Dynamic Efficiency Measurements on Hydraulic Turbomachinery Model Testing: Examples of Implementation and Validation

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Abstract

The implementation of the so-called "sliding-gate" dynamic method to measure in a faster way (up to ten times) the efficiency characteristics on hydraulic turbomachinery is presented. The new model testing measurement method is successfully applied on two different cases dedicated to recover the energy lost in release valves of water supply networks: a 2.65 kW double-regulated laboratory prototype of an in-line axial microturbine with two independent variable speed counter-rotating runners; a 11 kW multi-stage centrifugal pump as turbine (PAT) with variable speed.
The universal test rig of the HES-SO Valais/Wallis, Switzerland, dedicated to assess hydraulic performances of small-power turbomachines, has been employed. The applied procedure consists, in a first step, on measuring the 3D hill-chart of a given testing model (turbine or pump) by the classical static point-by-point method. Then, a second digitizer is 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 efficiency are performed at different constant speeds of the test rig recirculating pumps while increasing and/or decreasing the speed of the testing model runner(s) from zero to maximum, and vice versa, slowly enough in order to keep a steady-state operation. In the end, the resulting 3D efficiency hill-charts of the two tested machines obtained by the dynamic and by the classical static point-by-point measurement methods are compared, with the measurement precision and repeatability particularly emphasized.

Keywords: hydraulic efficiency, dynamic measurements, turbomachines, variable speed, model testing

1. Introduction

Historically, the main objective of model testing was to cast predictions on the performances of an industrial machine. Nowadays, it plays the additional role of addressing a large amount of contract guarantees, Fabre et al. [1]. Furthermore, while the progress of technologies employed on the test rigs has conducted to an increasing level of measurement accuracy, the role of the experimenter remains crucial to ensure reliable and accurate measurements.
In hydraulic turbine development, for both new- or refurbishment projects, either of large or of small scale, model tests remain often mandatory. Further, numerical simulation has established as a complementary relatively cost efficient tool to predict hydraulic efficiency and flow hydrodynamics mainly in the normal operating range. However, regarding the characteristics at off-design conditions, experimental investigation remains often irreplaceable. In addition, the dynamic behavior in transient operation can sometimes be drawn only experimentally.
According to the IEC [2] standard recommendations, hydraulic performance is retrieved by steady-state point-by-point measurements over the whole operating range of a turbomachine. This classical method, beyond the fact that is largely proved, it allows for reliable and accurate measurements. However, requiring a large time necessary to reach a steady operating condition and usually several additional dozens of seconds of acquisition time to calculate valid average values of different parameters, the method is relatively time-expensive.
Focusing on small hydro (as well as on mini-, micro-, and pico-hydro), Münch-Alligné & Avellan [3], as the budget allocated to the development is much more limited (compared to the large-hydro), the total men-

hours investment dedicated to the performance measurements must be reduced as much as possible. To this end, with the main purpose of reducing the time necessary to perform the full hydraulic performance tests of a turbomachine, a new dynamic method has been implemented and validated in Hasmatuchi et al. [4]. This technique is actually an adaptation of the so-called "sliding-gate" method, successfully used for index testing of Francis and Kaplan units (Almquist et al. [5], Almquist and Bickford [6]). It consists of applying the general procedure for standard testing, but instead of using discrete gate positions, data is collected while the guide vanes are moved slowly and continuously through the desired operating range. During the tests, the motion of the gates must be slow enough so that quasi- steady-state operating condition is ensured. The "sliding-gate" method uses the same instrumentation as the classical point-by-point one, with the advantage of obtaining continuous efficiency curves over the whole operating range and with a significant reduction of the total required time to perform the tests (Abgottspon and Staubli [7], Abgottspon et al. [8]).
The current work comes with two examples of implementation and validation of dynamic efficiency measurements on hydraulic turbomachinery model testing: a double-regulated axial counter-rotating variable speed microturbine and a centrifugal multi-stage variable speed pump as turbine. The experimental setup, including the employed hydraulic test rig and the two case studies are introduced. Then, the instrumentation and the experimental methodology are presented. The results focus first on the selection and validation of the optimal acceleration/deceleration ramps of the runners to ensure a steady-state condition for the dynamic measurements. Finally, the resulting hydraulic hill-charts obtained by the two methods are compared.

2. Experimental setup

2.1. HES-SO VS hydraulic test rig

The universal hydraulic test rig of the HES-SO Valais//Wallis - Switzerland (see Figure 1) dedicated to small-power turbomachines has been employed to perform the hydraulic performance measurements. Its configuration, instrumentation and operation follow the IEC 60193 [2] standard recommendations on hydraulic model testing. The closed-loop circuit of the test rig is supplied by three recirculating multistage centrifugal pumps connected in parallel. The variable speed pumps (2x18.5 kW and respectively 1x5.5 kW) can deliver a maximum discharge of about 100 m^3/h and a maximum pressure of 160 mWC. The testing variable-speed model is placed in the upper part of the circuit, upstream a free-surface pressurized reservoir. The latest allows simulating a given setting level of the model, either positive or negative, and thus investigating also the cavitation performances. The operation of the test rig is ensured by an automatic system. Its customized LabVIEW interface allows for real-time measurements and displays the instantaneous values of pumps speed, discharge, testing head, water temperature, Thoma number etc. The autonomous regulation system can keep constant the value of the pumps speeds, the testing head or the discharge. Finally, the wireless communication architecture between the hydraulic test rig and the measurement/monitoring systems (e.g. testing model control system) ensures safe centralization of data, storage and sharing, Hasmatuchi et al. [9].

