Task 3.1

Title

Innovative technologies

Projects (presented on the following pages)

DuoTurbo : Pilot Plant Commissioning and Monitoring
D. Biner, V. Hasmatuchi, C. Münch-Allігné

Prediction of unstable full load conditions in a Francis turbine prototype
J. Gomes Pereira Jr., E. Vagnoni, A. Favrel, C. Landry, S. Alligné, C. Nicolet, F. Avellan

RENOVHydro: Methodology to determine the parameters of the hydraulic turbine governor for primary control
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Numerical modelling of fish guidance structures
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DuoTurbo : Pilot Plant Commissioning and Monitoring

D. Biner [1], V. Hasmatuchi [1], C. Münch-Alligné [1]

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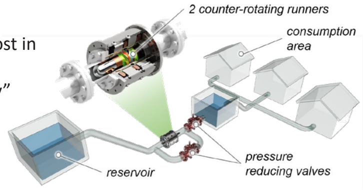
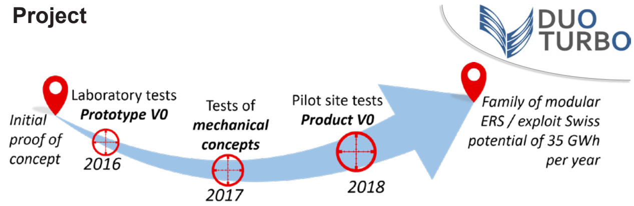
Context

- Recovering hydraulic energy lost in **drinking water networks**
- Modular in-line "plug and play" turbine from **5 to 25 kW**
- No environmental impact
- Low investment costs

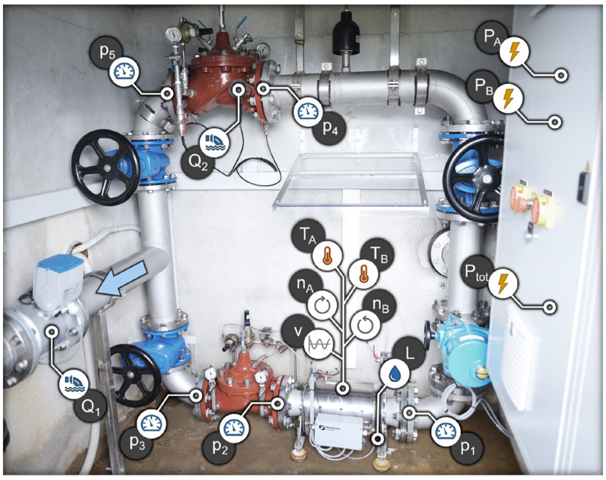


Project



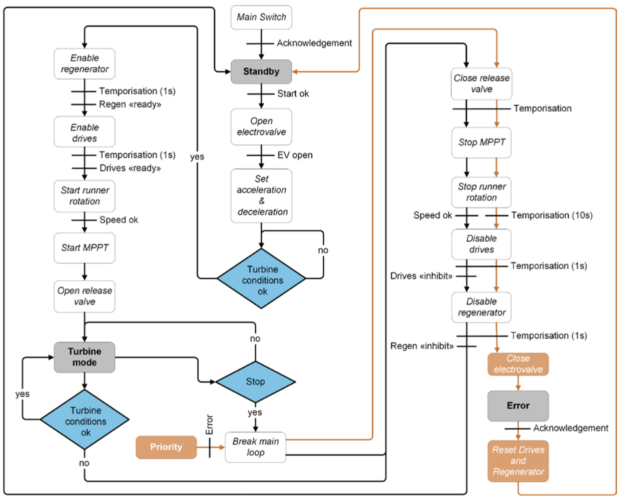
Pilot plant installation

The first DuoTurbo product has been installed in the drinking water supply network of Savièse, VS. Various hydraulic, mechanical and electrical parameters are monitored to study the long term behaviour of the DuoTurbo pilot plant. The installation was commissioned on 15th May 2019.



