ANALYTICAL HILL CHART TOWARDS THE MAXIMISATION OF ENERGY RECOVERY IN WATER UTILITY NETWORKS WITH COUNTER ROTATING MICRO-TURBINE

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ABSTRACT

Existing water utility networks provides a wide, distributed and unexploited hydropower potential. The installation of energy recovery stations on existing infrastructures limits both the required investment and the environmental impact. The counter rotating micro-turbine presented in this paper is a new technology dedicated to the harvesting of energy on drinking water network. An analytical model of the machine’s efficiency – the so-called hill chart – is used to investigate the runners’ speeds command laws that maximize the recovered energy on the wide range of consumer-driven flow discharge experienced on water utilities.

Keywords: Energy harvesting, Picohydropower, efficiency, Hermite polynomials

1. INTRODUCTION

1.1 Hydropower potential on water utilities

Almost all of the large scale hydroelectric potential is already harnessed in developed countries. An increase of sustainable hydraulic energy production mostly rely on the development of new technologies dedicated to harvesting the small scale hydroelectric potential. For instance, the Swiss yearly small hydropower production – generated on stations with a nominal power below 300 kW – is expected to grow from 0.3 TWh to 1.3 TWh between 2010 and 2050 as stated in SATW (2006).

The condition for such small power stations to meet economical profitability is to minimize the required capital expenditure while maximizing the efficiency of the installed hydraulic machine. Standard turbine types are usually not considered economical in these conditions Senior et al. (2010). Environmental impact should also be kept as low as possible to allow a massive and sustainable field implementation. Harvesting the hydroelectric potential on existing installation such as water utility networks limits both the investment and the environmental impact associated to new infrastructure.

Figure 1: Schematic view of the hydropower potential on drinking water utility.
The basic principle of the hydropower potential in drinking water networks is explained in Figure 1. The water is caught and stored in a reservoir near the spring catching area at an altitude \( z_i \) higher than the altitude \( z_c \) of the consumption area. The flow discharge \( Q \) in the pipe is driven by the water consumption by various mean. In this paper, it is only assumed that the discharge is fluctuating. The pressure in the consumption area must be limited to \( p_{\text{max}} \) to avoid damaging the equipment. This pressure regulation is commonly ensured by relief valves that dissipate the energy. However, the dissipated energy can also be recovered with an appropriate installation. With \( \rho \) the water density, \( g \) the gravity and neglecting the losses in the pipes, the available hydraulic power \( P_{\text{av}} \) is defined in Eq. [1]:

\[
P_{\text{av}} = \rho Q (g(z_c - z_i) - \frac{p_{\text{max}}}{\rho})
\]  

[1]

Several examples of successful implementations of conventional hydraulic machines technologies (Pelton turbine, Francis turbine, pump as turbines, etc.) to recover hydropower potential on drinking water networks are exposed in Bühler (2007). These conventional technologies applied to small hydropower face either a high capital expenditure or limited performance under the fluctuating discharge conditions commonly experienced on consumption-driven networks. Müller and Senior (2009) also consider the use of Archimedean screw as a relevant technology for energy harvesting in the case of low available head.

1.2 Counter rotating micro-turbine

Figure 2 presents the concept of counter rotating micro-turbine featuring two axial runners rotating in opposite directions. Its compact axial architecture ensures a lean in-line installation on existing facilities. Variable runners’ rotational speed allows to cope with the consumer-driven wide range of discharge usually experienced on water utility networks. It is therefore an interesting candidate for energy recovery on drinking water network.

Figure 2: Schematic representation of a single stage counter rotating micro-turbine.

Münch-Alligné et al. (2013) numerically investigated the behavior of this type of machine. They showed that altering the ratio \( \alpha \) between the rotational speed \( N_2 \) and \( N_1 \) leads to a shift of the efficiency curve of the machine.

