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# On the Dynamic Measurements of Hydraulic Characteristics

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**Abstract.** The present work introduces the implementation and validation of a faster method to measure experimentally the efficiency characteristics of hydraulic turbomachines at a model scale on a test rig. The case study is represented by a laboratory prototype of an in-line axial microturbine for water supply networks. The 2.65 kW one-stage variable speed turbine, composed by one upstream 5-blade runner followed by one counter-rotating downstream 7-blade runner, has been installed on the HES-SO Valais/Wallis universal test rig dedicated to assess performances of small hydraulic machinery following the IEC standard recommendations.

In addition to the existing acquisition/control system of the test rig used to measure the 3D hill-chart of a turbine by classical static point-by-point method, a second digitizer has been added to acquire synchronized dynamic signals of the employed sensors. The optimal acceleration/deceleration ramps of the electrical drives have been previously identified in order to cope with the purpose of a reduced measurement time while avoiding errors and hysteresis on the acquired hydraulic characteristics. Finally, the comparison between the turbine efficiency hill-charts obtained by dynamic and static point-by-point methods shows a very good agreement in terms of precision and repeatability. Moreover, the applied dynamic method reduces significantly (by a factor of up to ten) the time necessary to measure the efficiency characteristics on model testing.

## 1. Introduction

Hydropower remains one of the most important sources of renewable energy for electrical power production providing in 2013 about 16.6% of the world's electricity mix [1]. During the development process, either for a new- or for a refurbishment project, in both large and small hydro cases, hydraulic performance measurements are often still mandatory. Complementary, numerical simulation can be a relatively cost efficient tool to predict hydraulic efficiency and flow hydrodynamic stability especially in the normal operating range. However, the experimental tool remains essential to explore the whole operating range of a turbomachine, including the off-design conditions. Moreover, the dynamic behaviour in transient operation can sometimes be verified only by experimental means.

Standard hydraulic performance tests signify steady-state point-by-point measurements over the whole operating range of the machine [2]. The main advantage of this classical proved method comes from the fact that it allows for high-precision repetitive measurements and it often avoids possible hysteresis on the characteristics. However, the method is relatively time-expensive, mainly due to the large time necessary to reach a steady operating condition and to the long-enough acquisition time (usually several dozens of seconds) necessary to calculate valid average values of different parameters. In addition, the regulation, either manual or automatic, of a given constant hydraulic and/or



mechanical parameter (e.g. testing head, runner rotational speed, etc.) for each operating point is also relatively time consuming.

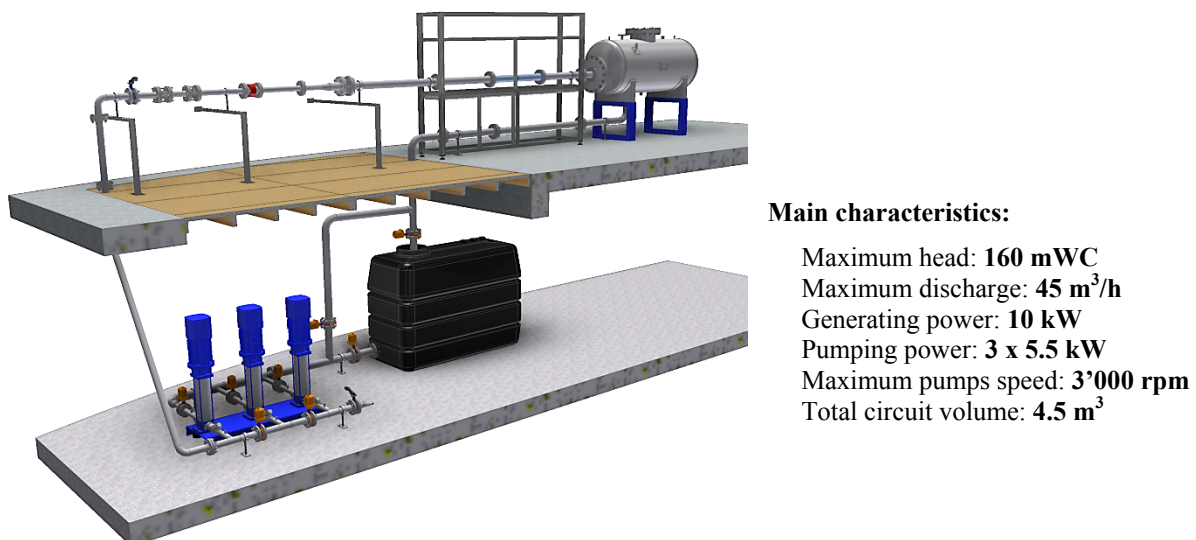
In the case of small hydro (installed power of less than 10 MW), mini-hydro (< 2 MW), micro-hydro (< 500 kW) as well as pico-hydro (< 10 kW), [3], for which the investment devoted to development is much more limited compared to the large hydro, the total time dedicated to perform a full performance measurements campaign must be as reduced as possible. In this context, a new dynamic method has been adopted with the main purpose of reducing the time necessary to perform the full hydraulic performance tests of a turbomachine. Indeed, this technique is inspired by the so-called “sliding-gate” method successfully used for index testing of Francis and Kaplan units [4],[5]. The general procedure for standard testing is applied with the exception that data is collected while the guide vanes are moved slowly and continuously through the desired operating range instead of using discrete gate positions. The trick is to move the gates slowly enough so the unit stays in quasi- steady-state condition. Relying on the same instrumentation as for classical point-by-point method, the “sliding-gate” method presents the advantage of obtaining continuous efficiency curves over the entire operating range while reducing significantly the total required time to perform the tests [6],[7]. This approach could be also used for fast detection of the different operating regions where hydraulic instabilities arise in the machine, such as part- and full-load cavitating vortex ropes, harmful rotor-stator pressure fluctuations and vibrations, or even flow rotating-stalls.

