Performance Measurements on the DuoTurbo Microturbine for Drinking Water Systems

Abstract
The present paper introduces the global concept of the DuoTurbo microturbine: the third generation of an axial microturbine with counter-rotating runners developed to recover the energy lost in the relief valves of drinking water supply networks. This new prototype is the result of the DuoTurbo project, led by HES-SO Valais/Wallis and performed in collaboration with EPFL-LMH and three Swiss industrial partners. The machine has been designed and constructed integrating custom-made synchronous rim generators. The electrical stators are placed around the rotors built with permanent magnets directly stuck on the bands of the runners. One-stage of this microturbine, with a maximal power of 5 kW, forms actually a compact independent unit with the possibility of stacking several stages in series. The resulting performance measurements of the single-stage prototype, installed in the HES-SO VS universal hydraulic test rig, comes to complete this work.

Keywords: DuoTurbo, counter-rotating, microturbine, rim generator, drinking water network, performance measurements

Introduction
In the actual context of Swiss nuclear phase-out strategy, harvesting the extensive potential of small hydropower (<10MW), in particular on existing infrastructure, is a priority. Considering the hilly and the mountainous area, the difference of altitude between the fresh water source and the consumers imposes relatively often the use of a relief valve (or of another energy dissipator) to regulate the pressure in the consumption area (Carravetta et al. [1-2], Samora et al. [3], Andolfatto et al. [4]). An inventory of Switzerland’s drinking water networks, provided by Hintermann [5], counted in 1994 no less than several hundred of potential hydropower sites with an installed power larger than 10 kW. In 2011, the Swiss Federal Office of Energy estimated an annual unexploited potential of drinking water networks of about 0.06 TWh, equivalent to the electrical consumption of 20’000 households (Eawag [6]).

Among different available technologies to recover the energy lost in drinking water supply networks, running a standard pump as turbine (PAT) may be often a cost-efficient solution (Orchard and Klos [7], Williams [8], Ramos and Borga [9]). However, despite the fact that a PAT may be operated at variable speed, its narrow operating characteristic is not always suitable for very large operating discharge ranges, typical for drinking networks. An alternative more efficient solution would be a counter-rotating multi-stage turbine, as the ones introduced by Nielsen et al. [10], Sonohata et al. [11], Lee et al. [12], or Nipp et al. [13].

In this framework, a new axial microturbine with counter-rotating runners has been developed to recover the energy lost in relief valves of drinking water supply networks. Conceptually, despite the fact that the axial turbines are generally used in cases of low head and high specific speed, the multi-stage configuration (similar to

Fig. 1. Electro-mechanical concept of the DuoTurbo microturbine prototype, Biner et al. [19].
multi-stage centrifugal pumps) allows working under high head operating conditions, a typical regime of Pelton turbines. Indeed, each stage recovers a fraction of the available head. The one-stage variable speed turbine is composed by one upstream runner followed by one counter-rotating downstream runner. Historically, the first prototype developed in the Laboratory for Hydraulic Machines - EPFL, Switzerland, consisted on an “elbow” configuration, with the two generators placed outside the elbows (Münch-Alligné et al. [14], Vagnoni et al. [15]). The second generation, developed by the HES-SO Valais/Wallis in collaboration with the EPFL-LMH within the framework of Hydro VS project, consists on an in-line configuration with the generators placed inside the upstream and respectively downstream bulbs of the turbine (Hasmatuchi et al. [16], Melly et al. [17], Biner et al. [18]). The main advantage of this fully instrumented prototype is that it offers also a good visual access to the flow field into the runners.

The present work focuses on the third generation of this turbine, developed in the framework of DuoTurbo project, led by HES-SO Valais//Wallis and performed in collaboration with EPFL-LMH and three Swiss industrial partners, Biner et al. [19]. The new prototype is designed and constructed integrating a custom-made permanent magnet synchronous rim generators; the electrical stators are placed around the rotors built with permanent magnets directly stuck on the bands of the runners. The one-stage of this microturbine, with a maximal power of 5kW, forms a compact independent unit with the possibility of stacking several stages in series.