Main characteristics:

Maximum head: **160 mWC**
Maximum discharge: **±100 m^3/h**
Generating power: **20 kW**
Pumping power: **2x 18.5 kW & 1x 5.5 kW**
Maximum pumps speed: **3'500/3000 rpm**
Total circuit volume: **4.5 m^3**

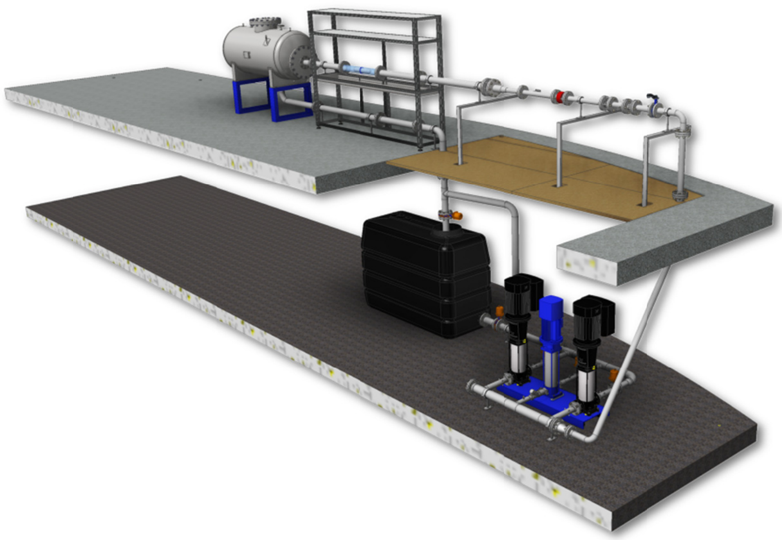


Figure 1. Hydraulic test rig (2015 version) of the HES-SO Valais//Wallis – Switzerland, Hasmatuchi et al. [9].

2.2. Case studies

2.2.1. *Counter-rotating microturbine – bulb version*

The first case study consists of a fully instrumented laboratory prototype of an axial microturbine with counter-rotating runners (see Figure 2) dedicated to recover the energy lost in release valves of water supply networks. Its multi-stage concept, similar to the one of centrifugal pumps, makes it appropriate for high head operating conditions, specific to the Pelton turbines. Indeed, each stage, composed by two counter-rotating axial runners, generally used for low head and large discharge conditions, recovers a fraction of the total head. The one-stage turbine of 2.65 kW is composed by two counter-rotating runners with respectively 5 and 7 blades. The outer diameter of the runners, as well as the pipe diameter, is 100 mm; the turbine inner diameter is 80 mm. The tested version includes a band, installed at the periphery of each runner, provided with a labyrinth in order to limit the leakage between the tip of the blades and the outer fixed wall. At the nominal operating point (a discharge of 37.5 m^3/h and a head of 20 m), for a ratio $\alpha = N_A/N_B = 1$ between the runners absolute rotational speed, its hydraulic efficiency, obtained by numerical simulation, reaches 85% (Biner et al. [10], Münch-Alligné et al. [11]). The optimal operation of the turbine is ensured by the relative rotational speed between the two runners along with their absolute rotational speed.
Further, the runners are driven by two independent electrical generators specially designed for this turbine, Melly et al. [12], placed into the upstream and downstream bulbs respectively. Two frequency converters are used to drive the variable speed electrical generators at the desired constant rotational speed value, whatever the sign of the mechanical torque. Finally, for each runner, an incremental encoder, used mainly for the rotational speed driving, along with a torque meter ensure the mechanical power measurements. In addition, the wet and the dry regions of the machine are separated using a sealed magnetic coupling.

