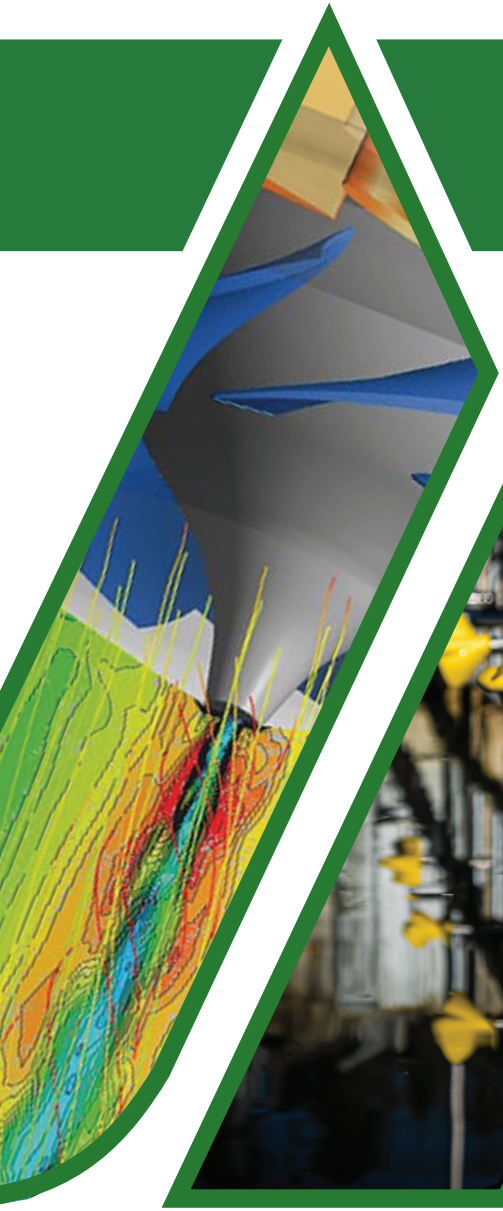
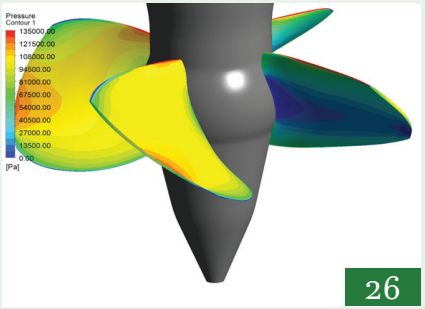
Prof. Dr.
Jaberg und Partner GmbH
Technologie und Strategie

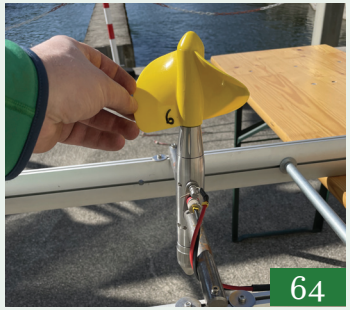


ACTIVITY REPORT 2020 – 2021

ACTIVITY REPORT
2020 – 2021

CONTENT





PROJECTS

Helmut Jaberg
O.Univ.-Prof. em. Dipl.-Ing. Dr.techn.

CEO Prof. Dr. Jaberg & Partner GmbH

Head of the Institute of Hydraulic Fluid Machinery (HFM) until 2020
Graz University of Technology

EDITORIAL

Dear Ladies and Sirs,

our new activity report for the years 2020 and 2021 gives you and your colleagues an overview of the latest developments at Prof. Jaberg und Partner GmbH - Technology - Strategy - Engineering Services. This summer 2022 we are already celebrating our 10 year anniversary and will be moving into new and larger premises.

Founded at the turn of the year 2011/2012 as a GmbH and “one-man-show”, the engineering consultancy has now grown to 10 employees. During this time, our business activities have expanded homogeneously from the Graz region and Austria, first to the German-speaking area, the famous DACH region, and then to Europe and now - almost unbelievably - also to Asia and “the Americas”, i.e. actually worldwide.

This small 10-year anniversary is therefore also a particularly good opportunity to say “Thank you very much!” to all our customers for their repeatedly confirmed trust in our reliability and the performance of our team.

And here we come to the Key Factor of Success: the employees and their knowledge and skills! “Many thanks to all the team members!” who have made this success story possible. All of our engineers learned their knowledge and skills from scratch at the Institute for Hydraulic Fluid Machinery at the Graz University of Technology, which I headed for over 25 years, and then honed and drastically expanded this knowledge in many years of research work. Successively, these top engineers moved to Prof. Jaberg and Partner after their TU activities and have recently also become co-partners. This alone shows that the success story of Jaberg und Partner will continue in the future: Top commitment and top know-how in the interest of our clients will continue to pave this way.

With this activity report we want to substantiate this claim and prove that we have an absolutely unique selling point in the segment of hydraulic fluid machinery, turbines, pumps, valves and plants - from the machines to the entire system, from inlet and outlet areas to the pressure surge - all of which of course is only possible as a team: no other engineering consultancy covers such a range of services from numerical calculations to plant measurements with all IEC measurement methods to the experimental IEC verification of our designs, whereby nota bene we prefer to outsource the performance of IEC acceptance tests to my former institute at Graz University of Technology.

The range of services thus extends without exaggeration from precision engineering to the gigawatt range - always on an engineering-scientific basis and always with implementation in mind and in the service of the global hydropower industry, the pump and turbine industry as well as water supply, process and thermal power plant technology.

Our range of services is rounded off by the Academy, under which we have combined the distance learning course “Pump Engineer” and the practitioner conferences “Pumps in Process and Power Plant Engineering” and “Hydropower, Turbines, Systems”. The Pump Conference in particular has established itself over 25 years as the technically most important and largest annual event in the German-speaking world, with always 100 to 150 participants.

The distant learning programme Pump Engineer I have been leading for almost 20 years, since 2012 it is also offered on a global scale. This contribution to lifelong learning is also literally unique worldwide. Soon, the title M.Eng. will be awarded with this training and in cooperation with an international academic educational institution.

We hope to have aroused your interest and remain

with best regards from Graz

Yours

Helmut Jaberg

Sehr geehrte Damen und Herren!

Unser neuer Tätigkeitsbericht für die Jahre 2020 und 2021 gibt Ihnen und Ihren Kolleginnen und Kollegen einen Überblick über die jüngsten Entwicklungen der Prof. Jaberg und Partner GmbH Technologie – Strategie – Engineering Services. In diesem Jahr begehen wir bereits das 10-jährige Bestehen als GmbH und beziehen in diesem Sommer 2022 neue und größere Räumlichkeiten.

Zum Jahreswechsel 2011/2012 als GmbH und „one-man-show“ gegründet ist die Ingenieur-Beratung auf jetzt 10 Mitarbeiterinnen und Mitarbeiter gewachsen. Unsere Geschäftstätigkeit hat sich in dieser Zeit und stets homogen von der Region Graz und Österreich zuerst auf den deutschsprachigen Bereich ausgedehnt, die berühmte DACH-Region, und im Anschluss auf Europa und mittlerweile – fast unglaublich - auch auf Asien und „the Americas“, also tatsächlich weltweit.

Dieses 10-Jahres-Jubiläum ist damit auch eine besonders gute Gelegenheit „Herzlichen Dank!“ an alle unsere Kunden zu sagen für das immer wieder bestätigte Vertrauen in das Leistungsvermögen des Teams und unsere Zuverlässigkeit.

Und hier sind wir beim Key Factor of Success: Die Mitarbeiterinnen und Mitarbeiter und deren Wissen und Können! „Herzlichen Dank!“ daher auch an alle unsere Teammitglieder, die diese Erfolgsgeschichte erst möglich gemacht haben.

Alle unsere Ingenieure haben ihr Wissen und Können zuvor am Institut für Hydraulische Strömungsmaschinen der Technischen Universität Graz, dem ich über 25 Jahre vorgestanden habe, von der Pike auf gelernt und danach in langjähriger Forschungsarbeit dieses Wissen geschärft und immens ausgeweitet. Sukzessive sind diese Top-Ingenieure nach ihrer TU-Tätigkeit zu Prof. Jaberg und Partner gewechselt und seit neuestem auch Mitgesellschafter geworden. Allein schon daran ist zu sehen, dass die Erfolgsgeschichte von Jaberg und Partner auch in Zukunft weitergehen wird: Top-Engagement und Top-Know-How im Interesse unserer Kunden werden auch weiterhin diesen Weg ebnen.

Mit diesem Tätigkeitsbericht wollen wir den Anspruch untermauern und belegen, ein absolutes Alleinstellungsmerkmal zu besitzen im Segment der Hydraulischen Strömungsmaschinen, Turbinen, Pumpen, Armaturen und Anlagen - und zwar von den Maschinen bis zum Gesamtsystem, von Ein- und Auslaufbereichen bis zum Druckstoß, all dies ist natürlich nur im Team möglich: Keine andere Ingenieurberatung deckt eine solche Leistungsbreite ab von Numerischen Berechnungen über die Anlagenmessung mit allen IEC-Messverfahren bis zur experimentellen IEC-Überprüfung unserer Auslegungen, wobei wir nota bene die Durchführung der IEC-Abnahmen bevorzugt an mein früheres Institut der TU Graz vergeben.

Die Leistungsbreite reicht also ohne Übertreibung von der Feinmechanik bis zum Gigawattbereich – immer auf ingenieur-wissenschaftlicher Basis und immer mit der Umsetzung im Blick und im Dienst der weltweiten Wasserkraftbranche, der Pumpen- und Turbinenindustrie sowie der Wasserversorgung, der Verfahrens- und der thermischen Kraftwerkstechnik.

Abgerundet wird unsere Leistungsbreite durch die Akademie, unter welchem Begriff wir das Fernstudium Pumpenfachingenieur und die Praktikerkonferenzen „Pumpen in der Verfahrens- und Kraftwerkstechnik“ und „Wasserkraft, Turbinen, Systeme“ zusammengefasst haben. Gerade die Pumpenkonferenz hat sich über 25 Jahre als technisch bedeutendste und größte jährlich stattfindende Veranstaltung im deutschen Sprachraum mit stets 100 bis 150 Teilnehmenden etabliert.

Das Fernstudium zum Pumpenfachingenieur leite ich auch schon seit fast 20 Jahren, die Ausbildung wird seit 2012 ebenfalls im weltweiten Maßstab angeboten. Auch dieser Beitrag zum lebenslangen Lernen ist im wortwörtlichen Sinne einzigartig. Demnächst soll mit dieser Ausbildung und in Zusammenarbeit mit einer internationalen akademischen Bildungsstätte der Titel M.Eng. vergeben werden.

Wir hoffen, Ihr Interesse geweckt zu haben und verbleiben

mit besten Grüßen aus Graz

Ihr

Helmut Jaberg

Prof. Dr.
Jaberg und Partner GmbH
Technologie und Strategie

ABOUT US

We offer high performance numerical simulation for multidimensional calculations and – as of our tight interlocking with research and educational institutions – extraordinary experimental and metrological competence and experience, especially test rigs in accordance to the IEC/ISO standard and highly performant and exact measurement technology for plant measurements.

With our expertise and knowledge in “Numerical Simulations / 3D - CFD” we support our partners (manufacturers, operators, planners and authorities) in the field of fluid mechanics in general – and for pumps and turbines in particular. Our well-founded Know-how gained over decades of extensive experience is based on numerical flow simulation (CFD) as well as on experimental experience.

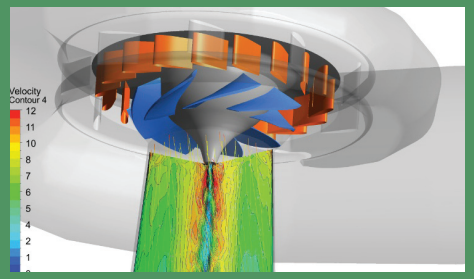
We subject our designs to experimental verification in accordance with IEC 60193 and preferably rely on the IEC test rigs of the Institute for Hydraulic Fluid Machinery of the Graz University of Technology, which Prof. Jaberg headed for over 25 years until December 2020. Prof. Meusburger has been his successor at the institute since 2020.

In our tailor-made trainings in all fields of hydraulic fluid machinery, we provide user-oriented and profound knowledge. These programmes are founded on our well-grounded Know-how and our expertise.

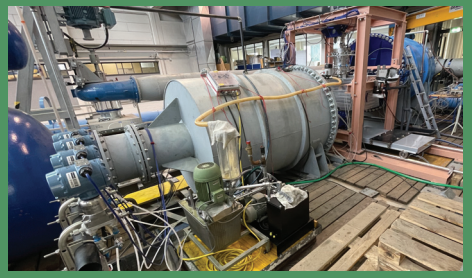
We are independent and work as a team with our partners and customers from industry, research and science. With our expertise, we cover the entire value-added process – from market requirements to implementation.

Numerical Simulation

Analysis, design and optimization of hydraulic fluid machinery, components and systems
Water hammer and pressure surge as well as transient conditions of fluid and gas flows
Fluid structure-interaction



Model Test



Approval test acc. to
IEC 60193, ISO 9906, IEC 60534
Setups for long-term tests- System, operation and life-cycle tests
Comparison of experimental data with numerical simulation results (CFD)

On-Site Measurement Technology

Measurements of industrial plants and power plants acc. to IEC 60041 and IEC 62006.

Thermodynamic efficiency measurement,
Acoustic Doppler, etc.
Sound measurement
Discharge measurement
Vibration measurement

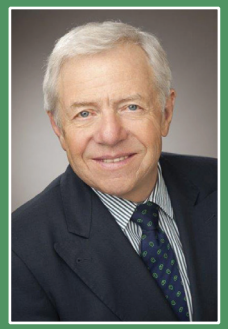
Assessments & Consulting

Our profound know-how, experience and well-established numerical as well as experimental methodology ensure the quality of our work on innovations and independent expertise.

- *Feasibility studies*
- *Modernisation of power plants and systems*
- *Product development potential*
- *Support with regard to Environmental Impact Assessments*
- *Damage events*

PEOPLE

Management



Helmut Jaberg
O.Univ.-Prof. em. Dipl.-Ing. Dr.techn.
CEO

Studied aerospace engineering in Stuttgart, Southampton and Munich and worked at MTU München GmbH and KSB AG, among others as development director of a German-French business unit and director of a business unit. Management trainings at INSEAD and MZ St. Gallen. From 1995 until 2020 he was full professor and head of the Institute of Hydraulic Fluid Machinery (HFM) at the Graz University of Technology. He created the Master of Engineering (MEng) Hydropower and was the academic director of this extra-occupational University Programme. He is also managing director of the Pumpenfachingenieur GmbH and organiser of the Practitioners‘ Conference Hydropower and the Practitioners‘ Conference Pumps. Convener of a CEN working group and lifting plants and deputy chairman of the DIN mirrow committee.

Engineers

Christian Bodner
Dipl.-Ing.

Areas of focus:

- Design and optimisation of hydraulic machines
- CFD simulation of hydraulic machines and plants
- Refurbishment of hydropower plants
- Test rig and laboratory measurements, IEC acceptance tests of hydraulic machines
- On-site measurements according to IEC

Stefan Höller-Litzlhammer
Dipl.-Ing.

Areas of focus:

- Design and optimisation of hydraulic machines
- CFD simulation of hydraulic machines and plants
- Refurbishment of hydropower plants
- Pressure surge calculations and transient processes
- Piping systems

Jürgen Schiffer
Dipl.-Ing. Dr.techn.

Areas of focus:

- Design and optimisation of hydraulic machines
- CFD simulation of hydraulic machines and plants
- Refurbishment of hydropower plants
- Displacement pumps
- Training and lecturing activities

Engineers

Bernhard Streitberger
BSc.

Areas of focus:

- Mechanical design
- Stress and strain calculation (FEM)

Lukas Zapf
Dipl.-Ing.

Areas of focus:

- CFD simulation of hydraulic machines and plants
- Piping systems

Administration

Margot Jaberg
Mag. soc. oec.

Areas of focus:

- Controlling
- Accounting
- Organisation and marketing of the practitioners' conference „Hydropower / Turbines / Systems“

Andreas Stachel
MSc

Areas of focus:

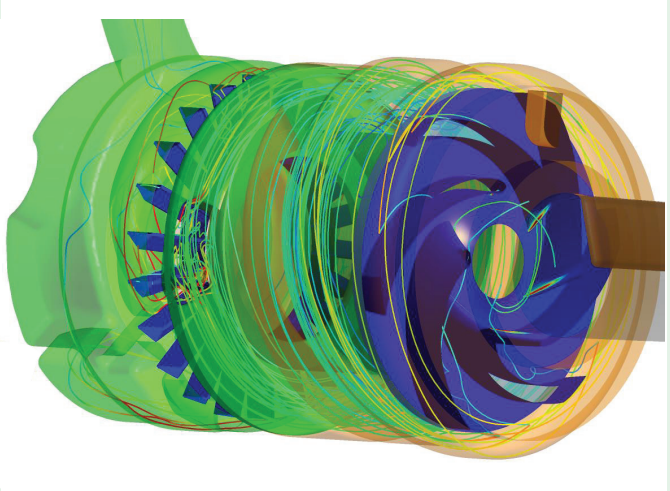
- Marketing

Karin Hermann
Mag. phil.

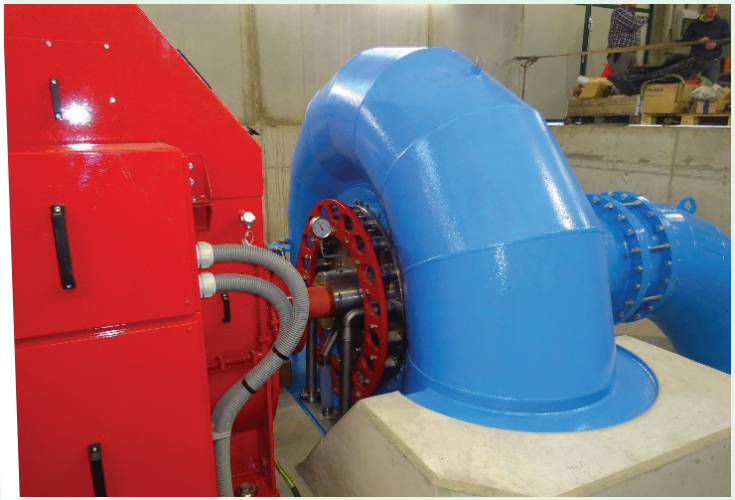
Areas of focus:

- Organisation and marketing of the practitioners' conference „Pumps in process engineering”
- Course-Management Pump Engineer

PROJECTS



Visualization of CFD-results



Turbine in a hydropower plant

Turbine test rig

Hydraulic Development of a Double Regulated Mixed Flow Turbine for Small Hydro Applications

The double regulated mixed flow turbine was initially developed by P. Deriaz in the 1950ies to be used as a pump turbine. The new concept with a diagonal flow path has allowed the inventor to utilise the idea of adjustable guide vanes and adjustable runner blades for the medium head operation range. This is of particular value in case of significant head variations, which are characteristic for medium head pumped storage installations. Furthermore, the additional runner blade adjustability has improved the unit performance in turbine and pump mode.

Since its first technical application known from the Niagara Falls hydropower plant, the Deriaz-type turbine has become more frequent in the subsequent decades. However, until now, only a few applications solely for the turbine operation are known, although the Deriaz-turbine can close a gap between the conventional Kaplan- and Francis-turbines – solving at the same time the problem of limited adjustability of Francis turbines. Double-regulated mixed flow turbines offer vital benefits, especially in the medium head range with H = 20 – 100 m and in the case of high flow variations. The advantage of Deriaz turbines compared to Francis turbines is the high efficiency level over a comparatively wide range of head and discharge and an extended region with limited pressure pulsations. However, these indisputable advantages are offset by the slightly higher susceptibility of cavitation and the complexity of the runner blades' adjustment mechanism.

The Deriaz-type turbine concept is especially suitable for small hydropower plants with medium head and comparably high discharge variations. Therefore, it was considered for a new power plant in St. Johann, Austria, where it has to cope with strongly variable discharge and needs to be operated all year. Thus, a turbine concept with excellent part-load behaviour was requested. Instead of using two rather costly small Francis turbines, the concept of a double-regulated mixed flow turbine was chosen as it offers the best economic operation with strongly variable discharge over the year.

According to the given database, the new turbine was designed for a net head of H_{Net} = 37 m and a maximum rated discharge of Q_{rated} = 1.5 m³/s. Furthermore, it shall be even capable of operating up to Q_{Max} = 1.7 m³/s for future applications. To cope with the intended suction head of H_S = + 2 m (tailwater level below reference axis of the turbine) at full load operation, the critical cavitation coefficient of the turbine needs to be below σ = 0.19 [-] (see Eq.1).

$$\sigma_{Plant} = \frac{10 - \frac{Altitude\ z}{900} - H_S}{H_{Net}} = 0.19$$

Eq.1

It has to be pointed out that the lack of data regarding the cavitation behaviour of Deriaz-turbines impedes the selection of the overall turbine dimensions. However, Hironaka et al. published a hydraulic study of Deriaz- and pump-turbines based on comprehensive model test data. According to the relation of the critical cavitation coefficient and the specific speed presented in this study, a specific speed of $n_{q,BEP}$ = 60 rpm, was chosen to achieve the cavitation target. To satisfy the equation of the specific speed (see Eq. 2), the turbine rotational speed was finally fixed with n = 750 rpm.

$$n_{q,BEP} = n \cdot \frac{\sqrt{Q_{BEP}/Q_{Ref}}}{\left(H_{Net}/H_{Ref}\right)^{0.75}} = 750 \cdot \frac{\sqrt{1.45/1}}{(37/1)^{0.75}} = 60\,rpm$$

Eq.2

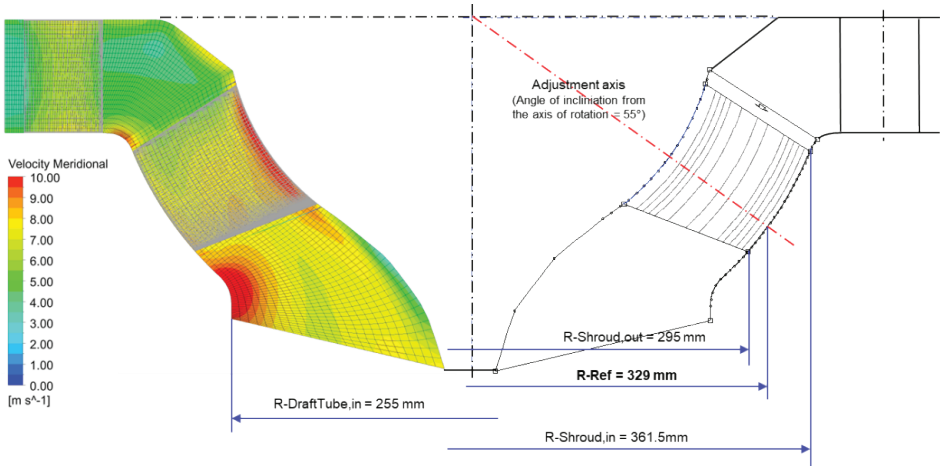


Fig. 1: Meridional section of the turbine (left: with the distribution of meridional velocity calculated utilising CFD / right: with overall runner dimensions)

Using the design recommendations given by Hironaka et al., an initial meridional section and a basic overall layout of the turbine were created. Furthermore, a CFD-based optimisation of the turbine design was performed. A turbine setup with 18 guide vanes and 10 runner blades was used for the first CFD simulations. Due to the lack of space for the adjustment mechanism, the number of runner blades had to be reduced to zRu = 8 in the design process. However, the reduction of the number of runner blades increases the blade loading, and it has an unfavourable impact on the cavitation performance, which made the optimisation of the runner even more difficult.

