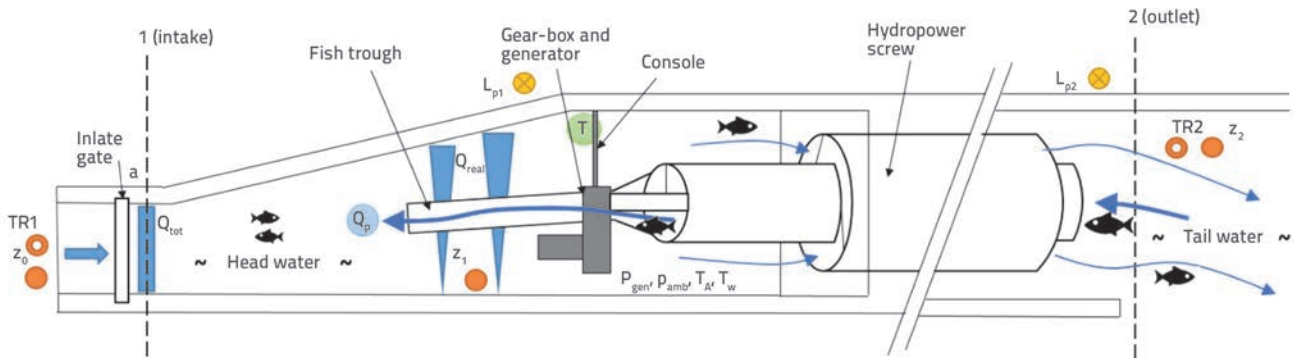

Figure 3. Schematic of power plant with main parts, measured variables are labelled

method according to ISO 748 [14] was implemented using the acoustic Doppler Velocimeter *Flow-Tracker* from *SonTec*. The *Flow-Tracker* was mounted on an automatic vertical carriage with a controlled speed and track, so that the mean flow velocity (u_m) at one vertical line was integrated uniformly over h_m = 82 cm in 117 seconds (speed: 0.7 cm/s) starting at h_b = 3 cm above the bed and finishing at h_s = 2.5-4 cm under the water surface (depending on the current water level), and vice versa. The acoustic principle of measurement itself was not influenced by traversal speed. The specific discharge (q_i [m²/s]) was measured at nine vertical profiles *i* and then integrated over the whole width (1.5 m) using the trapezoidal rule. A correction was made in each vertical according to the ISO standard for the regions adjacent to the bed and the water surface, which were not measured. The mean specific discharge in the zone between the wall and the next vertical was calculated using the velocity distribution method on the wall. All methods were implemented according to suggestions given in IEC41 [15]. The Flow-Tracker accuracy was also verified in the calibrating flume using reference measurements with LDA (Laser Doppler Anemometry), and simulating natural turbulent and turbid flow conditions in the hydraulic laboratory of BOKU. The final distribution of discharge over the width of the inlet channel is given in Figure 4 for six steady operating points. Circles are measured specific discharge values; short horizontal lines are mean values between measuring verticals and the wall.

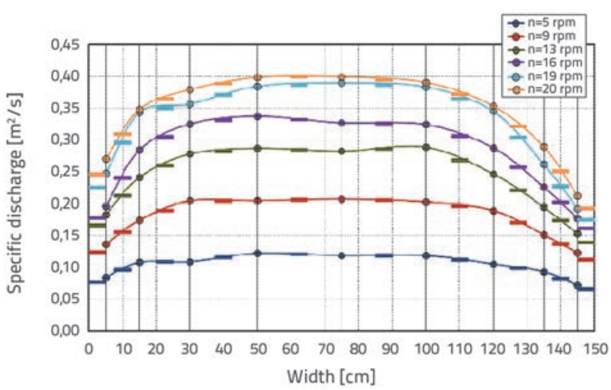

Figure 4. Results of the integrating method of discharge measurement

The pumping discharge (Q_p) was measured using the volumetric method according to ISO 8316 [16]. An open tank 507 litres in volume (the exact tank volume was precisely determined in our hydraulic laboratory) was used and quickly placed at the upstream end of the fish ladder trough under the falling water jet. The filling time of the tank was measured with a stopwatch.