Then, for given operating conditions, the best efficiency of the hydraulic machine is very likely to be reached with the two runners rotational speed set independently. Finding the optimal speed combination for each operating point is not a straightforward problem.

Section 2 presents the experimental campaign conducted to characterize a counter rotating micro-turbine model on its operation domain. The data gathered feed the method presented in section 3 to build an analytical model of the machine behavior, later called analytical hill chart. The benefit of the analytical hill chart is illustrated through the case study presented in section 0 in which the data collected on an instrumented pilot site with an average of 9.5 kW of available hydraulic power are used to estimate the expected annual energy production. Estimation with and without individually optimized runner speeds are compared.

2. EXPERIMENTAL CAMPAIGN

2.1 Efficiency measurement on the experimental test rig

The experimental test rig showed in Figure 3 has been developed to investigate the behavior of a single stage counter rotating micro-turbine model.

An external pump allows the reproduction of variable discharge in the water circuit. The discharge \( Q \) is measured with an electromagnetic discharge sensor. The specific energy \( E \) extracted from the flow is the difference of specific energy between the headwater side and the tailwater side. It is evaluated according to Eq. [2] thanks to two differential pressure sensors measuring the difference \( \Delta P \) of static pressure between the headwater section and the tailwater section.
The pump can be controlled either to regulate the discharge $Q$ in the network or the specific energy $E$. The hydraulic power $P_h$ extracted from the fluid is given by Eq. [3].

$$P_h = \rho Q E$$  \[3\]

From the machine side, the runners’ rotational speed $N_1$ and $N_2$ are controlled by speed drives and measured with inductive tachymeters. The torques $T_1$ and $T_2$ on the runners’ shaft are measured with strain gauges torque sensors. Then, the output mechanical power $P_m$ is measured according to Eq. [4].

$$P_m = T_1 \cdot \frac{2\pi N_1}{60} + T_2 \cdot \frac{2\pi N_2}{60}$$  \[4\]

Finally, the hydraulic efficiency $\eta$ of the machine is defined as the ratio between the extracted hydraulic power $P_h$ and the output mechanical power $P_m$:

$$\eta = \frac{P_h}{P_m}$$  \[5\]

2.2 Operating domain investigation

In order to capture the behavior of the micro-turbine in the widest possible operating domain, a full factorial plan Fischer (1926) is used with:

- 7 levels for $N_1$, uniformly distributed from 300 min$^{-1}$ to 3000 min$^{-1}$;
- 7 levels for $N_2$, uniformly distributed from 300 min$^{-1}$ to 3000 min$^{-1}$;
- 8 levels for $E$, uniformly distributed from 100 J/kg to 275 J/kg.

The rotational speed domain is limited by the capacity of the generators. The limits for the specific energy is given by the maximum power of the pump that feeds the micro-turbine model. For each of the 8 levels of specific energy $E$, the 49 combinations of rotational speeds are investigated. This gives 392 experimental points uniformly covering the operating domain of the single stage model. For each of these points, the discharge $Q$ and the efficiency $\eta$ are measured. The water temperature was monitored during the entire measurement campaign. The maximum temperature variation did not exceed 1°C.
3. ANALYTICAL HILL CHART MODELLING METHODOLOGY

3.1 Proposed method

As exposed in Münch-Alligné et al. (2013), the same operating point \((Q, E)\) can be reached with various combinations of settings \((N_1, N_2)\) yielding to different values for the efficiency. The method proposed to identify the efficiency hill chart of the machine on its entire \((Q, E)\) operating domain is described in Figure 4.

For each of the \(n_E\) levels \(E\) of investigated specific energy, the investigated set of 49 setting points \((N_1, N_2)\) and 49 measured responses \((\eta, Q)\) are extracted. Two polynomial chaos models \(\eta(N_1, N_2)\) and \(Q(N_1, N_2)\) are built according to the procedure described in subsection 3.2. Then, for an arbitrarily chosen, uniformly distributed, set of \(n_k\) discharge values \(Q_k\) on the experimental domain, the best efficiency \(\eta_k\) and the associated rotational speeds \(N_{1,k}\) and \(N_{2,k}\) are identified as explained in subsection 3.3. These \(n_E\) best efficiency ridges are finally used to build the initially targeted analytical hill chart \(\eta(Q, E)\) and the optimal rotational speed maps \(N_1(Q, E)\) and \(N_2(Q, E)\).