Apart from achieving an excellent efficiency level and acceptable cavitation characteristics, the particular challenge was creating a runner design that integrates the runner blades' adjustment mechanism. The relatively small diameter of the spherical hub contour ($D_{Hub,in}$ = 520 mm) hardly offers enough space for the pivot bearings and adjustment levers required to operate the highly loaded blades. Thus, it was necessary to maximise the hub space while keeping the runner blades' adjustment torque at the lowest possible level.

As there are many geometrical parameters to consider, the optimisation process of a Deriaz-turbine strongly differs from the one of Kaplan- or Francis-turbines. Furthermore, fewer design recommendations are available, making finding an ideal turbine design even more challenging. After many adaptations of the overall turbine dimensions and the shape of the runner blades, a quite compact runner design with a reference diameter of D_{Ref} = 658 mm was found. The meridional section of the final design is presented in Figure 1. While the left side of the picture shows the distribution of the meridional velocity calculated utilising CFD at the Best Efficiency Point (BEP), the right side presents the overall dimensions of the runner.

Referring to the final turbine design and a net head of H = 36.5 m, Figure 2 presents the hydraulic turbine efficiency (orange curve) and the cavitation coefficient (green curve) calculated using CFD. The intersection of the σ-curve of the plant and the turbine highlights the cavitation limit found at Q » 1.65 [m³/s]. Therefore, according to our experience with applying the so-called "σ-Histogram method" to evaluate the cavitation coefficient, a turbine operation up to the guaranteed maximum discharge of Q = 1.5 m³/s will be possible without any negative impact of cavitation on the turbine performance.

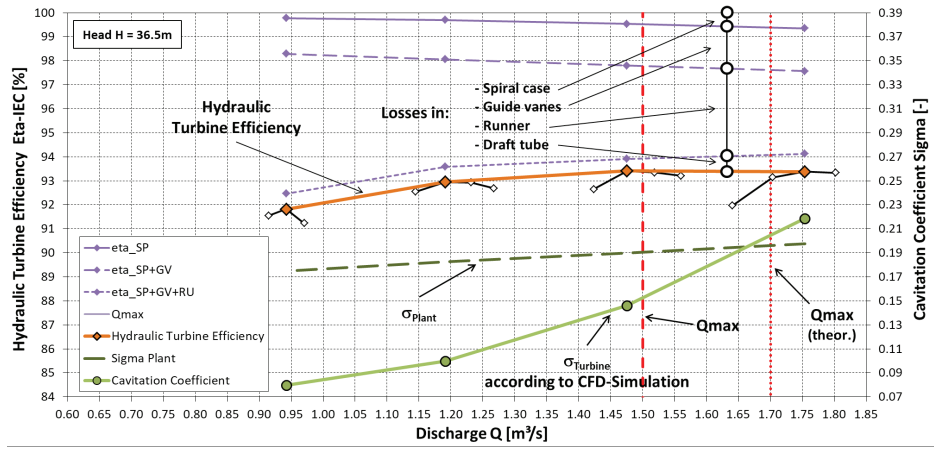
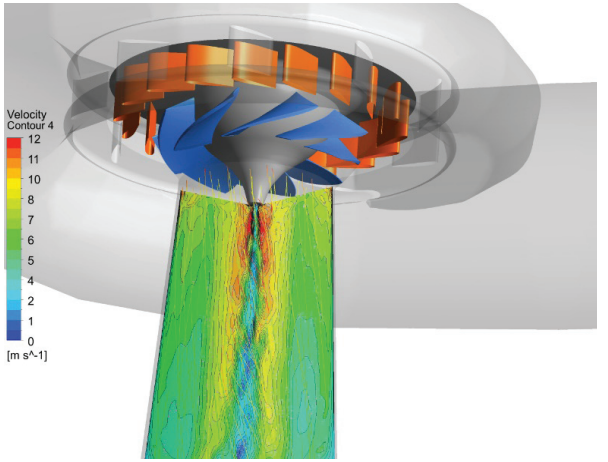
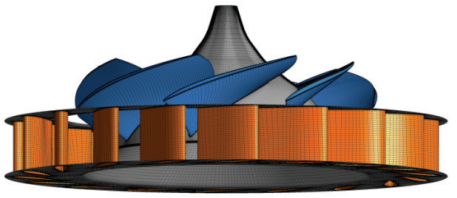


Fig. 2: Hydraulic turbine efficiency and cavitation coefficient calculated using CFD at H = 36.5 [m]





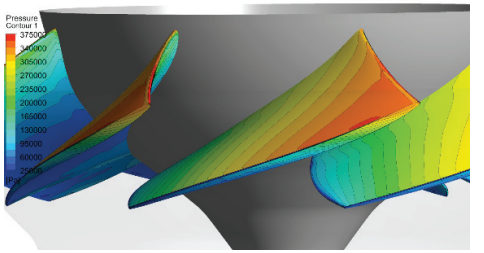


Fig. 3:
Visualization of CFD-results and CFD-meshes at the Best Efficiency Point of the final turbine design

According to the CFD simulation, the hydraulic peak efficiency of the turbine accounts for 93.5%. The loss composition also presented in Figure 2 shows, that the hydraulic losses in the draft tube account for only one percentage point over the whole investigated range of operation. This can be explained by the fact that the remaining swirl at the outlet of the runner can be kept close to Zero thanks to the adjustability of the runner blades. Compared to a conventional Francis turbine, this finding yields a relatively flat efficiency curve. It allows for highly efficient part-load operation, which underlines one of the benefits of the Deriaz-type turbine.

Referring to the final turbine design and the Best Efficiency Point Figure 3 presents a visualization of the CFD-results and the meshes required for the simulation. Furthermore, Figure 4 shows photographs of the turbine in the power plant.

Due to the tight schedule, it was necessary to carry out the CFD-based optimisation and the mechanical design process in parallel while managing the permanent exchange of information. As a result, within a relatively short development time of just 2.5 months, the hydraulic design of the double-regulated mixed flow turbine was completed. Only seven months later, at the end of 2020, the turbine was successfully put into operation. In order to provide proof of performance, a plant measurement was performed after a few months of operation. The comparison of on-site measurement data with the numerical simulation results showed that the prototype efficiency exceeded the predicted one over a wide range of operations. Furthermore, since the turbine was set into operation, it runs without any troubles related to noise emission or vibration to the customer's satisfaction.

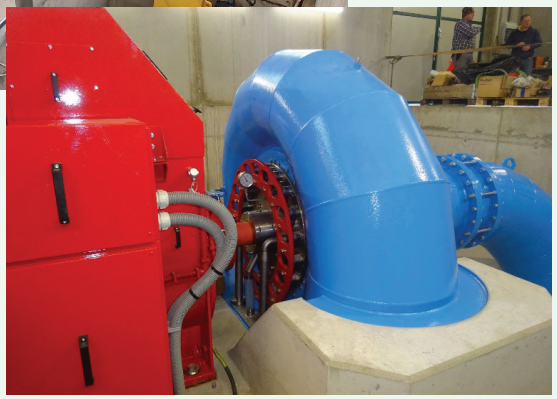


Fig. 4:
Photographs of the turbine in the power plant

Homologous Model and Acceptance Test of the Gratkorn Kaplan-Turbine

The homologous model test and acceptance test of the Kaplan-turbine developed by J&P-Team for a local hydropower plant took place at the end of 2021. The model test was carried out in cooperation with the Institute for Hydraulic Fluid Machinery (Graz University of Technology). The construction of the model machine for the institute's test rig was carried out by the J&P-team in close coordination with the HFM-team. After successful commissioning of the turbine model, extensive measurements were carried out.

The model test included the following measurements:

- Hill chart measurement (friction torque measurements by hydrostatic bearing - on the fly)
- Cavitation measurements (cavitation hill chart) including visualisation (image, video)
- Force measurement in the entire hill chart range as well as during the cavitation measurements
- Winter-Kennedy measurements during all measurement campaigns
- Torque measurement of 6 different guide vanes
- Torque measurement of one runner blade
- Throughput speeds (entire map as well as on-cam and of-cam operation points)

The homologous setup of the model machine at the HFM test rig is shown in Figure 1 and Figure 2.

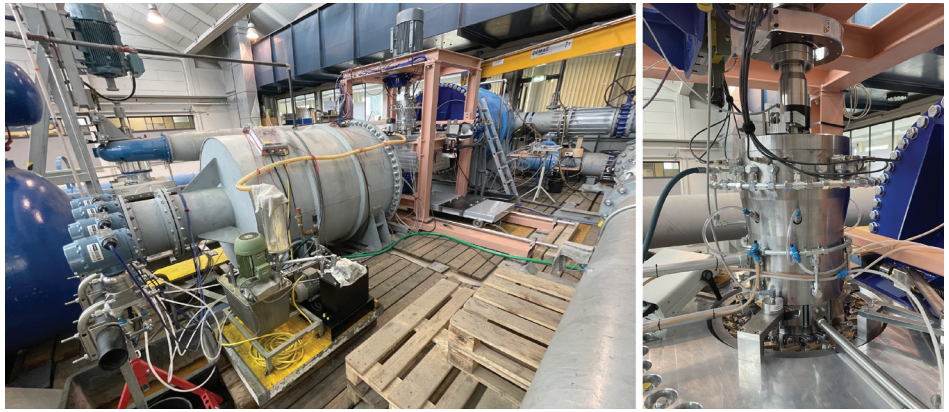


Fig. 1: Overview of the turbine test-rig (left); hydrostatic bearing (right)

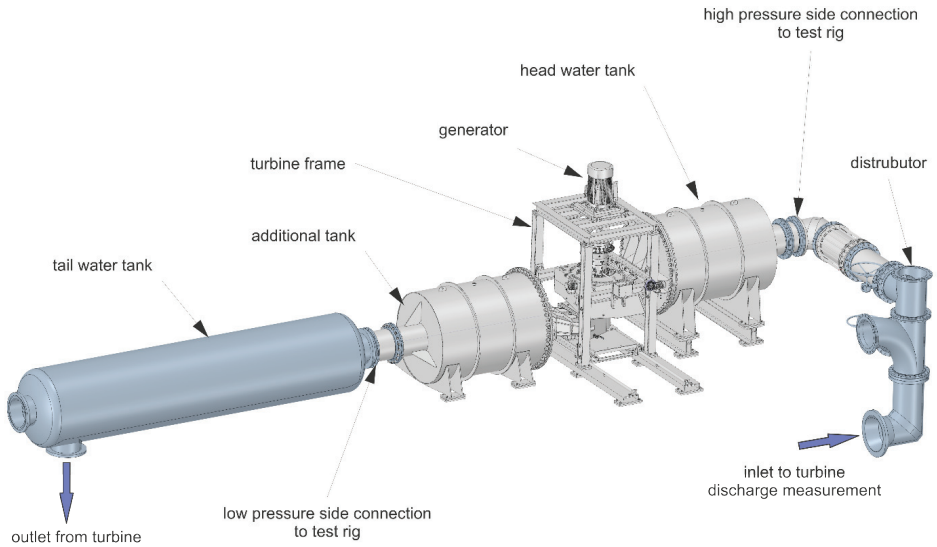


Fig. 2: 3-D Modell of the Kaplan-turbine test rig

Fig. 3: Calibration of the axial Force (top), guide vane torque (middle), runner blade torque (bottom)

A hydrostatic bearing (developed by the HFM-team) was used to measure the friction torque and the axial force on the model. This hydrostatic bearing allows the friction torque of the model to be measured under load during operation. In addition to measuring the friction torque, the hydrostatic bearing also allows the axial and radial thrust of the impeller to be measured. The design of the hydrostatic bearing can be seen in Figure 1 (right).

One of the key elements, apart from the measurement of all hydraulic characteristics and the behaviour during turbine runaway, was to measure the axial force and the torques of the guide vane and the runner blade in the entire hill chart range as well as during turbine runaway (on- and off-cam operation). For measuring the guide vane torque, 6 of the guide vanes were equipped with strain gauges. Likewise, one of the runner blades was equipped with strain gauges for determining the hydraulic runner blade torque. The transmission of the measurement signal of the runner from the rotating system was carried out by means of telemetry, which was housed in the specially constructed turbine shaft.

On the one hand, the complex measurements were used for dimensioning of the machine components such as the turbine bearings, the guide vanes and the runner blade adjusting mechanism including the necessary torques. In addition to the dimensioning, the measurements also served to verify the numerical calculation model of the turbine.

Figure 3 shows the calibration of the axial force measurement (top), the guide vane torque (middle) and the runner blade torque (bottom). Since the calibration is always carried out with the entire measurement chain, the runner blade torque had to be calibrated with the dismounted hydrostatic bearing, which contained the special turbine shaft (telemetry).

Figure 4 shows an example of the determined hydraulic runner blade torque from turbine operation to pump operation of the turbine at a prototype head of H=7m and different runner blade positions.

The acceptance test of the turbine was carried out in the presence of the client and the operator of the turbine. The acceptance test fully confirmed the numerical design which was carried out previously. All requirements and specification for the turbine design were fully confirmed by the model test.

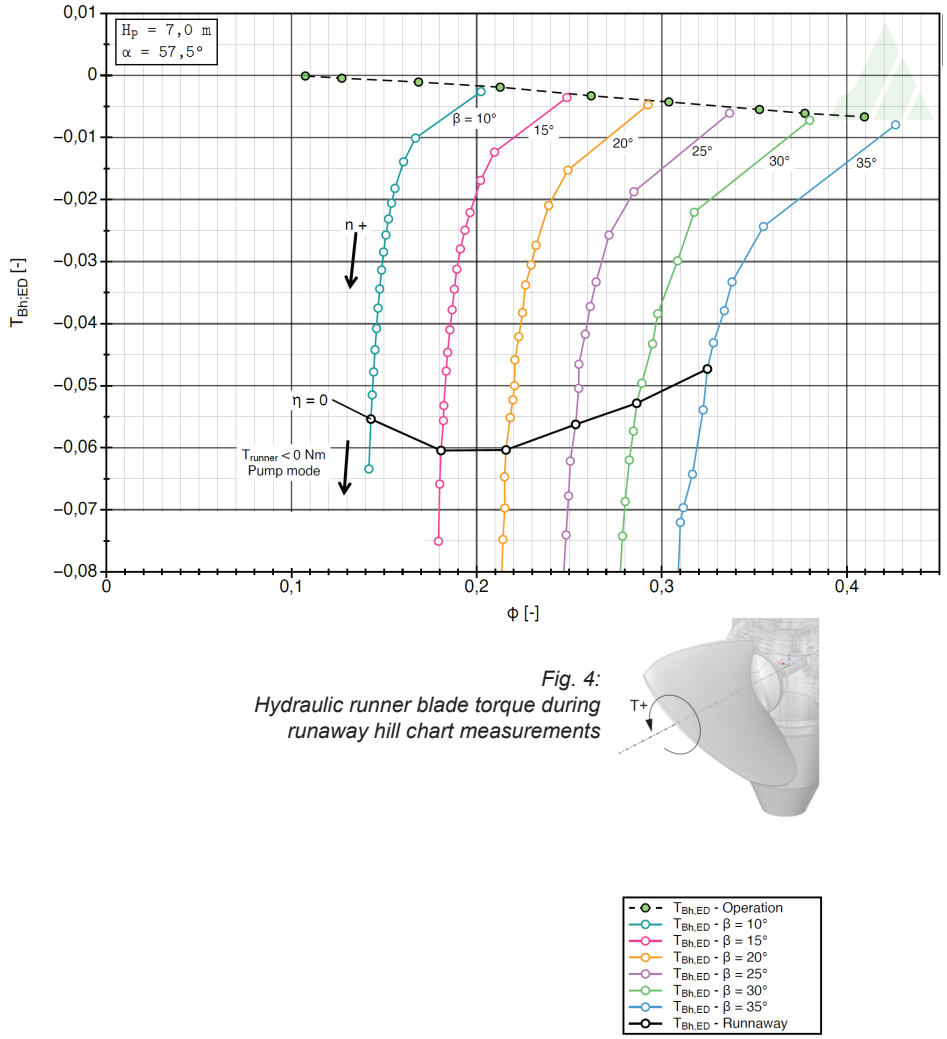


Fig. 4:
Hydraulic runner blade torque during runaway hill chart measurements

Impressive Performance Enhancement of an Austrian Small Hydro Power Plant

Around 15 years ago, two identical vertical Kaplan-Turbines were taken into operation after the completion of extensive rehabilitation works at a historical power plant site at the river Ybbs in Austria. The turbines were equipped with a semi-spiral case made of concrete, adjustable guide vanes and an elbow-type draft tube. The diameter of the 4-blade Kaplan runner was chosen with D = 1995 mm and the turbine speed accounts for n = 163 rpm. At a net turbine head of H = 4.5 m and a maximum discharge of Q = 17 m³/s a maximum power of P = 680 kW was achieved. As comparison: The maximum power of the original turbine configuration built in the year 1908 was only around 300 kW.

Some more years later, in 2017, an underwater deepening associated with an increase of the turbine head was undertaken. To enable this measure, the discharge- and outlet-part of the draft tube needed to be adapted (see Figure 1). As the generators were initially designed for a maximum power of 750 kW, the maximum discharge was consequently reduced to limit the power output of the turbines. In total, the net head of the turbines was increased by around 50%. Thus, it can be assumed, that – beginning with 2017 – the turbines were operated far off the ideal operation range in the hill chart. However, the limitation of the maximum discharge at least had a positive impact on the security against cavitation.

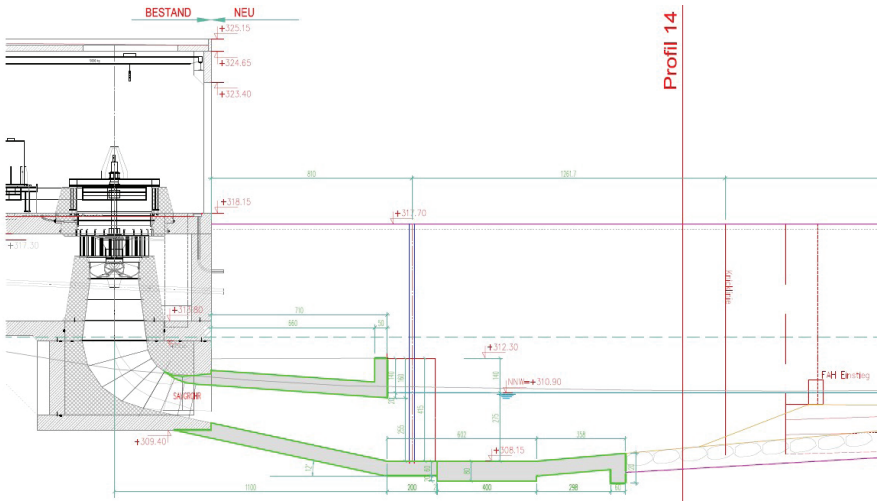


Fig. 1: Cross section oft he vertical Kaplan-turbine located at the river of Ybbs/Austria

A few years later, the operator had the idea, to replace the generator as well as the runner of one of the turbines by a completely new design, in order to utilize the original amount of water, to increase the bottleneck-output and to obtain a new feed-in-tariff. For this purpose, a speed adjustment also had to be made. Thus, the turbine speed was increased from n = 163 rpm to n = 176.5 rpm while the runner diameter remained unaltered. Associated with that, the specific speed of the Kaplan-turbine strongly changed (see equations below).

$$n_{q,Orig(2007)} = 163 \cdot \frac{\sqrt{0.78 \cdot 17/1}}{(4.5/1)^{0.75}} = 192.2 \ U/min$$

Eq.1

$$n_{q,New(2020)} = 176.5 \cdot \frac{\sqrt{0.75 \cdot 17.7/1}}{(6.7/1)^{0.75}} = 154.4 \ U/min$$

Eq.2

Fig. 2:
Assembly of the new runner

Taking into account all the measures mentioned, it was the target to increase the maximum power output of one single turbine to P > 1 MW. Compared to the initial situation back in 2007, this means a 50% increase of the turbine's power output.

The renewed change of the operation parameters did not only require a new turbine speed, but also a completely new design of the runner blades. For this purpose, a numerical simulation of flow in the turbine was performed by using the commercial software package ANSYS-CFX. The big challenges for the design of new runner blades were to adapt the design to the flow conditions in the existing draft tube and to achieve the highest possible security against cavitation over the entire range of operation.

Although solely stationary, single-phase simulations were carried out and the phase transition from liquid to gas is therefore not simulated, it is possible to use statistical methods to calculate the cavitation coefficient σ. In order to validate the cavitation performance, we use the so called „ σ Histogram method", which was cross-checked several times with experimental data gained from the model test rig. After several design iterations of the runner, it turned out that no serious problems with cavitation can be expected up to the maximum discharge of Q = 17 m³/s. Moreover, for a short period of time the new turbine could be operated even up to Q = 20 m³/s.

In this way, a new Kaplan-runner enabling an efficient and safe operation over the entire operation range of the single turbine, was developed within just a few weeks. In order to achieve this, the runner blades were additionally equipped with cavitation lips, which are usually only used for Kaplan-turbines working under a net head of above approx. 20 m.

Since the fall of 2021, the optimized turbine has already been successfully in operation. Figure 2 shows a photo taken in course of the assembly of the new runner. The visualizations shown in Figure 3 give an impression of the results of the CFD-simulation.

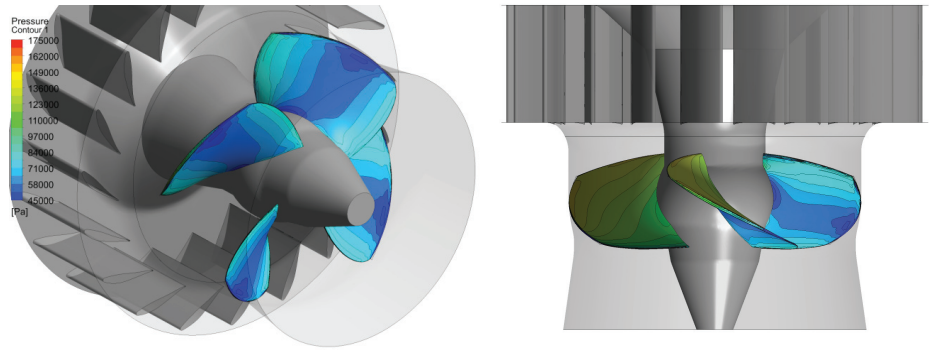


Fig. 3: Pressure contour plots in the runner of the optimized Kaplan-turbine at H = 6.5 m and Q = 18 m³/s

Optimisation of a Pelton Distribution Pipe

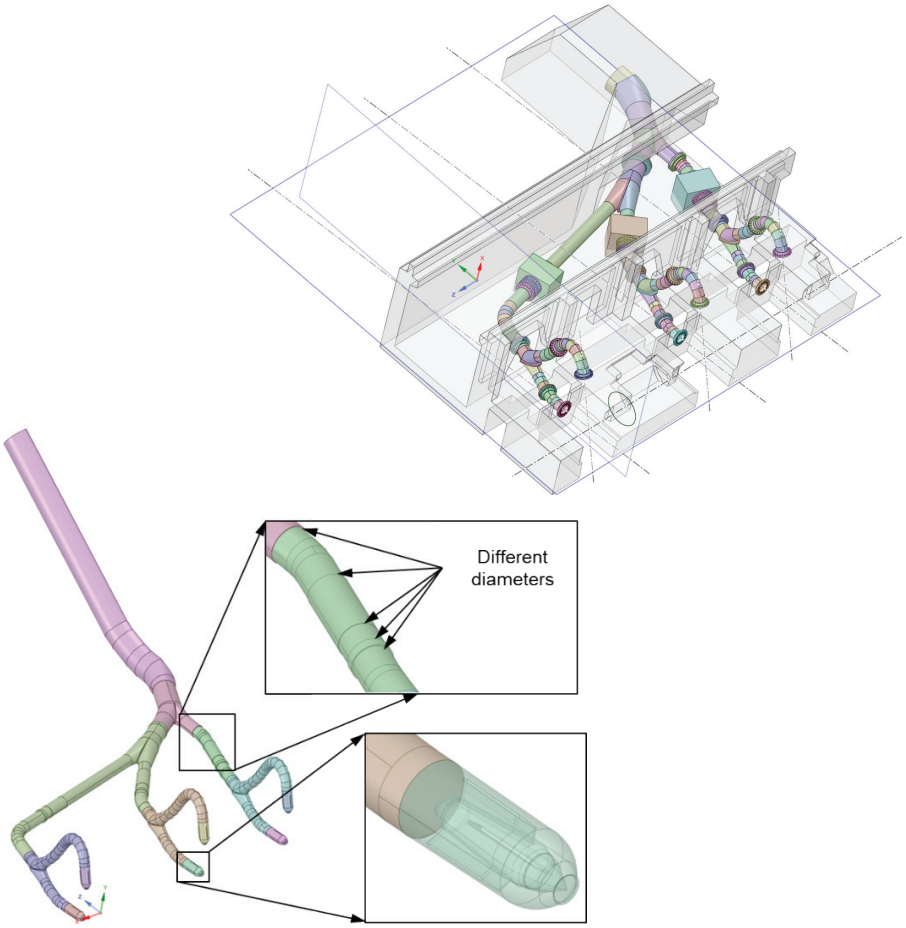


Fig. 1: Overview of the 3D-half-model of the powerplant with the new piping (above), fluid volume of the distribution pipe including the turbine nozzles (half model)

The Jaberg and Partner team was commissioned to carry out numerical calculations and optimisations of the new distribution pipes for the three tandem 2-nozzle Pelton turbines at a hydropowerplant in Switzerland in the Val d'Anniviers. It was commissioned in 1958 and turbines the water from further up the valley. The three existing machines of this were overhauled between 2020 and 2022. With a total capacity of 168 MW, the power plant complex is the fifth largest hydroelectric plant in Valais. Figure 1 shows the 3D-half-model of the powerplant structure with the distribution and the extracted fluid volume including the nozzles.