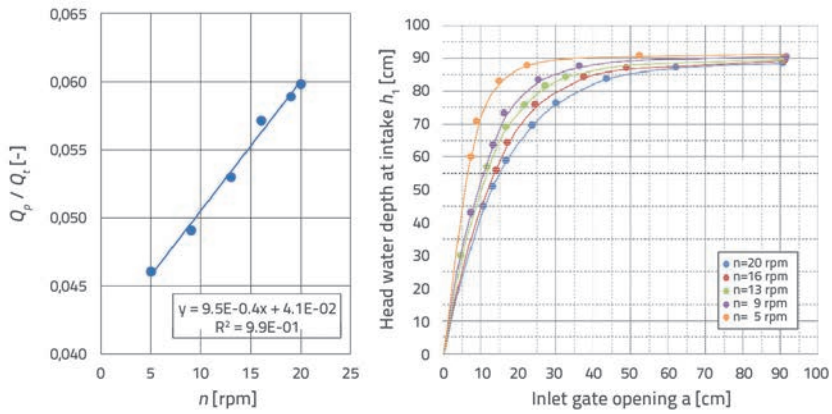

Figure 5. Pump-to-turbine discharge rate as related to rotating speed (left), and head water depth at the intake as related to inlet gate opening (right)

The measurement was repeated several times at different rotating speeds. During the measurements, the turbine water was blocked (inlet gate was closed) and the runner was motor-driven (with negative generator power). Results dependent of the speed corresponded quite well to a polynomial function of the second order [2]. Thus, the turbine discharge $Q_t = Q_{tot} + Q_p$ and the ratio of the pump to turbine discharge Q_p/Q_t, as related to the speed (*n*), could be determined for measured operating points as shown in Figure 5, left. The ratio was not constant but varied from 4.5 to 6 % over the operational range. The influence of the tail water level on the pumping performance could be neglected because its fluctuation was kept within an interval of 9 cm and the entrance cross-section of the inner pumping screw was submerged completely during all measurements.

Because of space limitations at the intake, the absolute measurement of total discharge was meaningful only if the inlet gate was fully opened. In case of a partly open gate, the relative measurement was implemented using two acoustic *Side-Looking-Doppler* (SLD) probes from *OTT-Hydromet*. They were mounted at various heights and cross-sections to better resolve the flow distribution under several flow conditions. The probes provided velocity means in horizontal paths over the width of the cross-section. The relative measurement was calibrated using the total discharge values from absolute hydrometric measurements. The inlet gate was used to lower the head water level at the intake to the turbine. At operating points with reduced intake depths due to reduced inlet gate opening, the turbine discharge was determined with regard to the relative depth of acoustic measurement paths and the log-law velocity distribution. The accuracy of the estimation was much lower compared to the one from absolute measurements because the SLD velocity data showed high fluctuations over the measurement time interval, which always amounted to approximately 5 minutes. Nevertheless, the results were consistent and allowed estimation of efficiency for reduced water depths at the intake. Since the pumping was independent of the head water level, the ratio Q_p/Q_t increased with closing of the inlet gate adequately. The relationship between the depth in the intake channel (h_1) and the inlet gate opening is shown in Figure 5, right, for several turbine speeds.

4.2. Head

The net head was first calculated between the cross-sections 1 and 2 (Figure 3) in case of absolute discharge measurements ranging from 5.13 to 5.16 m. The net head was then calculated with the SLD1 probe between the cross-section and the section 2 in case of lower water levels at the intake (relative measurements) varying between 4.60 and 5.18 m. The nominal gross head of 5.5 m was not available during the measurements because of higher tail water levels in the Mur River.