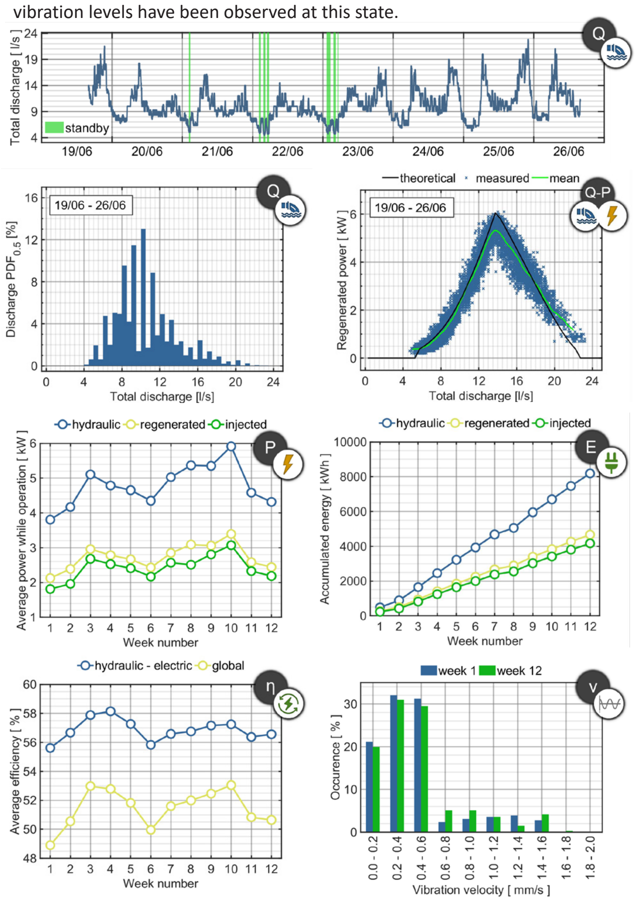
Automation flow chart

The realized micro-hydropower installation operates completely autonomously. The operator's intervention is required only in case of errors, failure and maintenance.



Monitoring results

The monitoring of the first 12 weeks of operation (15th May to 7th August 2019) shows a satisfying behaviour in terms of stability, operating regulation, efficiency and vibration. No significant drifts of the efficiency or vibration levels have been observed at this state.



Conclusion

The DuoTurbo pilot plant has successfully been installed and commissioned in May 2019. The turbine has recovered about 4.2 MWh of electrical energy during its first 12 weeks of operation. Furthermore, a very satisfying behavior in terms of system stability could be observed. Long term tests are ongoing for the final proof of the product's capability.

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Biner, D., Andolfatto L., Hasmatuchi, V., Rapillard, L., Chevailler, S., Avellan, F. and Münch-Alligné, C., **"DuoTurbo: A New Counter-Rotating Microturbine for Drinking Water Facilities"**, Proceedings, International Conference on Innovative Applied Energy (IAPE'19), Oxford City, United Kingdom, March 14-15, 2019

Development team of Duo Turbo (CTI Nr. 17197.1 PFEN-IW)

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EPFL LMH:
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Industrial partners:
Telsa SA, Jacquier-Luisier SA, Valelectric Farner SA

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Prediction of unstable full load conditions in a Francis turbine prototype

J. Gomes Pereira Jr., E. Vagnoni, A. Favrel, C. Landry, S. Alligné, C. Nicolet, F. Avellan

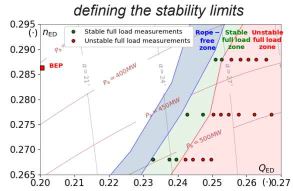
Introduction

Francis turbines operating in full load conditions feature an axisymmetric vortex rotating in the opposite direction of the turbine runner. This vortex rope may enter in an unstable self-exciting process, leading to large pressure pulsations and oscillations in the generating unit power output. In this research work, prototype on-site and reduced scale model test results are presented where the turbine changes from a stable to an unstable full load condition due to an increase in discharge. Measurements are compared in the frequency and time domain, where similarities are evidenced between model and prototype. Using the measurements on the reduced scale model and 1-D numerical models of both the reduced scale model and the turbine prototype, eigenvalue calculations are performed to predict the discharge value of transition from stable to unstable conditions. The transition point on the prototype is then predicted with a small deviation. Transient simulations in the time domain are performed replicating the self-exciting behavior of the unstable full load condition

