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EXPERIMENTAL INVESTIGATION OF A PUMP-TURBINE AT OFF-DESIGN OPERATING CONDITIONS

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ABSTRACT

Pumped storage power plants are key components for the development of new renewable CO_2-free primary energies and the security enhancement of electricity supply. They also offer interesting business opportunities in nowadays liberalized electricity market. However, the fast and frequent switching between pumping and generating modes as well as extended operations at off-design conditions poses technical challenges related to large unsteady hydrodynamic forces. In the present study, a reduced scale model of a low specific speed radial pump-turbine is investigated to identify the onset and development of flow instabilities. The focus is put on the generating mode at off-design conditions involving runaway and "S-shape". Wall pressure measurements in the stator are performed with the help of 30 miniature piezoresistive sensors. When starting from the best efficiency point and increasing the runner speed, a significant increase of the pressure fluctuation is observed mainly in the channels between wicket gates. The spectral analysis shows a rise of a low frequency component (about 70% of the runner rotational frequency) at runaway, which further increases as the zero discharge condition is approached. Phase analysis reveals that several instability sources, at least three, rotate with runner at sub synchronous speed. Although the nature of the rotating cells could not be described, it is thought that back flow may develop in the gap between runner and distributor with an alternate switch between generating and pumping modes of runner channels. Nevertheless, rotating flow separation may also develop in the runner channels leading to their blockage.

KEYWORDS

Instability, Off-design, Pump-turbine, Generating mode, Experimental investigation

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1. INTRODUCTION

Pumped storage power plants are key components for the development of new renewable CO_2-free primary energies and the security enhancement of electricity supply. They also offer interesting business opportunities in nowadays liberalized electricity market. However, the fast and frequent switching between pumping and generating modes as well as extended operations at off-design conditions poses technical challenges related to large unsteady hydrodynamic forces.

“In addition to the determination of hydraulic characteristics of a machine in a limited range of specific hydraulic energies and discharges, it is also important to know its complete characteristics covering possible operating conditions outside the normal operating range. Depending on the specific speed of a hydraulic reaction machine, the Q_{ED}-n_{ED} characteristics at constant guide vane opening can be S-shaped. Testing under steady state conditions can then be difficult”, [1]. These instabilities were studied by Pejovic et al. [2]. Tanaka and Tsunoda [3] and Oishi and Yokoyama [4] also reported this problematic phenomenon. Huvet [5] marked out charts to enable the designer as well as the operator to assess the steady oscillatory conditions. Nicolet [6], concluded that the stability criterion is intrinsically fulfilled for low head Francis turbines, but for high head Francis turbines or pump-turbines with a long penstock of small diameter, and low rotating inertia, the situation is more critical. Furthermore, high head pump-turbines are commonly of low specific speed types and therefore present “S-shapes” characteristic curves. Recently, Billdal and Wedmark [7] asserted that in the real life a single-stage reversible pump-turbine is by nature forced to be operated as a compromise between an optimal pump and an optimal turbine. Some penalty has to be paid on the overall performance to obtain the advantages of a simplified and cost-effective pump-storage design concept. The main reason is that the optimum speed is not the same in turbine and pump mode, and since speed is governed by the pump performance, the turbine is operating at off-design conditions in the whole operational domain. A linearized stability analysis was made by Martin [8] and [9] to predict the occurrence of the oscillations. Dörfler [10] solved the stabilization problem by changing the start-up procedure. So, he opened the turbine inlet valve only partially until synchronization. The artificial head loss rendered the hydraulic condition stable. Kuwabara et al. [11] developed an intelligent governor which was provided with an anti-S-characteristics control to be used upon load rejection. The wicket gates were once opened by the governor while being closed so that the discharge was reduced smoothly to the no-load value, neither with any detrimental hunting nor with any temporary reversals. As a solution to the difficulties with the synchronization, Billdal and Wedmark [7] introduced the concept of Multiflow Guide Vanes (MGV). In fact, it operates a few guide vanes independently from the rest of the guide vane mechanism.