4.3. Efficiency

Accurate efficiency measurements were conducted using absolute discharge measurement methods with the inlet gate fully opened and at six different speeds of the turbine. The absolute systematic uncertainty of efficiency was estimated at ± 1.8 % (at the conf. level of 95 %) with respect to standards IEC 41 and IEC 62006 [15, 17]. The efficiency values were normalized using the local maximum (η_{max}) and marked as relative efficiency: $\eta_{rel} = \eta / \eta_{max}$. The normalized efficiency curve with the corresponding uncertainty bandwidth based on the absolute discharge measurements is shown in Figure 6. The efficiency is plotted against the speed, which is an independent test variable. No conversion to a specified (nominal) head was made.

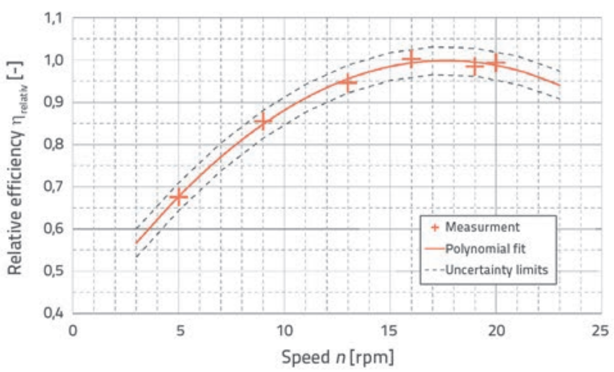

Figure 6. Relative efficiency curve determined using absolute hydrometric measurements of total discharge, with inlet gate fully opened and normalized with its maximum. The head of approximately 5.15 m was mostly constant

The dependences of the discharge, the electric power, the head and the water depth at intake on the inlet gate opening and speed were determined from measurements with reduced water levels at the intake. Using these results, efficiencies were calculated for a regular grid of speeds (5 – 20 rpm) and water depths (50 – 90 cm), normalized with a global efficiency maximum and plotted in the hill chart as shown in Figure 7.

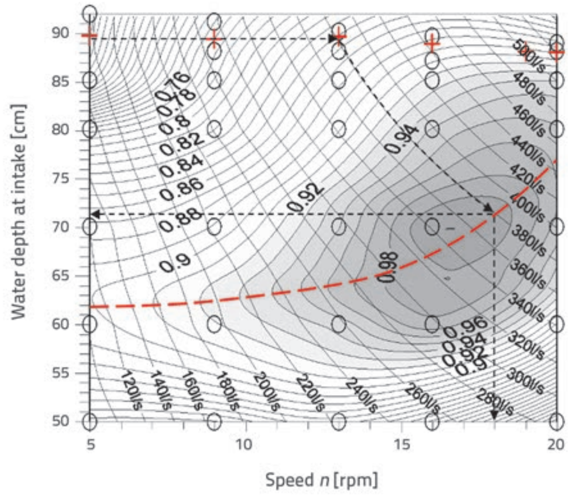

Figure 7. Hill chart with relative efficiency η_{rel} [-] = η / η_{max} and total discharge Q_{tot} [l/s]. The source data is marked with circles, while data points based on absolute hydrometric measurements are marked with crosses

The red dashed line in the hill chart (Figure 7) shows an optimum relationship between the intake depth and turbine speed. It is evident that water levels normally operated at the intake (fully opened inlet gate, $h_1 \cong 90$ cm) induce overfilling of the hydropower screw and a reduction in efficiency. An appropriate reduction of the intake water level or lift of the screw intake (by tilting it up) would decrease the intake depth and would thus enhance the efficiency. For instance, decreasing the intake water depth from approximately 90 to approximately 71 cm, and increasing the speed from 13 to 18 rpm (see black short dashed arrows), enhances efficiency by a factor of approximately 1.17, while preserving the discharge and power (compare with the power chart in Figure 10 further below). The range of water depth from 66 to 72 cm and the speed from 16 to 18 rpm can be considered as the operational optimum. Nevertheless, it should be remembered that the hill chart was based on the data from the relative discharge measurements and, furthermore, that it was determined by use of a bivariate local polynomial interpolation function of the second order, which further approximated the particular data points. Thus, the exact uncertainty of this efficiency data was not estimated.