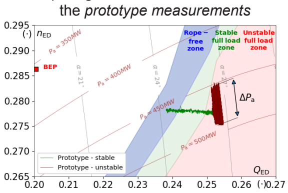
Predicting unstable full load conditions on the prototype

Hill chart with measurements results:

Reduced scale model measurements: *defining the stability limits*

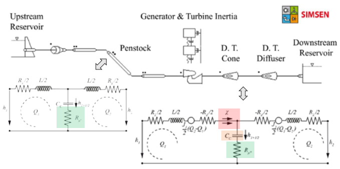


Comparing the reduced scale model and the *prototype measurements*

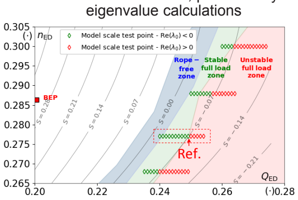


Eigenvalue calculations:

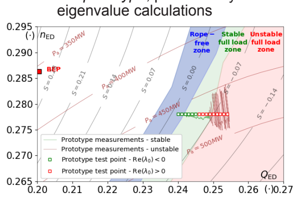
- *Pipes bulk viscosity [1]* $R_{\mu'} = \frac{\mu'}{\rho g A dx}$
- *Cavitation vortex bulk viscosity [2]* $R_{\mu''} = \frac{\mu''}{\rho g A dx}$
- *Mass flow gain factor (quantified by a new method)* $\chi = -\frac{\partial V_{Cav}}{\partial Q_1}$
- *Cavitation compliance [3]* $C_c = -\frac{\partial V_{Cav}}{\partial h}$



Stable and unstable conditions on the *reduced scale model*, predicted by eigenvalue calculations



Stable and unstable conditions on the *turbine prototype*, predicted by eigenvalue calculations



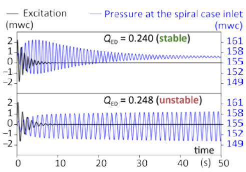
Transient simulation in time domain

Hydroacoustic parameters are *updated at each simulation time step*

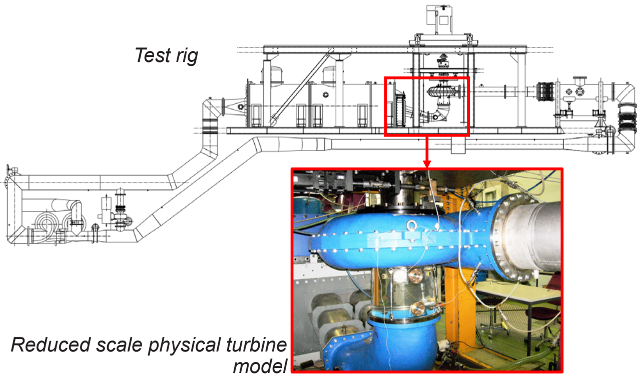
$$a_{Cone(t)} = \left[\frac{\Pi}{\rho}\cdot\left(\bar{p}_{Cone} + \Delta p_{(t)} - p_{Vap}\right)\right]^{0.5}$$

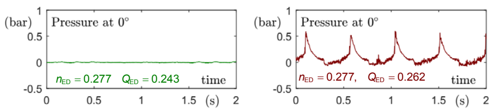
$$\mu''_{(t)} = M''\cdot\frac{\bar{p}_{Cone} + \Delta p_{(t)} - p_{Vap}}{f_0}$$

Self-excited behaviour of unstable full load conditions *are simulated*



Reduced scale model measurements



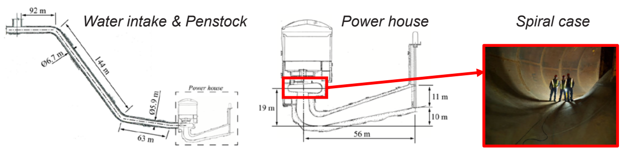


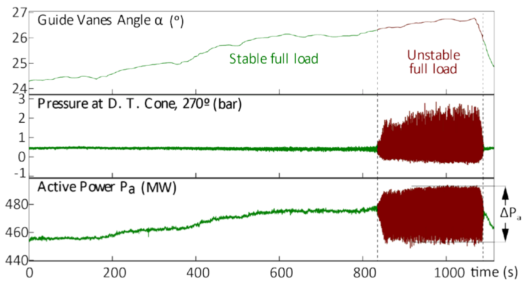
IEC speed & discharge factors

$n_{ED} = nD/\sqrt{E}$

$Q_{ED} = Q/D^2\sqrt{E}$

Hydropower plant featuring the full-scale Francis turbine prototype





Conclusions and future works

The occurrence of unstable full load operating conditions on the prototype was predicted by reduced scale model measurements and eigenvalue calculations on this specific test case. Further measurements for different test cases are expected to further validate the new methodology.