The present work is performed in the frame of HYDRODYNA project. It consists in the experimental investigation of a radial pump-turbine reduced model of low specific speed at off-design operating conditions in generating mode. After the experimental setup is described, the resulting “S-curves” are presented. Analysis of pressure measurements is used to identify and describe the onset and the development of flow instabilities, starting from the best efficiency point and increasing the runner speed until the runaway and turbine break mode are reached. Finally, concluding remarks are presented.

2. EXPERIMENTAL SETUP

The case study is a reduced scale of a radial pump-turbine, Fig.1, tested in the EPFL PF3 test rig in accordance with IEC standards, [1]. It is a low specific speed machine with 9 runner blades and 20 guide vanes. Two 400 KW centrifugal pumps in serial connection provide a maximum head of 100 m, and a maximum discharge of 1.4 m^3s^{-1}.

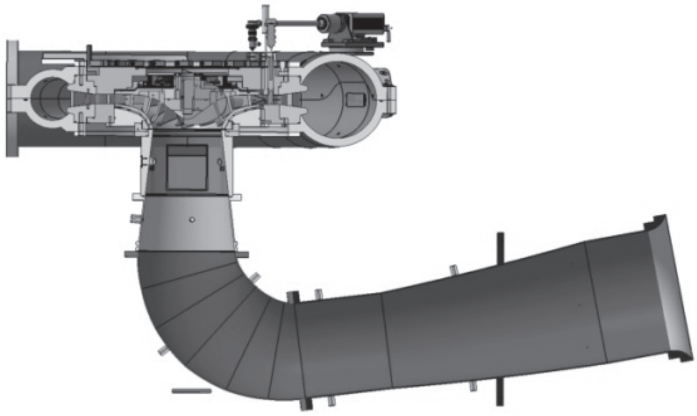

Fig.1 The HYDRODYNA reduced model

Off-design conditions, involving runaway and “S-shape” turbine brake curve, are investigated. Starting from nominal operation at fixed guide vane opening (5° and 10°), the rotation speed is gradually increased until occurrence of runaway (zero torque). At this point, the operation becomes unstable and the machine may switch back and forth from generating to reverse pumping modes. A specific procedure, commonly used in model testing of pump turbines, Dörfler [10], was adopted to stabilize the machine operation: once the runaway is reached, a butterfly valve, located in the main pipe upstream to the model, is closed up to 95% to provide additional hydraulic resistance. A bypass, equipped with a second valve is then used to finely adjust the head. This procedure improves significantly the stability of the machine operation and allows exploring the entire “S-curve”.

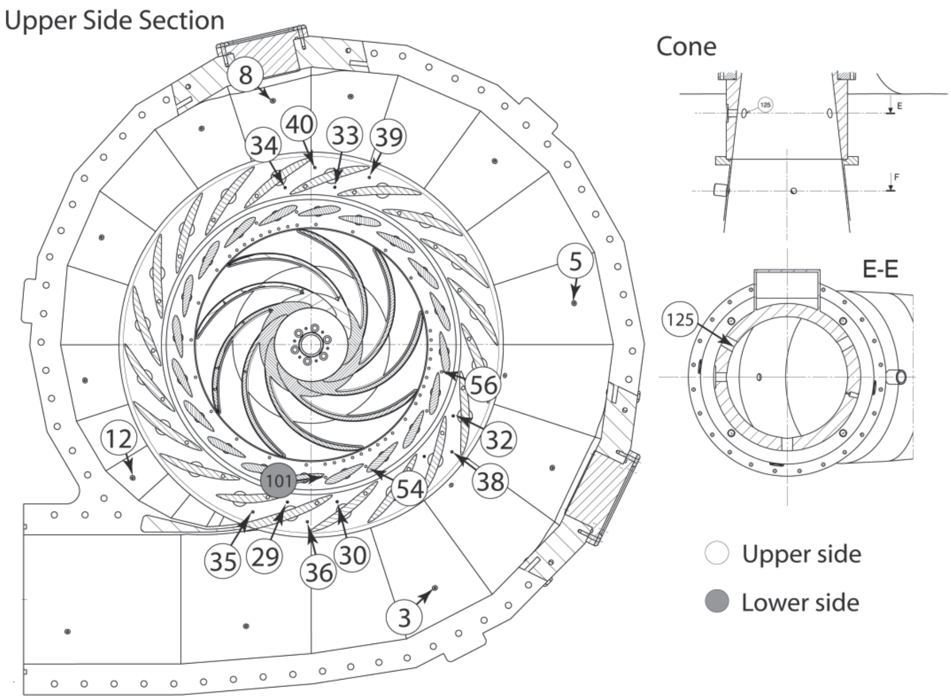

Fig.2 Pressure sensors location on the model (10° guide vanes opening)