3.2 Hermite polynomial chaos expansion

The polynomial chaos expansion has been introduced by Wiener in Wiener (1938) to study the effect of an input variables vector \(x\) of length \(n_i\) with probabilistic uncertainties on a quantities of interest vector \(y\) of length \(n_o\). It consists in providing a model function \(f\) so that \(f(x) = y^*\) is an approximation of the observed \(y\).

In the present case, the function \(f\) is a linear combination of Hermite polynomials \(H_{mn}\) that forms a complete and orthogonal basis of the function from \(\mathbb{R}^{n_i}\) to \(\mathbb{R}^{n_o}\) Laplace (1812). The general expression of \(f\) is given in Eq. [6].

\[
f(x) = \sum_{m=0}^{\infty} a_m \times H_{mn}(x)
\]

The training of the model consists in the identification of the set of coefficients \(\{a_m\}\) that provides the best approximation \(\{f(x)\}\) of the experimentally obtained \(\{y_j\}\) according to given fitting criterion. Considering a number of \(N\) experimental points, a maximum of \(N_b < N\) coefficient \(a_m\) can be identified: the Hermite polynomial basis is said to be truncated. In this work, the sequential cleaning strategy described in Blatman (2009) and implemented in the OpenTURNS library has been used to search for the optimal truncated Hermite polynomial basis. The fitting criterion used is the least angle regression described in OpenTURNS (2014).

![Flowchart of the analytical hill chart construction](image-url)
Figure 5 shows a graphical representation of the two models $\eta_E=175(N_1, N_2)$ and $Q_E=175(N_1, N_2)$ identified for the specific energy level $E$ of 175 J.kg$^{-1}$. Table 1 details the truncated basis size, the maximum absolute error and the standard deviation of the error between the experimental data and the model predictions for each investigated level of specific energy. All the standard deviations of the errors sit below 1.5% of relative error with respect to the predicted values, providing a satisfactory confidence into the identified models.

Table 1: Quality of the identified models for the efficiency and discharge models at each of the eight levels of specific energy.

<table>
<thead>
<tr>
<th>SPECIFIC ENERGY LEVEL [J.kg$^{-1}$]</th>
<th>$\eta$ TRUNCATED BASIS SIZE</th>
<th>$\eta$ MAXIMUM ABSOLUTE ERROR [%]</th>
<th>$\eta$ STD. DEV. OF THE ERROR [%]</th>
<th>$Q$ TRUNCATED BASIS SIZE</th>
<th>$Q$ MAXIMUM ABSOLUTE ERROR $\times 10^{-5}$ [m$^3$.s$^{-1}$]</th>
<th>$Q$ STD. DEV. OF THE ERROR $\times 10^{-5}$ [m$^3$.s$^{-1}$]</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>10</td>
<td>2.02</td>
<td>0.74</td>
<td>9</td>
<td>3.76</td>
<td>1.61</td>
</tr>
<tr>
<td>125</td>
<td>11</td>
<td>1.22</td>
<td>0.43</td>
<td>9</td>
<td>3.33</td>
<td>1.38</td>
</tr>
<tr>
<td>150</td>
<td>8</td>
<td>1.19</td>
<td>0.47</td>
<td>9</td>
<td>3.28</td>
<td>1.31</td>
</tr>
<tr>
<td>175</td>
<td>7</td>
<td>0.95</td>
<td>0.41</td>
<td>9</td>
<td>3.09</td>
<td>1.44</td>
</tr>
<tr>
<td>200</td>
<td>11</td>
<td>1.01</td>
<td>0.28</td>
<td>5</td>
<td>10.38</td>
<td>2.24</td>
</tr>
<tr>
<td>225</td>
<td>7</td>
<td>0.82</td>
<td>0.31</td>
<td>5</td>
<td>9.34</td>
<td>2.65</td>
</tr>
<tr>
<td>250</td>
<td>15</td>
<td>0.85</td>
<td>0.33</td>
<td>16</td>
<td>6.51</td>
<td>2.29</td>
</tr>
<tr>
<td>275</td>
<td>6</td>
<td>0.78</td>
<td>0.32</td>
<td>14</td>
<td>3.37</td>
<td>1.24</td>
</tr>
</tbody>
</table>