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[2] C. Landry, A. Favrel, A. Müller, C. Nicolet, F. Avellan, Local wave speed and bulk flow viscosity in Francis turbines at part load operation, Journal of Hydraulic Research 54 (2) (2016) 185–196 (2016). doi:10.1080/00221686.2015.1131204.
[3] C. Brennen, A. Acosta, Theoretical, quasi-static analysis of cavitation compliance in turbopumps., Journal of Spacecraft and Rockets 10 (3) (1973) 175–180 (1973). doi:https://doi.org/10.2514/3.27748.

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RENOVHydro: Methodology to determine the parameters of the hydraulic turbine governor for primary control

Christian Landry, Christophe Nicolet, João Gomes Pereira Junior, François Avellan

Motivation

The RENOVHydro project is dedicated to the **renovation** of an existing hydroelectric power plant with a systematic assessment of a **high number of civil and electromechanical potential modifications**. In order to automatically assess the primary control potential of the renovated hydroelectric power plant, it is necessary to have a **simple** and **robust** methodology to deduce the **parameters of a PID controller**.

1. Application to 40 different Francis turbine

- 40 Francis turbines are selected with head from 30 to 500 mWC.
- Mechanical power is fixed arbitrary to 50MW or 300MW.
- The Francis turbine is connected to electrical grid (f_{grid} = 50 Hz)
- The layout of the generic hydraulic power plant are defined by the following rules.

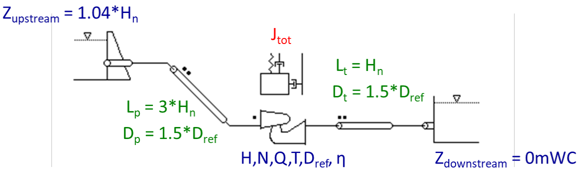


Fig. Dimensioning rules defining the layout of the hydraulic power plant

Dimensioning:

- The dimensioning of the turbine (spiral casing, runner and draft tube) are derived from statistical laws.
- A realistic performance hill chart are obtained with the new SIMSEN library.

$$\begin{cases} T_m = \dfrac{J_{tot}\omega^2}{P} = [5.5-9.6] \\ T_w = \dfrac{Q}{H}\sum \dfrac{L}{gA} = [0.9-2.6] \\ Hadley = \dfrac{T_m}{T_w} = [2.35-9.36] \end{cases}$$

2. Block diagram of the PID controller

The control system is a PID turbine governor with both speed and power control loops combined with the permanent droop Bs.

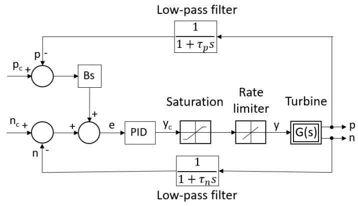


Fig. SIMSEN model of the control system

The PID controller is in series, where K is the proportional gain, *Ti* is the integral time constant and *Td* is the derivative time constant.

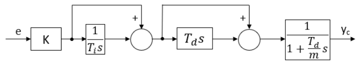
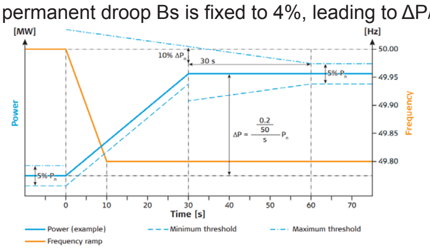


Fig. Block diagram of the PID controller in SIMSEN software

3. Primary control capability defined by Swissgrid

For each Francis turbine, the test defined by Swissgrid for primary control capability is based on a frequency linear variation of 200 mHz in 10 seconds. The output power variation must be delivered within 30 s and remain between minimum and maximum threshold. The permanent droop Bs is fixed to 4%, leading to ΔP/Pn = 10%.