Wall pressure measurements are performed using 30 piezoresistive pressure sensors flush mounted in the spiral casing, stay and guide vane channels as well as in the draft tube. As illustrated on Fig. 2, the sensors locations are chosen to cover stator channels up to rotor-stator interface. Pressure signals are simultaneously recorded using VXI HP1432A digitizers with 16 bits A/D resolution, 51200 Hz sampling frequency and a memory of 1 M-samples/channel, providing 20 seconds record length.

3. RESULTS

Owing to IEC standards [1], the speed, discharge, torque and power factors are defined by Eq.(1) to Eq.(4). The runner outlet diameter, D, is 0.25 m.

$$n_{ED} = \frac{nD}{\sqrt{E}} \qquad \text{- speed factor} \qquad (1)$$

$$Q_{ED} = \frac{Q_1}{D^2\sqrt{E}} \qquad \text{- discharge factor} \qquad (2)$$

$$T_{ED} = \frac{T_m}{\rho_1 D^3 E} \qquad \text{- torque factor} \qquad (3)$$

$$P_{ED} = \frac{P_m}{\rho_1 D^2 E^{1.5}} \qquad \text{- power factor} \qquad (4)$$

We present in Fig. 3 the turbine characteristics at 5° and 10° guide vane opening in (n_{ED},Q_{ED}), (n_{ED},T_{ED}), and (n_{ED},P_{ED}) system coordinates.