The present work introduces the implementation and validation of the dynamic “sliding-gate” method to measure the efficiency characteristics on hydraulic turbomachinery model testing. The employed methodology comes first with the experimental setup, including the employed hydraulic test rig and the double-regulated axial counter-rotating microturbine. Then, the instrumentation and the measurements protocol used for both the static point-by-point and the dynamic methods are presented. The results focus first on the selection and validation of the acceleration/deceleration ramps of the runners necessary for dynamic measurements. In the end, the hydraulic hill-charts obtained by the two methods are compared.

## 2. Methodology

### 2.1. Experimental setup

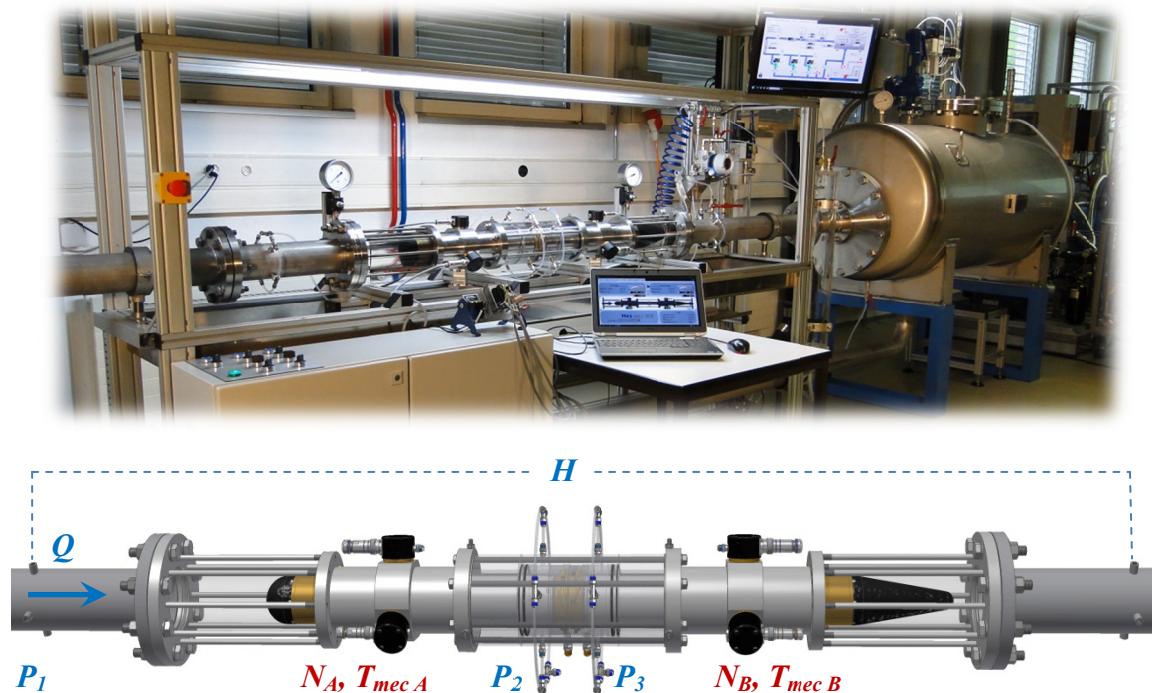
2.1.1. *Hydraulic test rig.* The hydraulic performance measurements are carried out on the universal