1. Electro-mechanical concept of the DuoTurbo microturbine prototype

The new axial counter-rotating microturbine prototype with a power of 5 kW per stage is illustrated in Fig. 1. The turbine has been developed from scratch including the design of the hydraulic profile, mechanical system, electrical generators, as well as of the electronic control system of the two independent variable speed runners. The hydraulic design is based on the characteristics of a pilot site on the drinking water supply system of Savièse, Valais, Switzerland, where the endurance tests of the new product will be later carried out. Starting from two operating conditions, the nominal point (a discharge of 9 l/s and a head of 24.5 m) and the point of maximum power, the hydraulic profile of the runners has been designed and optimised using numerical simulation (Biner et al. [19]). The one-stage variable speed turbine consists of a pair of one upstream 3-blade runner and one downstream 5-blade runner. The turbine inner diameter is 80 mm, whereas its outer diameter is 100 mm. At the nominal point, an energetic efficiency of 90% is reached for a ratio $\alpha = N_A/N_B = 1$ between the runners absolute rotational speed. The optimal regulation of the machine is ensured by changing the relative rotational speed between the runners.

The mechanical concept of the DuoTurbo microturbine deals with several technical challenges as the compactness, the concentricity of the rotating components, the bearing lifetime, the minimisation of volumetric losses into the labyrinths, as well as the compatibility of the materials with the drinking water. In order to ensure the compactness, each rotor combines both the hydraulic and the electrical generator rotating elements in one unit. Indeed, the band fixed at the periphery of the runner blades, serves as support for the permanent magnets of the electrical rotor. Then, a resin comes to fill the gap between the magnets and to give a cylindrical shape to the rotor unit. A polymer tube placed into the so-called “air gap” of the generator separates the electrical stator from the fluid region and ensures the static sealing of the machine. Moreover, the stators made of two separable sheet metal bundles leans the polymer tube in order to guarantee the mechanical resistance against the pipeline pressure. The radial and mainly the axial forces are taken up by ceramic ball bearings mounted on the central shaft. Finally, the different labyrinth seals come to minimize the volumetric losses both into the central and at the periphery regions of the machine. By this compact design, a total length of only 526 mm per stage, including the downstream diffuser, and an outer diameter of 300 mm, are obtained.

With the particular specifications imposed by the hydraulic characteristics and dimensional restrictions, a complete development of the custom-made the synchronous rim generators, has been done. This configuration of generators is also known in the literature as “Straflo” (Harza [20] and Miller [21]). The 8-poles rotors are built with Neodymium permanent magnets. The particular configuration of the stators distributed only over a half of the circumference ($2\times90^\circ$) ensures a minimum distance between the runners and avoids a conflict between the generator end windings. One may state here that the generators (with a nominal electrical power of 3.37 kW each), reach an efficiency of 92 %, obtained by experimental tests.

In the end, two M700 Emerson frequency converters are used to independently drive the runners, keeping constant their rotational speed, whatever the sign of the mechanical torque. The variable speed control of the generators is made in sensor-less mode, since the actual mechanical construction of the turbine does not allow for mounting an encoder. The energy produced by the microturbine is injected to the power grid by a third M700 converter installed on network side. Indeed, this architecture based on three converters interconnected by a DC bus ensures a four-quadrant operating control of the two independent generators.
2. Experimental setup

2.1 HES-SO VS model testing infrastructure

The hydraulic performance tests of the DuoTurbo prototype have been performed on the universal hydraulic test rig of the HES-SO Valais/Wallis – Switzerland, dedicated to small-power turbomachines (see Fig. 2). The configuration, the instrumentation and the operation of the test rig follows the IEC 60193 [23] standard recommendations on hydraulic model testing. The closed-loop circuit is supplied by three recirculating multistage centrifugal pumps connected in parallel. The variable speed pumps with a power of 2x18.5 kW and respectively 1x5.5 kW can deliver a maximum discharge of about 100 m³/h and a maximum pressure of 160 mWC. The variable-speed testing model is installed in the upper region of the circuit. The downstream free-surface pressurized reservoir allows simulating different setting levels of the model (positive or negative) and thus investigating also its cavitation performances. An autonomous regulation system ensures the operation of the test rig at constant pumps speeds, testing head or discharge, whatever the operating point of the testing model. The customized LabVIEW interfaces allow for real-time measurements and display the instantaneous values of different sensors (pumps speed, discharge, testing head, water temperature, Thoma number etc.) The wireless communication architecture between the hydraulic test rig and the measurement/monitoring testing model control system ensures safe data centralization, storage and sharing, Hasmatuchi et al. [22].

Main characteristics:

- Maximum head: 160 mWC
- Maximum discharge: ±100 m³/h
- Generating power: 20 kW
- Pumping power: 2x 18.5 kW & 1x 5.5 kW
- Maximum pumps speed: 3'500/3000 min⁻¹
- Total circuit volume: 4.5 m³

Fig. 2. Hydraulic test rig (2016 version) of the HES-SO VS – Switzerland, Hasmatuchi et al. [22].