Figure 5: Efficiency and discharge models for a specific energy level $E = 175$ J/kg.

3.3 Best efficiency ridge identification

The two polynomial chaos models $\eta_E(N_1, N_2)$ and $Q_E(N_1, N_2)$ of the efficiency and the discharge at a given specific energy level are assumed to be known. The explored rotational speed domain is written $\Omega = [300, 3000]$. The discharge domain, from $Q_{\min,E}$ to $Q_{\max,E}$ is defined by Eq. [7]:

$$Q_{\min,E} = \min_{N_1, N_2} Q_{E}(N_1, N_2)$$

$$Q_{\max,E} = \max_{N_1, N_2} Q_{E}(N_1, N_2)$$

The identification of the efficiency ridge consists in solving the $n_1$ optimization problems formulated in Eq. [8]:

$$\forall k \in 0, n_1 - 1 , (N_{1,k}, N_{2,k}) = \arg \max_{(N_1, N_2) \in \Omega} \eta_k(N_1, N_2)$$

With:

$$\forall k \in 0, n_1 - 1 , Q_k = Q_{\max,E} \cdot \left( 1 - \frac{k}{n_1 - 1} \right) + Q_{\min,E} \cdot \frac{k}{n_1 - 1}$$

And:

$$\forall k \in 0, n_1 - 1 , \eta_k = \eta_k(N_{1,k}, N_{2,k})$$

As the evaluation cost of both the objective function $\eta_k$ and the constraint function $Q_k$, known through their analytical polynomial expression, is low, the $n_1$ optimization problems are solved with a penalized particle swarm optimization built from the Inspyred Python library Inspyred (2014).
The ridge points identified for either $N_1$ or $N_2$ closer than 10 min$^{-1}$ of the boundaries of $\Omega$ are removed, as they do not depict the behavior of the machine but are only due to the limitation of the explored domain. Figure 5 shows the best efficiency ridge location for a specific energy level $E$ of 175 J/kg. Figure 6 shows a cut of the hill chart along the identified ridge, providing the best efficiency curve for this level of specific energy and with an optimal relative rotational speed $\alpha_{\text{opt}} = N_2\text{opt}/N_1\text{opt}$. The efficiency curve obtained with $\alpha = 1$ is also plotted, showing the efficiency improvement provided by optimal relative rotational speed at this level of specific energy. Figure 7 shows the identified best efficiency ridges (solid lines) and the efficiency curves for $\alpha = 1$ (dashed lines) with respect to the experimentally obtained data points.

### 3.4 Global analytical hill chart

The collection of eight efficiency curves for the eight explored levels of specific energy serve as input data to build the global analytical hill chart. The input vector $X$ gathers the discharges and the specific energy. The output vector $Y$ gathers the efficiency and the two rotational speeds, respectively the common rotational speed when $\alpha$ is kept constant. According to these data, three polynomial chaos models $\eta_{\text{opt}}(Q, E)$, $N_{1\text{opt}}(Q, E)$ and $N_{2\text{opt}}(Q, E)$ are built to model the hill chart with independent optimal setting of the rotational speeds and two models $\eta_{\alpha=1}(Q, E)$, $N_{\alpha=1}(Q, E)$ are built for the case when $\alpha$ is kept constant. These two hill charts are presented in Figure 8 and Figure 9 respectively.