**Figure 1.** Hydraulic test rig of the HES-SO Valais//Wallis – Switzerland, [8].

hydraulic test rig of the HES-SO Valais//Wallis - Switzerland (see Figure 1) adapted to small-power turbomachines and/or different hydraulic components with complex geometries. The IEC 60193 [2] standard recommendations on hydraulic model testing are implemented. The closed-loop circuit is supplied with hydraulic power by three recirculating multistage centrifugal pumps connected in parallel. The variable speed pumps (3x5.5 kW) can deliver a maximum discharge of 3x15 m<sup>3</sup>/h and a maximum pressure of 160 mWC. The testing variable-speed model is installed in the upper part of the circuit in a comfortable position, in terms of instrumentation, operation and experimental observations. The free-surface pressurized reservoir placed downstream the test section allows simulating different setting levels of the model and therefore determining the cavitation performances as well. Several valves, solenoid and/or manual, are employed to control the hydraulic circuit operating configuration as well as its filling, cooling and draining. The operation of the test rig is driven with an automatic system through a customized LabVIEW interface that allows for real-time measurement and display instantaneous values of pumps speed, flow discharge, testing head, water temperature and Thoma number. To complete, the autonomous regulation system is programmed to keep constant the value of the desired parameters, as the pumps speed, the testing head or the discharge, while ensuring the security (in terms of overpressure, reservoir water level, etc.) In addition, the implemented wireless communication architecture between the hydraulic test rig and further measurement/monitoring systems (e.g. testing model control system) allows for safe data centralization, storage and sharing [8].

**2.1.2. Counter-rotating microturbine – bulb version.** The case study consists of a fully instrumented laboratory prototype of an axial microturbine with counter-rotating runners dedicated to recover the energy lost in release valves of water supply networks, illustrated in Figure 2. Axial turbines are generally used in cases of low head and high specific speed. However, the multi-stage concept (similar to multi-stage centrifugal pumps) allows working under high head operating conditions (a typical regime of Pelton turbines), each stage recovering a fraction of the available head. The one-stage variable speed turbine is composed by one upstream 5-blade runner followed by one counter-rotating downstream 7-blade runner. The turbine inner diameter is 80 mm, whereas its outer diameter is



**Figure 2.** Experimental setup of the counter-rotating microturbine installed on the test rig.

100 mm; the total length of the prototype is 1.5 m. It has been designed to provide about 2.65 kW for a discharge of 37.5 m<sup>3</sup>/h and a head of 20 m [9],[10]. A best efficiency of 85% is reached by numerical simulation for a ratio  $\alpha = N_A/N_B = 1$  between the runners absolute rotational speed. The optimal regulation of the turbine beyond the best efficiency point is ensured by changing the relative rotational speed between the two runners.

Each of the inlet and outlet bulbs houses an independent electrical generator, specially designed for this turbine [11], a torque meter and an incremental encoder. The variable speed electrical generators are independently driven by frequency converters, capable to keep constant a given rotational speed value whatever the sign of the mechanical torque. Further, a magnetic coupling and a hydrodynamic bearing relay the runner to the generator, while separating the wet and the dry regions of the machine. The inlet, central and outlet sections, made of Plexiglas, allows for cavitation and flow visualization. Finally, in order to limit the leakage between the tip of the blades and the outer fixed wall, the tested version includes a band of Plexiglas, provided with a labyrinth, which has been installed at the periphery of each runner. The annular cross section of the machine has been kept unchanged.

## 2.2. Instrumentation

The test rig is equipped with several measurement instruments with the aim of recovering the full testing conditions and particularly the hydraulic power of the testing model. The characteristics of the main measurement instruments that equip the test rig are provided in Table 1, including the output signal type, the measurement range and the absolute error. The discharge  $Q$  is measured with the help of an electromagnetic flowmeter. The head  $H$  and the setting level  $H_s$  are measured with differential pressure transducers, whilst the static pressure at the wall  $M_{1,2,3}$  is measured with capacitive absolute pressure transducers. All pressure sensors are connected to the measurement section through wall static pressure collectors (see Figure 2). The temperature  $T$  is retrieved with the help of a PT100 transducer. Then, the rotational speed of the recirculation pumps  $N_{p1, 2, 3}$  is measured with optical tachometers, while vibrating tuning fork detectors are used for the minimum, maximum and security water level  $L_{\min, \max, s}$  measurements. Finally, the manometers and the mano-vacuum-meters are used as visual indicators for the test rig operator.

In addition to the measurement instrumentation already contained on the test rig, each of the two runners of the microturbine prototype is equipped with an incremental encoder and a torque meter necessary for the driving of the electrical generators and, of course, for the computation of the mechanical power.

**Table 1.** Characteristics of the main instruments.

Acronym	Measured/displayed quantity	Sensor type	Output signal	Range	Precision
$Q$	Discharge	Electromagnetic flowmeter	4..20 [mA]	0..50 [m <sup>3</sup> /h]	± 0.5 [%]
$H$	Head	Differential pressure sensor	4..20 [mA]	0..16 [bar]	± 0.1 [%]
$H_s$	Setting level	Differential pressure sensor	4..20 [mA]	0..5 [bar]	± 0.2 [%]
$M_{1, 2, 3}$	Absolute static pressure	Capacitive pressure transducer	4..20 [mA]	0..10/20 [bar]	± 0.05 [%]
$T$	Temperature	PT100 transducer	4..20 [mA]	0..100 [°C]	± 0.1 [%]
$N_{p1, 2, 3}$	Pump rotational speed	Tachometer	24 [V] pulse	0..1000 [Hz]	-
$T_{\text{mec A, B}}$	Mechanical torque	Torque meter	0..±10 [V]	0..±7.5 [Nm]	± 1 [%]
$N_{A, B}$	Turbine rotational speed	Incremental encoder	24 [V] pulse	0..7500 [rpm]	2048 [ppr]