The aim of the numerical calculations was to evaluate the initial distrubution pipe design. Furthermore extensive optimisation of the initial design were carried out to ensure minimal head loss of the new piping. The main focus beside the head loss reduction was to minimize secondary flow at the nozzle and nozzle outlet to guarentee minimal deviation of the water jet before entering the pelton bucket. The investigations regarding the water jet deviation were carried using complex two-phase numerical calculations. Figure 2 showns the full 2-phase flow numerical setup as well as some of the cross section for flow-field evaluation at the nozzle and at the water jet between dthe nozzle outlet and the turbine runner.

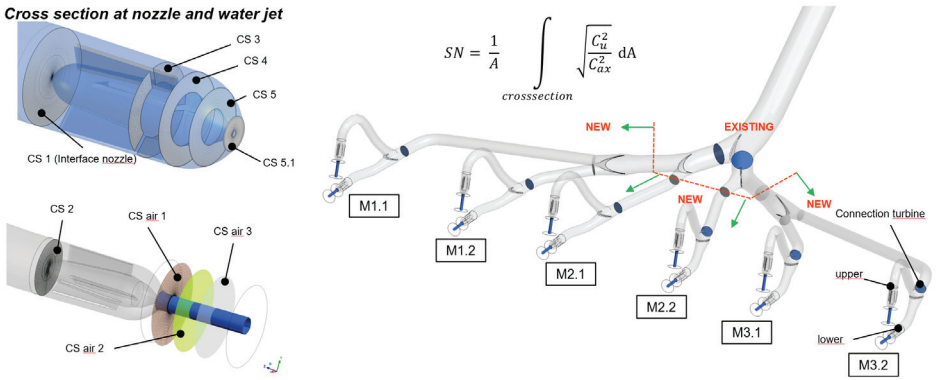


Fig. 2: Overview of the evaluation cross section at the nozzle and at the water jet (left), setup of the full 2-phase-flow numerical model (right)

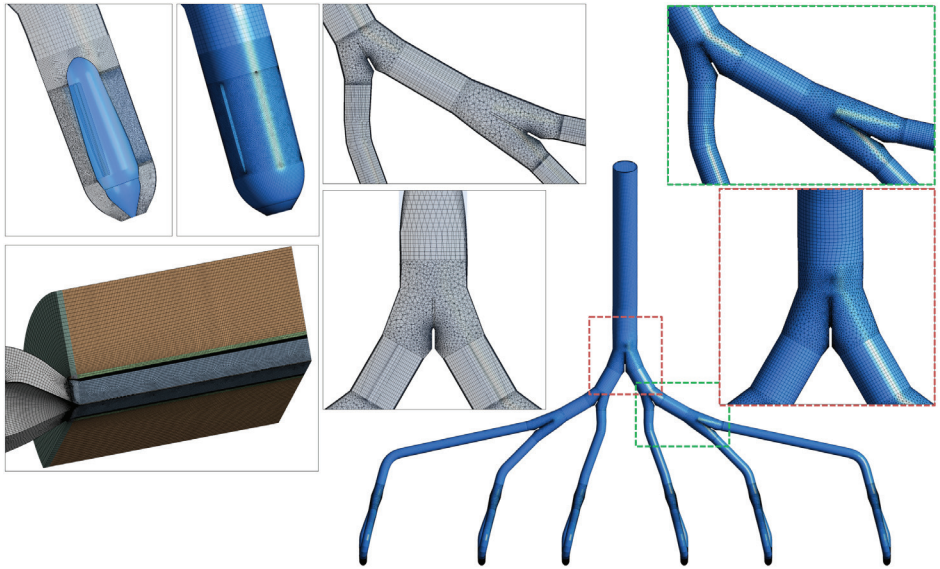


Fig. 3: Overview of the meshed used for the numerical calculations

Partially structured grids were used for the CFD analysis. The bifurcation pipe as well as partial areas of the nozzles were mapped with unstructured grids. All other geometries of the distribution pipe and the nozzles as well as the “outblock” of the water jet (2-phase calculations) were gridded with structured meshes. In the Figure 3, the surface meshes (blue) and the centre cross section through the computational domain (grey) are shown respectively.

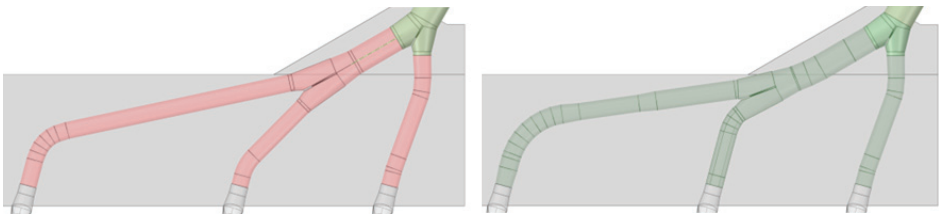


Fig. 4: Initial mid-section of distribution pipe (left), new optimised design (right)

With the final optimised version, the head loss of the new distribution pipe could be reduced by up to 20% (outside turbines). Figure 4 shows the comparison of the pressure pipe (middle section) of the original design (left) compared with the final optimised design. The validation of the secondary flows in the cross-sections were carried out via contour plots (Figure 5) and with the use of the swirl number (SN). Figure 6 shows the swirl number of the original design compared to the final optimised design of all six machines. The secondary flow could be reduced, in some cases drastically, compared to the initial design. After the optimisations, the secondary flows are at the same low level for all machine sets.

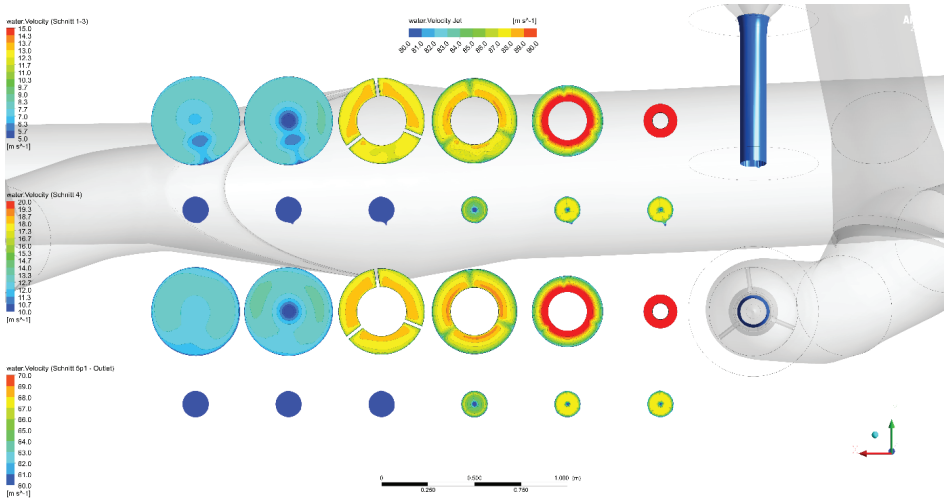


Fig. 5: Detailed view of the velocity fields and the shape of the water jet for one turbine at each cross section of the nozzle

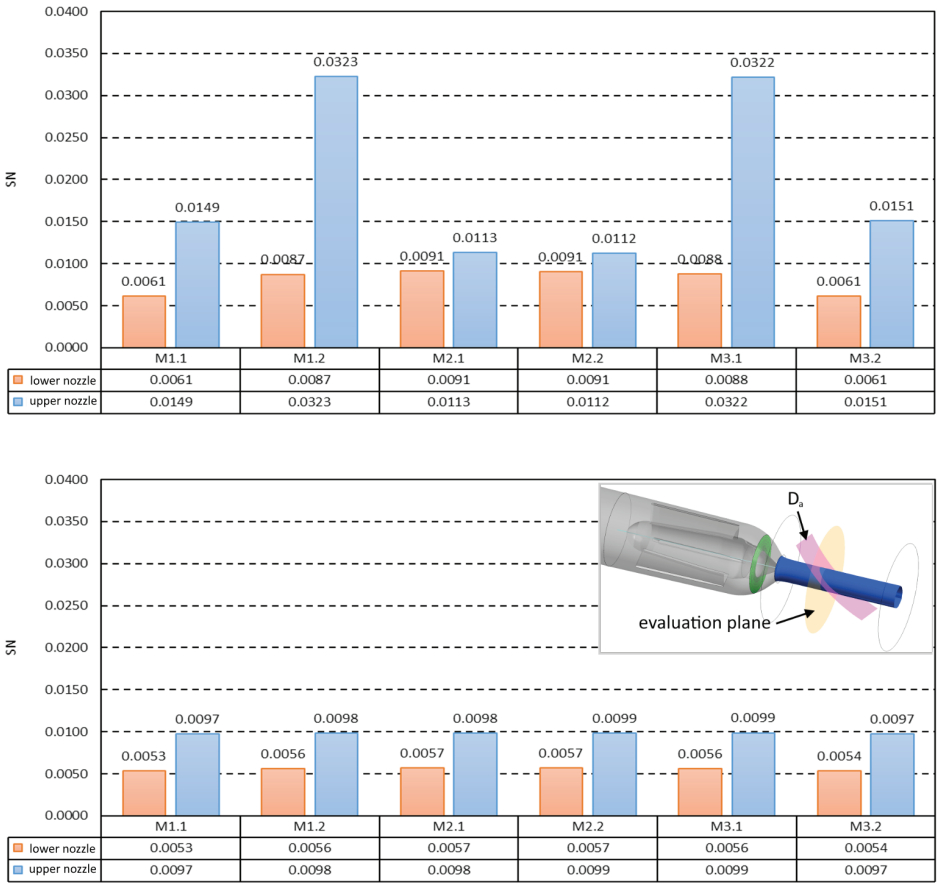


Fig. 6: Swirl number (SN) at one cross section of one turbine of the initial distribution pipe (top) compared to the finally optimized geometry [distribution pipe, bifurcation pipe, nozzle geometry (bottom)]

Beside the swirl number, the jet evaluation was also carried out using cross section though the water jet. In Figure 7 the cross section of the water jet with different volume fraction criteria is shown.

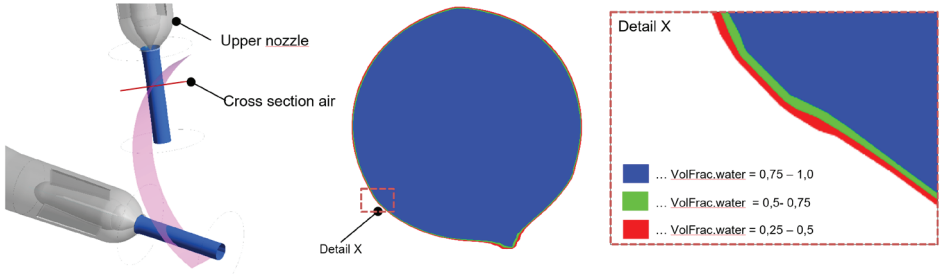


Fig. 7: Detailed view of the water jet shape at different water volume fractions (0,25 - 1,0)

Optimisation of the Francis-Runner Design at a Turkish HPP

At a dam located east of Ankara, a hydropower plant with 3 Francis units is located. The dam was built between 1964-66 and the units were brought in operation in 1966. The annual production of the hydropower plant is approx. 86 GWh (2003-2016) with a nominal power of P = 27 MW installed. Due to massive cavitation damages on the runner blades and shroud ring (see Figure 1) a new runner design needed to be developed. A detailed analysis of the actual situation and a subsequent geometry optimisation of the Francis runner was performed by applying Computation Fluid Dynamics (CFD).

According to the operational data of the power plant transmitted at the beginning of the project, the net head varies between H » 53 m (3-unit-operation) and H » 72.5 m (1-unit-operation). The maximum discharge to be reached at 3-unit-operation accounts for Q = 21 m³/s. At H = 72.5 m (1-unit-operation) the maximum discharge accounts for only Q = 19 m³/s. The best efficiency point of the new runner design shall be reached at a head level in between H = 53 m and H = 72.5 m and at a discharge of Q_{BEP} = 14 m³/s. By using the turbine speed of n = 300 rpm the specific speed $n_{q\text{-}BEP}$ of the optimized Francis-turbine is calculated as:

$$n_{q-BEP} = n \cdot \frac{\sqrt{\frac{Q_{BEP}}{Q_{Ref}}}}{\left(\frac{H_{Net}}{H_{Ref}}\right)^{0.75}} \cdot 300 \cdot \frac{\sqrt{\frac{14}{1}}}{\left(\frac{62.75}{1}\right)^{0.75}} = 50.4\ rpm$$

Eq.1

However, the meridional section of the turbine shown in Figure 2 is typical for a specific speed of around $n_{q,BEP}$ = 70-80 rpm. Thus, it can be assumed that the turbines were originally designed for much higher discharge and/or lower head.

Fig. 1:
Photograph of the original Francis-runner in the build-in-state

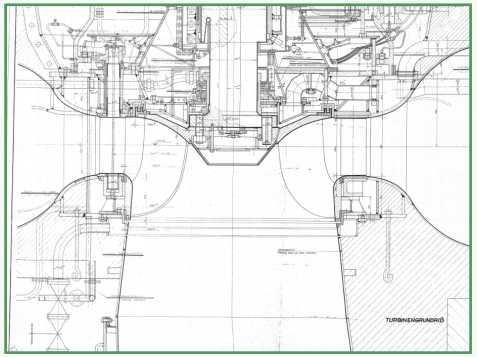


Fig. 2:
Meridional section of the turbine

The CFD-simulation of the initial state showed that the efficiency curves of the original runner are located at too high discharge. As expected the runner was obviously designed for full load operation at H = 53 m, which results in a very low part load efficiency and problems with vibrations and cavitation due to the occurrence of the typical part load cavitation vortices. Moreover, it was found that the cavitation performance is very poor. To reach an operation without extensive cavitation on the runner blades, the σ-curves of the turbine have to be below the σ_{Plant}-values. Obviously there is a very high risk of cavitation over a wide range of operation, which also explains the extensive cavitation damages found on the original runner design. These damages can be well explained with the help of the CFD-results.

Figure 3 presents the pressure distribution calculated at full load operation at H = 53 m (3-unit-operation, see left pictures) and the pressure distribution calculated at deep part load at H = 72.5 m (1-unit-operation, see right pictures). It turns out that the regions with the lowest pressure values correlate with the cavitation damages visualized on the photographs.

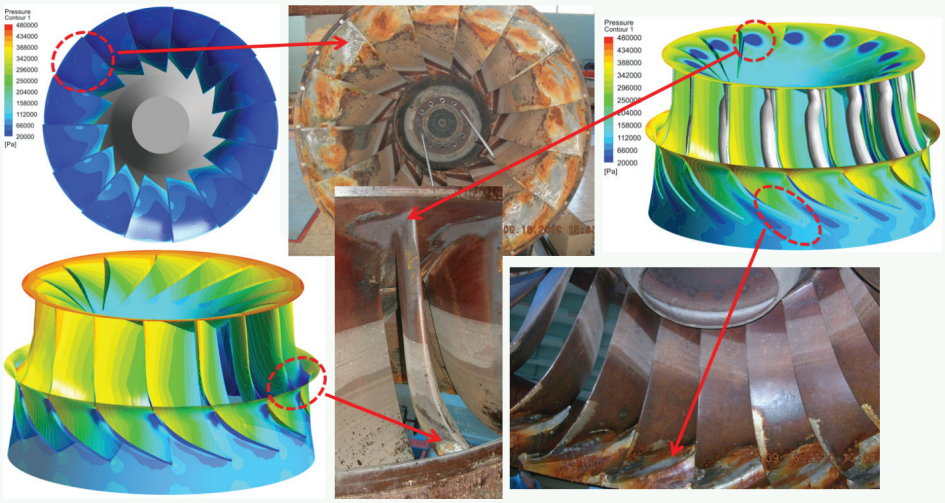


Fig. 3: Visualization of low-pressure zones on the runner and pictures of cavitation damages

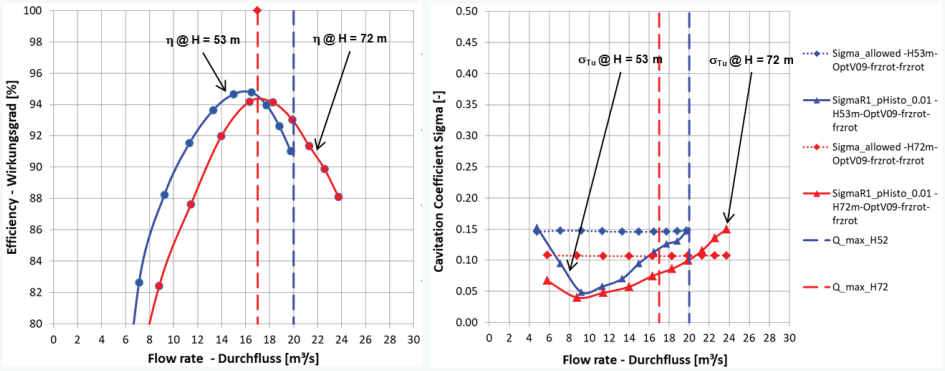


Fig. 4: Efficiency- and σ-curves calculated for a H = 53 m and H = 72 m

With the help of an optimized runner design the Best Efficiency Point was shifted to lower discharge and the cavitation characteristics were significantly improved, which is shown with the η- and σ-curves calculated for a H = 53 m and H = 72 m and plotted in the diagrams shown in Figure 4.

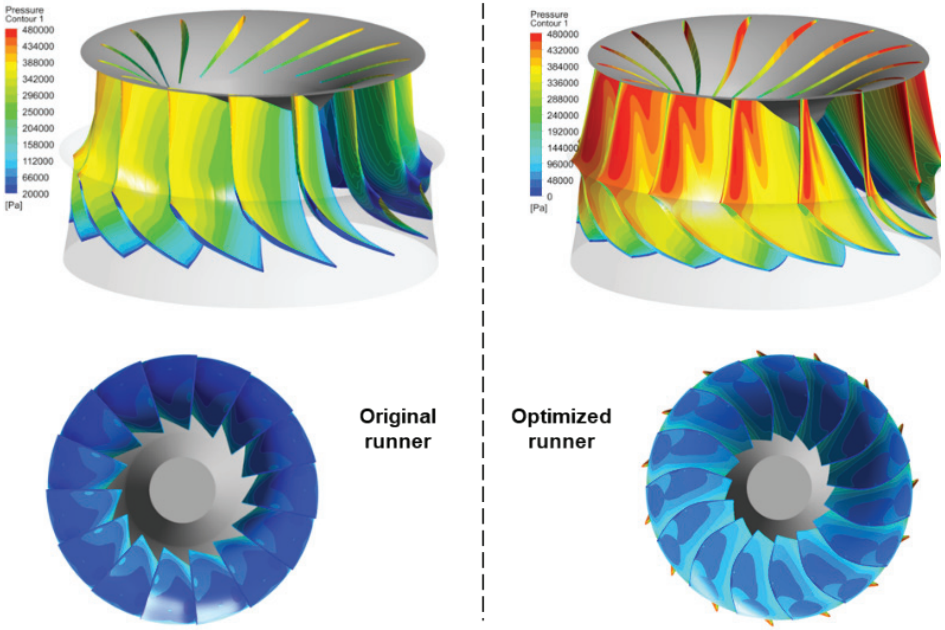


Fig. 5: Pressure contours plotted on the original (left side) and optimized runner (right side) at the example of the full load conditions at H = 72.5 m

The improvements achieved with the new runner design can also be visualized with the pressure distribution plotted on the runner. At the example of the full load conditions at H = 72.5 m (1-unit-operation) Figure 5 presents a comparison of the pressure contours plotted on the original (left side) and optimized runner (right side).

At the beginning of 2022 the first of three runners was taken into operation and the operator is impressed by the improved operational behaviour of the new runner design.

Homologous Model Test of a Single Stage High Performance Storage Pump

The J&P-team was commissioned by the Verbund Hydropower GmbH to carry out a homologous model test of a high performance 1-stage storage pump. The 1-stage storage pump is based on an already existing design. The already existing pump design was adapted to the new location, and CFD-based development was applied to it by the manufacturer. The new storage pump design replaces three multistage pumps to transport water from the Kreuzeck side to the Reisseck power plant side. It is installed in the valley of the two reservoirs and has impressive performance data, achieved in single-stage operation at n = 3000 rpm.

The characteristics of this new pump should be confirmed by the J&P-team as an independent organisation in cooperation with Institute of Hydraulic Fluid Machinery of the Graz University of Technology. Within the scope of a model test, the performance data was measured, including the inflow and outflow situation. The pressure pulsations had to be verified. Subsequently, on behalf of Verbund, these pulsations had to be reduced by installing air vessels, both on the suction side and on the pressure side, and also through modifications in the bladed areas of the impeller and the guide vane section. The new storage pump will be integrated into the existing Kolbnitz powerhouse and connected to the current distribution pipeline network. This setting results in a suction-side connection of the pump (via a bend and a suction branch), which influences the suction-side inflow of the pump. To obtain measurement results as representative as possible and to be able to take the influences mentioned above into account, the model test will be carried out with a homologous design of the prototype inflow pipework.

All measurements were carried out in a closed circuit; this allowed for measurements at different system pressures levels. In this setup, the central frequency-controlled test rig pump was also used as a booster pump for measuring the 4-quadrant pumps curve.

For the measurements of the pressure pulsations and the hydraulic characteristics in pump mode, the test rig's main pump was not required and was bypassed. The model pump was driven by a frequency-converter-controlled motor connected to the pump's drive shaft utilising a torque measuring flange. Figure 1 shows the homologous assembly of the pump test bench.