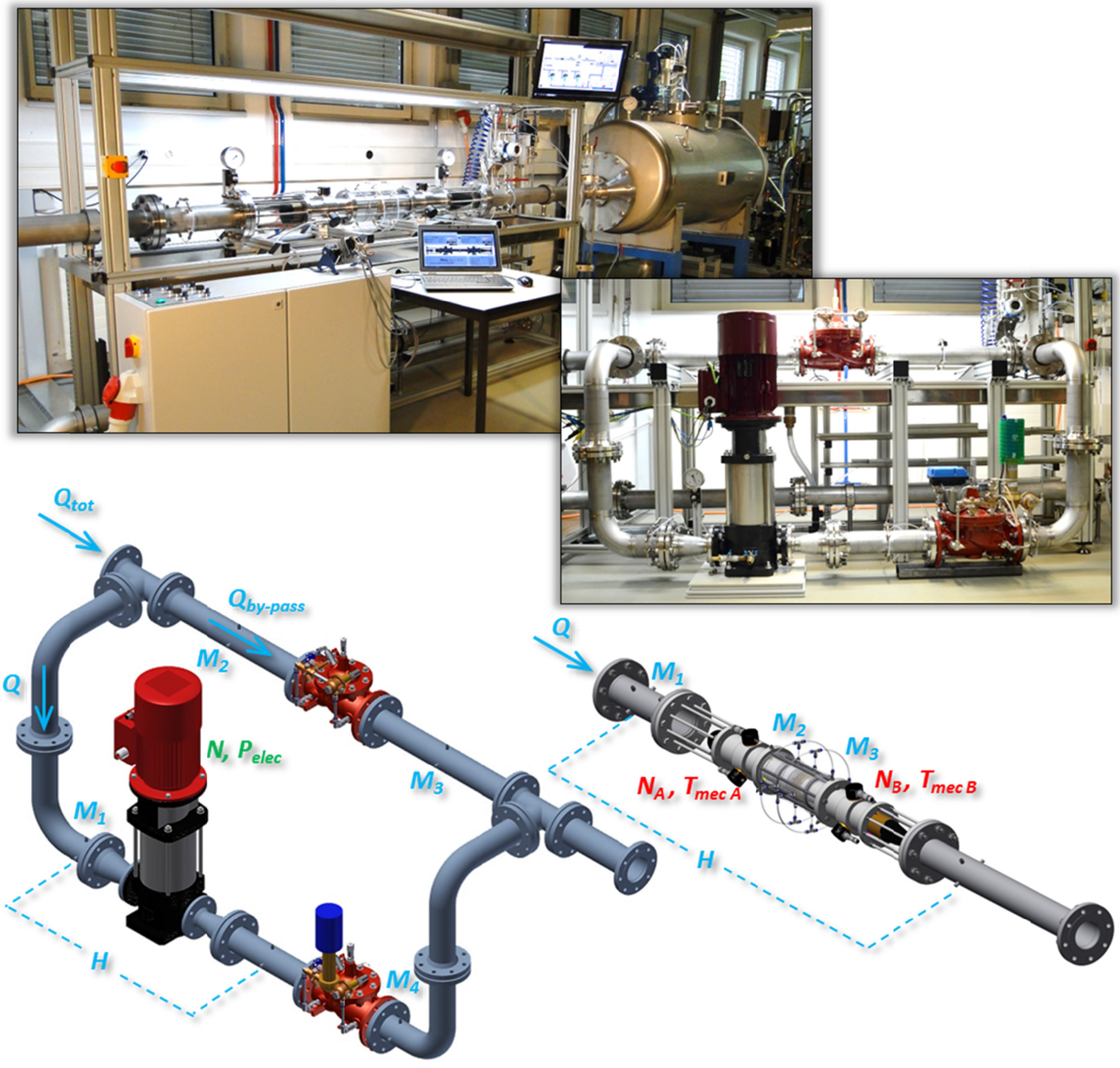


Figure 2. Experimental setup of the counter-rotating microturbine and of the PAT installed on the hydraulic test rig.

2.2.2. *Multi-stage centrifugal pump as turbine (PAT)*

Running a standard pump as turbine is often a cost-efficient solution to recover the extra available energy of the water supply systems, and not only - Orchard and Klos [13], Williams [14], Ramos and Borga [15], Motwani et al. [16]. However, prior to the installation, its operation and control in turbine mode must be investigated, Garay [17]. Indeed, if the hydraulic characteristics may be derived using, for example, the experimental data provided by Chapallaz et al. [18], or the method presented by Derakhshan and Nourbakhsh [19], the hydrodynamic stability mainly outside the normal operating range remains difficult or even impossible to predict without experimental tests.

In this context, a second case study dedicated to the energy recovery in water supply networks, consisting of an 11kW multi-stage centrifugal pump used as turbine (Ebara EVMG32 5-0F5/11) has been considered - see Figure 2. Its five radial-axial runners (with 5 blades each) have an inlet diameter of 136 mm and an outlet diameter of 67.5 mm. The turbine is regulated only with the help of the variable rotational speed of the runners. The permanent magnet synchronous generator, Leroy-Sommer LSRPM 132 M, with a nominal power of 15.8 kW, has been driven, in this case as well, by a frequency converter able to keep constant the desired rotational speed value, whatever the sign of the mechanical torque or the rotational speed. Concerning the instrumentation, the machine is equipped only with an incremental encoder, necessary for the rotational speed driving. Therefore, for this case, only the electrical power of the generator has been measured.

One may state here that the turbine has been tested in a hydraulic configuration similar to the one of the pilot site, where it has been already installed. The release valve installed on the by-pass branch, which plays the role of an existing release valve on a drinking water supply network, ensures the extra flow passage when the requested discharge is larger than the maximum flowrate of the turbine. The second release valve, placed downstream the turbine, limits the pressure at the outlet of the main branch, and therefore secures the network, whatever the operating conditions of the turbine. Moreover, in the case of discharges lower than the minimum value allowed for the turbine, or in the case of a failure, this release valve is able to cut completely the main branch, the whole discharge passing automatically through the by-pass one. Anyway, during the performance tests on the turbine, the by-pass release valve remained closed all the time, while the one of the turbine branch has been kept completely opened.

2.3. Instrumentation

The characteristics of the main employed instruments necessary to recover the hydraulic performances of the testing model along with the testing conditions are provided in Table 1. On the one hand, the test rig is equipped with an electromagnetic flowmeter for the discharge Q, two differential pressure transducers for the head H and the setting level H_S respectively and three additional capacitive absolute pressure transducers for the static pressure at the wall $M_{1,2,3}$. The latest are connected through wall static pressure collectors, as may be seen in Figure 2. A PT100 transducer for the water temperature T and three optical tachometers for the rotational speed of the recirculation pumps $N_{p1,2,3}$ come to complete the list of instruments.

On the other hand, the axial microturbine is equipped with two incremental encoders and two torque meters, one for each counter-rotating runner, necessary for the driving of the electrical generators and for the computation of the mechanical power. Finally, an incremental encoder and a high-precision electrical multimeter are used to drive the PAT's electrical generator and to compute the electrical power.

Regarding the measurement of quantities related to the test rig (including the hydraulic power), a National Instruments (NI) cRIO 9074 autonomous digitizer, equipped with various acquisition/control modules, is used. A NI cDAQ 9174 digitizer is then employed to drive the testing model and to record the values of the rotational speed of the runners, of the mechanical torque of the runners (microturbine) and of the electrical power of the generator (PAT).

Table 1. Characteristics of the main instruments.