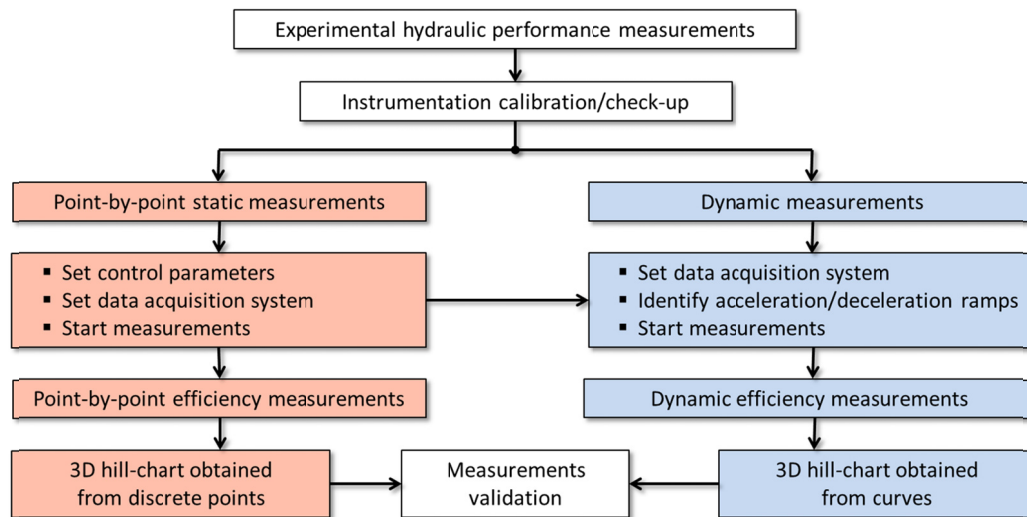
A National Instruments (NI) autonomous digitizer CompactRIO 9074 equipped with different acquisition/control modules is used to measure all the quantities related to the test rig (including the hydraulic power). Then a NI CompactDAQ 9174 digitizer is used to drive the testing model and to measure the mechanical torque and the rotational speed of the runners. To cope with the classical static point-by-point method, by default, both digitizers perform measurements over 8 seconds at 50 Hz (configurable by the user depending on the stability of the operating condition) and compute the average and the standard deviation values for all parameters. Finally, a second CompactDAQ 9174 digitizer is added to acquire synchronized dynamic signals of the sensors in parallel with the existing



acquisition/control system of the test rig. The acquisition time is set long enough in order to get measurements over the full variation of the micro-turbine runners speed from minimum to maximum. The acquisition frequency is set to 100 Hz, fast enough considering the frequency response of the employed sensors.

### 2.3. Measurements protocol

The experimental measurements protocol employed in this study is provided in Figure 3. Prior to the beginning of measurements, all the employed instruments have been calibrated and/or rechecked.



**Figure 3.** Flowchart of the employed experimental protocol.

In the case of classical point-by-point method, measurements have been performed at 11 different testing head values of the turbine (see Table 2). For each constant testing head, different combinations of runners rotational speeds were chosen over the whole possible operating range of the turbine; in total, 9 values of constant ratio  $\alpha$  between the runners absolute rotational speed were systematically considered. In total, more than 1'000 operating points were acquired. As a result, the efficiency diagrams for obtained for 11 heads have been used to build the full 3D hill-chart on the whole operating range of the turbine.

**Table 2.**Combinations of runners rotational speeds for point-by-point measurements.

$H = 0.5 / 0.75 / 1 / 1.25 / 1.5 / 1.75 / 2 / 2.25 / 2.5 / 2.75 / 3$ [bar]									
$N_A$ [rpm]	$N_B$ [rpm]								
0	0								
250	0	62.5	125	187.5	250	332.5	500	1000	250
500	0	125	250	375	500	665	1000	2000	500
750	0	187.5	375	562.5	750	1000	1500	3000	750
1000	0	250	500	750	1000	1334	2000		1000
1250	0	312.5	625	937.5	1250	1662.5	2500		1250
1500	0	375	750	1125	1500	2000	3000		1500
1750	0	437.5	875	1312.5	1750	2327.5			1750
2000	0	500	1000	1500	2000	2667			2000
2250	0	562.5	1125	1687.5	2250	3000			2250
2500	0	625	1250	1875	2500				2500
2750	0	687.5	1375	2062.5	2750				2750
3000	0	750	1500	2250	3000				3000
$\alpha = N_B/N_A$ [-]	0	0.25	0.5	0.75	1	1.33	2	4	$\infty(N_A=0)$