**Figure 6:** Identified best efficiency ridge and efficiency curve for $\alpha=1$ for a specific energy level $E = 175$ J/kg.

**Figure 7:** Identified best efficiency ridges (solid) and efficiency curve (dashed) for $\alpha=1$ as envelopes of the experimental points.
Table 2: Quality of the identified models for the global analytical hill charts.

<table>
<thead>
<tr>
<th>MODEL</th>
<th>TRUNCATED BASIS SIZE</th>
<th>MAXIMUM ABSOLUTE ERROR</th>
<th>STANDARD DEVIATION OF THE ERROR</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \eta_{opt}(Q, E) )</td>
<td>17</td>
<td>0.84%</td>
<td>0.30%</td>
</tr>
<tr>
<td>( N_{1, opt}(Q, E) )</td>
<td>8</td>
<td>59.8 min(^{-1})</td>
<td>23.9 min(^{-1})</td>
</tr>
<tr>
<td>( N_{2, opt}(Q, E) )</td>
<td>14</td>
<td>92.2 min(^{-1})</td>
<td>22.4 min(^{-1})</td>
</tr>
<tr>
<td>( \eta_{\alpha=1}(Q, E) )</td>
<td>17</td>
<td>2.31%</td>
<td>0.60%</td>
</tr>
<tr>
<td>( N_{\alpha=1}(Q, E) )</td>
<td>12</td>
<td>57.0 min(^{-1})</td>
<td>16.8 min(^{-1})</td>
</tr>
</tbody>
</table>

Figure 8: Hill chart of a single stage micro-turbine with independent optimal setting of the rotational speeds.

Figure 9: Hill chart of a single stage micro-turbine with constant relative rotational speed \( \alpha = 1 \).
4. EVALUATION OF THE PRODUCTIVITY ON A PILOT SITE

4.1 Instrumented pilot site

A pilot site has been identified and instrumented on a drinking water network in the Swiss Alps. The reservoir is situated approximately 190 m higher than the relief valve that has been instrumented. This relief valve ensures a constant pressure of 6.5 bar (or 650 J.kg\(^{-1}\)) at its tailwater side. A discharge sensor and two pressure sensors at the headwater and tailwater side of the relief valve have been installed on the site. The average measured values have been recorded and stored every 15 minutes from July 22, 2013 to July 21, 2014. The installation is pictured in Figure 10 and the measurements are represented in Figure 11.

![Instrumented pilot site](image)

Figure 10: Instrumented pilot site.

4.2 Modelling of the recoverable specific energy

The net specific energy \( gH_n \) at the headwater side of the relief valve is evaluated by subtracting the energy losses in the upstream pipe \( gH \), from the gross specific energy \( gH \). The energy losses are assumed to be regular, following the model of Eq. [10], where \( Q \) is the measured discharge.

\[
gH_n = gH - gH_e = gH - k_q \cdot \frac{Q^2}{2} \quad [10]
\]

The gross specific energy \( gH \) and the equivalent regular loss coefficient \( k_q \) are identified by a least square curve fitting on the measurement data, assuming the net specific energy is deduced from the measured headwater pressure using Eq. [2]. The identified model is pictured in Figure 11, where \( gH = 1878 \) J.kg\(^{-1}\) and \( k_q = 4.12 \times 10^6 \) J.kg\(^{-1}\).m\(^6\).s\(^{-2}\). The 35040 measured points are represented by their joint density probability function: the darkest area covers the most frequent operating conditions.