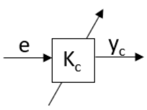
$$Bs = \frac{\dfrac{\Delta f}{f_n}}{\dfrac{\Delta P}{P_n}} = 4\%$$

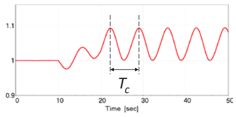
Fig. Test for primary control capability defined by Swissgrid (2011)

4. Methods to define the PID controller parameters

A. Ziegler-Nichols Method

Limit of stability → Kc and Tc → Parameters for PID controller





$$\begin{cases} K = 0.3 \cdot K_c \\ T_i = 0.5 \cdot T_c \\ T_d = 0.114 \cdot T_c \end{cases}$$

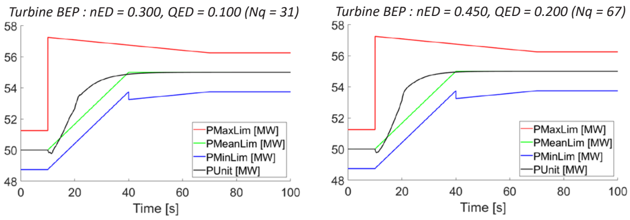


Fig. Time response to the frequency variation of 200 mHz (Pm = 50MW)

B. Time constant Method

Tm and Tw → Parameters of PID controller in parallel → Parallel to series

$$\begin{cases} T_m = \dfrac{J_{tot}\omega^2}{P} \\ T_w = \dfrac{Q}{H}\sum \dfrac{L}{gA} \end{cases}$$

$$\begin{cases} K_p = 0.0193 \cdot \dfrac{T_m^3}{T_w} + 3.8126 \\ K_I = 0.4362 \cdot \dfrac{T_m}{T_w} + 0.3983 \\ K_d = 1.6356 \cdot T_m - 6.6442 \end{cases}$$

$$\begin{cases} K = \dfrac{K_p + \sqrt{K_p^2 - 4K_i K_d}}{2} \\ T_i = \dfrac{K}{K_i} \\ T_d = \dfrac{K_d}{K} \end{cases}$$

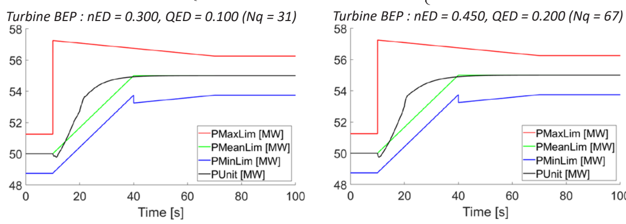


Fig. Time response to the frequency variation of 200 mHz (Pm = 50MW)

5. Conclusion

- The Ziegler-Nichols method is robust and can be applied regardless of the mechanical power of the Francis turbine.
- The time constant method is based on the geometric quantities of the layout and avoids a search for the limit of stability. A correction constant must be applied depending on the power of the hydraulic turbine ($K_{300MW} = 0.6*K_{50MW}$)

Acknowledgments

The RENOVHydro project is granted by CTI, Commission for Technology and Innovation (Grant funding 19343.1 PFEN-IW) and by SFOE, Swiss Federal Office of Energy (Grant funding SI/501436-01).

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Numerical modelling of fish guidance structures

Claudia Leuch, VAW, ETH Zürich

Introduction

Fish guidance structures (FGS) are implemented at hydropower plants to reduce fish mortality during downstream migration. Their design is crucial for their guidance efficiency and the losses caused to the power production. The objective of this thesis was to set up and test a numerical model. The model was then used to analyse FGS configurations.

Background

Fish Guidance Structures

Vertical FGS consist of a bar rack implemented at an angle to the flow. The bars create hydraulic cues, which trigger an evasive behavior of the fish. Traditionally, rectangular, angled bars are used. However, they cause high hydraulic losses and an asymmetric admission flow to the turbines. Curved bars are currently tested at VAW, ETH, as an alternative design to mitigate these issues. Two additional bar shapes were also analyzed numerically (Fig. 1).