Then, a second digitizer has been added to acquire synchronized dynamic signals of several sensors (e.g. discharge, head, mechanical torque and runner rotational speed) in parallel with the existing acquisition/control system. The dynamic measurements of hydraulic efficiency are performed at different constant speeds of test rig recirculating pumps while increasing and decreasing the speed of the turbine runners from minimum to maximum and vice versa for different constant runners absolute rotational speed ratios (see Table 3). Prior to the dynamic tests, the optimal acceleration/deceleration ramps of the electrical drives have been identified in order to cope with the purpose of a reduced measurement time while avoiding measurement errors and hysteresis on the acquired characteristics. In the end, the results obtained by classical point-by-point measurements method are used to validate the ones obtained by the dynamic method.

**Table 3.** Minimum-to-maximum runners rotational speeds variation for dynamic measurements.

$N_{p1,2,3}$ [rpm]										$\alpha = N_B/N_A$ [-]
	720	1000	1250	1500	1750	2000	2250	2500	2750	
$N_A$ [rpm]	0÷3000	0÷3000	0÷3000	0÷3000	0÷3000	1000÷3000	2250÷3000	2500÷3000	2750÷3000	<b>0</b>
	0÷3000	0÷3000	0÷3000	0÷3000	1250÷3000	-	-	-	-	<b>0.125</b>
	0÷3000	0÷3000	0÷3000	0÷3000	1250÷3000	2500÷3000	-	-	-	<b>0.25</b>
	0÷3000	0÷3000	0÷3000	0÷3000	1000÷3000	1750÷3000	2750÷3000	-	-	<b>0.5</b>
	0÷3000	0÷3000	0÷3000	0÷3000	750÷3000	1500÷3000	2250÷3000	-	-	<b>0.75</b>
	0÷3000	0÷3000	0÷3000	0÷3000	750÷3000	1250÷3000	1750÷3000	2500÷3000	-	<b>1</b>
	0÷2250	0÷2250	0÷2250	0÷2250	562.5÷2250	937.5÷2250	1500÷2250	-	-	<b>1.33</b>
	0÷1500	0÷1500	0÷1500	0÷1500	375÷1500	750÷1500	1375÷1500	-	-	<b>2</b>
	0÷750	0÷750	0÷750	0÷750	187.5÷750	375÷750	-	-	-	<b>4</b>
	0÷375	0÷375	0÷375	0÷375	125÷375	-	-	-	-	<b>8</b>
$N_B$ [rpm]	0÷3000	0÷3000	0÷3000	0÷3000	1000÷3000	-	-	-	-	$\infty$ ( $N_A=0$ )

### 3. Results

#### 3.1. Dynamic measurements – runners speed acceleration/deceleration ramp selection

Considering the reduced size and the geometrical complexity of the microturbine, only the hydraulic-to-mechanical efficiency  $\eta_{h-m}$  could be measured. As expressed in eq. (1), this efficiency is the result of the product between the hydraulic  $\eta_h$  and the bearing efficiency  $\eta_m$ . Further, the hydraulic efficiency includes the efficiency of the disc friction  $\eta_{rm}$ , the energetic efficiency  $\eta_e$  as well as the volumetric efficiency  $\eta_q$ .

$$\eta_{h-m} = \eta_h \cdot \eta_m = (\eta_e \cdot \eta_q \cdot \eta_{rm}) \cdot \eta_m [\%] \quad (1)$$

$$\eta_{h-m} = \frac{P_{mec}}{P_h} = \frac{(\omega_A \cdot T_{mecA}) + (\omega_B \cdot T_{mecB})}{\rho \cdot Q \cdot E} [\%] \quad (2)$$

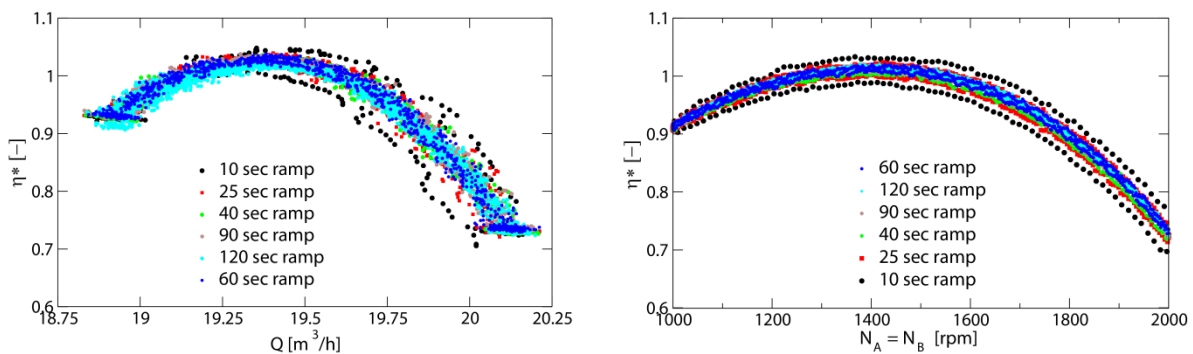
The final formulation of the hydraulic-to-mechanical efficiency is given by the ratio between the mechanical power recovered by both runners and the hydraulic power of the whole one-stage turbine (eq.(2)).  $\omega_{A,B}$  and  $T_{mecA,B}$  represents respectively the angular speed and the mechanical torque of the first (A) and of the second (B) runner. For the hydraulic power, whilst the discharge  $Q$  is measured with the electromagnetic flowmeter, the total specific energy  $E = gH$  is recovered with the differential pressure sensor between the inlet and the outlet sections of the machine (see Figure 2). Indeed, considering the Bernoulli's equation, the calculation of the specific energy only from the difference of