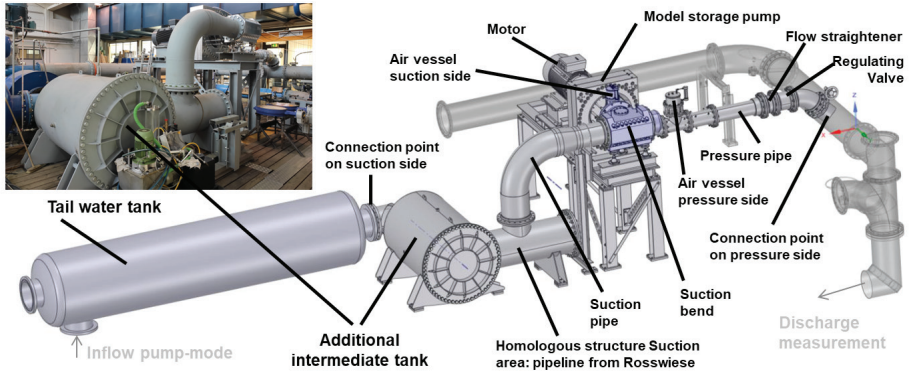


Fig. 1: Homologous assembly on the test bench

In addition to the pressure measuring holes for stationary pressure measurements (pump head), additional pressure measuring taps for highly dynamic pressure measurements (pressure pulsations) have been integrated into the pump model. A total of more than 60 pressure measuring holes were provided in the suction area, in the pump housing and on the pressure side of the pump. Preliminary investigations allowed for the reduction of the number of measuring points to 12 sensor positions. These evaluations of the pressure sensors' signals included peak-to-peak evaluation and signal filtering with and without a high-pass filter, as well as the associated FFT analyses. In Figure2, only those measurement boreholes are visualised, which are later evaluated in detail.

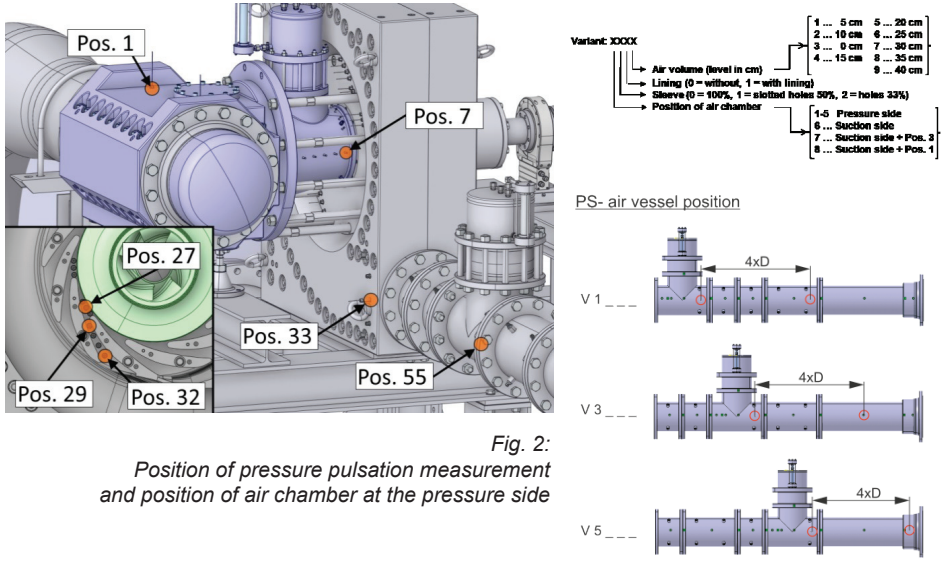


Fig. 2:
Position of pressure pulsation measurement and position of air chamber at the pressure side

The pump curves were measured at different speeds (n = 750, 1000, 1250, and in the best efficiency point also with 1500 rpm) with both increasing and decreasing flow rates. This was done to exclude any influence of the speed or to record the impact of the flow variation on the pump instability (hysteresis) and to vary the Reynolds number. Figure 3 shows the measured pump curve (prototype) with the pressure pulsations (Hdyn) of four pressure pulsation positions of the modified geometry.

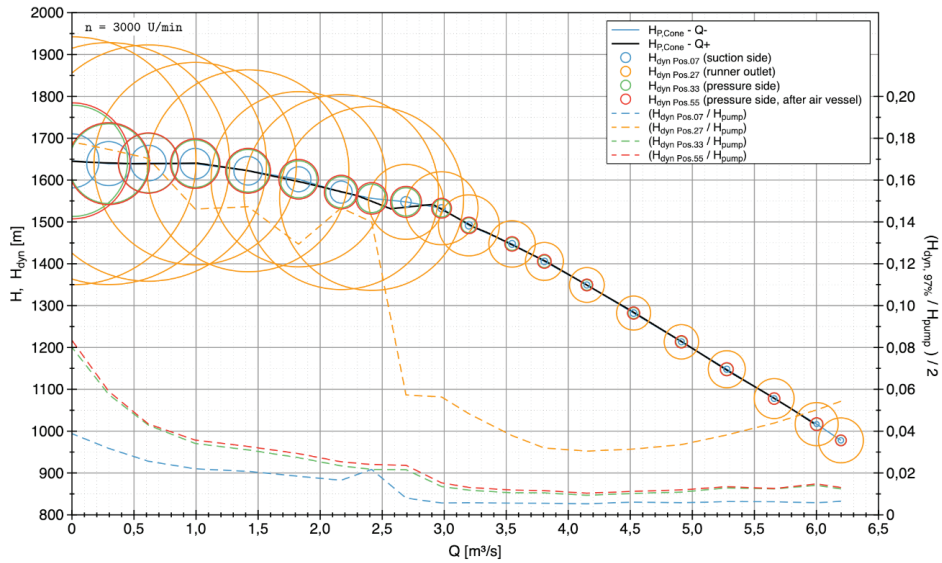
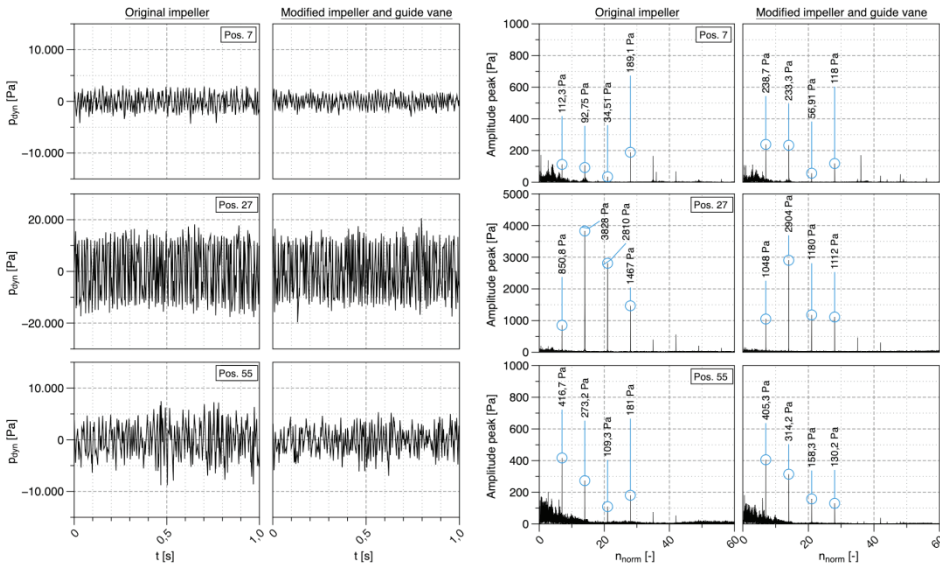


Fig. 3: 97%-peak-to-peak pressure pulsation for different sensor positions, operation without air vessel, prototype values calculated from model test rig results, modified geometry



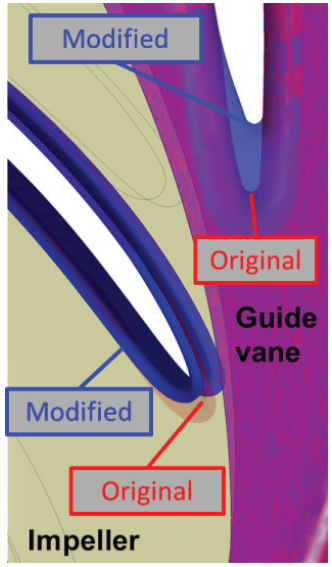


Fig. 4:
Pressure pulsation for different sensor positions, operation without air chamber, model test n = 1000 rpm

Right: Modification of impeller and guide vane

One option of reducing the pressure pulsations is to increase the distance between the outlet of the impeller vanes and the inlet of the guide vanes. The J&P-team modified the original distance between the impeller and guide vanes (modified geometry). With this modification, the delivery head of the pump is just reduced to the height required at the guarantee point at nominal speed. Figure 4 shows an example of the pressure pulsations before and after this modification of three positions of the pulsation sensors.

For further possible reduction of the pressure pulsations, a second option of reducing the pressure pulsations along the suction and pressure side is to add air vessels at the pump in- and outlet. Therefore, a considerable number of variations of the air vessel setup were measured. For each parameter variation of the air vessel setup, the pressure pulsations were measured along the entire pump curve. During the pulsation measurements, the air volume of the air vessel (suction and pressure side), the connection of the air vessel to the pipeline (free cross-section) and the position of the pressure-side air vessel were varied. Different insert sleeves provided the free passage of the connection of the air vessel. Thus, 100% free passage, as well as 50% (long holes) and 33% (round holes) free passage were examined. The perforated sleeves were manufactured with different patterns.

In Figure 5, the results for sensor position 55 (in the pressure pipe after the air vessel at the pressure side) are shown.

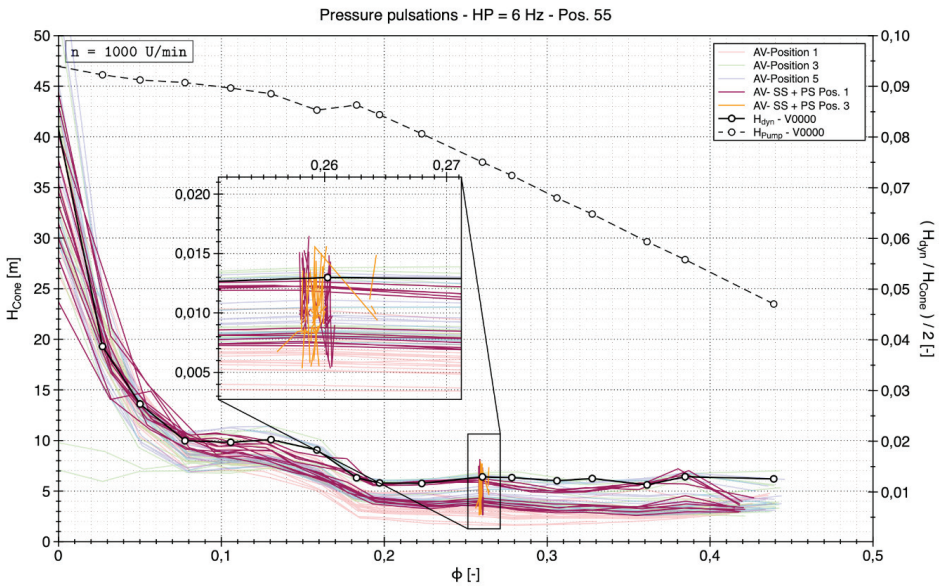


Fig. 5: 97%-peak-to-peak pressure pulsation for different air chamber positions and air fill levels, results from model test

Additionally, an air volume variation was carried out in the suction and pressure side air chambers for the optimum operating points. The variation of the air volumes includes the suction side air volume, the pressure side volume, and the typical variation of the filling level of both air chambers. Figure 6 for example shows the influence of air height in the air chambers variations on the amplitude of the 1st harmonic in the pump optimum.

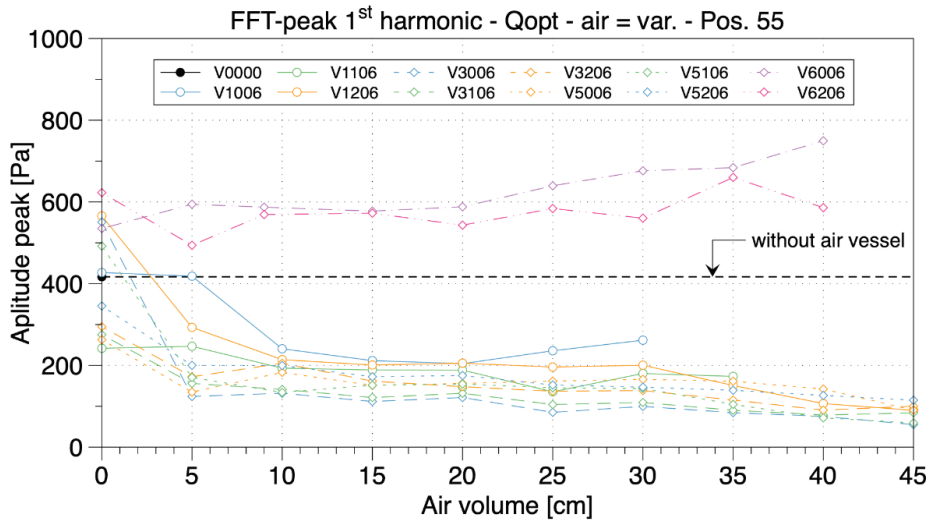


Fig. 6: FFT peak of 1st harmonic at optimum discharge for sensor 55, variation of air volume

In the prototype version (see Figure 7 right for the aligned pump before setting in concrete), installing air chambers at different positions identical to the model test will be possible. Thus, the pressure pulsations can be dampened should they have a negative acoustic influence on the environment.

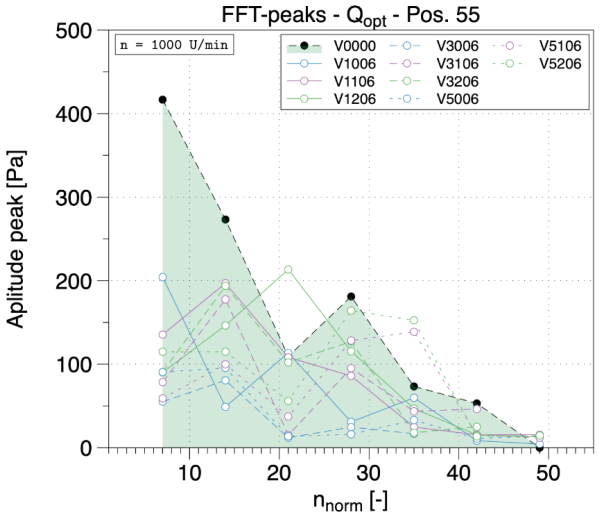


Fig. 7:
Above: Lowest FFT peak of 1st to 7th harmonic at optimum discharge for sensor 55

Right: Prototype pump during installation (Source: Verbund)

Fig. 8: Test rig installation of the pump with the high pressure taps at the suction cone (top left), the overview of the pump (top right) and the high pressure with attached air vessel at the position 1 (bottom)

A Systematic Optimisation of 7 Different Kaplan-Turbine Configurations and Development of a Design Tool

To enhance the competitiveness of a European manufacturer of hydraulic turbines for the small hydropower market, it was intended to optimise the hydraulic design of the offered range of turbines to increase the hydraulic turbine efficiency and to improve the cavitation performance. For different turbine configurations (Kaplan turbines with spiral casing and radial inflow, axial Kaplan turbines in S- / Z- / bulb- and pit-type configuration) 3-/ 4- and 5-blade runners and their corresponding guide vanes needed to be developed for specifically given design heads. For future projects, the optimised turbines shall be scaled to a wide range of head and flow rate by using appropriate turbine hill charts which needed to be provided based on CFD-simulations.

As an example, the Figure shown below presents a flow visualization of a 4-blade Kaplan runner design with a specific speed of $n_{q,\ opt}$ = 165 rpm which was optimised for the use in a z-type, bulb-type and S-type turbine configuration.

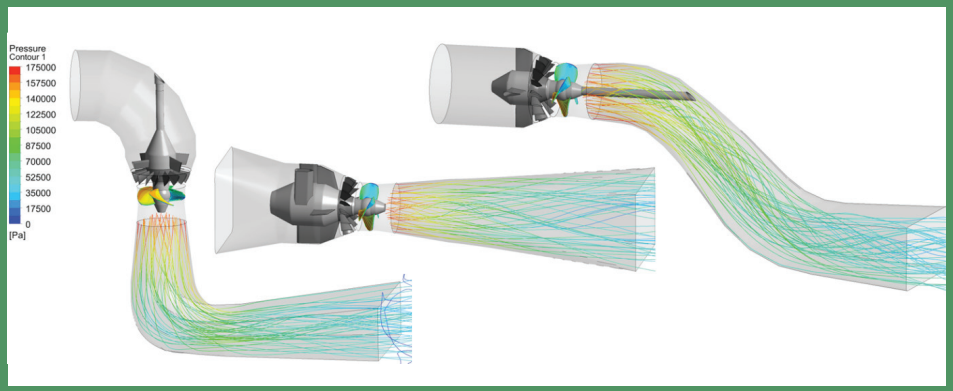


Fig. 1: Flow visualizations of a 4-blade Kaplan runner design with a specific speed of nq, opt = 165 rpm which was optimized for the use in a z-type, bulb-type and S-type turbine configuration

Between July 2019 and December 2020, the Jaberg&Partner GmbH carried out a hydraulic development of 7 different Kaplan turbine configurations plus an appropriate design tool to adapt these turbines for different operation points. For each of the turbines, the customer submitted a proposal for the overall turbine geometry as CAD-file including the turbine intake, the flow path of the guide vanes and runner blades as well as the draft tube. Within a cycle time of 8 to 10 weeks per turbine configuration, the work packages were finished step by step.

The target of each stage was to design new runner blades and guide vanes in order to achieve the best possible hydraulic efficiency and cavitation performance. In some cases, the given shape of the hub and shroud contour needed to be adapted as well. In case of the S-type turbine configuration, it turned out that also the shape of the draft tube needed to be improved. Referring to the Best Efficiency Point the Figure shown below presents a comparison of the original (left) and optimised (right) draft tube design. With the new S-type draft tube a separation of flow can be avoided and the efficiency can be improved.

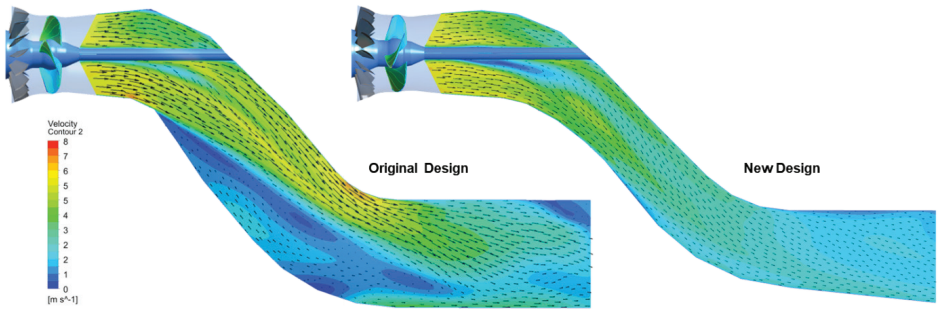


Fig. 2: Velocity distribution on a mid section of the original (left) and new (right) design of the S- draft tube at BEP

By using in-house developed design tools initial designs of the runner blades and guide vanes were created in the first step. Only in case of the radial flow 5-blade runner equipped with a full spiral case, the whole design from intake to draft tube was elaborated by Jaberg&Partner. In the next step CFD-meshes were generated and a CFD-model was prepared at each stage. The evaluation of the turbine performance as well as the optimisation of the turbine geometry was carried out by using the commercial CFD software package ANSYS-CFX release 17.1.

In order to validate the cavitation performance, the so-called „Sigma-Histogram" method was applied which was cross-checked several times with experimental data. According to our experience the σ-values calculated thereby typically correspond to the σ_0-values detected in course of a model test.

For all 7 Kaplan turbine configurations investigated, a semi spherical shape was used for the shroud contour of the new runner to facilitate the installation of the turbine. According to empirical data used in the field of small hydro, the gap at the runner tip was modelled with s = 1/1000*D. In case of larger units the gap may be reduced to s = 1/2000*D which would yield a slightly increased efficiency level. Another focus was set on the hub ratio (= DRunner-Hub/DRunnerShroud) of the investigated turbine designs. Typically, the hub ratio is a function of the specific speed nq and decreases with increasing specific speed nq. In order to reduce the number of hub-patterns and to ease the fabrication process it was decided to use only 3 different hub ratios (0.35 / 0.40 / 0.45).

By combining various positions of the runner blades and guide vanes, steady-state and single-phase flow simulations were performed over the entire range of operation at a fixed design head level. For the optimisation of the blades a simple single-channel model with periodic boundary conditions was used. The final turbine designs were then evaluated by using a full 360° turbine model. The full turbine simulations of the final design versions were then extended to a head of +/- 35% of the design head. Thus, the turbine characteristics could be plotted as function of the head H and the flow rate Q. Finally, hill charts containing the hydraulic turbine efficiency, the cavitation coefficient, the ideal opening of the runner blades and guide vanes as well as the hydraulic blade adjustment torque and the axial forces were created. A conversion into non-dimensional parameters (e.g. pressure coefficient ψ instead of head H and discharge coefficient φ instead of flow rate Q) facilitates to convert the runner diameter D and the turbine speed n to projects with different nominal data which was used as background for the creation of the Excel-based design tool.

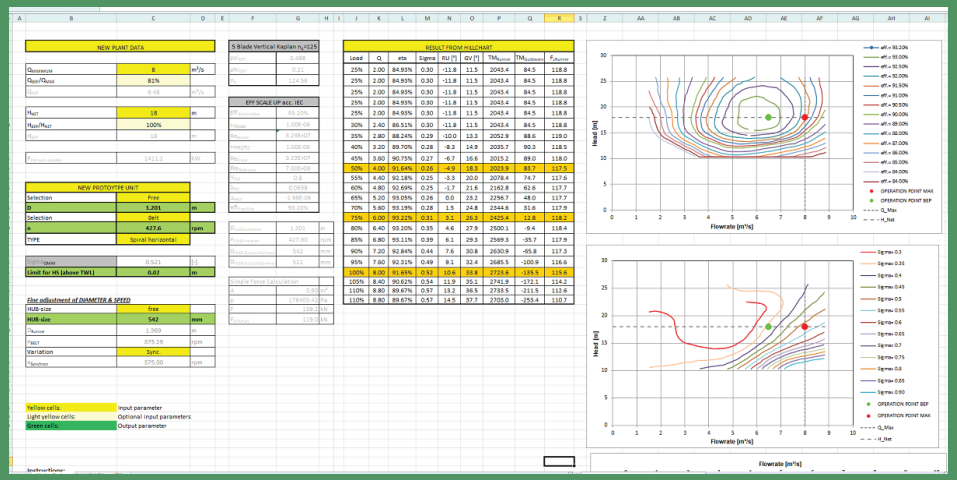


Fig. 3: Overview of the input mask and the hill charts of the Excel-based design tool

The hill charts prepared based on the CFD-results cover the typical operation range of a turbine but do not include information on the runaway curve. Typically, this curve is measured in the course of a model test. Especially in case of a double regulated Kaplan turbine, it is very time-consuming and costly to estimate the runaway curve by means of CFD. Thus, guidelines taken from literature can be used alternatively.

In order to convert the runner diameter D and the turbine speed n for different projects, to find out the operational range in the hill-chart and to convert the efficiency level from rather small to rather large units, a table-based design tool was prepared by using MS-Excel cf. Figure 3. By defining the flow rate Q and the head H of future projects all required turbine parameters can be calculated. Additionally, the expected hydraulic turbine efficiency, the ideal position of the runner blades and guide vanes as well as their hydraulic adjustment torques and the axial force acting on the runner are plotted versus the discharge in separate diagrams. The Figure presented below shows an overview of the input mask and the hill charts presented in the design tool.

Field Measurements HPP Truchtlaching

photo by Elektrizitäts-Genossenschaft ALZGRUPPE eG

Parts of the power plant in Truchtlaching on the Alz in Germany are to be modernised. Out of this reason, measurements of the existing turbines were conducted. In addition to the two old Kaplan turbines (Storek-turbines), the site also has a Kaplan bulb turbine (Escher Wyss) and a hydroelectric screw turbine. The two Storek-Kaplan turbines were commissioned in 1963 and have been in operation since then without any major shutdowns. The generators and transformers were supplied by the company AEG and together they generate a combined output of 600 kW.