Then, as the relief valve ensures a constant tailwater specific energy \( gH_t = 650 \) J.kg\(^{-1}\), the specific energy \( E_{net} \) that can be recovered on the pilot site is estimated as a function of the discharge \( Q \) defined by Eq. [11]:

\[
E_{net}(Q) = gH - k_q \cdot \frac{Q^2}{2} - gH_t
\quad [11]

4.3 Modelling of the density probability function of the discharge over the year

The yearly-averaged available hydraulic power \( P_{h,av} \) is estimated considering the density probability function of the discharge \( f_Q \) according to the probabilistic definition given in Eq. [12]:

\[
P_{h,av} = \int f_Q(q) \cdot P_{h}(q) \cdot f_Q(q) \, dq
\quad [12]

And the yearly-averaged available mechanical power \( P_{m,av} \) extracted by the counter rotating micro-turbine is estimated by integrating the efficiency of the machine in this integral:

\[
P_{m,av} = \int f_Q(q) \cdot \eta(q) \cdot E_{net}(q) \cdot f_Q(q) \, dq
\quad [13]

Then, Eq. [13] allows the comparison between the energy production with constant relative rotational speed using \( \eta_{rel} \) model and with independent optimal setting of the rotational speeds using \( \eta_{opt} \) model.

This approach therefore requires to model the density probability function of the discharge \( f_Q \). The non-parametric kernel smoothing method is used with a normal kernel. Considering a reduced centered normal distribution \( N \), its probability density function \( f_X \) and a set \( \{Q_i\} \) of measured discharge, the expression density probability function of the discharge \( f_Q \) is given by Eq. [14]:

\[
f_Q(x) = \frac{1}{nh} \sum_{i=1}^{n} f_X \left( \frac{x - Q_i}{h} \right)
\quad [14]
where \( n = 35040 \) is the number of points and \( h \) is a parameter called *bandwidth*. It is estimated with the Silverman rule as detailed in Silverman (1986) and implemented in the OpenTURNS library. The resulting probability density function is pictured in Figure 12.

![Figure 12](image)

**Figure 11:** Available specific energy and probability and joint probability density function of the measured points on the pilot site.

**Figure 12:** Marginal probability density function of the discharge measured on the pilot site over one year.

### 4.4 Comparison of the energy production with and without optimal setting of the rotational speeds

The probability density function of the discharge \( f_Q \) pictured in Figure 12 is used to evaluate both the average available hydraulic power \( P_{\text{av}} \) according to Eq. [12] and the average extracted mechanical power \( P_{\text{m,av,opt}} \) with optimum rotational speeds and \( P_{\text{m,av,1}} \) with fixed relative rotational speed.

The range of specific energy that can be harvested by the tested single stage counter rotating micro-turbine lies from 100 J.kg\(^{-1}\) to 275 J.kg\(^{-1}\) and the range of available specific energy \( E_{\text{rec}} \) lies from 400 J.kg\(^{-1}\) to 1200 J.kg\(^{-1}\). It is assumed that a serial installation of four stages operating under the same conditions, where each stage recovers \( E_{\text{rec}}/4 \).

As the range of discharge experienced on the pilot site is wider than the capacity of the micro-turbine, the micro-turbine is assumed to be by-passed when it is not capable to operate properly due to the limitation on the rotational speed. The efficiency models used to evaluate the power as defined in Eq. [13] are corrected to account for these limitations:

\[
\eta_{\text{corr}}(Q,E) = \begin{cases} 
\eta(Q,E) & \text{if } 300 \leq N_1(Q,E) \leq 3000 \text{ and } 300 \leq N_2(Q,E) \leq 3000 \\
0 & \text{else}
\end{cases}
\]

Table 3 summarizes the average power, estimated annual energy, considering that the micro-turbine operates during the 8760 hours of a year, and average efficiency for the two investigated operation mode. For both modes, the average efficiency is rather poor and this can be explained by two main causes:

\[ \text{[15]} \]
The efficiency measured on the test rig is itself rather low due to the elbowed architecture of the test rig leading to sizeable mechanical losses, evaluated around 15%, and volumetric losses estimated around 15%;

The micro-turbine discharge operation domain is tight compared to the discharge domain experienced on the pilot site. The machine is effectively operating only 29.0% of the time for the optimum mode and 30.8% for the fixed mode.