static pressures is possible since the machine is placed horizontally and the inlet and outlet cross sections are equal and far enough from the central section changes.

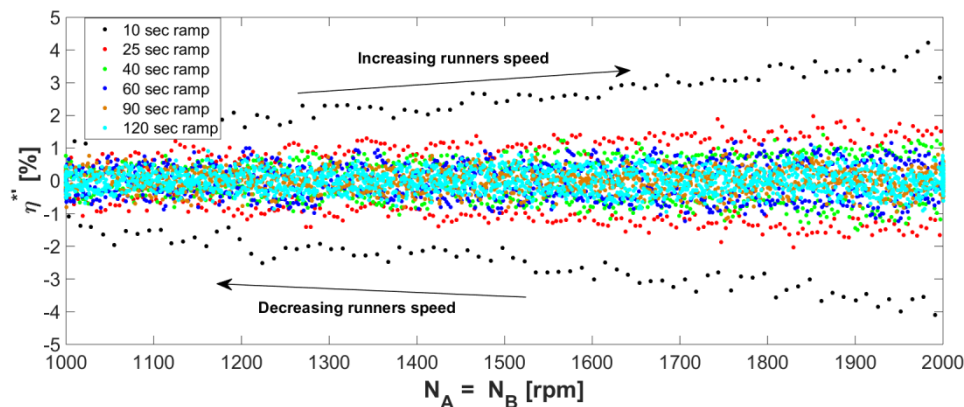
When performing dynamic measurements, one of the most important steps is to find the optimal speed of the applied variation in order to keep the turbine in a quasi- steady-state operating mode during the whole measurement process. For our case, this step consists in establishing the optimal acceleration/deceleration ramp of the runners speed .To this end, the speed of the three recirculating pumps was set and maintained constant at a value of 1500 rpm. For a constant runners absolute rotational speed ratio  $\alpha = 1$ , the runners speed was uniformly increased from 1000 rpm to 2000 rpm and then decreased back to 1000 rpm. In total, 6 different acceleration/deceleration ramps of respectively 10, 25, 40, 60, 90 and 120 sec/1000 rpm have been addressed. The resulting influence of the acceleration/deceleration ramp of the runners speed on the efficiency hysteresis for fixed inflow conditions is presented in Figure 4. The dimensionless efficiency is computed with the eq. (3), taking into account the maximum efficiency determined from all static measurements. The dependency of the efficiency on both the discharge and the runners rotational speed shows hysteresis and large measurement errors for speed ramps below 40 sec/1000 rpm.

$$\eta^* = \frac{\eta_{h-m}}{\max(\eta_{h-m})} [-] \quad (3)$$

$$\eta^{*'} = \eta^* - \eta_{interp}^* [-] \quad (4)$$



**Figure 4.** Influence of the acceleration/deceleration ramp of the runners speed on the efficiency hysteresis for fixed inflow conditions (pumps speed  $N_p = 1500$  rpm).

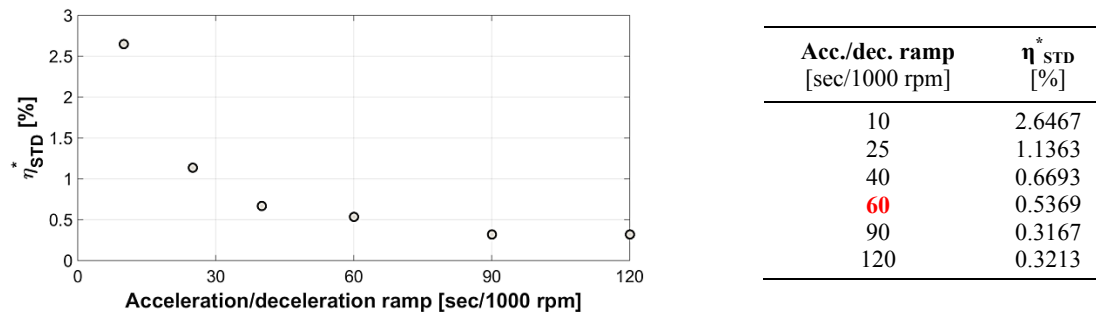


**Figure 5.** Efficiency fluctuation during an increasing/decreasing cycle at fixed inflow conditions (pumps speed  $N_p = 1500$  rpm) for different speed acceleration/deceleration ramps.