4.4. Torque

An additional objective of the tests was to measure the torque on the shaft without constructive adaptations, and to learn about mechanical behaviour of the machine. A relative method for measuring torque by means of stick-on strain gauges was developed and successfully implanted. Two simple strain gauges were placed on the console and on the pull bar, which hold the gear-box with motor-generator in position, as shown in Figure 8. The gauges were calibrated exerting a calibrating force F_c on the console (a step-wise filled water tank inducing up to 5.5 kN). Because the calibration loading scheme was different from the operational one, and because the gear-box was equipped with a brake that could not be de-activated during the calibration, two loading states had to be considered – one during the calibration (brake is on) and the second one during the measurement (brake is off and the turbine is running). The recalculation of calibration functions to measurement functions was the crucial part of the method. It should be noted that this method was not a standard method and, furthermore, that an unknown original torque T_0 was always present because of the water that remained in screw pockets, and because of the preloaded torque at the moment the brake was turned on. Nevertheless, implementing the loading model as shown in Figure 8, the original torque could be approximately reduced so that consistent torque data could be obtained for the main operation points and for all relative discharge measurements as described above.

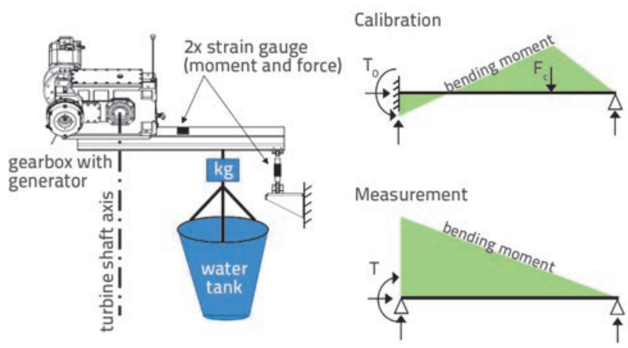

Figure 8. Loading scheme for torque calibration and measurement

Using torque data, the mechanical power, P_m, was determined according to Eq. (2). The generator, mechanical and hydraulic power, and the torque were plotted, as related to speed, on the chart shown in Figure Figure 9, up, for the main operating points where the absolute discharge was measured. Hydraulic efficiencies between 80 and 90 % were obtained based on Eq. (3). However, it should once again be noted that the uncertainty of this result cannot be estimated because some unknown original torque still remains. Therefore only a blind contour hill-chart (without showing absolute values) is plotted in Figure 9, down, showing the mechanical efficiency as computed from the measurement data according to Eq. (4). The distribution of the torque and generator power in the investigated range of speeds and water depths at intake is also presented in the chart shown in Figure 9, down. The tendency of mechanical efficiency is evident – the higher the speed and intake depth (and finally also the discharge), the higher the generator power and mechanical efficiency. The torque increases with the decrease in speed. The crest of hydraulic efficiency would be nearly independent of the speed at the intake depth level of about 60 cm, marked with the dashed line ellipse. The multiplication ($\eta_h \cdot \eta_m$) yields the plant efficiency hill-chart (related to P_{gen}) with an efficiency optimum, as shown above in Figure 7.

