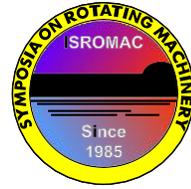


Experimental Analyses and Numerical Simulation of a Pump-Turbine Transient Load Following Process in Pump Mode

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Long Abstract

Introduction

The possibility of storing electric energy is a key issue when the production of electricity is based, for a substantial portion, on renewable sources. In fact, it is known that these sources are intermittent and not-programmable, so that they cannot meet the electricity user's demand.

Today, one of the most promising and tested way for storing electricity is the Pumped Hydro Storage Power Plant which is characterized by good dynamics (rapid changing from pumping mode to turbine mode) and provides high-quality energy.

The traditional pump-turbine equipment design is the reversible single-stage Francis pump-turbine, which acts as a pump in one direction and as a turbine in the other. On a conventional unit, pumping occurs at a fixed speed and almost fixed wicket gate opening. The power input is nearly constant at the input rating of the pump and the discharge varies with the pumping head. Although this technology is proven and has worked well for over six decades, there are limitations to its performance. Adjustable-speed machines enable the power consumed in the pumping mode to be varied over a range of outputs. Consistently, an adjustable speed pumping unit can operate over a larger head range and pumping power input range than a single speed unit.

Because adjustable-speed technology is well suited to integration of variable renewable generation, many of the proposed new pumped storage projects are considering adjustable-speed machines. The principal feature of the adjustable-speed units is that the input power is adjustable when carrying out automatic frequency control (AFC) while filling the upper reservoirs. This flexibility is frequently employed by adjusting the speed of units during light load periods over a load range of 50 to 60% of rated pumping power. In addition, pump operation with adjustable-speed units is extended in comparison to single-speed units, enabling more real-time response to grid conditions (Henry J M et al., 2012).

With Adjustable Speed Hydro the limits of the pumping range are normally defined by cavitation (low head, high speed operation), motor-generator output (medium to high head, high speed operation), turbulence or reverse flow (high head, low speed operation), and range of operating speed.

Although these limits seem to be very restrictive, in reality the improvement in operating range is extremely impressive, especially when considering the possibilities of how the extended range can be used to avoid cavitation and reduce input power at low heads, adjust pumping power, avoid reverse flow at high head operation, and provide frequency control in the pumping mode.

With adjustable speed, it is possible to operate the pumps in the range of 60% to 100% of rated capacity. This means that a unit does not have to wait until a block of power equal to its full pump capacity rating is available before pumping can begin. Thus there are more opportunities to purchase blocks of pumping power.

In that context it is requested to develop pump-turbines that reliably stand dynamic operation modes, fast changes of the discharge rate.

Furthermore, the overall operating range of a pump-turbine should be well balanced.

Stability limits should be positioned considerably away from the normal operating range in pump operation mode.

To enable smooth transient behaviour during rapid variations of energy level (output or consumption) the stable operation in off-design, start-up and transient conditions is a key issue for pump-turbines. At off design conditions in pump mode, the wicket gates channel and the draft tube do not work properly and give awkward boundary conditions to the impeller, together with a strong fluid-dynamical interaction between rotor and stator parts (Yuekun Sun, et al., 2014, Hui Sun et al., 2013, Li W., 2012 and Rodriguez C G et al., 2014, Gentner C et al., 2012). The flow features such as separation and recirculation occur severely in an unsteady manner. Non-rotating components of the turbine, such as guide vanes, stay vanes, head cover, draft tube cone, and also on the hydraulic system, especially the penstock may experience strong dynamic load and high cycle fatigue stress that may result in the propagation of cracks and the failure of the shear pin or the guide vanes stem.

Several experimental and numerical analyses have been carried out to identify a possible connection of unsteady flows and pressure fluctuations developing inside centrifugal pumps with runner/guide vane geometries and operating conditions.

Guo and Maruta (2005) investigated the onset of resonance phenomena as a consequence of the circumferential unevenness of the pressure fluctuations, whereas Rodriguez et al. (2007) presented an interesting theoretical method to predict and explain the possible harmonics that could appear in a pump-turbine as a consequence of the interaction between moving and stationary blades.

The frequency content of the pressure fluctuations was analysed both in frequency and in the time-frequency domains by Pavesi et al. (2008), Cavazzini et al. (2009), Yang J et al., (2013) and Yang J et al., (2015), whose study presented a spectral analysis of the unsteady phenomena developing in a pump-turbine. Their analyses highlighted the existence of a rotating structure of pressure pulsations at the runner exit appearing and disappearing in time, having greater intensity at part loads. This strong rotor stator interaction (RSI), at off-design conditions, resulted to be further emphasized in multi-stage pump-turbines in which a 'full-load-instability' (FLI) develops in the range from 60 to 90% of the design flow rate (Yang J et al., 2013) whereas Pavesi et al (2008) analyzed the influence of two rotational speed on the inception and evolution of the pressure instabilities.