In order to quantify the influence of the speed ramp on the obtained measurement errors, the efficiency fluctuations  $\eta^{*'}$  compared to the average efficiency obtained by curve fitting (see eq. (4)) are



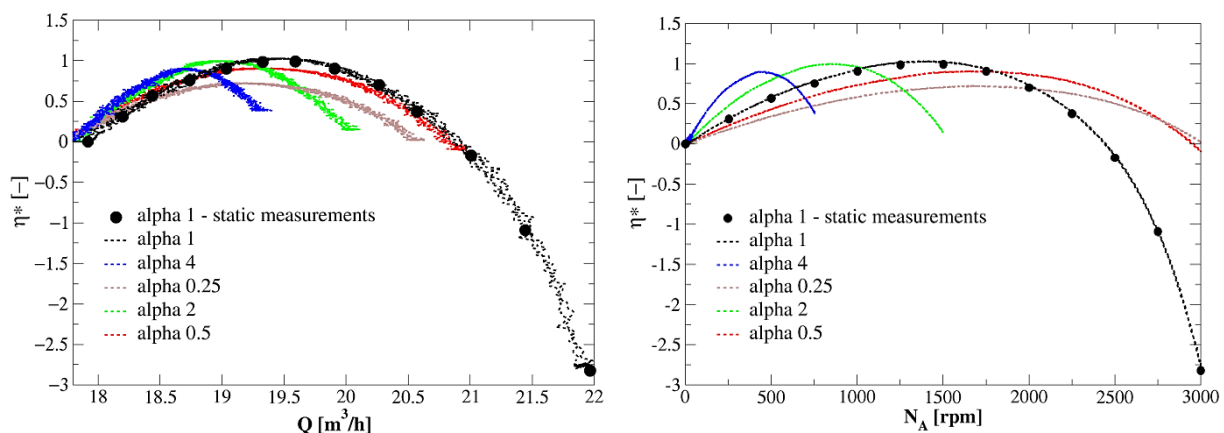
plotted in Figure 5. Accordingly, for speed ramps larger than 40 sec/1000 rpm the hysteresis effect seems to disappear. Indeed, the resulting standard deviation (STD) of the efficiency fluctuation  $\eta^*_{STD}$  depending on the speed ramp, provided in Figure 6, comes to confirm this statement. For a speed acceleration/deceleration ramp of 60 sec/1000 rpm, selected for the further dynamic measurements as a good compromise between a reduced measurement time and an acceptable measurement precision, the efficiency errors are below 1%.



**Figure 6.** Resulting efficiency STD depending on the speed acceleration/deceleration ramp.

### 3.2. Dynamic measurements results and validation

In Figure 7, a comparison between the efficiency obtained by dynamic and static measurement methods is performed for fixed inflow conditions (speed of recirculating pumps of  $N_p = 1500$  rpm) and a runners absolute rotational speed ratio of 1. For the discrete point by point method the runners speed has been modified in steps of 250 rpm from 0 to 3000 rpm. For the dynamic method, the speed of the runners has been continuously increased and then decreased from 0 to 3000 rpm and vice versa. One may state here a good agreement between the obtained results with a maximum error of 1% between the mean values. Moreover, the results of the dynamic method fit very well the ones of the static point-by-point method on the whole operating range, including off-design conditions. The negative values of efficiency indicate actually a negative mechanical torque on the runners, corresponding to the turbine brake operating mode.

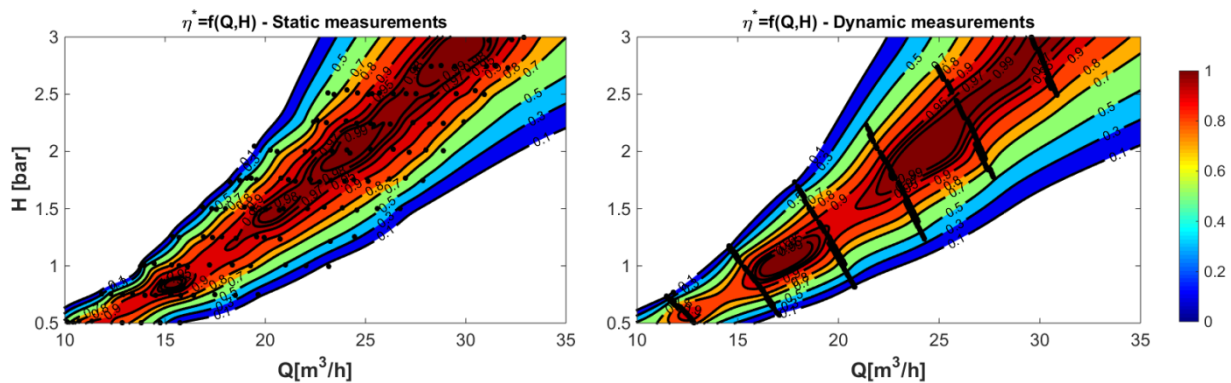


**Figure 7.** Comparison between dynamic and static efficiency measurements at fixed inflow conditions (pumps speed  $N_p = 1500$  rpm) for different runners rotational speeds ratio.