2.2 Instrumentation setup

The Table 1 presents the characteristics of the main instruments employed to recover the hydraulic performances of the testing model. Accordingly, an electromagnetic flowmeter is used to recover the discharge Q, whilst two differential pressure transducers are used for the testing head H and the setting level Hs. The wall static pressure at the inlet of the turbine M1 is measured with a capacitive absolute pressure transducer connected through a collector, as illustrated in Fig. 3.

<table>
<thead>
<tr>
<th>Measured/displayed quantity</th>
<th>Sensor type</th>
<th>Range</th>
<th>Precision</th>
</tr>
</thead>
<tbody>
<tr>
<td>Discharge (Q)</td>
<td>Electromagnetic flowmeter</td>
<td>8..280 [m³/h]</td>
<td>± 0.2 [%]</td>
</tr>
<tr>
<td>Head (H)</td>
<td>Differential pressure sensor</td>
<td>0..16 [bar]</td>
<td>± 0.1 [%]</td>
</tr>
<tr>
<td>Setting level (Hs)</td>
<td>Differential pressure sensor</td>
<td>0..5 [bar]</td>
<td>± 0.2 [%]</td>
</tr>
<tr>
<td>Absolute static pressure (M1)</td>
<td>Capacitive absolute pressure sensor</td>
<td>0..20 [bar]</td>
<td>± 0.05 [%]</td>
</tr>
<tr>
<td>Rotational speed (Nₐ,A,B)</td>
<td>Precision electrical multimeter</td>
<td>0.50 [kHz]</td>
<td>0.025 [%]</td>
</tr>
<tr>
<td>Mechanical torque (Tₘₐ,A,B)</td>
<td>Precision electrical multimeter</td>
<td>0.32 [Aₘₐ]</td>
<td>0.025 [%]</td>
</tr>
</tbody>
</table>

Table 1. Characteristics of the main measurement instruments.

Data acquisition and control

<table>
<thead>
<tr>
<th>NI cDAQ 9174 digitizer</th>
<th>- Dedicated to the values related to the testing model</th>
</tr>
</thead>
<tbody>
<tr>
<td>NI CompactRIO 9074 controller</td>
<td>- Dedicated to the values of the test rig</td>
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</table>
For the mechanical power, since the generators are equipped neither with an encoder, nor with a torquemeter, an indirect measurement method is used. Indeed, a precision electrical multimeter connected between the frequency converters and the generators allows recovering both the rotational speed of the runners and the mechanical torque through respectively the frequency and the true-rms values of the current.

The measurements of parameters necessary to compute the hydraulic power are performed with a National Instruments (NI) cRIO 9074 autonomous digitizer equipped with several specific acquisition/control modules. An additional NI cDAQ 9174 digitizer is employed to record the values necessary mainly for the computation of the mechanical power and of the electrical power of the generators. Finally, the recorded average and standard deviation values of all parameters are computed using an acquisition time of 4 seconds at 200 Hz (user-configurable, depending on the operation stability).

### 2.3 Experimental methodology

The employed experimental protocol to measure the hydraulic performances consists in a first step in calibrating/rechecking all the employed instruments. In Fig. 4, the calibration curves and the absolute error for the torque measurement from the current true-rms values are presented. One may notice here the scattering of the error values as a result of the generator temperature variation. This effect has been actually neglected in this case, considering that the obtained error values do not exceed the measurement precision of the reference torque sensor.

![Calibration curves and the absolute error for the torque measurement from the current true-rms values.](image)

Then, the measurements have been performed for 16 different testing head values necessary to build the full efficiency hill-chart of the microturbine. Several operating points have been systematically addressed for each constant testing head value, varying different combinations of the runners rotational speed from 500 to 3'500 min⁻¹. Indeed, this represents a standard testing procedure typical for a double-regulated axial turbine with variable angle of the stator and rotor blades.
3. Experimental results

3.1 Performance characteristics at constant testing head

The hydraulic-to-mechanical efficiency $\eta_{h-m}$ of one stage composed by two counter-rotating runners can be expressed through the product between the hydraulic efficiency $\eta_h$ and the bearing efficiency $\eta_m$, and computed as a ratio between the sum of mechanical powers $P_m$ of the two runners and the total hydraulic power $P_h$:

$$\eta_{h-m} = \eta_h \cdot \eta_m = \frac{P_{mA} + P_{mB}}{P_h} = \frac{\omega_A T_{mA} + \omega_B T_{mB}}{\rho Q E},$$

where $\omega_A$ and $T_{mA}$ are respectively the angular speed and the mechanical torque of the first runner, and $\omega_B$ and $T_{mB}$ the angular speed and the mechanical torque of the second runner.