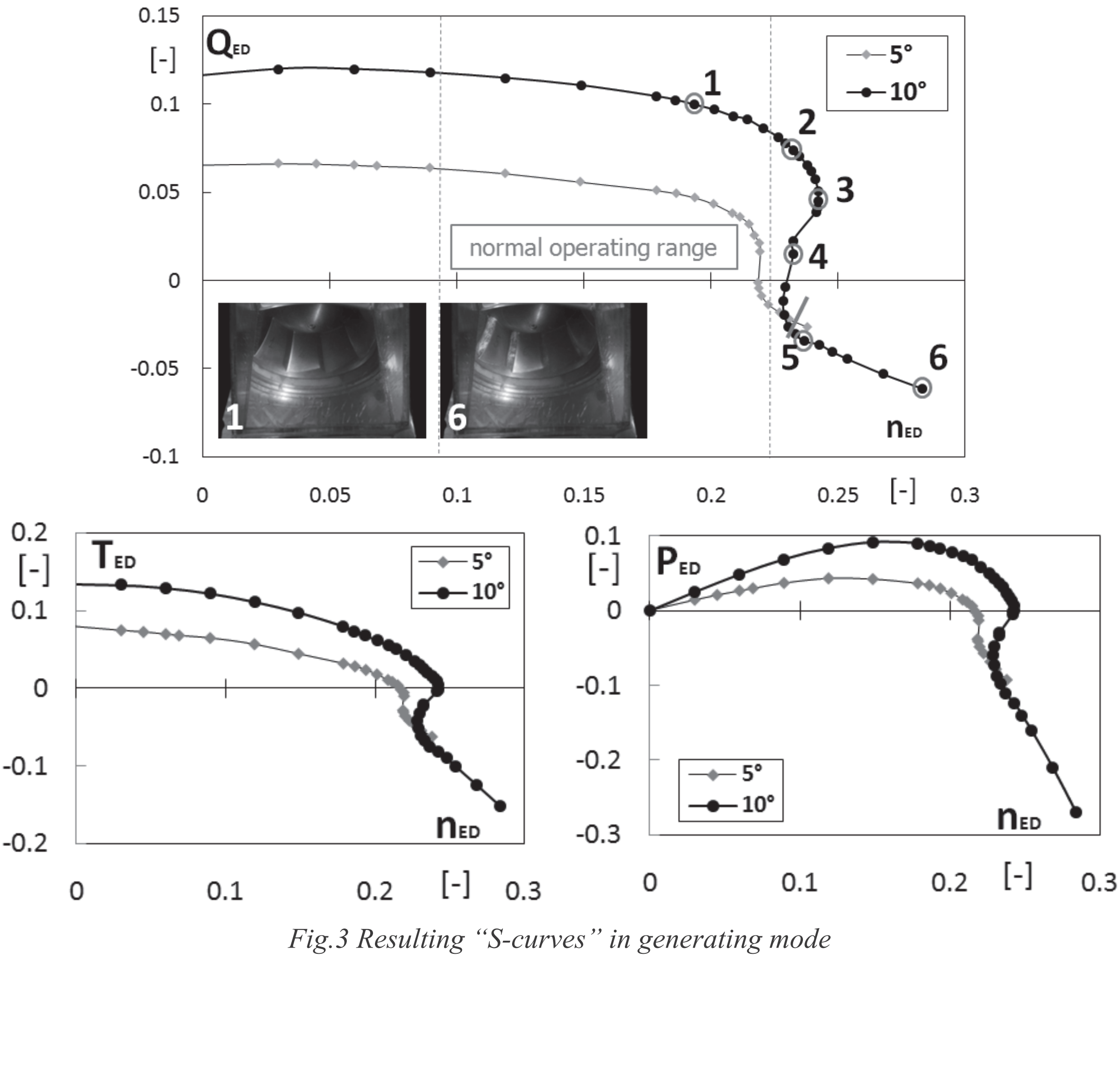

Fig.3 Resulting “S-curves” in generating mode

At 10° guide vane opening, the turbine characteristic exhibits a positive slope after the runaway speed (see Fig. 3, OP. 3). When a pump-turbine prototype is brought in such a situation, the operation suddenly switches to reverse pumping mode. The discharge as well as torque and power are reversed with a substantial increase of structural vibrations driven by flow instabilities. In our case, the use of the stabilizing procedure described above prevents such unstable operation and let us explore the positive slope part of the characteristic curve. Once is in reverse pumping quadrant, an increase of the rotation speed (OP. 6) leads to strong cavitation occurrence on the runner blades as illustrated by the photograph in Fig. 3. For this operating point, a large fluctuation of the guide vane opening and discharge is also observed. This unstable operation is even magnified by the test rig automation system, which fails in correcting the parameters fluctuation in real time. It is noticed that a significant hysteresis is systematically observed for the "S-shape" curve: a different path is obtained when the machine is brought back from reverse pumping to generating mode.