A performance enhancement of the power plant site by a Kaplan bulb turbine was realized in 2001. In this year, the Kaplan bulb turbine was commissioned with an output of approx. 380 kW. The installation of the Kaplan bulb turbine was accompanied by the installation of a central computer-based control system for the entire power plant. The hydroelectric screw with a capacity of approx. 29 kW was taken into operation in 2015 and was installed beside to the Kaplan bulb turbine.

Figure 1 shows the inflow channel of the head water with all four turbine inlets. In addition in Figure 2 all three turbines types are shown.

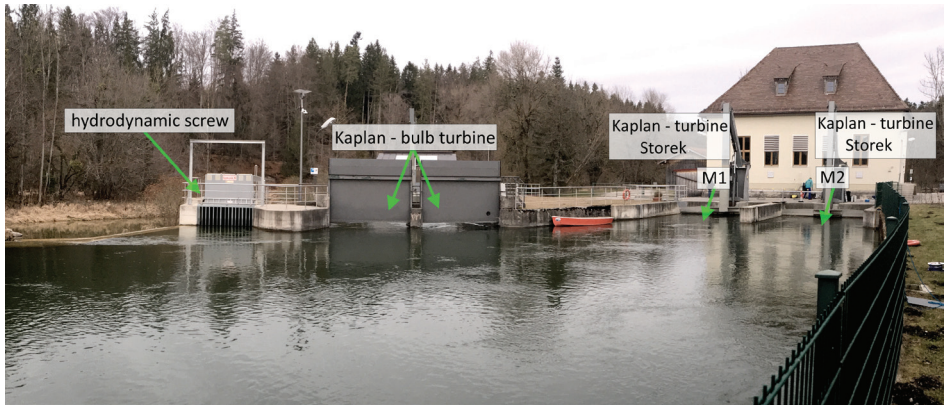


Fig. 1: Overview of the powerplant with all 4 turbines (head water view)

One of the main challenges by conducting field measurements at powerplants is to determine the flowrate with high accuracy. In this case the direct integration method with current meters (propeller-type) according to the IEC 41 standard was used to determine the discharge of each turbine. Overall eight current meters in two different sizes were mounted on a frame (hight adjustable) which was guidance by the gate slots. In the centre of the measuring section larger current meters were used. For capturing the higher velocity gradient near the side walls of the cross sections two smaller current meters were attached to the frame.

Fig. 2: The three different turbine types of the power plant:
Kaplan turbine – Storek (top left), Kaplan bulb turbine (middle), hydrodynamic screw (right)

In Figure 3 the installed frame in the gate slots of the Storek turbine M2 and the assembling of the frame with one of the centre current meters is shown.

Fig. 3: Installed frame at the Kaplan turbine M1 (left), assembling of the frame (middle) and detail of one current meter (right)

In addition to the flowrate, the turbine head was measured with multiple level probes at the head and tail water of each turbine. The power of the generator was measured with a DEWETRON power analyser, which guarantees the highest accuracy as well as detailed information like current and voltage of each phase and power factor of the generator. Furthermore, the guide vane openings, the water temperature and additionally the environmental data (ambient pressure, air temperature, air humidity,) were recorded as well. Figure 4 shows the connection points for measuring the generator power as well as the measuring equipment during operation.

In order to determine the efficiency curves of the turbines over the entire operating range, a total of eight operation points were measured for each turbine.

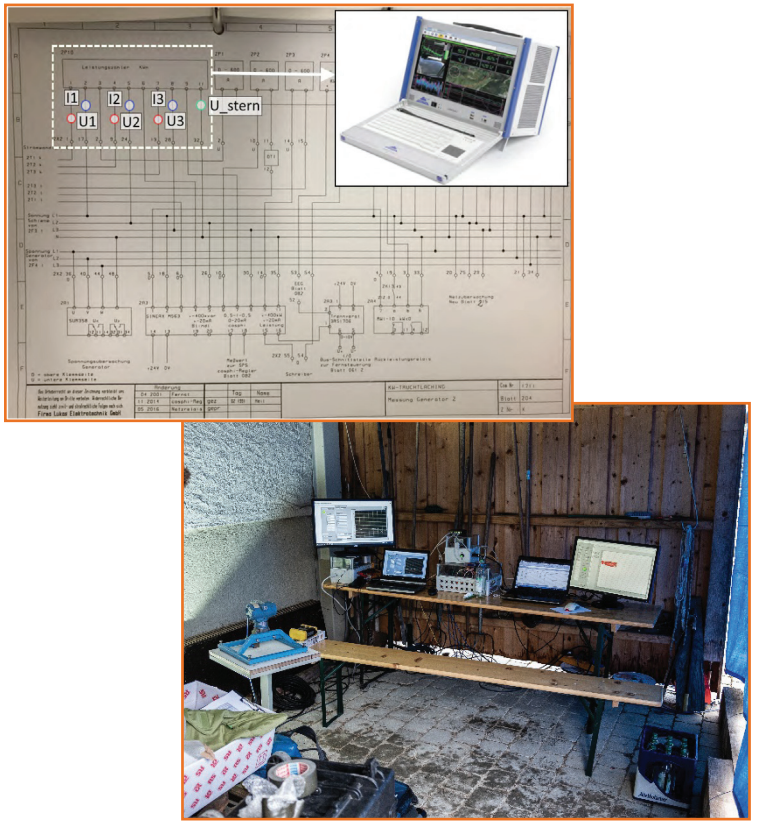


Fig. 4: Connection points for measuring the generator power (top), installed measuring equipment

Pressure Surge Investigation of Industrial Water Systems

Pressure surges, also known as water-hammer, occur in every pumping system, even with correct design, and must be considered at the planning stage. If the loads that occur are not known, damage to the system is a frequent consequence.

Hydraulic systems (new design, but also adaptation or an upgrade) and their components are usually designed or dimensioned for stationary operation only. But transient flow phenomena already occur when the pumps are switched on or off, or in case of a valve malfunction. Much higher system loads during these transient operating conditions are to be expected compared to stationary operation. Transient flow processes with partly higher system loads than in stationary operation are already caused by operation-related start-up or shut-down of the system. Even undesirable events that cannot be excluded during plant operation often lead to a multiple of the loads determined for steady-state operation. Examples include a sudden pump failure (for example due to an electrical defect) or the closing of a control valve due to an incorrect control signal. By a numerical pressure surge investigation (water-hammer analysis), these events and their effects (overpressure and underpressure or even cavitation) can be reliably determined and any safety measures derived. In principle, the longer the (unprotected) pipeline and the faster the change in flow, the more severe the pressure surge. The magnitude of the pressure surge is therefore not dependent on the applied static system pressure, but is added to it.

Case study 1:

Damage analysis of district heating system

In the course of a re-design and extension of a district heating pipeline and its connection to the grid (Figure 1), a pressure surge analysis was carried out. Its aim is to determine the maximum and minimum possible pressure loads in the hydraulic system in case of a total or partial failure of the circulation or booster pump stations due to a power failure for different operating scenarios (summer, winter operation, ...) as well as for the emergency closure of the control valve in the converter station (UFO). Based on network diagrams (Figure 1 - left) and documents on the components of the plant (pumps, valves, drives, ...), the district heating plant to be investigated (Figure 1 - right) can be modelled in the CFD simulation environment. In the best case, calibration at selected reference operating points is possible on the basis of measurement data. If these are not available (as in the case of a new plant in planningphase), careful assumptions must be made. In the example given, a verification at randomly selected operating points (Figure 2 - left) confirmed the modelling approach by very good agreement between measured data and simulation results. The calculations subsequently carried out showed no inadmissible pressure surge loads on the system in the event of drive failure of the pumps.

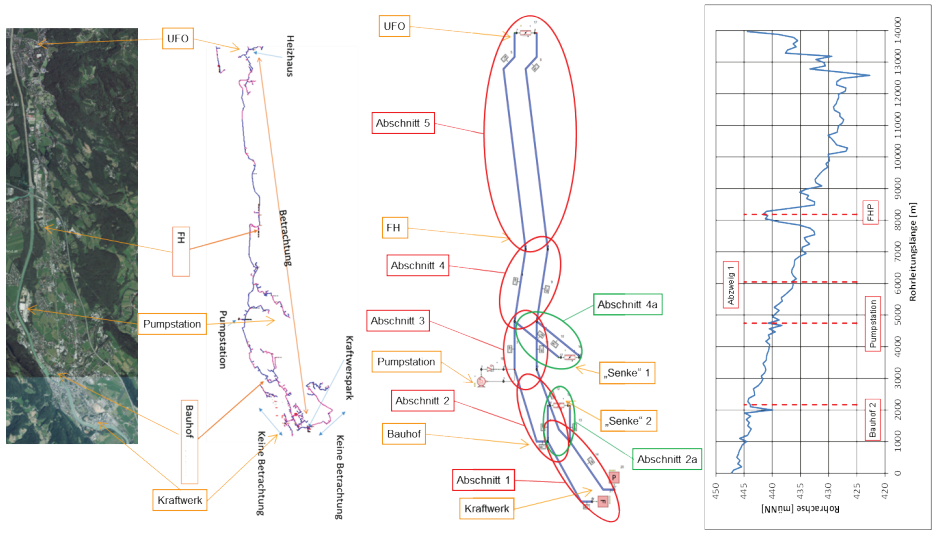


Fig. 1: District heating network

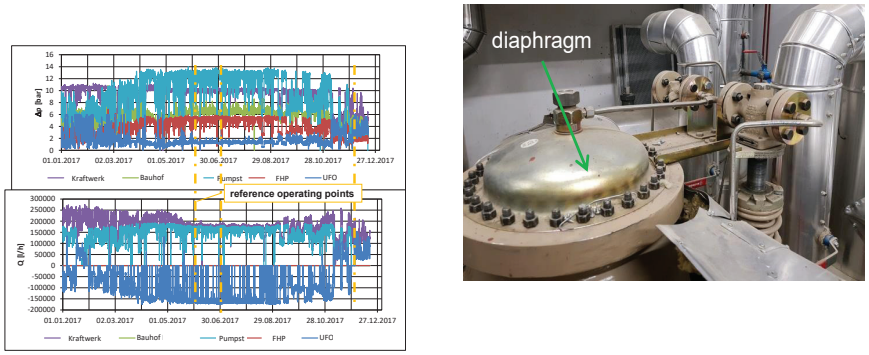


Fig. 2: Operating data - left; diaphragm-controlled bypass valve (right).

The situation is quite different for the emergency shutdown of the UFO control valve! In this case, based on the simulation results and the assumptions made, there is an impermissible system load - both with regard to the maximum and the minimum pressure. This finding was also confirmed by operating experience. The overpressure protection currently installed in the plant by means of a diaphragm-controlled bypass valve (Figure 2 - right) does not function satisfactorily or is damaged when triggered. The diaphragm cannot withstand the stresses in the event of a malfunction.

With the help of the simulation model, the cause of the impermissible loads could be quickly determined. The bypass valve was set to respond in case of exceeding the maximum permissible pipe-line pressure. For the load case of an emergency closure of the control valve, it was clearly demonstrated by the numerical analyses that with differential pressure control instead of maximum pressure control, the pressure surge can be greatly reduced and lowered to an acceptable level.

Case study 2:

Clean water network of a steelworks including upstream water treatment / gravel filter plant

The investigation of the transient system behaviour of the new water treatment plant, designed as a gravel filter plant, in a steelworks is the subject of case study 2. The drive failure of the clean water pumps, which provide the system pressure for the downstream clean water network of the entire steelworks, was identified as a critical load case. Although the steelworks has an emergency power supply, in the event of a power failure, a short-term voltage drop and thus drive failure is to be expected in case of a full operation of the clean water system. In order to analyse this scenario, it is necessary to model and prepare the entire clean water network (Figure 3).

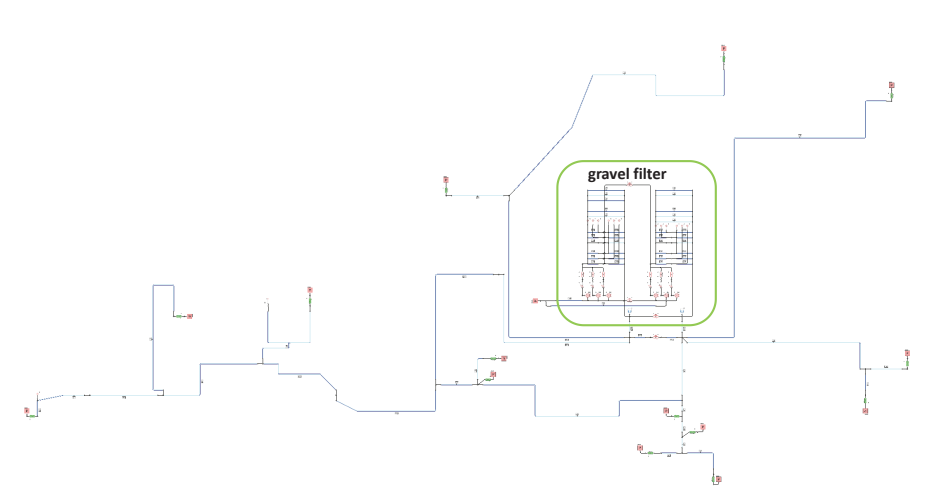


Fig. 3: Simulation network of the pure water network including the gravel filter plant.

In contrast to the modelling of the gravel filter plant, which was fully documented due to the new construction (Figure 4), the data collection of the existing pure water network turned out to be a busy task. On the one hand, it is de facto impossible to record the entire system due to its complexity, but on the other hand it is not necessary, as no potential pressure surge risk is to be expected from very small consumers. The failure of larger consumers, on the other hand, cannot be neglected. For this reason, it is advisable to agree in advance on the smallest nominal pipe size to be surveyed. The most challenging part of the work is then the data preparation and modelling of the network to be investigated. It is well known that process plants have the property of changing over the years due to various modifications, expansions, etc. However, these changes are not always documented or may not have been implemented as originally planned. Thus, modelling is also a welcome occasion to bring the plant documentation up to date.

If the analyses carried out reveal inadmissible plant loads under certain load cases, it is not at all uncommon that these loads could also be possible prior to the upgrade or expansion of a plant. Fortunately, such operating scenarios had never occurred in the past, although it can by no means be definitively ruled out that such conditions may arise in the future. Thus, a pressure surge analysis in the course of a plant expansion is also a good possibility to check the plant safety at the status quo.

In the investigated load case of a drive failure of the pure water pumps and due to the construction-related height difference in the gravel filter plant and the insufficient pressure of the water supply, an impermissible negative pressure with cavitation was detected in the plant. The only feasible remedy proposed was to ventilate the 2nd floor of the gravel filter (Figure 4 - right). In addition, energy storage in the form of membrane pressure vessels had to be provided for pressure surge protection.

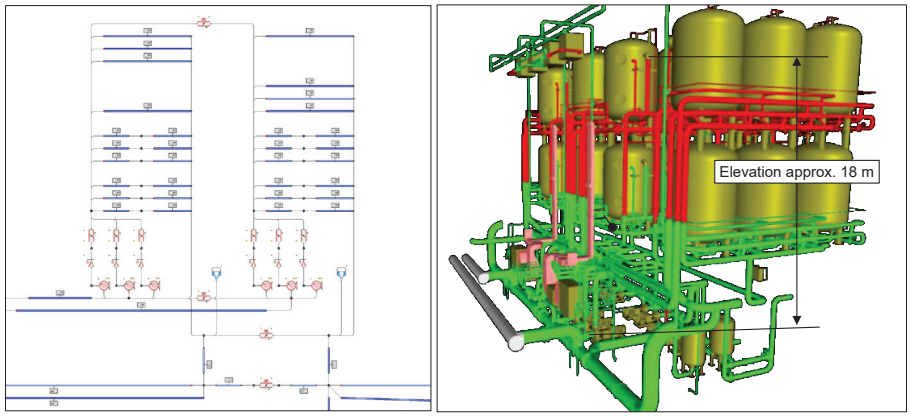


Fig. 4: Gravel filter system - simulation model (left) and 3D CAD model (right).

These measures reduce the pressure drop in the event of a drive failure and thus not only protect the plant from impermissible negative pressure (risk of cavitation!), but also dampen any occuring pressure pulsations. This in turn reduces the dynamic load on all pressurised components. Due to the limited installation space for the pressure vessels, great care had to be taken in the dimensioning of these as well as their connecting pipework. In the end, an air vessel was found to be acceptable, which is already used in identical dimensions in other plants of the steelworks. This also minimises maintenance and servicing costs.

Finally, a note on aeration valves in pumping plants: In general, the operator one wants to keep air out of a water pumping system. Aeration valves are therefore usually an unloved emergency solution if no other remedy against negative pressure, i.e. cavitation with the subsequent dreaded cavitation shocks, can be realised. Once installed, you have the unloved air in the system every time they respond.

Apart from the other case studies described above examples of investigated pumping systems in the last two years cover the range from gas exploration fields over the fire water system of a tank farm / fuel depot (figure 5), Austria's biggest sewage treatment plant (figure 6) to the upgrade of Vienna's district cooling network.

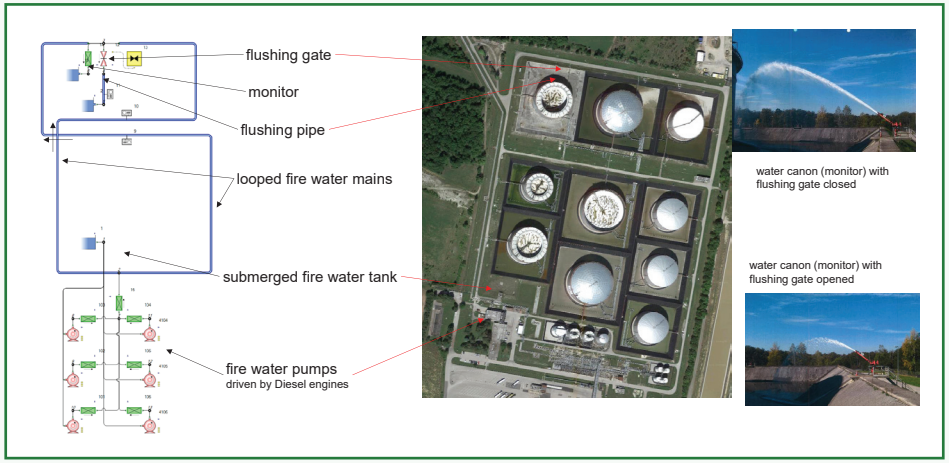


Fig. 5: Fire water system of a tank farm

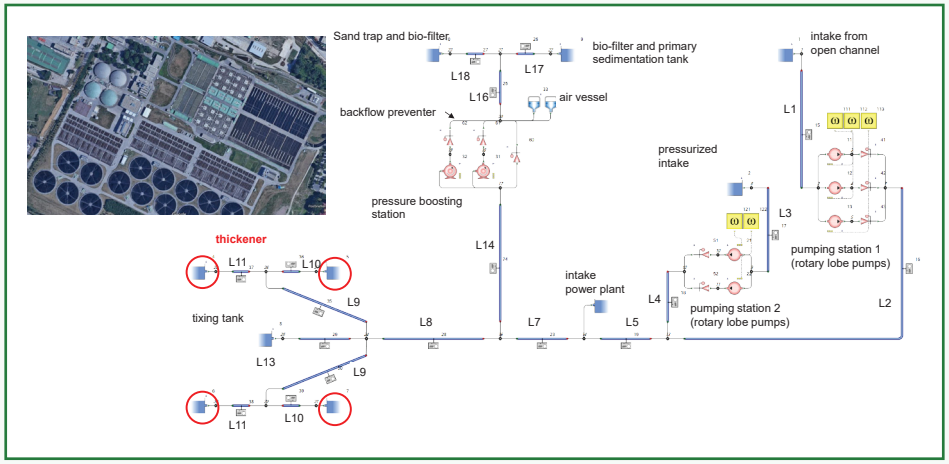
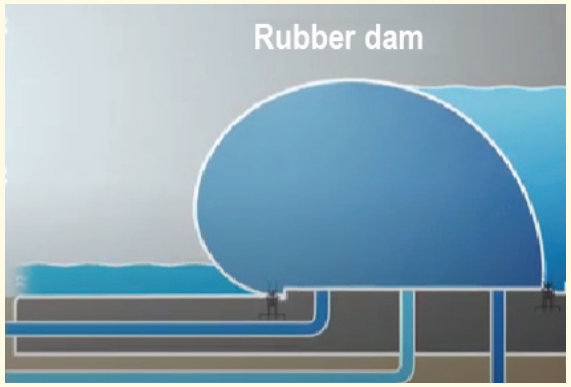
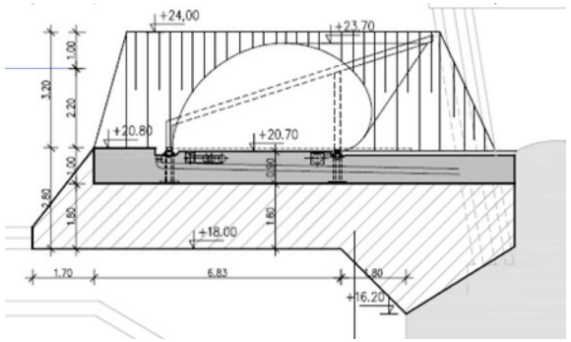


Fig. 6: Water distribution in a sewage treatment plant

FE-Simulation of a Rubber Dam Clamping System

In a rubber dam, a rubber hose is inflated either with water or with air. This creates a dam with variable height, which can be completely lowered if necessary. Weir heights of more than 4 meters are already possible.





The clamping device, which fixes the hose-diaphragm to the foundation, plays thereby an important role. In addition to secure clamping, this must also ensure leak-tightness over many decades.

Clamping

For clamping, the hose is clamped with a stainless-steel rail over a series of threaded bolts co-cast in the foundation. The bolting itself is rather complex since the connection consists of different materials each with its own elastic characteristics. Steel, concrete and the membrane made of a rubber-fabric composite result in a connection of different Young's-moduli and frictional contacts. Hence, a purely analytical calculation of this connection is hardly possible or only possible with considerable uncertainties.

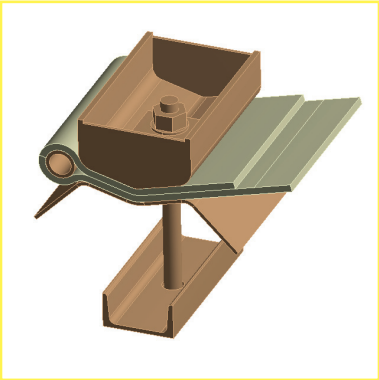
Solution

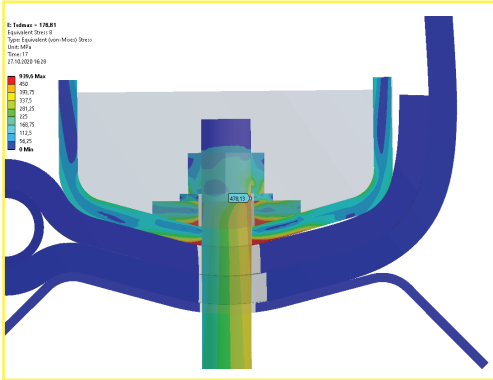
By means of a finite element simulation on a section of the clamping rail, the stresses, deformations and clamping forces can be estimated very well. The FE simulation allows a view into the interior of the components. Not the occurring stresses only can be determined, but also the existing clamping forces. Hence, it is possible to analyze the sealing effect as well as the movements of the hose-diaphragm in the clamping point..