Acronym	Measured quantity	Sensor type	Range	Precision
Q	Discharge	Electromagnetic flowmeter	0..±100 [m^3/h]	± 0.5 [%]
H	Head	Differential pressure sensor	0..16 [bar]	± 0.1 [%]
H_s	Setting level	Differential pressure sensor	0..5 [bar]	± 0.2 [%]
$M_{1,2,3,4}$	Absolute static pressure	Capacitive pressure transducer	0..10/20 [bar]	± 0.05 [%]
T	Temperature	PT100 transducer	0..100 [°C]	± 0.1 [%]
$N_{p1,2,3}$	Pumps rotational speed	Tachometer	0..1000 [Hz]	-
$T_{mec\ A,B}$	Mechanical torque	Torque meter	0..±7.5 [Nm]	± 1 [%]
$N_{A,B}$	Turbine rotational speed	Incremental encoder	0..7500 [rpm]	2048 [ppr]
P_{elec}	Electrical power	Precision electrical multimeter	0..1000 [Vtrms] 0..32 [Atrms]	± 0.03 [%]
N	Turbine rotational speed	Incremental encoder	0..6000 [rpm]	4096 [ppr]

In the case of classical static point-by-point method, both digitizers perform measurements and compute the average and the standard deviation values for all parameters over 8 seconds at 50 Hz (user-configurable, depending on the stability of the operating condition). In the case of dynamic measurements, a second cDAQ 9174 digitizer has been installed to acquire synchronized dynamic signals of the sensors in parallel with the existing acquisition/control systems of the test rig and of the testing model. Its acquisition frequency has been set to 100 Hz, considered fast enough to cope with the frequency response of the employed sensors.

2.4. Experimental methodology

The employed experimental protocol to measure the hydraulic performances of both the microturbine and the PAT is provided in Figure 3. First, all the instruments have been calibrated and/or rechecked before the beginning of measurements. Considering the classical point-by-point method, the measurements have been performed for 11 different testing head values to build the full efficiency 3D hill-chart of the microturbine. Moreover, 9 values of constant ratio α between the runners absolute rotational speed were systematically considered for each constant testing head. That makes a total of more than 1'000 acquired operating points. In the case of the PAT, the measurements have been performed at 10 different constant rotational speed values of the recirculating pumps. For each recirculating pumps constant rotational speed value, more than 12 different runner rotational speed values of the PAT have been acquired, which makes a total of more than 120 operating points necessary to draw the full-3D efficiency characteristic of the machine.
Then, focusing on the dynamic method, first the optimal acceleration/deceleration ramps of the electrical drives have been identified in order to ensure a quasi- steady-state operation of the testing model. The main objective was to reduce at maximum the measurement time, while avoiding measurement errors and hysteresis on the acquired characteristics. Then, the hydraulic efficiency measurements have been performed at 9 different test rig recirculating pumps constant rotational speed values while increasing and decreasing the speed of the microturbine/PAT runner(s) from minimum to maximum, and vice versa. In the case of the microturbine, for each recirculating pumps constant rotational speed value 11 different constant runners absolute rotational speed ratios have been tested. Finally, the results obtained by the dynamic method have been validated with the ones obtained by the classical static point-by-point measurements method.

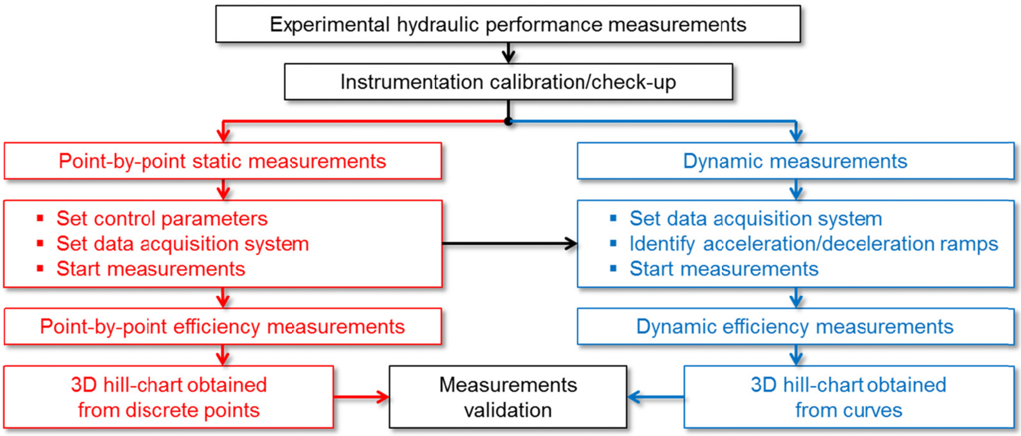


Figure 3. Flowchart of the employed experimental protocol, Hasmatuchi et al. [4].

3. Results

3.1. Acceleration/deceleration ramp selection

The hydraulic-to-electrical efficiency $\eta_{h\text{-}elec}$ can be expressed as the product between the hydraulic-to-mechanical efficiency $\eta_{h\text{-}mec}$ and the electrical efficiency of the generator η_{elec}:

$$\eta_{h-elec} = \eta_{h-mec} \cdot \eta_{elec} = \eta_h \cdot \eta_m \cdot \eta_{elec} = \left(\eta_e \cdot \eta_q \cdot \eta_{rm}\right) \cdot \eta_m \cdot \eta_{elec}\ [\%] \tag{1}$$

Further, $\eta_{h\text{-}mec}$ results from the product between the hydraulic η_h and the bearing efficiency η_m, with the hydraulic efficiency η_h including the efficiency of the disc friction η_{rm}, the energetic efficiency η_e as well as the volumetric efficiency η_q. In the case of the microturbine, considering its reduced size and geometrical complexity, only the hydraulic-to-mechanical efficiency $\eta_{h\text{-}m}$, given by the ratio between the mechanical power P_{mec} recovered by both runners and the hydraulic power P_h of the whole one-stage turbine, could be measured:

$$\eta_{h-mec} = \frac{P_{mec}}{P_h} = \frac{\sum(\omega_i \cdot T_{mec\,i})}{\rho \cdot Q \cdot E}\ [\%] \tag{2}$$