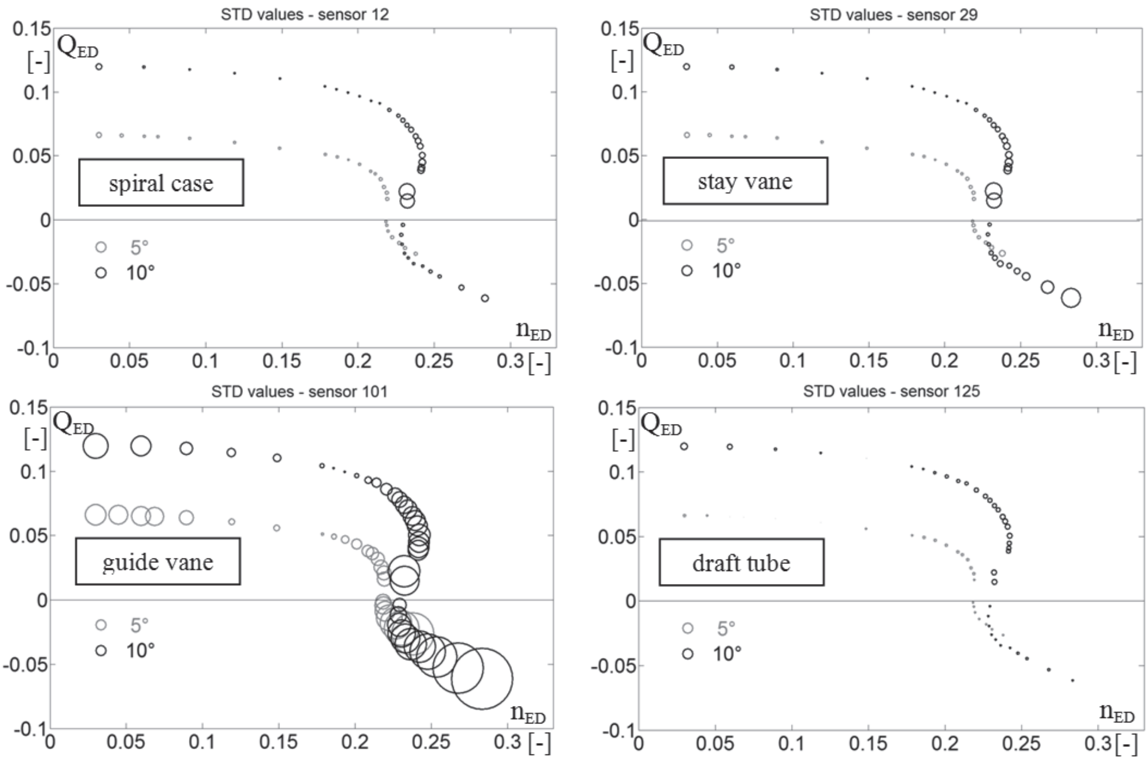

Fig.4 Pressure fluctuations STD – 5° and 10° comparison between cases

To provide a global view of the flow unsteadiness at 5° and 10° guide vane openings, we have superposed on the (n_{ED},Q_{ED}) characteristic curves, the standard deviation of the pressure fluctuation in the spiral casing, stay vane, guide vane and draft tube (see Fig. 4). The size of the circles is proportional to the standard deviation of pressure fluctuation and a same scale is adopted for all graphs to allow for straightforward comparison between different operating points and different locations. One may easily observe that the pressure fluctuation in the guide vane channels, close to rotor/stator interface, is by far the most important. In contrast, the lowest fluctuations are observed in the draft tube. Moreover, for any given location, the pressure fluctuations are increased for low rotation speed, around the "S-shape" and in reverse pumping mode. At these conditions, a substantial increase of the structural vibration was observed. It should be noticed that in the particular case of guide vane, the maximum pressure fluctuation is at least 25 times larger than in nominal conditions. The present analysis let us believe that the source of flow unsteadiness at off-design conditions is likely located in the runner or in the vaneless space between the runner and the guide vane.

In the following, further analysis is carried out for six operating points (OP. 1 through OP. 6) for 10° guide vane opening, as illustrated on Fig. 3 and Tab. 1. They are distributed as follows: normal operating range region (OP. 1), runaway speed (OP. 3), very low positive discharge (OP. 4) and high runner speed with negative discharge (OP. 6). Two other intermediate points (OP. 2 and OP. 5) are selected in the transition regions between those already presented.

OP.	n_{ED}	Q_{ED}	T_{ED}	P_{ED}	Comment
1	0.19335	0.09949	0.06797	0.08258	best efficiency
2	0.23226	0.07400	0.02549	0.03719	low torque
3	0.24209	0.04493	0.00033	0.00049	runaway
4	0.23244	0.01483	-0.02262	-0.03303	low discharge
5	0.23648	-0.03446	-0.07492	-0.11133	reverse pump
6	0.28327	-0.06139	-0.15100	-0.26877	reverse pump

Tab.1 Selected operating points for analysis

The pressure coefficient fluctuation is computed in accordance with the equation Eq. 6.

$$c_p^{\sim} = \frac{p - \overline{p}}{\rho E} \qquad \text{- pressure coefficient fluctuation} \qquad (5)$$