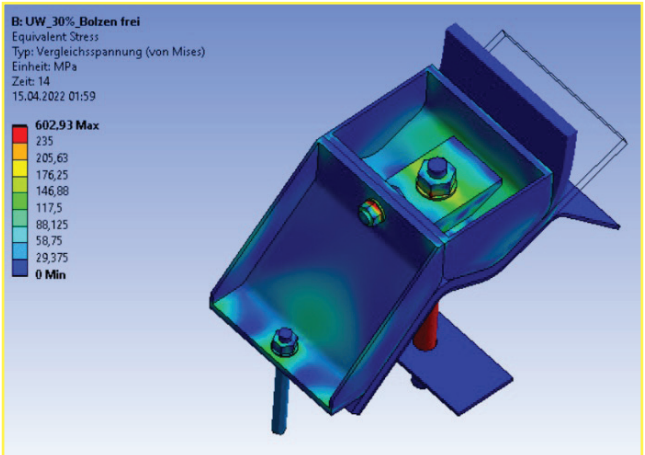
Case studies and service lifetime calculation

A wide variety of investigations were carried out via parameterization. For example, which bolt diameters are required for which weir heights, or how much is the loss of pretension force due to thermal influences, and what is the affects on the clamping in the end. The hose weir clamping must still function reliably in 50 years' time, so it is often impossible to rule out fatigue failure even for components that at first glance are not subject to dynamic loads. In one specific case, therehas been a vibration that occurred during the filling of the weir at a certain water level.

By means of finite element simulation, however, it was possible to determine the stress excursions caused by the load fluctuation, which could then be used to perform an operational strength calculation in accordance with DIN EN 1993-1-9.





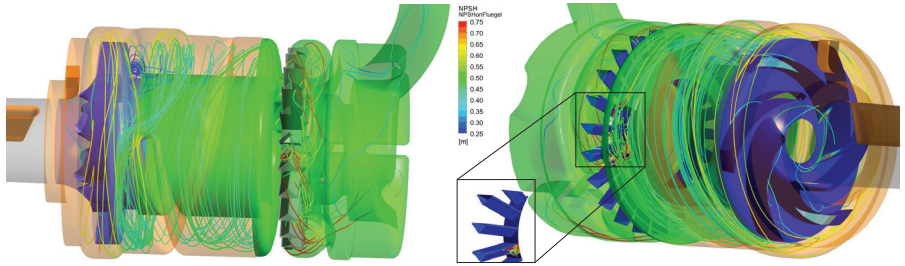


Experimental and Numerical Optimisation of Side Channel Pumps

If the choice is between a positive displacement pump and a centrifugal pump, a side channel pump is often selected. These pumps can deliver highest pressures at relatively low flow rates.

Besides the main advantages of excellent suction performance and the ability to pump liquids with high gas loads, the disadvantage of pumps with such low specific speeds is a rather poor efficiency. In the past, when efficiency was often negligible, this fact was simply accepted. Nowadays, were energy consumption being the topic of many discussions companies are also pushed by law to increase the efficiency of their products, e.g. by the European Union's Energy Efficiency Directive or various climate agreements.

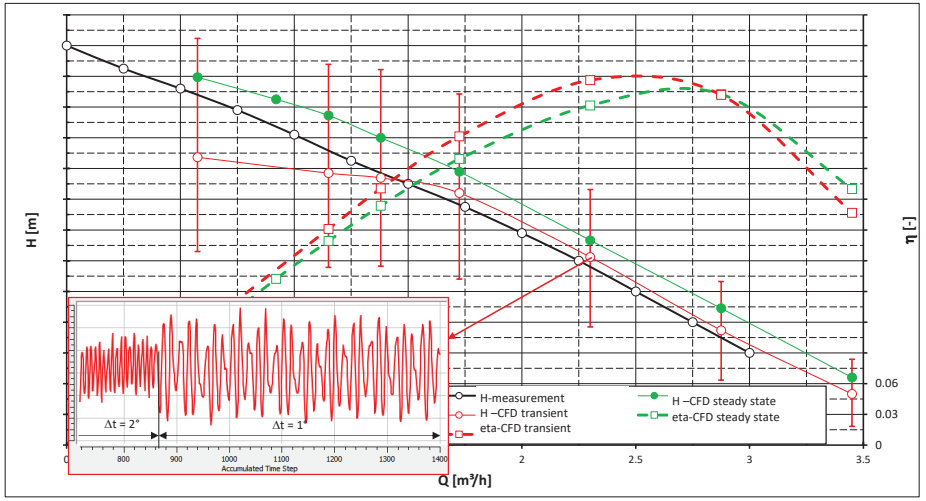
In the course of this project, single-stage side channel pumps of different size, each with an additional radial impeller were analysed in detail by means of CFD simulations, with the aim of determining the main losses and unlocking their optimisation potential. In addition to the challenge of increasing efficiency, it should be mentioned that the overall dimensions must remain identical in order to ensure the compatibility of the pump as a spare part. Another challenging task was found in maintaining the self-priming capability (suction behaviour) of the pump when optimising head and efficiency. Last but not least the cavitation performance of the pump had to be improved.



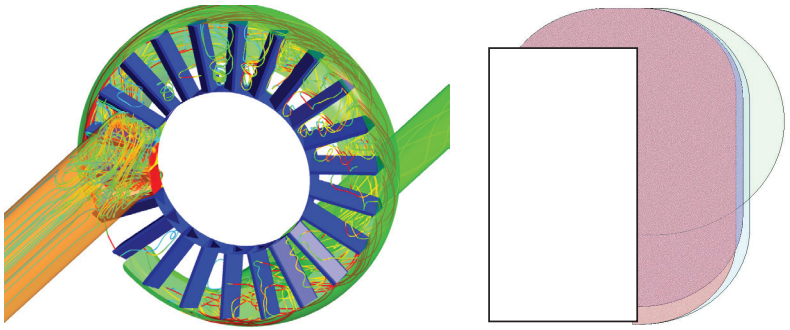
Simulation results (streamlines and pressure distribution) of the original pump design

In a first step, a numerical model was created successively and it was already clear at a very early stage that it had to include all details such as suction impeller (360° model), main stage (360° model) as well as the pressure casing and all narrow gaps in order to provide usable simulation results. The numerical simulations were performed using the commercial CFD package ANSYS CFX. Mainly structured grids were used for the impeller, the side channel, the suction nozzle and the gaps. The final model of the original pump design consisted of approx. 15 million nodes.

The CFD simulations were validated with model tests and the behaviour of the main components were analysed in detail. In addition to the loss analyses of the aforementioned components, special attention was paid to the behaviour of the fluid in the inlet and outlet duct and the pressure generation in the side duct.



Comparison of numerically and experimentally determined pump head curves



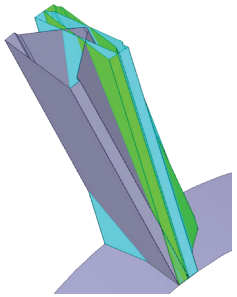
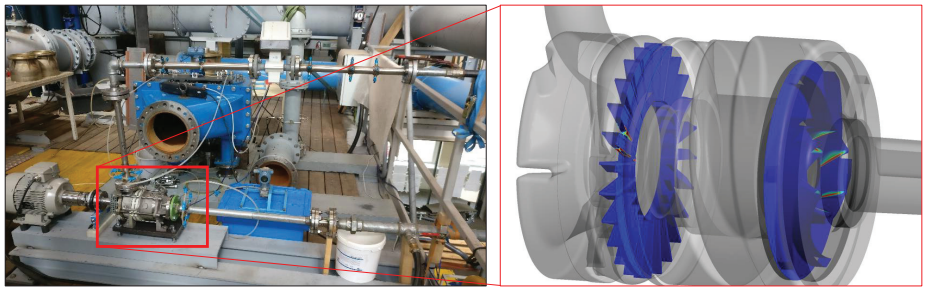


Fig.: Reduced numerical model; examples of investigated geometries for side channel and impeller

For the optimisation process, it was necessary to create a reduced numerical model to analyse the effects of dozens of geometry variations. After analysing the effects of single and multiple geometry variations, the results were combined to define the desired objectives and validated again with the full numerical model. The numerical simulations predicted a relative increase in both head and efficiency in a wide operating range of approximately 30% with respect to all given constraints.

The analysis and optimisation of the cavitation and priming performance of the pump were performed mainly experimentally with CFD-support. Complex and multi-phase flow phenomena occur during these operating conditions. However, no state-of-the-art numerical model is available nowadays to capture all effects and provide reliable numerical results in a comparatively time effort.



Test rig with mounted side channel pump and visualisation of potential cavitation zones from CFD

With the help of this hybrid optimisation approach it was revealed that the geometry of the inlet opening into the side channel stage has a very strong influence of both the priming as well as the cavitation performance of the side channel pump. In combination with a new designed radial impeller the cavitation performance could be significantly optimised together with best priming capability.

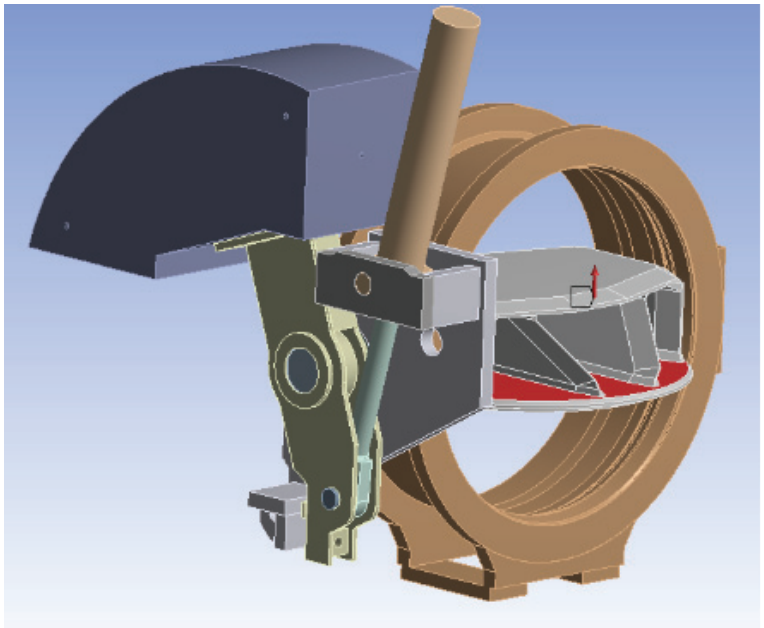
Lifetime Analysis of a Butterfly Valve for a Power Plant in Angola

In cooperation with a well-known international manufacturer of power plants and hydraulics, we have carried out a vibration analysis to calculate the resulting service life of the closing weight of a butterfly valve.

The vibrations measured on the real component were re-modelled on the 3D model, where the resulting stress deflections were then subsequently determined.

Vibrations can affect the service life of individual components in complex hydraulic systems. A simple modal analysis can be used to determine the natural frequencies. If these are close to an excitation frequency, a resonance problem can often already be avoided by changing the masses and stiffnesses.

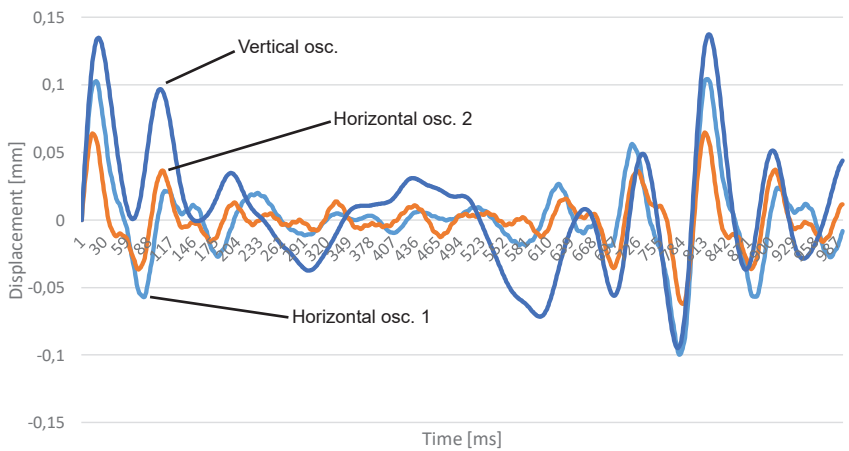
Vibrations can often occur at unexpected locations. This is what happened at a closing weight for a shut-off device – see right page. When the turbine was running, the closing weight of the butterfly valve was excited to vibrate to such an extent that it had to be feared that a fatigue fracture could occur at the lever arm, since this was only designed for a static load. In cooperation with a well-known manufacturer, we carried out a service life analysis based on real measured vibrations.



Butterflyvalve – 3D Model

Preparation and examination of the measurement data

In the first step, the measurement data supplied by the customer was processed, visualized and analyzed. The analysis of the measurement data already revealed a kind of beat, which was attributed to the fact that an excitation frequency had to be very close to a natural frequency.

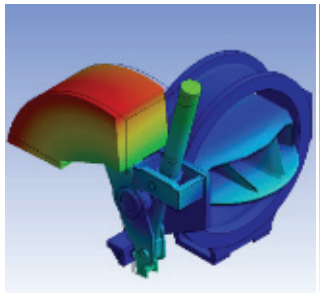


Examination of the measurement data

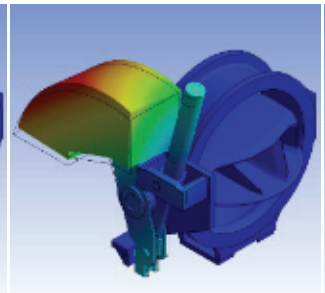
Modal analysis

A modal analysis on the 3D model yielded a total of 11 natural frequencies in the measurement range from 1-120Hz, of which the first three modes affected the closing weight at around 9, 12 and 20Hz. This also coincided very precisely with the measurement data.

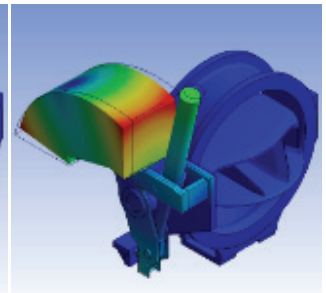
1. Natural vibration



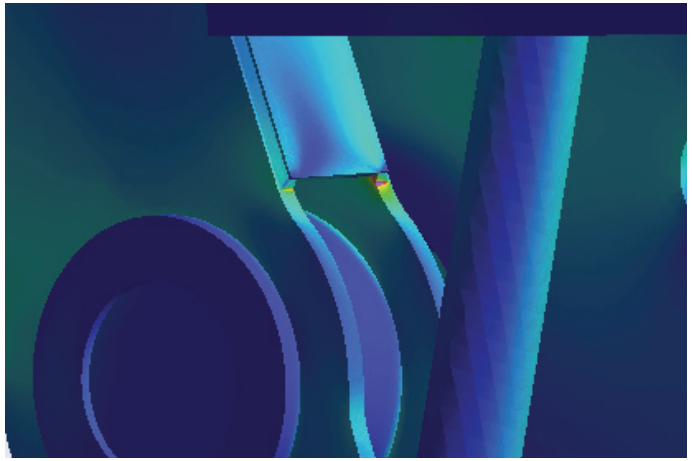
2. Natural vibration

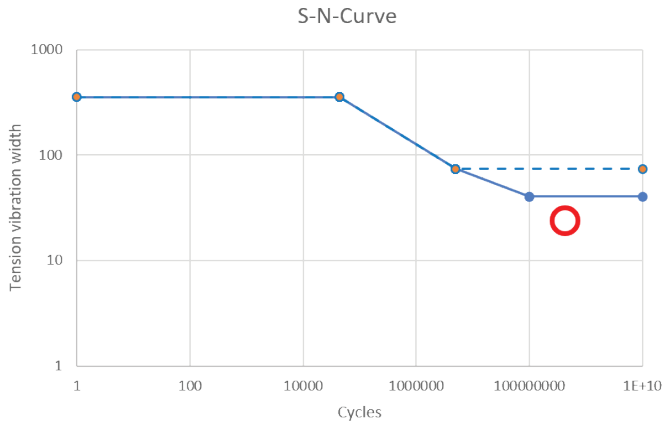


3. Natural vibration



Critical eigenmodes





Stress results and lifetime estimation

Re-modelling of the vibration

Since the source of the excitation and its frequency was not known exactly, it was assumed that the vibration was transmitted through the pipe via the valve door to the closing weight. Therefore, in the next step, we added a harmonic excitation to the valve door in the simulation, which was optimised until the resulting oscillations at the closing weight matched the measured data.

It turned out that there are 2 possible excitation frequencies that provide 2 different vibration modes, both of which fit the measurement data. This was once a circular (elliptical) oscillating motion of the closing weight and once a diagonal pendulum motion.

It was observed that the plane of the main diagonal of the elliptical oscillation was almost identical to the plane of the pendulum motion, which is why the stress maxima caused by these two oscillations were almost identical.

Stresses and service lifetime

With the determined stresses, endangered areas could be identified and a service life calculation according to DIN EN 1993-1-9 could be performed. The results show that no fatigue failure is to be expected based on the measured vibrations.

Development and Homologous Test Rig Measurements of a 6-Nozzle Pelton-Turbine

For the ongoing development and optimisation of a 6-nozzle Pelton-turbine with a vertical shaft, test bench measurements are being carried out in parallel. The test rig measurements were carried out in cooperation with the HFM-Institute.

On the 4-quadrant test rig of the HFM-Institute the horizontal 6-nozzle Pelton turbine has been investigated. These investigations include the overall performance measurements (hill chart) as well as the flow visualizations of the jet and the bucket-jet interaction. The model turbine of the test rig was built and manufactured by the industry partner. The following Figure 1 shows the parts of the 3D-turbine model test rig (top) and the turbine model (bottom). The distribution pipe casing (black coloured) was made of a two-piece aluminium block. At the top of the casing Plexiglas inlays were placed to conduct the flow visualization.

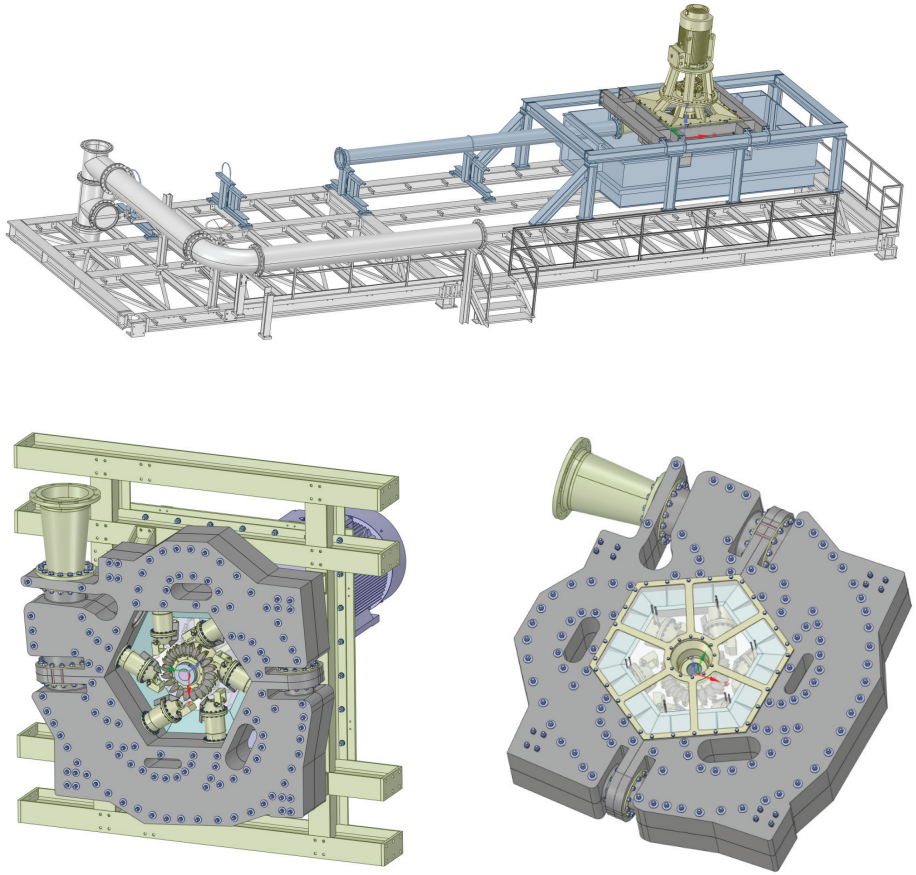


Fig.1: Pelton test rig setup (top) and detail of the turbine model (bottom)

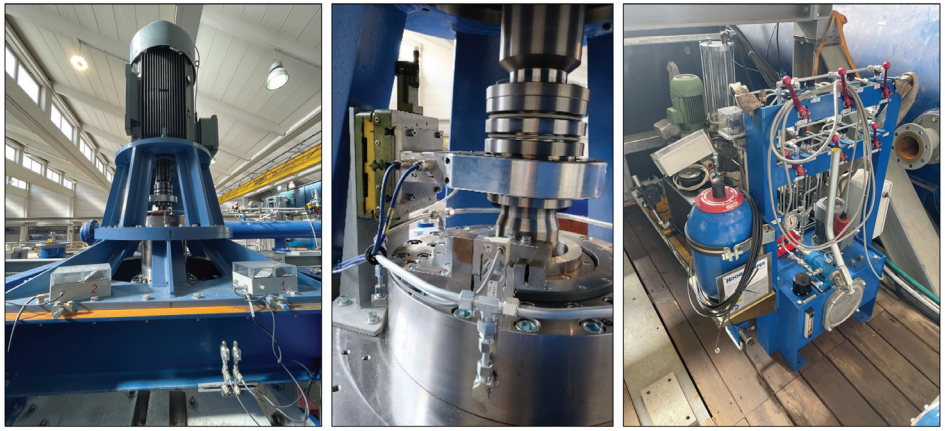


Fig.2: Motor-generator on top of the turbine (left) with torque transducer and load cell for measuring the bearing friction (middle) and hydraulic power unit for adjusting the needle position for each nozzle (right)

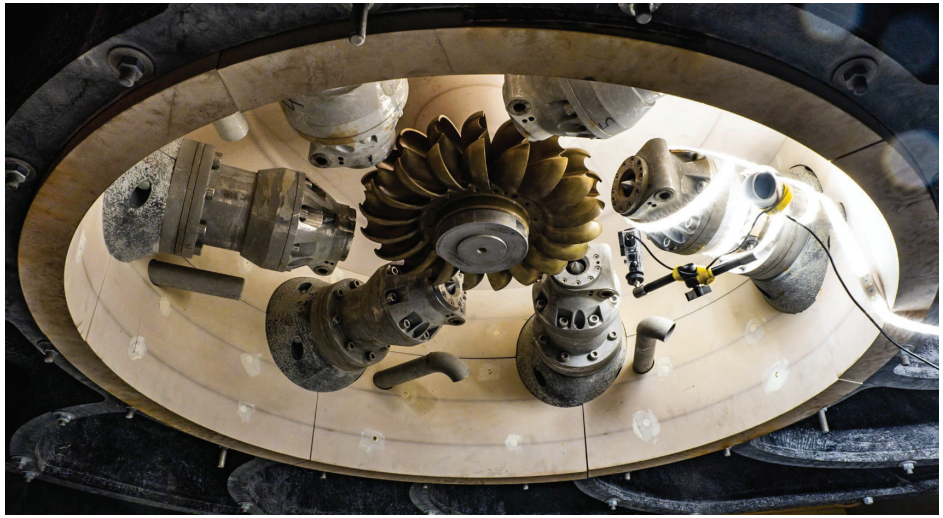


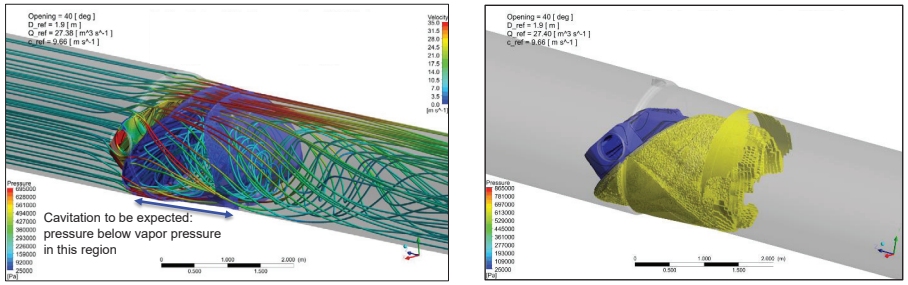
Fig.3: Example picture of an already conducted flow visualisation of a 6-jet Pelton-turbine. Courtesy: HFM-team

As this is a fully homologous model test, the nozzle needle is adjusted via the same hydraulic unit as in the prototype version of the turbine. Likewise, the position measurement of the nozzle needle stroke is carried out via laser distance measuring devices, also analogous to the large-scale turbine. Figure 2 shows the structure of the motor-generator with the torque transducer and the measuring equipment for the nozzle needle position as well as the hydraulic unit for adjusting the nozzle stroke.