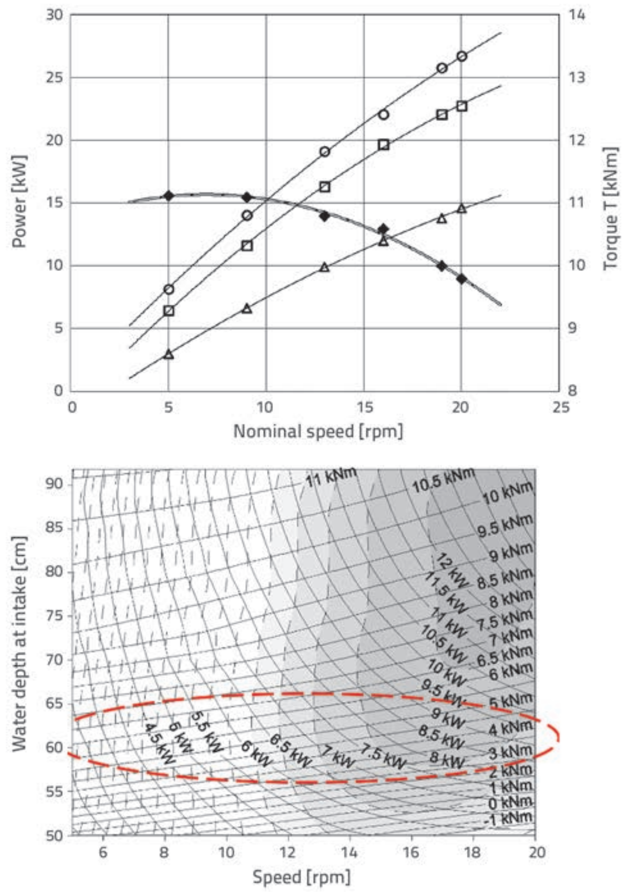

Figure 9. Hydraulic, mechanical and generator power and torque as related to speed (up) and mechanical efficiency blind hill-chart with generator power and torque lines (down). Dashed-line ellipse shows region of maximum hydraulic efficiency

The torque (strain) and other measurement signals were acquired and saved with high data rates (200 Hz) on a computer. An example of measurement in case of n = 16 rpm and stepwise reduced head water depth at intake is given in Figure 10, left. Three vertical lines are plotted in this figure to better show the correspondence between strain gauge signals and the generator power. Zooming into the curve (Figure 10, right), a very good time response can be observed. It shows the rotational frequency due to watering/dewatering of the pump screw, and even a good reproduction of three turbine screws.

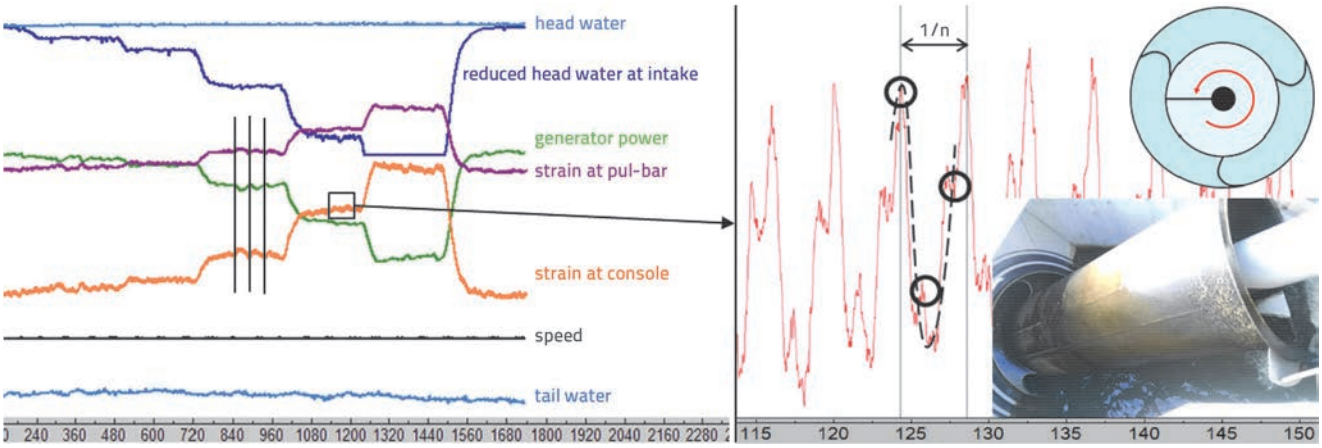

Figure 10. Time curves of measurement signals (left, not to scale) and strain response curve detail showing good reproduction of pump and turbine screws (right)

5. Hydro-energetic evaluation of the first year of continuous operation

After the commissioning in Q1/2015 and some tests with different equipment, the year 2016 was the first year with continuous operation of the screw. Therefore the following data are shown for the period from January 2016 until December 2016, Table 2. The Retznei Hydropower Plant consists of the main turbine (Kaplan) on the right bank, a weir in the middle, and the hydropower screw for fish migration on the left bank.