Numerical analysis were carried out by Cavazzini G et al., (2011) identifying the pulsating onset of reserve flow cells in the runner, moving along the blade length and from one channel to another. This unsteady behaviour in the runner resulted to be associated with a perturbation of the wicket gates channel flow field, characterized by an unsteady flow rate migrations between passages and by unsteady flow jets.

Liu et al. [2012] investigated the hump characteristic of a pump turbine based on an improved cavitation model, and the calculation results are in agreement with the experimental data. Braun [2005] carried out calculations for the flow distribution in pump mode and an head discharge curve was obtained. The results showed that there was strong vortex between the guide vanes and flow became worse when entering the hump region. Yan [2010] obtained the same fluctuation results as the testing in the vaneless region by using compressible model. Iino [2004] considered that the hump characteristic was related to complex vortex structure in the runner inlet and centre region of the tandem cascade through simulation and experimental investigation.

More recently, Li Deyou et al (2015) focused the numerical analyses into the hump region trying to correlate the hump characteristics to the vortex motion in the tandem cascade.

Numerical analysis were also carried out by Gentner et al. (2012) highlighting the dependence of the flow behaviour in the head drop from the specific speed of the pump-turbine.

The results at constant flow rate of both experimental and numerical analyses highlighted the existence of a spatial fluctuation pattern concentrated close to the runner exit, whose fluctuations levels increases at off-design conditions.

Even though these studies have allowed to obtain interesting information on the unstable behaviour of pump-turbines, to solve instability problems and to significantly enlarge the working range of pump-turbine, an in-depth understanding of the unsteady flow mechanism during power regulation is crucial for the production stabilization.

The aim of this investigation will be to analyse the development of the unsteady phenomena of a reversible-pump turbine operating in pump-mode load following control function. The experimental research included the dynamic pressure measurements and high-speed flow visualizations from design to part flow rate at a fixed gradient of the power reduction. The analysis of pressure fluctuations will be conducted both in frequency and time frequency domains and the flow visualization will be focused in the wicket gates and in return channels.

A 3D transient flow simulation in the entire pump-turbine will be conducted to investigate the rotor–stator interaction by adopting the DES turbulence model. A commercial CFD programme will be utilized to study the flow through this pump-turbine in its stationary and transient passages, from 100% optimum load to 50% of part load reduction by an adjustable-speed approach.

Methods

The experimental study will be carried out in the test rig for turbines and pumps at the Department of Industrial Engineering of the University of Padova.

The accuracy that can be achieved in the calculation of the efficiency and speed of the machines was 0.2% and ± 0.5 rpm respectively. The calibration of the instruments will be performed on site. The pump-turbine model is the low-pressure stage of a two stages $n_s = 37.6 \text{ m}^{0.75} \text{ s}^{-1}$. (dimensionless design specific speed $\omega_s = 0.71$) pump-turbine with a seven 3D backward swept blades with a discharge angle of 26.5° runner (Fig. 1).

Refeeding channels are used to guide the flow that leaves the impeller to the inlet of the subsequent channel. The channels are made up of twenty two adjustable guide-diffuser vanes and eleven continuous vanes. The guide-diffuser allows continuous and independent adjustment of the vane angle and of relative azimuthally position with the return channel vanes. The relative azimuthally position of the guides' vanes is fixed rotating the system of 8 degrees from the face to face configuration (reference position with $\lambda=0^\circ$).

Geometry characteristics of the tested pump-turbine is listed in table 1.

Fig. 2 shows the gray guide vane and the return vane where the unsteady pressure will be measured at the mid-height by 12 piezoresistive transducers Kulite XCL-072 (sensitivity of about 29.3 mV/bar), which are faced mounted. Details of this configuration and of the complete machine geometry can be found in Yang J et al., (2013).

The pressure signals will be analyzed in both the frequency domain and the time–frequency domains (Torrence C and Compo G, 1998, Farge M., 1992) to identify and characterize the unsteady phenomena at the part load. The

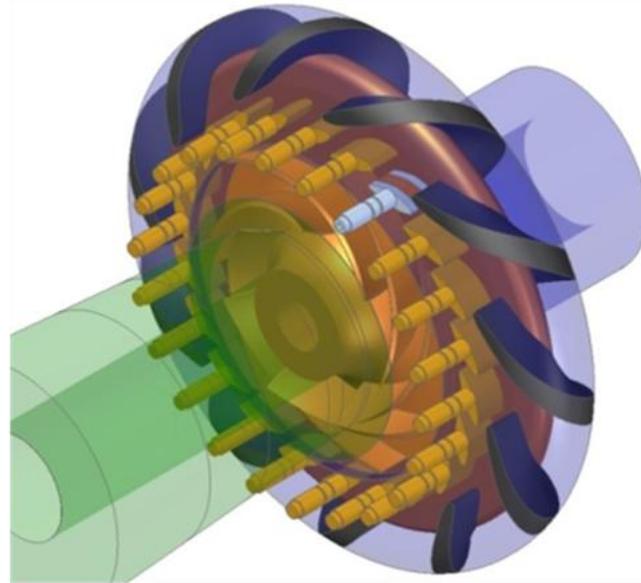


Fig. 1 3D scheme of the tested configuration.