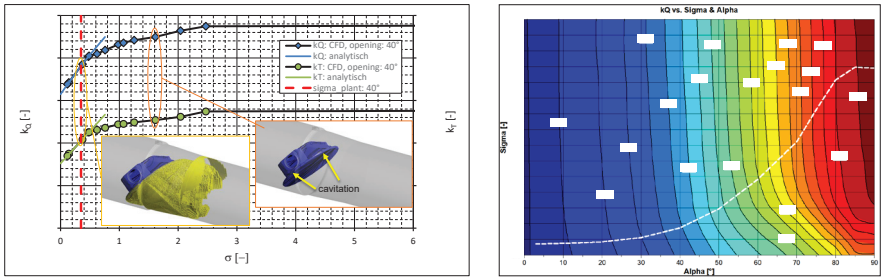
With the main pump upgrade of the 4-quadrant test rig (500kW), the test rig is now able provide the pressure and the discharge for the Pelton model for the fully-opened 6-nozzle operation (H>80m at Q=250m³/s). The test rig investigations included the following at least 3 different Turbine runner geometries (so far, potentially more runner geometries will measured in the future). Beside the measurements of the full hill charts for the turbine, flow visualisations to better understand the interaction of the runner with the water jet are planned. Further measurements with new runner geometries are planned for 2022.

Figure 3 shows one example of a deployed camera system with light stripe, which were used to observe the jet shape and parts of the jet interaction with the runner (Photo: HFM-Team).

Investigation of the Emergency Shut-Off Behaviour of a Butterfly Valve under Cavitation in the Event of a Penstock Rupture



Predicted cavitation region from single phase simulations (left) and with cavitation model



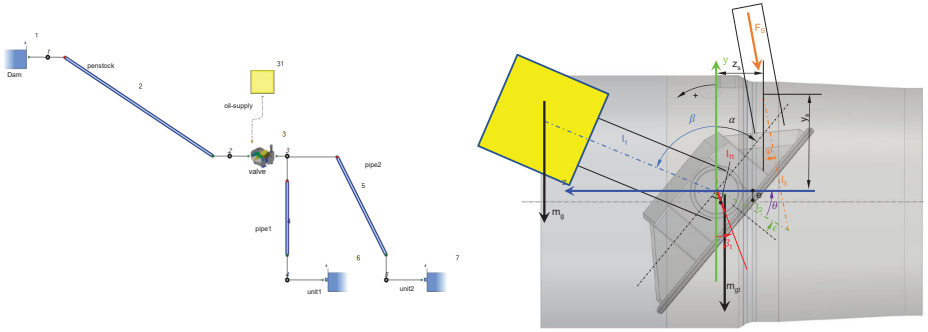
3D-CFD-results with cavitation model:
constant valve opening at different sigma values (left),
hill chart of loss characteristics kQ for all investigated valve openings and pressure levels

We are pleased that we could serve with our know-how for comprehensive flow analyses and water-hammer investigations for a power plant of Grand Dixence SA.

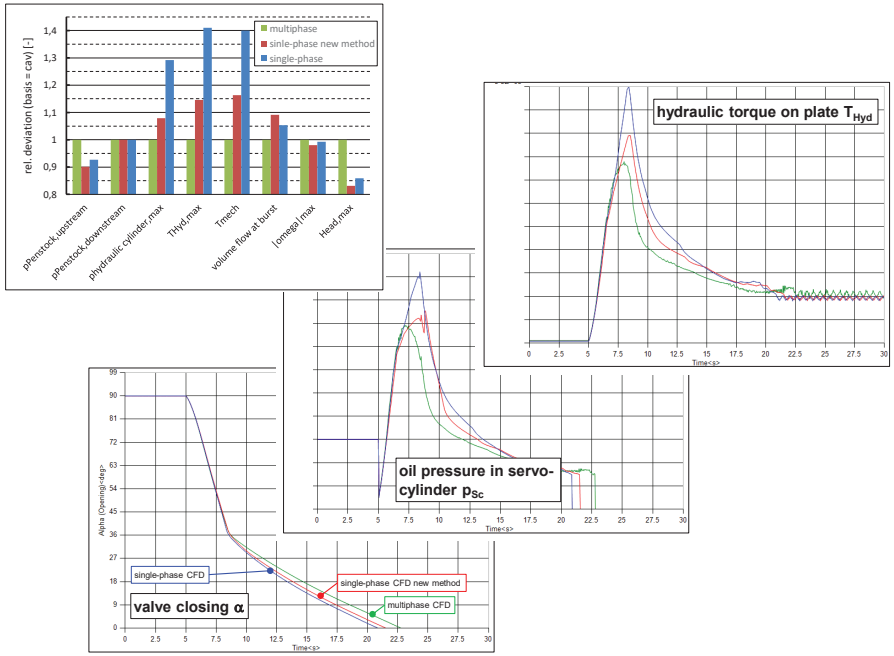
The aim of this investigation was to determine the behaviour of the butterfly valve in a catastrophic case of a penstock burst, which hopefully will never occur, but cannot be ruled out. If such a disaster occurs, the butterfly valve provides the only remaining possibility to prevent the entire volume of the dam from flooding the valley below. Since the flow-rate through the valve in case of a penstock burst will be several times higher than the nominal flow rate, cavitation is to be expected in case of an emergency closure.

Using 3D CFD simulations, the hydraulic properties of the flap during emergency closure were investigated both in single-phase and under cavitation influence (multi-phase). With these simulations the loss characteristics as well as the resulting mechanical loads in various valve positions were calculated. The simulations with the multi-phase model need to be carried out at different pressure levels i.e. different values of sigma available for each position. Hence, these simulations are very costly in terms of computational power and time.

These 3D-CFD results were mandatory information for the subsequent water-hammer investigation. With our tailor-made numerical models for hydro-power-plants we are able to calculate the forces and loads on the mechanical components in these catastrophic load case of a penstock burst.



Numerical network of the power plant for the water-hammer simulation (left) and mechanical model of the emergency shut-off valve



Results of water-hammer simulations during emergency shut-off – comparing different input data of the hydraulic valve characteristics

In the investigated emergency load case it was assumed that a simultaneous pipe rupture occurs at all pumps downstream of the butterfly valve. In order to determine the dynamic closing behaviour of the valve in this event and the resulting loads respectively, the headrace upstream of the valve to the dam and downstream to the respective rupture areas had to be modelled. The numerical model of the butterfly valve was created separately for this project since the cinematics had to modelled correctly. To calculated the dynamic behaviour of the valve in case of an emergency shut-down, all loads acting on the mechanic structure had to be considered. To do so, the torque equilibrium around the valve axis had to be evaluated for each time step in the simulation of the emergency shut-off.

To reduce the numerical effort in gaining the valve characteristics under cavitation a method to estimate these data out of single-phase 3D-CFD results was implemented in the course of this project. It could be shown that the loss in accuracy with the new method is considerably low compared to the costly multiphase simulations. On the other hand, if cavitation is not considered at all, the calculated load would be far too big. The new method could therefore be validated as a reasonable compromise considering calculation accuracy and numerical effort.

Hydraulic Development of the Kaplan-Turbine of HPP-Gratkorn

In the north of the city of Graz, the largest operator of hydropower plants in AUSTRIA (VERBUND) is going to build the new power plant HPP-Gratkorn at the river of Mur. The invitation to tender was provided for two vertical Kaplan turbines, as the construction costs of which are lower than those of the bulb turbines frequently used on the river of Mur.

On behalf of the company manufacturing the new Kaplan-units, Jaberg & Partner carried out a CFD-based development of the turbine hydraulics. The special challenge was the design of an efficient and compact vertical Kaplan-turbine with a comparatively high specific speed n_q for highly restricted geometry ratios, which were specified by VERBUND.

For the design of the turbine, a nominal head of H = 6.5 m (mean head for the two-unit operation) and a maximum flow rate of Q_{Max} = 102.5 m³/s were used. The speed was set to n = 107.14 rpm. The Best Efficiency Point should be at Q_{Opt} = 0.685 * Q_{Max} = 70 m³/s. The specific speed n_q is therefore calculated as follows:

$$n_{q,BEP} = n \cdot \frac{\sqrt{Q_{Opt}}}{H^{0,75}} = 107.14 \cdot \frac{\sqrt{70}}{6.5^{0,75}} = 220.2\ U/min$$

Eq.1

Experience has shown that the maximum achievable specific speed n_q-value for vertical Kaplan turbines in relation to the Best Efficiency Point of the hydraulics is about 180 rpm. This value is limited by the head losses occurring in the elbow-type draft tube, which steeply increase with increasing discharge and significantly reduce the efficiency, especially at low head. The Best Efficiency Point of the new Kaplan-design will therefore probably be at a somewhat higher head.

In order to achieve the Best Efficiency Point at Q_{Opt} = 70 m³/s and the highest possible full-load efficiency, it was necessary to find the most efficient impeller design possible, which needed to be ideally matched to a tailor-made draft tube. For the custom-designed new Kaplan-turbine, CFD simulations were used.

The initial design proposal from the customer was scaled to the required runner diameter of D = 4 m and the overall turbine design was optimised step by step. Following design features were required to reach the challenging targets of the project:

- The tip-clearance was adjusted with s = 2 mm which is equivalent to 1/2000 * D.
- Reduction of the hub ratio of the turbine in two steps from 0.4 to 0.365 and later to 0.35. Thus, the full load efficiency as well as the cavitation behaviour could be improved. After all, the critical position in terms of pronounced cavitation is the hub region of the blade!
- A customized optimisation of the draft tube design was required. Step by step the geometry of the draft tube was adapted to the design specifications given by VERBUND – especially regarding the maximum construction depth, the maximum width, the position of the outlet cross section and the twist from the axis of symmetry.
- Increase of the originally planned guide vane height from b_0 = 1505 mm to b_0 = 1700 mm (approx. 13% more). Furthermore, the pitch diameter of the guide vanes was slightly shifted outwards. Both measures contribute to the improvement of the efficiency. Note: An even larger pitch circle diameter would have been beneficial to increase the turbine performance. However, this conflicts with the limited maximum installation width of the spiral housing, which leads to an already high inlet swirl at the inlet of the guide vanes which increases the losses there.

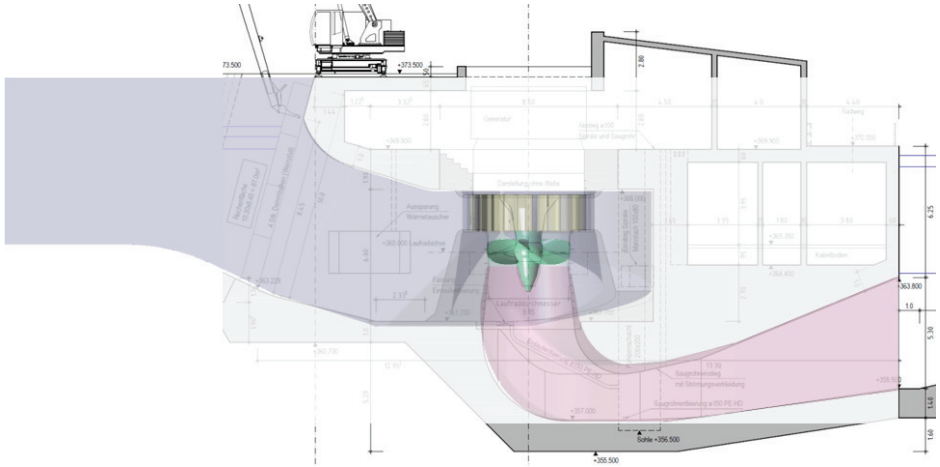


Fig.1: Superposition of the structural limitations given by VERBUND (grey colours) and the final turbine design (bright colours)

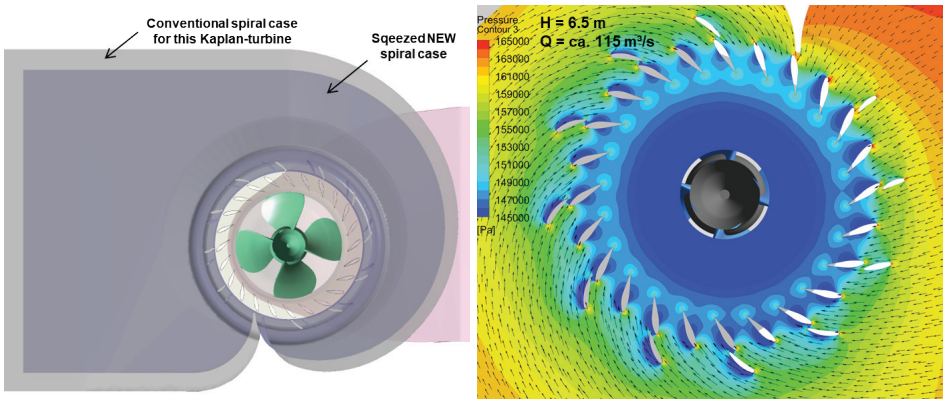


Fig.2: Visualizations of the spiral case
Left side: Superposition of a conventional spiral case and the new custom-designed spiral case
Right side: Velocity distribution plotted on a mid-section in the spiral case at high load operation

- In order to minimize the flow losses in the spiral case and to reduce the swirl at the inlet of the guide vanes (especially important to reduce the losses at high-load operation) the dimensions of the spiral case had to be maximized, taking into account the structural limitations.

Figure 1 presents a cross-section of the powerplant in side view with a superposition of the structural limitations given by VERBUND (grey colours) and our final turbine design (bright colours).

Furthermore, the given design specifications caused a reduction of the construction width of the power plant. Figure 2 presents a superposition of a conventional spiral case for this Kaplan-turbine and the new custom-designed “squeezed” spiral case design (left side). Not only the height of the spiral case needed to be reduced (from H = 6.75 m to H = 6.25 m) but also the width (from B = 12.75 m to B = 10.5 m). This design deviation results in an increased swirl generated by the spiral case. As consequence, the inflow conditions are not ideal (see right side) which results in comparatively high losses.

The blade geometry was optimised to achieve minimum losses and a sufficiently good cavitation behaviour at full load. Therefore, the ideal inflow conditions of the blade profiles are achieved at full-load operation. Referring to the full load operation point at H = 6.5 m, the Figure shown below presents the pressure distribution plotted on the suction- and pressure-side of the final runner blade V42. The regions most prone to cavitation are located at the blade tip close to the trailing edge and especially at the blade root close to the hub.

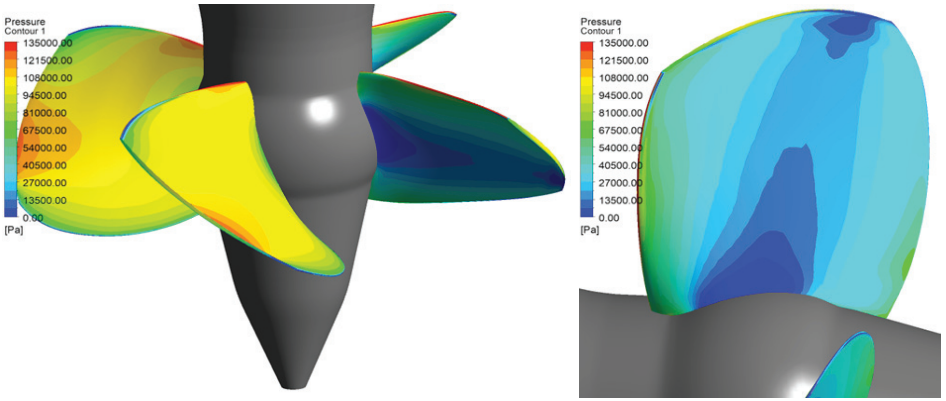


Fig.3: Pressure distribution plotted on the runner blade at high load conditions

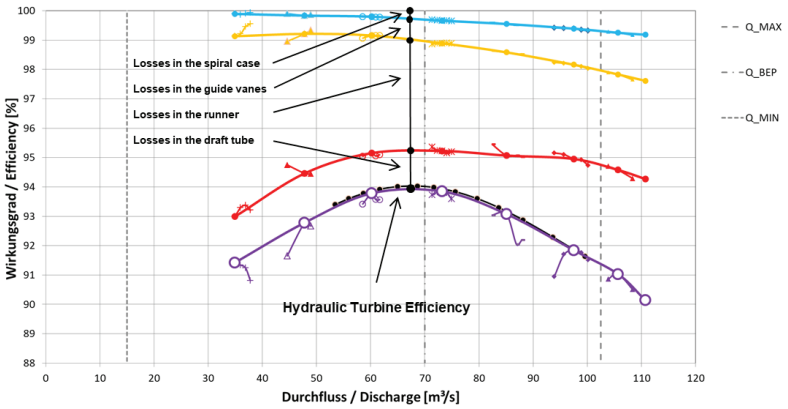
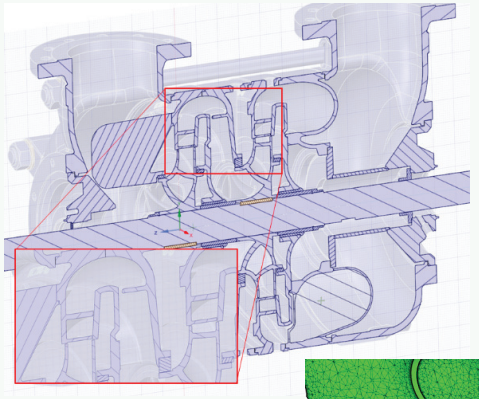
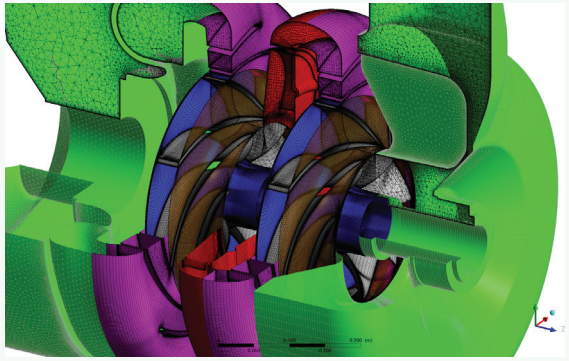


Fig.4: Hydraulic turbine efficiency and loss composition of the final turbine design at H = 6.5 m

At this point, it should be noted that the runner blade optimisation was necessarily carried out together with the optimisation of the draft tube. Every change in the blade geometry causes a change in the draft tube flow, which of course again affects the overall turbine performance.

Finally, the hydraulic turbine efficiency achieved in course of the optimisation as well as a loss composition for all components of the turbine is presented in the diagram shown below. The presented results show, that the guaranteed peak efficiency as well as the overall shape of the efficiency curve was exactly met with the customized turbine design. To achieve the targets, the runner design needed to be optimised for high load operation, which is clearly visible with the decreasing runner losses with increasing discharge. Referring to the full load operation point (Q = 102.5 m³/s) the majority of losses results from the flow in the draft tube.



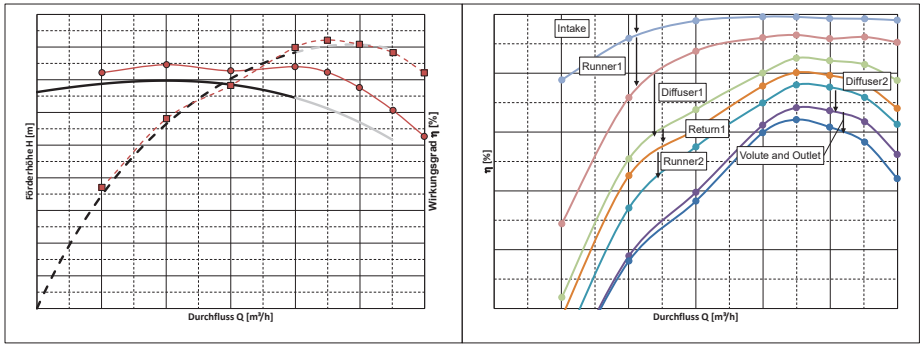


CAD model of the pump (two-stage model) and corresponding CFD model

Optimisation of a Multi-Stage Centrifugal Pump w.r.t. Head Curve Stability, Efficiency and Cavitation

In the course of the present project, a multi-stage ring-section pump was analysed in terms of its optimisation potential. Multiple targets had to be met: maintaining a stable head curve in combination with increasing head and efficiency as well as improving the cavitation behaviour. It should be mentioned that the overall dimensions must remain identical in order to ensure the compatibility of the pump as a spare part. An extracted fluid volume with many details was necessary to enable the highest possible accuracy of the simulations. A comparison with existing experimental results enabled the possibility to ensure a maximum reliability of the numerical results.

The original geometry of the pump was rebuilt from a CAD model and meshed using “reverse engineering”. Two models were generated and analysed before optimisation started. The single-stage model consists of 1 runner and 1 guide vane (each as a 360° circular segment). The double-stage model consists of an additional return part with 8 blades after the guide vane and a second runner and second diffusor. Each model also includes a suction side and a spiral and outtake section with a full 360-degree model. To analyse each component separately, a head loss analysis was performed to calculate a cumulative distribution of the total unit.

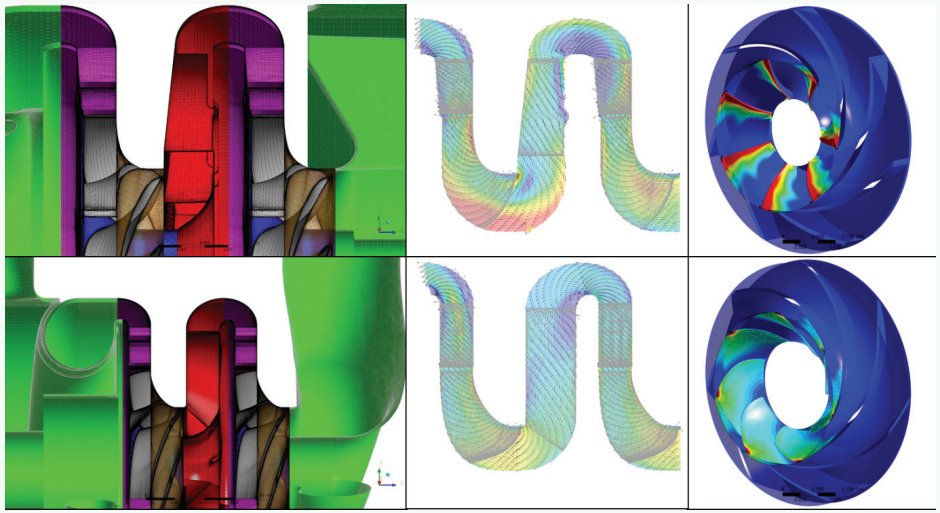


Comparing numerical and experimental results for the initial geometry and corresponding loss analysis

Visualised vortex structures at different operating points (part load, optimum and overload)

With the help of the Q-criterion Vortex structures could be visualised within the transient calculations of the initial situation. These vortices showed a high dissipation rate in the suction area at the lowest flow rates and in the pressure area at the highest flow rates. The area of the guide vane, on the other hand, is filled with vortex structures whose intensity, however, is significantly lower.