In the case of the PAT, since the machine is not equipped with a torquemeter, only the hydraulic-to-electrical efficiency, given by the ration between the electrical power of the generator P_{elec} and the hydraulic power has been measured:

$$\eta_{h-elec} = \frac{P_{elec}}{P_h} = \frac{P_{elec}}{\rho \cdot Q \cdot E}\ [\%] \tag{3}$$

$$E = g \cdot H\ [J \cdot kg^{-1}] \tag{4}$$

Regarding the hydraulic power, whilst the discharge Q is directly measured, the total specific energy E (eq. (4)) is calculated with the value of the head H. As illustrated in Figure 2, for both tested cases, the inline inlet and outlet cross sections of the machine are equal. Considering the Bernoulli's equation, for this particular case the calculation of the total head can be performed only from the difference of the static pressure between the inlet and the outlet, directly measured with the help of the differential pressure sensor.

$$\eta^* = \frac{\eta_{h-mec\ /\ h-elec}}{max(\eta_{h-mec\ /\ h-elec})}\ [-] \tag{5}$$

$$\eta^{*\prime} = \eta^* - \eta^*_{interp}\ [-] \tag{6}$$

The first step before performing dynamic measurements is to find the optimal acceleration/deceleration ramp of the runner(s) speed in order to stay in quasi- steady-state operating condition during the measurement process. To this end, the speed of the three recirculating pumps of the test rig has been set and maintained constant at a value of 1500 rpm in the case of the microturbine, or 2000 rpm in the case of the PAT. Then, for the microturbine, considering a constant runners absolute rotational speed ratio α = 1, the runners speed was uniformly increased from 1000 rpm to 2000 rpm and then decreased back to 1000 rpm. The same procedure was applied in the case of the PAT for the range from 750 to 2250 rpm. In total, 6 (for the microturbine) and respectively 4 (for the PAT) different acceleration/deceleration ramps of respectively 10, 25, 30, 40, 60, 90 and 120 sec/1000 rpm have been addressed.

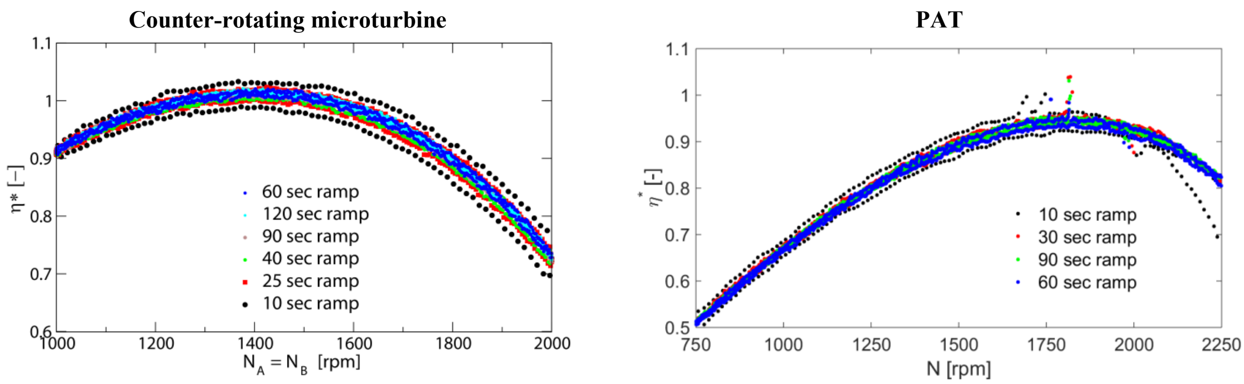


Figure 4. Influence of the acceleration/deceleration ramp of the runner(s) speed on the efficiency for fixed inflow conditions (pumps speed N_p = 1500/2000 rpm).

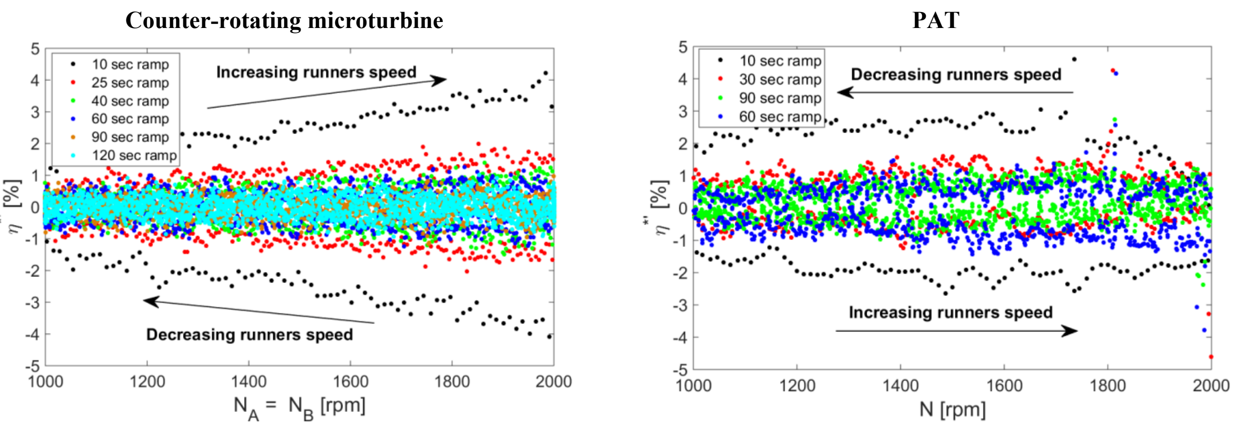
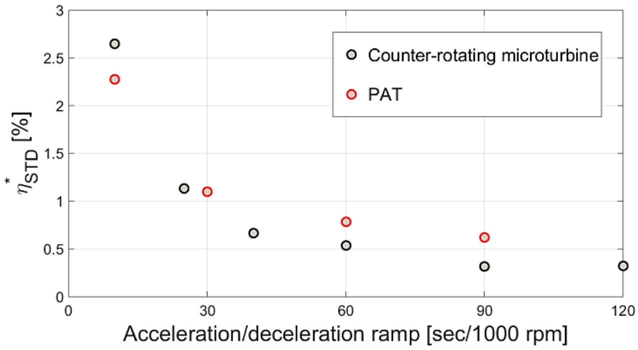


Figure 5. Efficiency fluctuation during an increasing/decreasing cycle at fixed inflow conditions (pumps speed N_p = 1500/2000 rpm) for different speed acceleration/deceleration ramps.