Averaged spectra of pressure fluctuation in the spiral casing (sensors 5) and in guide vanes region (sensor 56) are presented in Fig. 5 for previously selected operating points. The plots scales are differently chosen in order to facilitate the visualization. The pressure fluctuation in the guide vanes channels is dominated by the blade passing frequency and its first harmonic with lower amplitude ($f=9 \cdot f_n$ and $f=18 \cdot f_n$) except at low discharge operating point. The spectral analysis shows also a low frequency component (~70% of the runner rotational frequency) rising at runaway (OP. 3), which further increases in amplitude as we approach the zero discharge condition (OP. 4). At this point it even represents the dominant frequency and is found to modulate the blade passing frequency. In the spiral casing (Fig. 5, left column), as reported by Tanaka [12], the blade passing frequency has a lower amplitude than its first harmonic. The low frequency component is also visible in this region and becomes dominant in turbine brake mode (OP. 3 ÷ OP. 5). We believe that it represents the signature of the flow unsteadiness at off-design operating conditions. When starting from the nominal conditions and increasing the runner speed, a significant increase of the pressure fluctuations amplitude is observed in the channels between guide vanes as well as in the spiral casing. At best efficiency point (OP. 1) the amplitude of fluctuations is at least 10 times smaller than any of operating points OP. 2 through OP. 6. In the spiral casing, the pressure fluctuations amplitude is significantly lower (almost 10 times) than in the guide vane region, except for the zero discharge condition (OP. 4). These observations confirm that the source of flow unsteadiness is likely located either in the runner or in the vaneless space between runner and distributor.
Time history over 15 runner revolutions corresponding to pressure fluctuations in 3 guide vane channels (sensors 101, 54 and 56) are presented in Fig. 6 at zero discharge condition (OP. 4). The sensors are located on the same circumference at 0°, 18° and 72° angular position. Surprisingly, the pressure fluctuations at this very unstable operating point are not random but exhibit a remarkable periodicity. We may observe how the rotor-stator interaction frequency is carried by the low frequency ($0.7 \cdot f_n$). The phase analysis, derived from cross correlation between the signals, indicates that several instability sources rotate with the runner at sub synchronous frequency. A minimum of three rotating cells was found but the actual instrumentation does not allow determining their exact number. As a result, the maximum frequency of rotating cells is the third of the one revealed by pressure spectra ($0.7 \cdot f_n$).

The nature of rotating flow instabilities remains unknown. Flow separation (stall) may occur on part of runner channels leading to their blockage. On another hand, due to the large vaneless space between the runner and the guide vane at 10° opening, back flow cells may also develop with an alternate switch between generating and pumping modes of runner channels. This explanation is likely more appropriate since it is in accordance with zero discharge condition.