Fig. 1: Analyzed bar shapes: angled bar, curved bar, slim bar and fish bar

Numerical modelling

Numerical simulations can be used as an alternative to expensive and laborious physical experiments. Turbulent flow is often modelled using Reynolds averaging on the flow equations to reduce computational costs. As this leads to an under-determined set of equations, a turbulence model is needed as a closure relation (Fig 2). Several different models exist, and it is difficult to know a priori which one is suitable for a given problem.

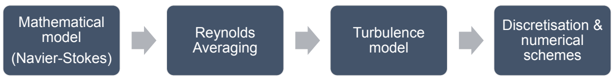

Fig. 2: Schematic approach of the numercial modelling of turbulent flow

Turbulence model evaluation

Five common turbulence models (*standard k-ε*, *realizable k-ε*, *RNG* k-ε*, *standard k-ω* and *k-ω SST***) were analysed for their applicability on the FGS set-up. In a preliminary assessment, the performance of the turbulence models was tested on standard scenarios (flow over flat plate, flow around a cylinder). Grid convergence was studied on a single bar and a 10-bar set-up for the drag coefficient and the overall pressure difference. The *k-ε* models could not capture well the boundary layer behaviour (Fig. 3). The *k-ω* SST model showed the best performance and was chosen for the FGS model set-up.

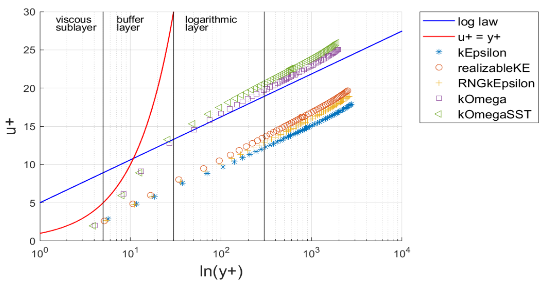

Fig. 3: Boundary layer behavior of the tested turbulence models displayed by the dimentionless parameters for velocity (u+) and wall distance (y+)

Bar rack simulation

2D Set-Up

Loss coefficient (ξ_{FGS}) and flow distribution downstream of the rack of the numerical model were determined for two approach velocities. They were compared to empirical data for the angled and the curved bar to validate the model. The model proved to depict both parameters well. The deviation of the loss coefficient was 12 % for the angled bar and 7 % for the curved bar set-up (Fig. 4).

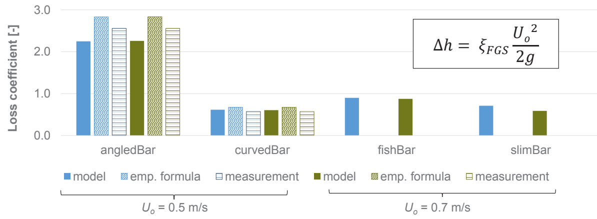

Fig. 4: Comparison of loss coefficients of the numerical model, empirical formula and physical measurement for the different bar shapes and two different approach flow velocities

Both additionally tested bar shapes performed much better than the original angled bar and indicated to be comparable alternative designs to the curved bar layout.

The numerical simulation was used to analyze the flow field in close vicinity of the bars where physical measurements were not possible. Regions of flow detachment or high shear stress can thus be detected (Fig. 5), and flow features might then be correlated to observed fish behavior.

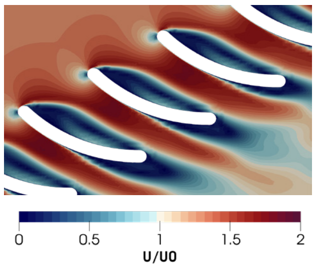

Fig. 5: Flow field around the bars of the curved bar rack set-up. The velocity U is normalised by the approach velocity U_o

3D Set-Up

To assess the flow variation in vertical direction, the model was extended to a 3D setting. Near the bottom, the flow was influenced by wall friction. In the water column, however, there was only small variation of the vertical flow field.

Conclusions

The choice of a fitting turbulence model is a crucial part of numerical flow simulations. It could be shown that the *k-ω* SST turbulence model was suitable for the numerical simulation of the bar rack configuration. Both flow field and loss coefficient could be reproduced well. A 2D model seems to be appropriate for a simple bar rack set-up. Further analysis should be done on the use of 3D models for simulations of FGS with additional structures such as overlays, which introduce stronger vertical flow components.

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