In Figure 4, the resulting influence of the acceleration/deceleration ramp of the runner(s) speed on the efficiency, for fixed inflow conditions, is provided. The efficiency is scaled with the maximum measured value determined from the static measurements (see eq. (5)). One may notice here hysteresis and large measurement errors on the resulting efficiency for speed ramps below 40 sec/1000 rpm.



Acc./dec. ramp [sec/1000 rpm]	η^*_{STD} [%]	
	Microturbine	**PAT**
10	2.6467	2.2774
25	1.1363	-
30	-	1.0995
40	0.6693	-
60	0.5369	0.7863
90	0.3167	0.6246
120	0.3213	-

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

Then, the Figure 5 presents the influence of the speed acceleration/deceleration ramp on the magnitude of the efficiency fluctuations (due to measurement errors) η^{*} compared to the average efficiency obtained by curve fitting (see eq. (6)). Accordingly, for both tested cases, the hysteresis effect tends to become insignificant for speed ramps larger than 40 sec/1000 rpm. The evolution of the resulting standard deviation (STD) of the efficiency fluctuation η^*_{STD} with the speed ramp, provided in Figure 6, confirms this assertion. Consequently, as an acceptable compromise between the measurement time and the measurement precision (efficiency errors below 1% for both cases) a speed acceleration/deceleration ramp of 60 sec/1000 rpm has been selected for the further dynamic efficiency measurements.

3.2. Dynamic measurements results and validation

The comparison between the efficiency obtained by dynamic and by static point-by-point measurement methods, for fixed inflow conditions (recirculating pumps speed of N_p = 1500/2000 rpm), is given in Figure 7. In the case of the discrete point-by-point method the speed of the turbine runner(s) has been modified in steps of 250 rpm from 0 to 3000 rpm, whilst is the case of the dynamic method it has been continuously increased, and then decreased, from 0 to 3000 rpm, and vice versa. A good agreement between the results obtained by the two methods, with a maximum error of 1% between the mean values, may be observed for both tested cases, on the whole operating range, including off-design conditions. In the case of the microturbine, the negative efficiency values are given by a negative mechanical torque of the runners, corresponding to the turbine brake operating mode.

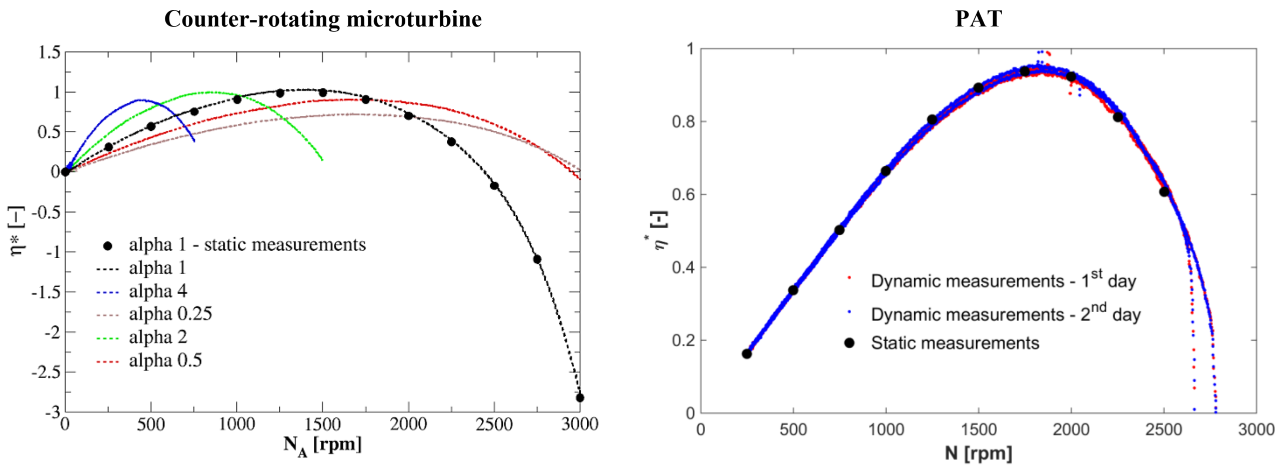


Figure 7. Comparison between dynamic and static efficiency measurements for fixed inflow conditions (recirculating pumps speed N_p = 1500/2000 rpm).

In the case of the PAT, an accentuated hysteresis, specific to S-shaped characteristic curves, is noticed for high rotational speed values. To this end, considering the speed, the discharge and the power factors (eqs. (7) to (9)), the resulting S-shaped 4-quadrants characteristic curves of the PAT measured through the dynamic method are provided in Figure 8. Focusing on the discharge-speed characteristic, all the measurements are clustered on only one S-shaped curve. Indeed, this is explained by the fact that the machine has not been designed with any distributor, its characteristic being similar to the one of a pump-turbine at fixed guide vanes opening angle. However, on a power-speed representation, depending on the recirculating pumps speed, several S-shaped characteristics and hysteresis are noticed. Concerning the hydraulic configuration of the test rig for these 4-quadrants measurements, no special adjustment has been applied, the machine running smoothly between the turbine and reverse-pump modes, and vice versa.