Even though its apparent lowest versatility, the optimum rotational speeds operating mode proposed in this paper allows to harvest 18.2 MWh per year while 17.4 MWh can be harvested with the fixed relative rotational speed investigated in Münch-Alligné et al. (2013).

Figures 13 to 16 shows the evolution of the efficiency, extracted mechanical power, rotational speeds and relative rotational speed along with the discharge considering the characteristic of the pilot site as modelled in Eq. [11] and pictured in Figure 11 considering the optimum rotational speeds mode and the fixed relative rotational speed mode.

Figure 13 and Figure 14 show the better performance of the optimum mode. However, this better performance is available on a tighter range of discharge than for the fixed relative rotational speed mode. Figure 15 and Figure 16 show that an extension of the accessible rotational speed domain over 3000 min$^{-1}$ may lead to an extension of the operational domain of the micro-turbine with optimized speeds.

Table 3: Yearly-averaged power, estimated annual energy and average efficiency of the micro-turbine under optimum and fixed relative rotational speeds.

<table>
<thead>
<tr>
<th>OPERATING MODE</th>
<th>AVERAGE POWER [kW]</th>
<th>ESTIMATED ANNUAL ENERGY [MWh]</th>
<th>AVERAGE EFFICIENCY [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Available hydraulic energy</td>
<td>9.54</td>
<td>84.2</td>
<td>-</td>
</tr>
<tr>
<td>Optimum rotational speeds</td>
<td>2.07</td>
<td>18.2</td>
<td>21.6</td>
</tr>
<tr>
<td>Fixed relative rotational speed</td>
<td>1.99</td>
<td>17.4</td>
<td>20.7</td>
</tr>
</tbody>
</table>
Figure 13: Efficiency curves for a 4-stages micro-turbine on the pilot site operating under optimum and fixed relative rotational speeds.

Figure 14: Mechanical power curves for a 4-stages micro-turbine on the pilot site operating under optimum and fixed relative rotational speeds.

Figure 15: Rotational speeds for a 4-stages micro-turbine on the pilot site operating under optimum and fixed relative rotational speeds.

Figure 16: Optimum and fixed relative rotational speed compared for a 4-stages micro-turbine on the pilot site.
5. CONCLUSIONS

This paper presents the concept of counter rotating micro-turbine dedicated to the harvesting on hydropower potential on drinking water networks. It is more particularly dedicated to the proposal of a strategy to independently set the rotational speed of each runner in order to reach optimum operating conditions, as a previous study suggested that it can lead to the enhancement of the machine’s performance.

The proposed approach consists in building a multilayer analytical model of the machine behavior: the so-called hill chart. Several intermediate models are built for different levels of specific energy allowing to identify the optimum operating conditions. A global model is finally built on the identified best efficiency ridges.

This model is used to evaluate the energy production on an instrumented pilot site. The performance under the identified optimum conditions is compared with the performance obtained with the baseline mode with the two runners rotating at the same speed.

The proposed method provides a 4% relative increase of energy production compared to the baseline mode. Furthermore, the proposed study allows to draw several conclusions. First, the main limiting factor towards a higher annual average efficiency is the limited discharge operating domain of the machine. This is very likely to be overcome by increasing the range of rotational speed of the machine. The efficiency with the proposed mode is higher than with the baseline mode but the operating domain obtained is smaller. The proposed method can therefore be improved in order to identify the optimal rotational speed also considering cases when one of the runner as reached its speed limit. This must enlarge the operating domain at least up to the baseline one and therefore increase the annual energy production.

Finally, this study only focuses on the mechanical energy extracted by the micro-turbine. Under the same framework, the efficiency of the electric machine and of the connection to the grid should also be integrated. This way, the method would no longer allows to maximize the extracted mechanical energy but will instead maximize the energy provided to the grid, which is the actual objective.

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