Table 2. Main parameters Retznei HPP in 2016

Parameter	Unit	Hydropower screw	Kaplan turbine
Work W	MWh	72,93	3896,3
Mean power P (FC)	kW	10,50	467,22
Mean head H	m	5,13	5,13
Cumulated unavailability	h	22,5	-
Maintenance and inspection	h	160	-

The hydropower screw installed at Retznei was exclusively designed for fish migration and not for enlargement of nominal flow of the hydropower station. The unavailability of the hydropower screw is explained by the typically high flows of the Sulm during the floods. Because of the drop of the upstream level and the rise of the downstream level the turbines are to be shut down, see hydrograph in Figure 11.

The company Hydroconnect GmbH is currently developing several projects in Canada and Central Europe. The biggest double rotating screws are currently being planned for heads of approximately 40 m. In this case, 3 screws should be installed in series. They are designed to transport fishes up to 1.5 m in length. The first fish ascent (single rotating) screw will be built in Austria at the end of 2017. Some more photographs from the efficiency measurements are shown in Figure 12.

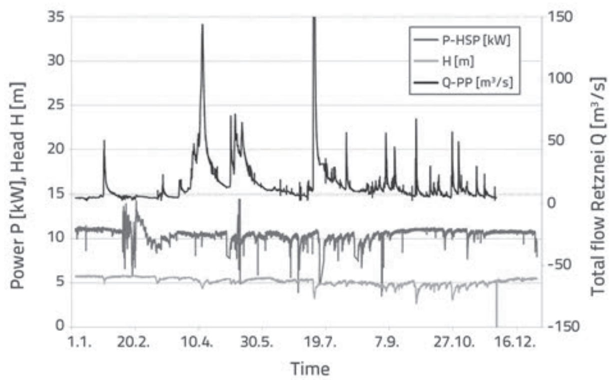

Figure 11. Hydropower screw hydrograph with generator power, head and total flow at the Retznei hydropower site in 2016, without short time unavailability

Figure 12. View of power screw from tail water (top left), view of intake channel towards head water, intake gate is closed (top right), mounting SLD probes and water level meter in intake channel and schematic illustration of the operating mode (bottom left), and filling gauging barrel with pumped water at the end of the fish trough (bottom right)

6. Conclusion

The double rotating screw installed at Retznei is the first bi-directional fish pass of this design, based on the double rotating screw technology. The first two years of operation have shown that this system works really well. Fishes and other water animals can migrate through the system without any excessive effort. It is possible to migrate from the head water to the tail water, and also backwards from the tail water to the head water, without any remarkable risk of injury [1]. Various measurements demonstrate that the double rotating screw is not only a fish pass, but also an energy generating device. In addition, the hydraulic efficiency is quite good. It is possible and profitable to use this system as a small power plant under proper conditions. To maximise the efficiency, it is important to run the double rotating screw with an optimum speed for the rated flow, and also to design an optimum water depth at the intake, as indicated in the measurement results. Torque measurement results suggest a very good hydraulic performance of the double rotating screw (including the hanging flat belt bearing losses), amounting to more than 80 %. There is still a considerable potential for improvement of the mechanic-electrical transition. In total, losses of the generator, frequency converter, power electronics, and the gear mechanism exceed 20 %, mostly at lower speeds. Because of unknown uncertainty in the torque measurement method, further tests should be conducted before making final conclusions about the ratio between the mechanical and hydraulic efficiencies.

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