$$N_{ED} = \frac{\mathrm{N} \cdot D_{ref}}{60 \cdot \sqrt{E}} \; [-] \tag{7}$$

$$Q_{ED} = \frac{\mathrm{Q}}{D_{ref}^2 \sqrt{E}} \; [-] \tag{8}$$

$$P_{ED}^* = \frac{P_{elec}}{\rho \cdot D_{ref}^2 E^{1.5}} \; [-] \tag{9}$$

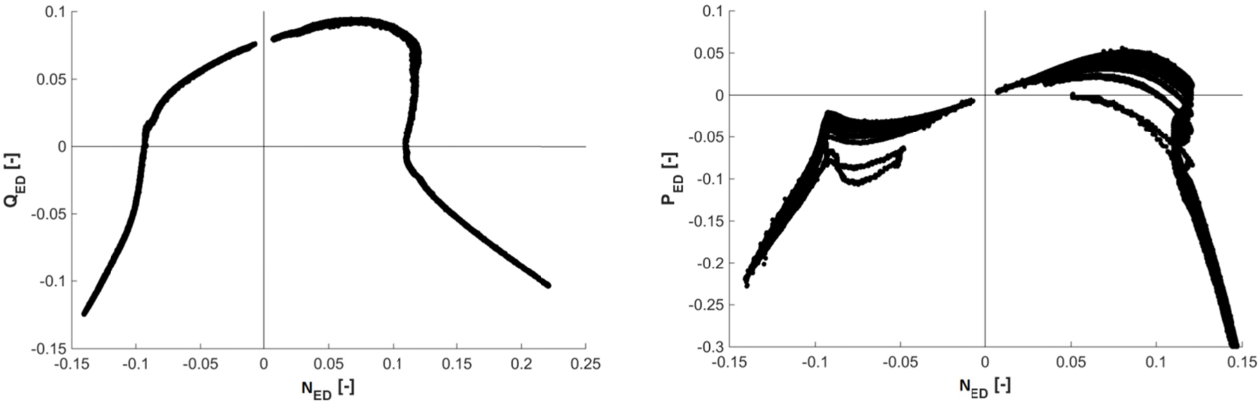


Figure 8. Resulting 4-quadrant characteristic curves of the PAT obtained with the dynamic measurements method.

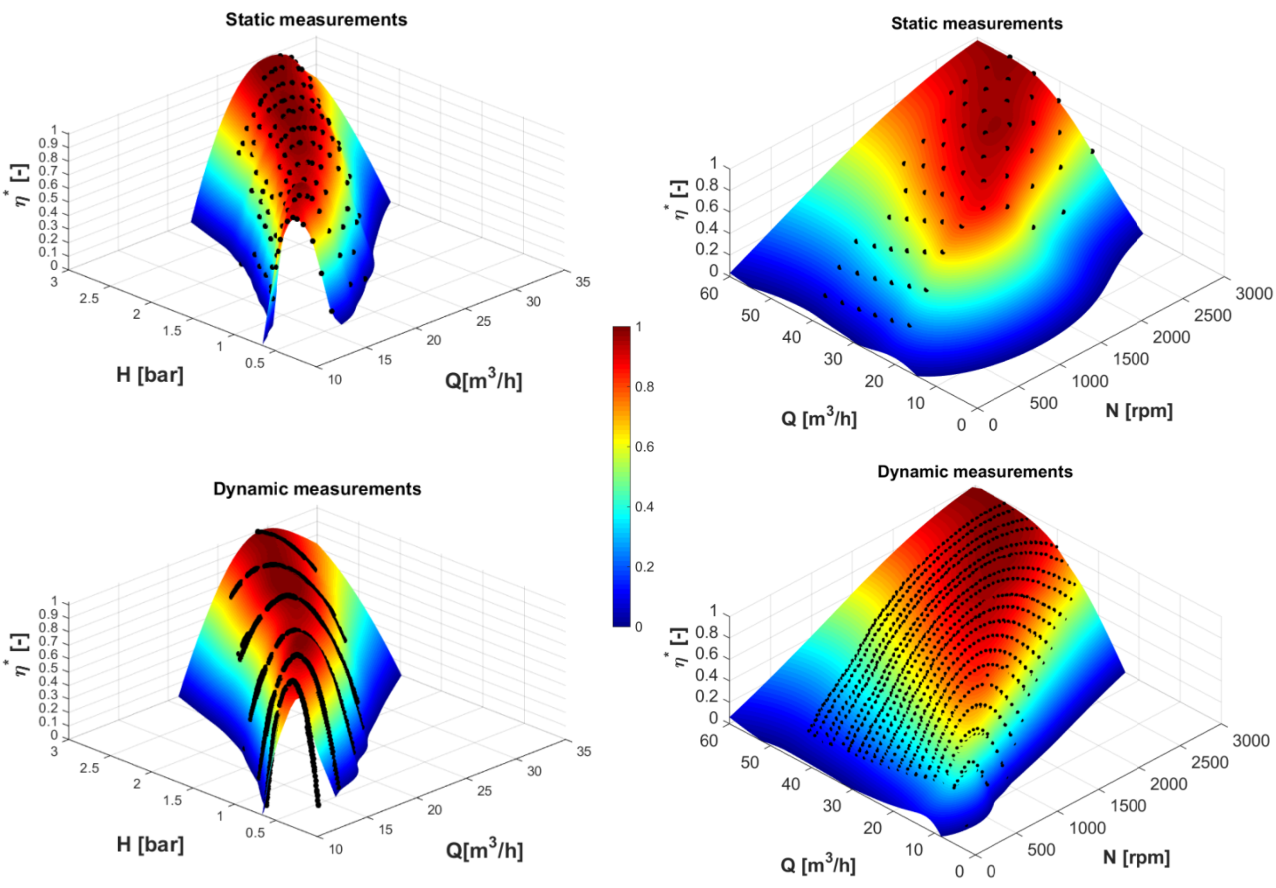


Figure 9. Resulting efficiency 3D hill-charts obtained by the classical static point-by-point and by the dynamic measurement methods.