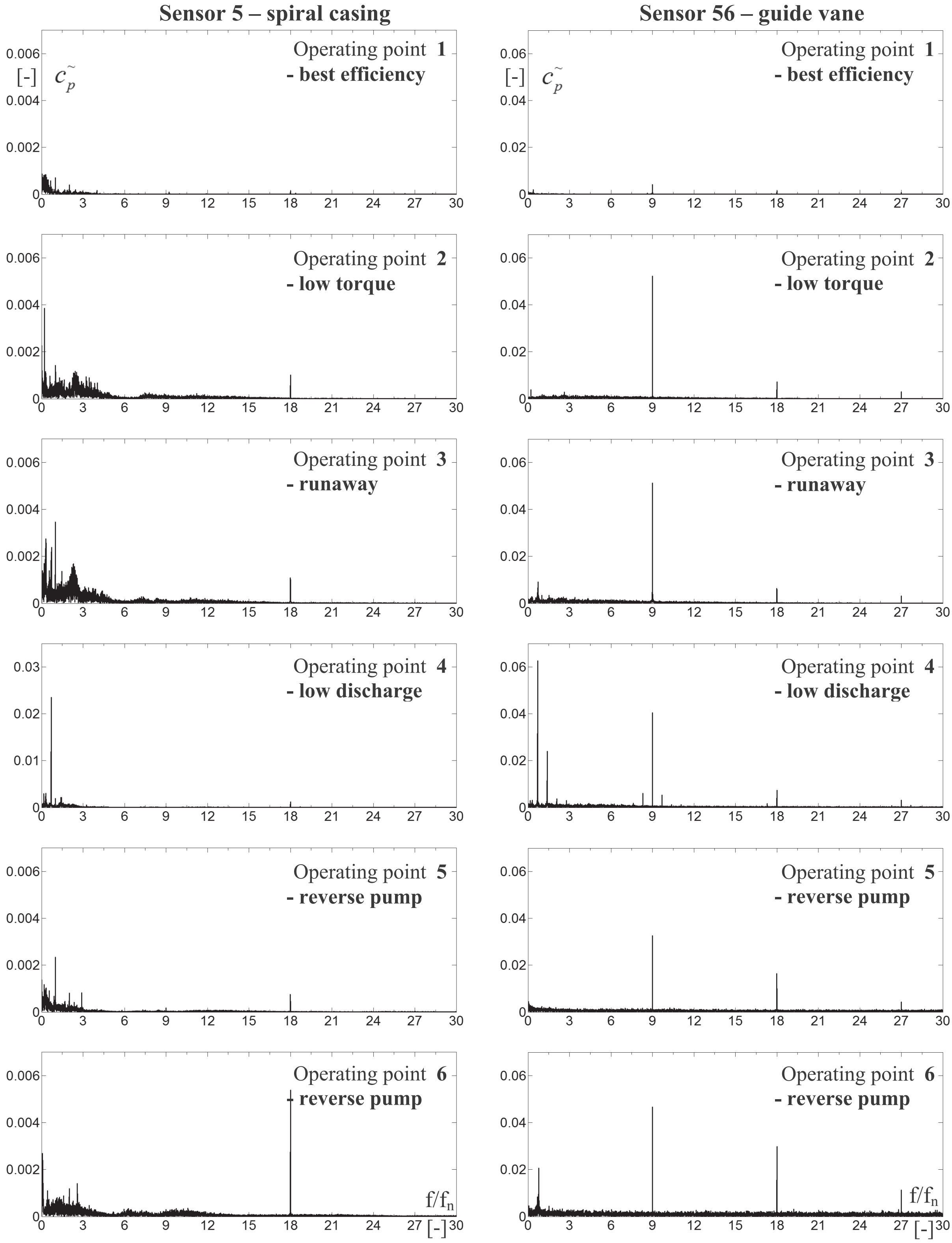

Fig.5 Pressure fluctuations STD – 5° and 10° comparison between operating points

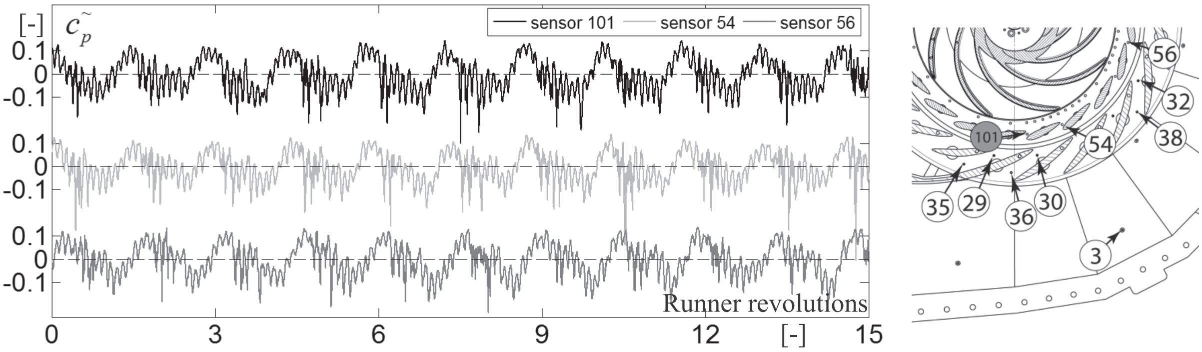

Fig.6 Guide vane region pressure coefficient fluctuations – low discharge operating point (OP. 4)

4. CONCLUSIONS

In the present study, we have investigated off-design operations of a centrifugal pump-turbine at reduced scale. The experiment, carried out in the EPFL test rig, involves wall pressure measurements in the stator with the help of miniature sensors. The focus is put on the generating mode at off-design conditions involving runaway and "S-shape" turbine brake. Starting from the best efficiency point, the runner speed is gradually increased until the flow is totally reversed. The main conclusions may be summarized as follows:

- At the best efficiency point, the pressure fluctuation is very low and mainly dominated by the blade passing frequency and its first harmonic.
- As the runner enters the "S-shape" domain, a substantial increase of the pressure fluctuation is observed everywhere in the stator. This increase is particularly pronounced in the guide vane channels while it is minimum in the draft tube.
- At runaway, the spectral analysis shows a rise of a low frequency component at about 70% of the runner rotational frequency, which further increase in amplitude as we approach the zero discharge condition. Phase analysis reveals that several instability sources, at least three, rotate with runner at sub synchronous speed.
- Although the nature of the rotating cells could not be described, we believe that due to the large vaneless space between the runner and the guide vane at low opening, back flow cells may develop with an alternate switch between generating and pumping modes of runner channels. Nevertheless, rotating flow separation may also develop in the runner channels leading to their blockage.