The resulting efficiency hill-chart contours obtained by classical static point-by-point and dynamic measurement methods are presented in Figure 8. The obtained results are from qualitative point of view the same. However, a significant difference between the two methods is observed on the distribution of measured operating points: for the static measurements method the obtained operating

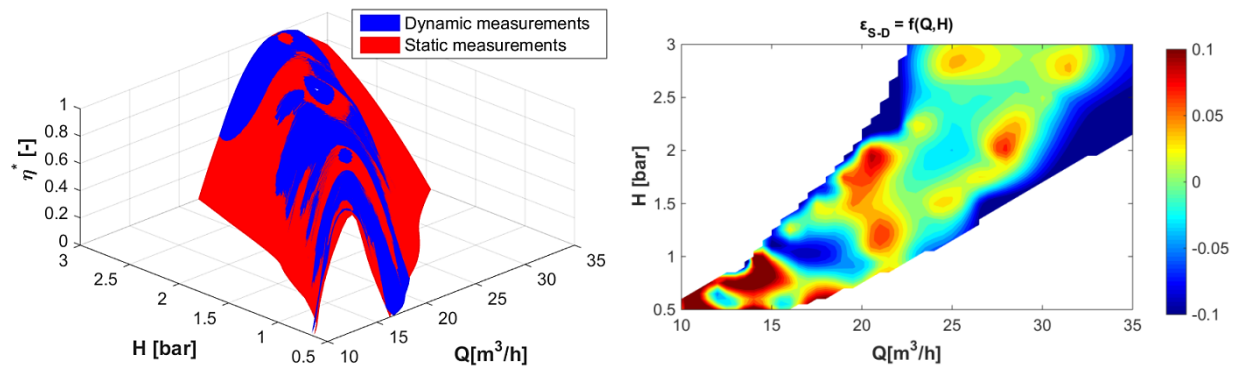
points crosses horizontally the final Q-H turbine characteristic, while the intersection with a typical characteristic of a pump is noticed from the distribution of operating points on the dynamic method.

Finally, the superposition of the two resulting 3D hill-surfaces (see Figure 9) shows a satisfactory match between the results and validates the new proposed dynamic measurements method. The differences between the two hill-charts, computed with the eq. (5), are mainly due to the quality of the obtained interpolated surfaces. A more refined measurements grid and/or a different surface interpolation method should remove these artificial differences.



**Figure 8.** Resulting efficiency hill-charts contours obtained by classical static point-by-point and dynamic measurement methods.

$$\varepsilon_{S-D} = \eta_{Static}^* - \eta_{Dynamic}^* [-] \quad (5)$$



**Figure 9.** Validation of resulting hill-chart obtained by the dynamic measurements method with the results of the classical static point-by-point method.

#### 4. Conclusions and perspectives

The implementation and validation of a faster method to measure the efficiency characteristics of a hydraulic turbomachine has been introduced. Indeed, the so-called “sliding-gate” dynamic method has been adapted and successfully tested. The universal test rig of the HES-SO Valais/Wallis - Switzerland has been employed. A laboratory prototype of an in-line axial microturbine with counter-rotating runners has been selected as case study. The one-stage variable speed turbine is composed by one upstream 5-blade runner followed by one counter-rotating downstream 7-blade runner with a hub/shroud ratio of 80/100mm, designed to provide about 2.65 kW.

The applied protocol consisted in a first step on measuring the 3D hill-chart of the turbine by the classical static point-by-point method. Then, a second digitizer has been added to acquire synchronized dynamic signals of the sensors in parallel with the existing acquisition/control system of the test rig. The dynamic measurements of hydraulic efficiency have been performed at different

constant speeds of the test rig recirculating pumps while increasing and/or decreasing the speed of the turbine runners from zero to maximum, and vice versa, for different constant speed ratios. The optimal acceleration/deceleration speed ramps of the electrical drives have been previously identified in order to find an optimum that ensures a reduced measurement time while avoiding measurement errors and hysteresis on the acquired characteristics.

In the end, the efficiency hill-chart of the turbine obtained by dynamic measurements has been successfully compared (in terms of precision and repeatability) to the one measured by static point-by-point method. Moreover, the new applied dynamic method has shown a significant reduction (by a factor of up to ten) on the time necessary to measure the efficiency characteristics of a testing model. This approach is actually particularly gainful for small-hydro, for which the investment devoted to development is limited. In perspective, the same method could be also applied to detect hydrodynamic instabilities within the operating range of a turbomachine machine, or for fast detection of hydrodynamic instability operating regions.

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