Table 1 Geometry characteristics and performance parameters of the tested pump-turbine.

Impeller data				
D_2 (mm)	B_2 (mm)	n_b	β_{2b} ($^\circ$)	ϕ_{Des}
400	40	7	26.5	0.125
Guide vanes data				
D_3 (mm)	B_3 (mm)	n_b	α_{3b} ($^\circ$)	λ ($^\circ$)
410	40	22	10÷30	-8÷8
Return channel vanes data				
D_4 (mm)	B_4 (mm)	n_b	α_{4b} ($^\circ$)	
516	40	11	30	

power spectra will be computed by partitioning each time signal into 2^8 segments of 2^{10} samples with no overlapping, filtered with a Hanning window for avoiding aliasing and leakage errors. The frequency resolution was 0.125Hz. To determine the non-linear and linear components in the frequency domain, bispectrum analysis will be carried out (Rosenblatt M and Van Ness J W., 1965).

To allow high-speed flow visualizations between stay vanes and guide vanes, the casing was manufactured in Plexiglas. A Photron FASTCAM PCI digital camera will be used and the video camera recorded images at resolution 512x512 pixels with a frame rate 10000 fps and a shutter 1/10000. Two tungsten halogen bulbs with 1000W were equipped to provide the light of the scene. Needle

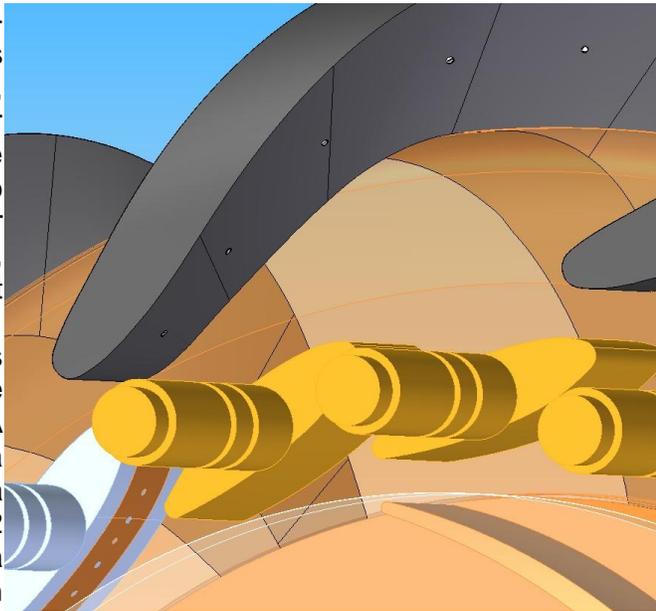


Fig. 2 Detail of the tested configuration.

valves were employed to control the amount of injected air throughout holes of 0.5 mm diameter located in the mid span of the guide and return channel vanes located in the same positions of the pressure transducers, shown in Fig. 2. The injection pressure will be maintained at a value slightly above the mean pressure at the injection location.

Moreover a numerical analysis will be carried out by means of the commercial software ANSYS 15.

The numerical model of the entire machine will include inlet duct, return channel, guide vanes (distributor), runner, draft tube and leakage system.

The draft tube will be discretized by a structured mesh of about 339500 elements with a y^+ values lower than 20. As regards the runner, an O grid was performed. The stage head H and the stage efficiency will be evaluated in preliminary tests to assure grid independent solution for the runner and to guarantee the capacity of numerical solution to capture the local pressure pulsations as well. Even if the sensitivity analysis highlighted a grid independent solution with about 200 000 elements per passage, the intensity and the extent of the local pressure pulsations appear to be correctly evaluated only with grid greater than 300 000 elements per passage. To be sure of the capacity of the numerical solution to capture local pressure pulsations in the whole domain, as a precautionary measure, the adopted number of elements will be further increased to 565 000.

The resulting runner computational domain that will be used have a total of 3.96 million of cells with y^+ values below 15. O-type grids are adopted for both the diffuser and the return channel discretization with about 4.3 and 5.4 million of cells, respectively. The leakage from the labyrinth seal are also considered and several H-blocks are built to describe the cavities.

The choice of the turbulence model is a key issue in CFD. According to the large flow separations expected at part load, a Detached Eddy Simulation (DES) model will be adopted. On both blades and end walls, the boundary layer will be assumed to be fully turbulent. All the interfaces between stator-rotor blocks will be standard transient sliding interfaces.

At the inlet Boundary Condition (BC) the total pressure will be imposed with the values taken from the experimental data. At the outlet BC, due to the highly disturbed flow field, an opening condition with an average static pressure will be fixed.

In order to simulate the load following it will be carried out a simulation with a time-varying boundary condition in which the impeller will be slow down from 100% to about 90% rpm corresponding to a power reduction from full load to about 55% of the full load with a ramp rate of 2.5%.

A second order implicit time stepping will be adopted for the time discretization and a time step of 1° with a maximum number of coefficient loops equal to 5 will be defined.

The numerical performance curves and the unsteady pressure in the wicked gate will be

compared with those experimentally acquired.

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