As may be noticed from the instrumentation list presented in Table 1, if the hydraulic-to-mechanical efficiency can be measured, it is not the same for the hydraulic and the bearing efficiency. Concerning the hydraulic power, whilst the discharge is measured with the electromagnetic flowmeter, the total specific energy $E = gH$ is computed with the value of the differential pressure sensor installed between the inlet and the outlet sections of the turbine (see Fig. 3). The specific energy is calculated only from the difference of static pressure considering that the turbine is installed horizontally and its inlet and outlet cross sections are equal and sufficiently far from the central section changes.

$$E \approx 300 [J-kg^{-1}]$$

$$E \approx 400 [J-kg^{-1}]$$

Fig. 5. Measured 3D hill-charts of the counter-rotating microturbine at two different constant testing head values.

Fig. 6. Distribution of the mechanical power between the two runners for a constant testing head value of 4 bars.
In Fig. 5, the resulting measured 3D hill-charts of the counter-rotating microturbine for two different constant testing head values are presented. The dimensionless hydraulic-to-mechanical efficiency is computed using eq. (2), with the maximum efficiency corresponding to the maximum value measured over the whole operating characteristic of the turbine. One may see here that the two efficiency characteristics are very similar and that turbine exhibits a wide range of relatively high efficiency values.

In Fig. 6, the distribution of mechanical power between the two runners of the turbine for a constant testing head value of 4 bars is provided. The values are scaled with the maximum value of total mechanical power measured for this constant testing head (eqs. (3) (4)). The total mechanical power characteristic exhibits also a wide range of high values, as an optimal combination between the two different mechanical power characteristics of the runners.

3.2 Resulting 3D hill-chart

The final global efficiency characteristic of the DuoTurbo microturbine is obtained from several sets of measurements at constant testing head value. The result, illustrated in Fig. 7, shows a relatively wide region of high efficiency on the Q-H characteristic. Actually, this is explained by the possibility of regulation of this type of turbine, which makes it more adapted in terms of energy recovery potential (compared to a standard centrifugal PAT) especially in the case of drinking water supply systems, where the inflow conditions vary a lot.

![Fig. 7. Resulting efficiency hill-chart contour and 3D surface of the DuoTurbo microturbine.](image)

3.3 Measurements repeatability

Finally, in Fig. 8, the result of measurements performed on two consecutive days at fixed inflow conditions (constant recirculating pumps rotational speed values) for a ratio $\alpha = 1$ between the runners absolute rotational speed, show a good repeatability. This can actually be noticed on the hydraulic characteristic, represented with the help of the discharge-specific energy coefficients ($\phi$-$\psi$), as well as on the efficiency characteristic ($\phi$-$\eta_{h-m}$).

$$\phi = \frac{Q}{A_1 \cdot U_{1e}} = \frac{\frac{Q}{4} \cdot (D_{1a}^2 - D_{1b}^2) \cdot 2 \cdot \pi \cdot N}{2 \cdot g \cdot H}$$

$$\psi = \frac{2 \cdot E}{U_{1e}^2} = \frac{2 \cdot \pi \cdot N}{60} \cdot \frac{D_{1a}}{2}$$

![Fig. 8. Repeatability of hydraulic characteristic measurements of the turbine.](image)
Conclusions & perspectives
The present work presented the performance measurements of the third generation of an axial microturbine with counter-rotating runners dedicated to recover the energy lost in relief valves of water supply networks. The experimental tests have been performed on the HES-SO VS test rig. This new prototype, developed in the framework of DuoTurbo project in collaboration with EPFL-LMH and three Swiss industrial partners, has been designed and constructed integrating a custom-made permanent magnet synchronous rim generators. The electrical stators have been installed around the rotors built with permanent magnets directly stuck on the bands of the runners. The one-stage of this microturbine, with a maximal power of 5 kW, forms actually a compact independent unit with the possibility of stacking several stages in series. The optimal regulation of this type of turbine under variable inflow conditions, typical for water supply networks, is ensured by changing the relative rotational speed between the runners.

To conclude, the undergoing development of the DuoTurbo microturbine prototype is performed in view of an industrial fabrication. Beyond the aspects of functionality, efficiency and lifetime, this in-line compact multistage concept targets low investment (production and maintenance) costs in order to ensure a profitable industrial fabrication. Beyond the aspects of functionality, efficiency and lifetime, this in-line compact multistage concept targets low investment (production and maintenance) costs in order to ensure a profitable exploitation of sites with an available hydraulic power of 5 to 25 kW. In addition, an appropriate command strategy based only on measurements of the generators electrical power should maximize the recovered energy.

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