The multi-stage centrifugal pump optimisation was mainly done in a simplified model to minimise the calculation effort. Impeller, inflow area and volute were optimised manually. Automated optimisation was realised for the guide vane and return channel. A meta-model assisted by multi-objective optimisation method with evolutionary algorithms was applied to find the best hydraulic geometry.



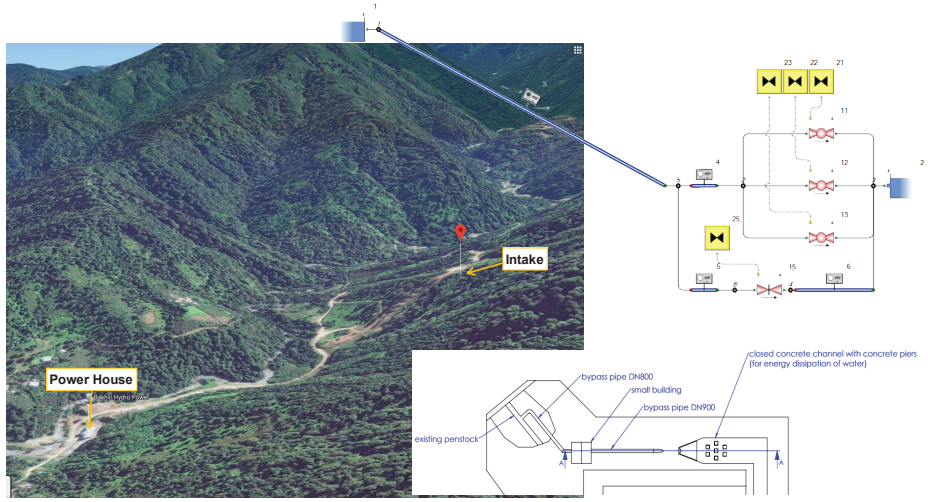
Comparison of existing geometry (top) and optimized pump design (bottom)

It turned out that the head curve stability is mainly affected by the shape of the intake part. The spiral outlet was left at the lowest point for self-priming. The new shape for both, the guide vanes and the return vanes are the result of the parametric optimisation method. There were about 50 geometric degrees of freedom. The outer diameter of the stage dramatically influences the head and efficiency at all the operating points studied and it is found: the lager, the better. The diameter of the trailing edge of the return blades has a major influence on the head and efficiency of the downstream impeller. Here, it turns out: the smaller, the better.

An efficiency breakdown of the optimised two-stage pump geometry and a comparison with the original geometry were made as well. As a result of the optimisation, the efficiency is increased in the entire operating range; in the single-stage variant by approx. 10%, and in the two-stage variant by approx. 7%! Hence the geometric specifications of the pump connection dimensions (retrofit) are adhered to, and the characteristic stability based on the CFD simulations is improved.

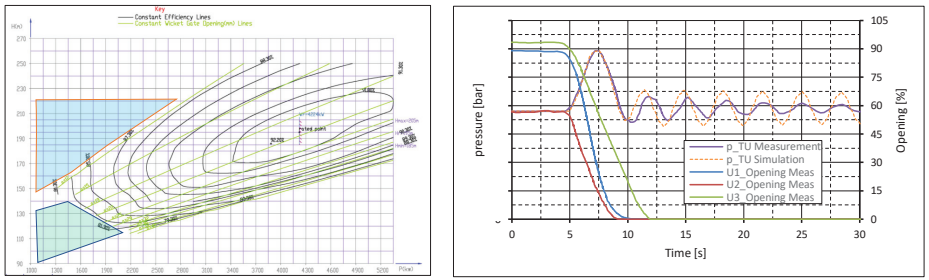
Comprehensive Hydro Power Plant Analysis: Water-Hammer Investigations in Combination with Economic Assessment of Unit Replacement

We were commissioned to investigate a pressure-peak problem and correspondingly develop a suitable remedy for a hydro power plant in Georgia. It is equipped with 3 Francis units operating at a net Head of 195 m. In case of an emergency shutdown from more than 50% load, there existed a serious water-hammer problem. The maximum allowable pressure was exceeded by more than 20% in case of an emergency shut down from full load operation. Additionally, the runaway speed exceeded a critical limit in this case. The initial solution suggested by the plant operator was to install a bypass to reduce the pressure peak in case of a turbine shutdown.



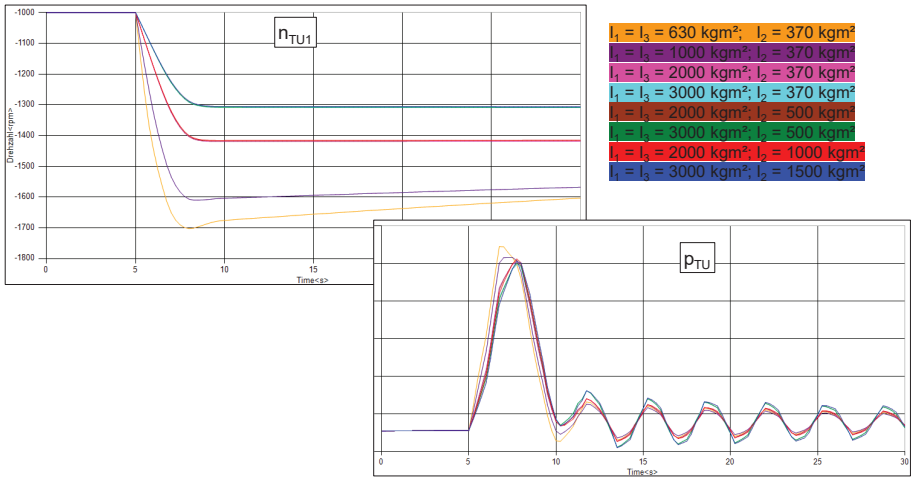
Project site and numerical model

Unfortunately, no reliable performance data of the installed turbines and generators (inertia) were available. The accessible hill chart did not provide sufficient information for a reliable transient power plant analysis. Only the turbine behaviour near normal operating conditions is documented. Regions of high interest in case of an emergency shut down were missing. But there were measurement data from shut-off tests. Hence, a calibration of our numerical model for the transient analysis of the initial situation and the investigation of the effect of the bypass on exceeding the maximum permissible pressure was possible. For this purpose, the performance characteristics of a Francis-Turbine from our database with similar specific speed had been used.



Available hill-chart and simulation results of calibrated numerical model compared to measurement data

Another issue in the available data was the unknown mass and inertia of the installed generators as well as uncertainties in the closing speed of the guide vanes. However, both (generator inertia and guide vane speed) are essential information for an accurate calculation of the maximum speed and pressure values in case of an emergency shut down. In the end, several hundred simulation runs were performed to calibrate the numerical model of this hydro power plant. The results of the calibrated numerical model showed very good agreement with available measurement data for all investigated load cases.



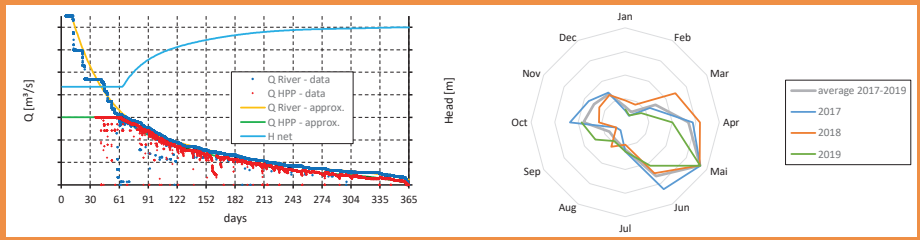
Variation of Inertia has a huge effect on nmax but very little influence on pmax

As the presently installed Francis-Turbines would have to undergo comprehensive maintenance and repair in the near future, an alternative possibility that would solve the water-hammer problem at the same time avoiding a by-pass valve was found in modifying the closing behaviour of the guide vanes. The analysis showed, that by increasing the guide vane closing speed a safe power plant operation at full load is possible. However, a standalone operation of unit one in it's existing configuration was not feasible with the limitation of the maximum allowed penstock pressure. Since the operating conditions were in general not ideally for the installed turbines their replacement possibly stepwise by Pelton-Turbines was investigated additionally.

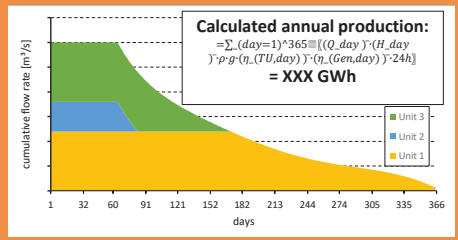
To analyse different machinery concepts in terms of their economic efficiency, hydrology and production data was analysed. Based on the submitted data an annual load duration curve was determind. From these data it was clearly obvious that there is potential in unit operation to gain maximum power production. Several times, the smallest unit was not in operation but the power plant output was below the maximum power output of this particular unit. This means that in those cases the remaining units were operating in deep part load (P < 50% Pmax) with low efficiency and other known issues like high mechanical load, vibrations, fatigue, etc.

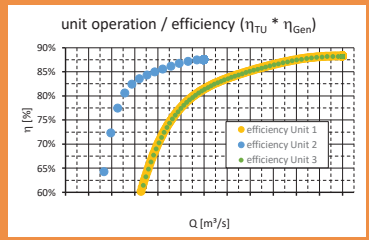
Approximation curves for both the river flowrate and the power plant flowrate were generated based on the submitted data set. The net head was then calculated according to the losses in the penstock and the distributor. These losses were known from the shut-down measurements and were assumed to be the same for each unit. Since no detailed information was available, the tailwater level was treated to be constant for every flowrate. In combination with constant water level in the intake a constant gross head for all operating conditions resulted.

A daily based calculation of the annual electricity production was then applied to the existing machinery concept and compared with the available averaged measurements. The power plant flow rate was split between the units and the turbine efficiency for each individual unit was determined based on the available hill-chart. The generator efficiency was calculated based on the turbine output and reference generator data from our database. In the end, a summation of 365 determined individual daily values results in the calculated annual production.

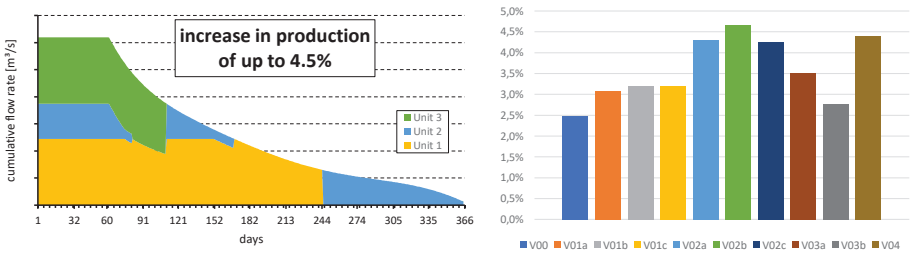


Calculated annual duration line and net head as well as monthly production





Calculation of annual production



Additional annual production of different investigated new machinery concepts

Different aspects concerning the installation of a Pelton-Turbine had to be considered. No adjustments to the existing penstock were permitted and the civil work had to be kept as low as possible. Compared to the existing Francis-Turbines an overhung to the tailwater level is necessary which reduces the available net head. The weight of any new unit must not exceed the limit of the power house crane. This is of special interest when selecting the generator speed of the new unit which itself has a huge impact on the machinery concept. The maximum power generation of the power plant must remain below a certain limitation due to existing electrical equipment (transformer) as well as for contractual reasons.

Considering a replacement of the smallest unit only (with a new Pelton-Turbine) and keeping the other units in existing configuration, it was found out that there is no markable difference in terms of annual production if the maximum flowrate of the new Pelton-Turbine would be increased compared to the initial situation. If all existing units would be replaced with new turbines and generators, the best concept in terms of annual production was found to be 2 Francis-Turbines and one Pelton-Turbine.

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pump.ing
Vocational Training for Pump Experts

pump.ing. is an extra-occupational distance learning study programme for pump-specific expertise regarding pump technology, operational issues and systems. The study course lasts 400 hours and is divided into the chapters “Pump basics”, “Pump unit”, “Pumps in systems” and “Fields of application (optional subjects)”. Out of eight arbitrary fields of application offered (process engineering, refineries, power plants, water, sewage, food and sterile industry, paper industry and vacuum technology) two more subjects have to be chosen according to individual interest or industry.

Via the TU Graz teaching and learning platform TU Graz TeachCenter all students have access to the study material and are totally free with regard to their self-study time management. In addition, they have to participate in eight attendance events of two days each (Friday and Saturday) at various locations in Austria and Germany. The attendance event “Pump assembly” takes place at the laboratory of the Institute of Hydraulic Fluid Machinery at Graz University of Technology. The duration of the course is about 14 months. The acquired knowledge on the optimum design and operation of pumps and systems and the ability to conduct energy counselling are documented by the certificates “Certified energy consultant for pumps and systems” and “Pump Engineer” or respectively “Pump technician”.

Practical training by renowned pump specialists

The German study course for pump engineers starts every year in July. In 2022 already the 18th study course has been launched. In 2005 the idea of setting up a special study programme for pump specialists arose, on the one hand, on the initiative of Professor Helmut Jaberg, Head of the Institute of Hydraulic Fluid Machinery at Graz University of Technology.

On the other hand, its setup was based on considerations by the pump section of the German Association of the chemical industry VCI with its then chairman, Dr. Friedrich Wilhelm Hennecke, longstanding head of BASF’s Pump Centre.

At the annual Practitioner’s Conference on pumps – “Pumpen in der Verfahrens-, Abwasser und Kraftwerkstechnik” – held in Graz, the participants were asked about the project to install the study programme “Pump engineer” respectively “Pump technician” (depending on the level of professional qualification a participant presents at the beginning of the programme). The proposal was received enthusiastically. Manufacturers were hooked and operators could finally acquire specialist knowledge about pumps. Moreover, plant operators were very pleased as they could finally count on qualified pump salesmen. Fortunately, participants from both the manufacturers’ and the operators’ side attend the study course for pump engineers in large numbers. Prof. Helmut Jaberg and Dr. Friedrich Wilhelm Hennecke also brought Professor Paul-Uwe Thamsen from the Technical University of Berlin and Dr. Walter Schicketanz, then head of a planning unit at BASF, on board.

pump.ing – theory and practice in the laboratory

The Pumpenfachingenieur GesmbH was founded and a high-ranking management board, in which the pump section of the the VCI is still prominently represented, was installed and has since then defined the teaching content. The first study course was launched in 2005 with around 30 lecturers – all of them leading pump experts with well-founded and relevant industrial experience. More than 500 pump experts from the top industry representatives have since been exclusively trained as pump engineers and pump technicians. Especially in the last three years the German-speaking pump engineer achieved record numbers of up to 49 participants.

pump.ing. is highly appreciated by the industry. The practical relevance and the high standard of the teaching content probably contribute to this. Ambitious goals and challenges – also of international significance – are a welcome incentive for us, and of course we aim at keeping the programme at the latest state of the art.

Attendance event

Pump assembly in the laboratory

pump.ing. – International study course in English

Since 2012 already five international study courses for pump engineers have been launched. The idea of offering the course for an international audience is based on the demand of globally operating companies wishing to employ qualified pump experts worldwide. Companies such as Netzsch, Sulzer, Andritz AG, Shell Petroleum, Grundfos, Egger Turo Pumps are just some of our international customers. The English-language programme corresponds to the syllabus of the German study course for pump engineers.

Attendance event of the English language study course

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Our Two Practitioners' Conferences

Practitioners' conferences are an integral part of Jaberg & Parnter's calendar of events. The attractiveness of the concept developed by Prof. Jaberg in the last 26 years is unbroken, as each conference attracts up to 150 participants to visit Graz. In autumn 2021, the "Pumps in Process Engineering" practitioners' conference celebrated its 25th anniversary. Its mission statement remains "by practitioners for practitioners" and means that renowned lecturers from the industry will take part in the programme – manufacturers and planners as well as operators. The fact that lectures have to be rejected every year because the agenda is overflowing proves the importance of the conference in the German-speaking world as a forum for exchange between experts in the field of hydraulic fluid machinery. Mario Hübner, e.g., emphasizes "I am not aware of any similar successful event in the entire pump industry in German-speaking countries." He is the responsible Manager System-Engineering at WILO SE and has already spoken at several conferences.

Practioner conference Graz 2021 - Pumps in process engineering

What distinguishes practitioners' conferences from similar events? First of all, the speakers come from the front row of companies, the presentations are given by leading engineers and the focus is on technical topics. The generous time frame allows to go into depth during the lectures and do not just remain on the surface of challenging points. Another advantage is that the conference is held in German. And this will not change, despite a growing number of visitors and speakers from non-German speaking countries: An open dialogue is important to us, and the lively discussions after the lectures confirm that this path is well accepted.

Presentation and open dialogue between speakers and attendees

Knowledge exchange and discussion

Every year, representatives of operators find the time to report on their experiences and problems. After all, the attending audience of experts could often provide suggestions for essential solutions to solve various operating problem. Since 2020, the practitioners' conferences have also been offered online. An additional plus to be mentioned is the framework of the conferences: a trip to Graz should be worth it in every respect! "Regular guests" never leave Graz without a bottle of pumpkin seed oil, and the atmospheric evenings in the historic buildings of Styria and the city of Graz offer plenty of time to pursue ideas which emerged during the day and make new contacts.

Practioners' Conference Graz 2021 - Pumps in process engineering

Late afternoon
guided tour through Graz's old town

Practioners' Conference
Congress Graz

Practitioners' Conference
"Pumps in process engineering"

The conference "Pumps in process engineering" organised by practitioners provides a unique platform for practitioners to share experiences, new developments and ideas for coping with the everyday challenges of pumps and pump systems.

The Practitioners' Conference Pump especially addresses operators as well as planners and manufacturers of pumps and pump systems.

Praktikerkonferenz
Graz

Practitioners' Conference Graz
"Pumps in process engineering"
Graz, Austria
More information and registration:
www.praktiker-konferenz.com

Practioners' Conference Hydropower Graz 2021

Practioners' Conference Hydropower at the Graz University of Technology

Presentation

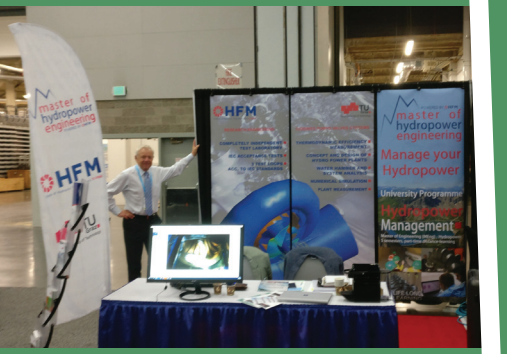
Practitioners' Conference "Hydropower"

This German-language conference provides the ideal platform to build up networks as well as for the exchange of experience and knowledge by practitioners for practitioners – manufacturers, operators, planners and researchers – focussing on hydro power plants and core technical / economic topics such as turbine efficiency, pressure surge, turbine in- and outtake, pump storage, control units, surge chamber, penstock, service life, refurbishment etc. covering the range from small to large hydro power.

Praktikerkonferenz Wasserkraft

Practitioners' Conference
Hydropower | Turbines | Systems
Graz, Austria
More information and registration:
www.wasserkraft-graz.at

PUBLICATIONS & CONFERENCE CONTRIBUTIONS



Prof. Helmut Jaberg at the HYDROVISION

Presentation by Stefan Höller
at the Practitioners' Conference "Pumps in process engineering"

Prof. Helmut Jaberg & Christian Bodner
at the Jaberg & Partner Booth at the HYDRO

Publications

2021

Mehrziel-Optimierung von mehrstufigen vertikalen Halbaxialpumpen für API-Anwendungen - Der Weg zum stabilen Pumpbetrieb

Höller, S., Schiffer, J, Oct 2021, In: Chemie Technik, 50.Jhg. 10/2021, p.32-34

Sicherer Anlagenbetrieb durch Druckstoßuntersuchung

Höller, S., Schiffer, J,, Oct 2021, In: CIT Plus, 24.Jhg, 10/2021, p.28-31

Doppelt geregelte Diagonalturbine für Kleinwasserkraftanwendungen

Schiffer-Rosenberger, J., Pretis, G., Benigni, H. & Jaberg, H., 17 Sep 2021, In: Wasserkraft & Energie. 21, 3, p. 2-11

2020

Assessing the hydraulic behaviour of a reverse radial gate with a flap

Höller, S., Jaberg, H., Galehr, J., Colic, D., Feb 2020, In: The International Journal on Hydropower & Dams. 27, 2, p. 59-64

Drehzahlvariable Diagonalturbine - von der hydraulischen Entwicklung bis zur Umsetzung

Schiffer-Rosenberger, J., Benigni, H., Jaberg, H., Fella, G., Winkler, C. & Winbeck, M., 1 Apr 2020, In: Wasserkraft & Energie. 26

Turbineninnovation in steirischem Kraftwerk sorgt für Aufsehen in der Branche

Schiffer-Rosenberger, J., Benigni, H., Jaberg, H., Fella, G., Winkler, C. & Winbeck, M., Okt 2020, In: ZEK Fachmagazin für Wasserkraft

Research for the upgrade of a Kaplan turbine at Krippau, Austria, and operational experience

Benigni, H., Jaberg, H., Becker, M., Krappel, T., Juhrig, L., Schiffer-Rosenberger, J., Penninger, G., Artmann, M. & Weichselbraun, C., 20 Dec 2020, In: The International Journal on Hydropower & Dams. 27, 6, p. 74-80

Conference Contributions

2021

Druckstöße in Pumpsystemen und Kraftwerksanalgen – Praxisbeispiele

Höller, S., 6 Sep 2021, Praktikerkonferenz Graz Pumpen in der Verfahrenstechnik, Kraftwerks- und Abwassertechnik. Graz: Prof. Dr. Jaberg & Partner GmbH

Schadensanalyse und Laufradoptimierung am Beispiel einer Francis-Turbine mit starken Kavitationsschäden

Schiffer-Rosenberger, J., Benigni, H., Prirschl, R., Giersemehl, I. & Jaberg, H., 7 Aug 2021, Tagungsband Anwenderforum Kleinwasserkraft. Pforzheim: Conexio PSE, p. 47-52

Kostenoptimierte Modellversuche für die Kleinwasserkraft

Benigni, H., Leithner, S., Bodner, C., Jaberg, H. & Meusburger, P., 8 Aug 2021, Tagungsband Anwenderforum Kleinwasserkraft. Pforzheim: Conexio PSE, p. 104-117

Numerische Untersuchungen des Turbinendurchganges von Fischen

Schneider, J., Benigni, H., Jaberg, H., Zenz, G., Tuhtan, J. & Reckendorfer, W., 2021, Wasserbau-Symposium 2021: Wasserbau in Zeiten von Energiewende, Gewässerschutz und Klimawandel. Boes, R. (ed.). Zürich: Eigenverlag der Versuchsanstalt für Wasserbau, Hydrologie und Glaziologie, ETH Zürich, p. 477 - 485 (VAW-Mitteilungen; vol. 263).

2020

Downstream Fish migration in a Kaplan turbine – Part 2: Simulation practices and new post-processing method

Benigni, H., Schneider, J., Reckendorfer, W., Zenz, G. & Jaberg, H., 26 Oct 2020, Conference Proceedings Hydro 2020. The International Journal of Hydropower and Dams, Vol. 2020. p. 1-6

Cavitation as a limiting factor for the empowering of a Kaplan turbine – CFD calculations, test rig results and operational experience

Benigni, H., Schiffer-Rosenberger, J., Penninger, G., Weichselbraun, C., Artmann, M., Juhrig, L., Becker, M., Krappel, T. & Jaberg, H., 26 Oct 2020, Conference Proceedings Hydro 2020. p. 1-10

Potenzial für Wasserkraft & Energiespeicher

Benigni, H. Höller, S., Feb 2020, Proceedings des 16. Symposium Energieinnovation

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