The resulting efficiency 3D hill-charts obtained by the classical static point-by-point and by the dynamic measurement methods are provided in Figure 9. From a qualitative point of view the results obtained by the dynamic method are in a good agreement with the ones obtained by the static method for both tested cases. Indeed, a satisfactory match between the two resulting 3D hill-surfaces (see the superposition of the fitted surfaces in Figure 10) validates the new proposed dynamic measurements method. The differences between the two hill-chart surfaces are mainly due to the quality of the interpolation and due to the number and the position of the available operating points. A more refined grid of operating points along with a more adapted surface interpolation method should reduce these artificial differences. Finally, in the case of the microturbine, if for the static method the measured operating points cross horizontally the final Q-H characteristic, in the case of the dynamic method the intersection with a typical characteristic of a pump is noticed from the distribution of operating points.

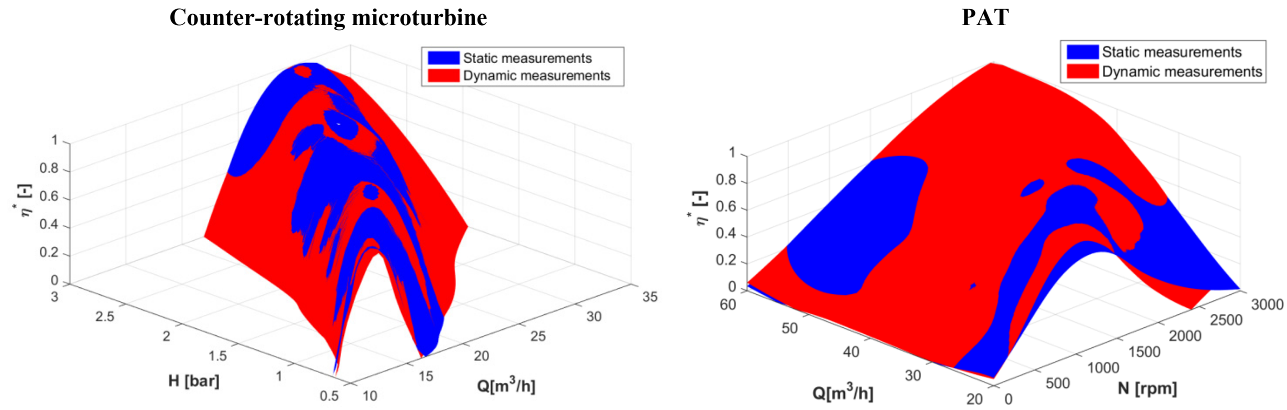


Figure 10. Validation of resulting 3D hill-charts obtained by the dynamic measurements method with the results of the classical static point-by-point method.

4. Conclusions

The implementation of the so-called "sliding-gate" dynamic method to measure in a faster way (up to ten times) the efficiency characteristics on hydraulic turbomachinery has been presented. The new model testing measurement method has been successfully applied on two different cases dedicated to recover the energy lost in release valves of water supply networks: a 2.65 kW double-regulated laboratory prototype of an in-line axial microturbine with two independent variable speed counter-rotating runners and a 11 kW multi-stage centrifugal pump as turbine with variable speed.

The measurements have been carried out on the universal test rig of the HES-SO Valais/Wallis, Switzerland, dedicated to assess hydraulic performances of small-power turbomachines following the IEC standard recommendations. The applied procedure consisted, in a first step, on measuring the 3D hill-chart of a given testing model (turbine or pump) by the classical static point-by-point method. Then, a second digitizer has been installed to acquire synchronized dynamic signals of the sensors in parallel with the existing acquisition/control systems of the test rig and of the testing model. The dynamic measurements of efficiency have been performed at different constant speeds of the test rig recirculating pumps while increasing and/or decreasing the speed of the testing model runner(s) from zero to maximum, and vice versa, slowly enough in order to keep a steady-state operation. Finally, the resulting 3D efficiency hill-charts of the two tested machines obtained by the dynamic have been validated with the results obtained by the classical static point-by-point measurement method. Moreover, the S-shaped 4-quadrants characteristics of the PAT have been also successfully measured using the new dynamic method.

In conclusion, this approach is particularly gainful for small-hydro due to the reduced time required to perform the hydraulic efficiency tests. The same method could be interesting to rapidly identify the regions dominated by hydrodynamic instabilities within the operating range of a turbomachine.

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Nomenclature

D_{ref}	[m]	Runner outlet diameter
E	[J·kg^{-1}]	Specific energy
g	[m·s^{-2}]	Gravity
H	[m]	Head
N	[rpm]	Runner rotational speed
N_{ED}	[-]	Speed factor
P	[W]	Power
P^*_{ED}	[-]	Power factor (based on electrical power)
Q	[m^3·s^{-1}]	Discharge
Q_{ED}	[m^2]	Discharge factor
T_{mec}	[N·m]	Runner mechanical torque
η	[%]	Efficiency
η^*	[-]	Dimensionless efficiency
$\eta^{*\prime}$	[-]	Dimensionless efficiency fluctuation
η^*_{interp}	[-]	Interpolated dimensionless efficiency
η^*_{STD}	[-]	Efficiency Standard Deviation
ρ	[kg·m^{-3}]	Water density
ω	[rad·s^{-1}]	Runner angular speed

Subscripts

A, B	1st, 2nd independent runners
e	Energetic losses
$elec$	Electrical
h	Hydraulic
m	Bearing losses
mec	Mechanical
q	Volumetric losses
rm	Disc friction losses

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