The rotating instability is expected to generate hydraulic unbalance and strong structural vibrations. Further investigations are required to assess the proposed explanation of flow instability at off-design conditions.

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6. REFERENCES

[1] International Standard: *IEC 60193 Second Edition 1999 – 11*. 1999.

[2] Pejovic, S., Krsmanovic, L., Jemcov, R., Crnkovic, P.: *Unstable Operation of High-Head Reversible Pump-Turbines.* Proceedings of 8th IAHR Symposium on Hydraulic Machinery and Cavitation. pp. 283-295. Leningrad. 1976.

[3] Tanaka, H., Tsunoda, S.: *The Development of High Head Single Stage Pump-Turbines.* Proceedings of 10th IAHR Symposium on Hydraulic Machinery and Cavitation. pp. 429-440. Tokyo. 1980.

[4] Oishi, A., Yokoyama, T., *Development of High-Head Single- and Double-Stage Reversible Pump-Turbines.* Proceedings of 10th IAHR Symposium on Hydraulic Machinery and Cavitation. pp. 441-452. Tokyo. 1980.

[5] Huvet, P.: *Steady Oscillatory Conditions between Pump-Turbine Operating at Partial Flow and Surge Shaft*. Proceedings of 16th IAHR Symposium Hydraulic Machinery and Systems. Sao Paulo, Brazil. 14-19 September 1992.

[6] Nicolet, C.: *Hydroacoustic Modeling and Numerical Simulation of Unsteady Operation of Hydroelectric Systems*. PhD thesis No. 3751. EPFL Lausanne, Switzerland. 2007.

[7] Billdal, J. T., Wedmark, A.: *Recent Experiences with Single-Stage Reversible Pump Turbines in GE Energy's Hydro Business*, Hydro 2007. Paper 10.3. Granada. 15-17 October 2007.

[8] Martin, C. S.: *Stability of Pump-Turbines during Transient Operation*. The 5th International Conference on Pressure Surges. Hannover, Germany. 22-24 September 1986.

[9] Martin, C. S.: *Instability of Pump-Turbines with Shaped Characteristics*. Proceedings of 20th IAHR Symp. Hydraulic Machinery and Systems. Charlotte, USA. 2000.

[10] Dörfler, P. K.: *Stable Operation Achieved on a Single-Stage Reversible Pump-Turbine Showing Instability at No-load*. Proceedings of 19th IAHR Symposium Hydraulic Machinery and Systems. Singapore. 9-11 September 1998.

[11] Kuwabara, T., Katayama, K., Nakagawa, H., Hagiwara, H.: *Improvements of Transient Performance of Pump Turbine upon Load Rejection*, Power Engineering Society Summer Meeting. Vol. 3. pp. 1783-1788. Seattle, WA, USA. 2000.

[12] Tanaka, H., Vibration Behavior and Dynamic Stress of Runners of Very High Head Reversible Pump-Turbines. The 15th IAHR Symposium. Belgrade, Yugoslavia. 1990.

7. NOMENCLATURE

n_{ED}	(-)	speed factor	Q_{ED}	(-)	discharge factor
T_{ED}	(-)	torque factor	P_{ED}	(-)	power factor
n	(rot.s^{-1})	runner speed	D	(m)	runner outlet diameter
E	(kg.s^{-1})	specific energy	Q	(m^3.s^{-1})	discharge
T_m	(N.m)	runner torque	P_m	(W)	mechanical power
ρ	(kg.m^{-3})	water density	f	(Hz)	frequency
f_n	(Hz)	runner frequency	p	(Pa)	wall pressure
$\tilde{c_p}$	(-)	pressure coefficient fluctuation	$\bar{p}$	(Pa)	time average wall pressure