Modelling and experimental studies on hydro-pneumatic energy storage using pump-turbines

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Résumé:

Ce document vise à présenter les travaux réalisés sur l'étude d'un système de stockage hydro-pneumatique d'énergie utilisant une pompe-turbine rotodynamique multi étagée. Le papier introduit en premier lieu un modèle fonctionnel d'une structure de stockage en cycle fermé (voir Figure 1) basé sur le comportement dynamique des domaines mécanique, hydraulique et thermodynamique. Les points clés du système sont présentés : 1-le rendement d'accumulation, dont l'analyse nous conduit à privilégier dans le futur la recherche d'un cycle isotherme ; 2- la gestion de la puissance qui implique l'étude de structures de stockage alternatives. Pour finir on présente une comparaison entre les mesures faites sur une expérience de cycle stockage/restitution et le modèle qui semble prédire correctement le comportement global du prototype.

Abstract:

Studies on a hydro-pneumatic energy storage system are the main goal of this paper. Firstly a functional modelling of a closed cycle storage structure (Figure 1) is introduced. The paper first introduces the model based on the dynamic behaviour of the mechanical, hydraulic and thermodynamic domains. The key points of the system are introduced: 1- the analysis of the accumulation efficiency led us to favour an isothermal cycle in the future; 2- power rate management optimization requires the study of alternative storage structures. To finish, a comparison between a storage/recovery cycle measurements and modelling results shows that observations were quite correctly predicted.

Key words: energy storage, rotodynamic pump-turbine, compressed air

1 Introduction

Energy storage is one of the widely studied solutions considered by electrical network managers for the current and future increasing stresses on these networks [1]. Energy storage techniques and applications are several. One of the most exiting applications could be contributing in a renewable energy sources higher integration in the existing electrical networks.

Hydro-pneumatic energy storage (HYPES) aims to combine the good efficiency and technological maturity of hydraulic energy conversion and the potential spatial flexibility and energy densities of compressed air energy storage in vessels. HYPES basic idea is to compress air in a closed vessel by means of a liquid piston during storage phase and use this potential energy when needed to drive a turbine (Figure 1).

Thus, HYPES has been under development by different teams in Switzerland [2] and the USA [3], [4]. Structures proposed by these teams use closed or open gas cycles, i.e. single compression and storage volume (Figure 1) or separated compression and storage volumes implying compressed air displacement into separated storage vessels. Another aspect in these works is the use of a physical separation between air and the compressing liquid piston. One of the objectives for these works is to be close to an isothermal cycle in order to obtain relatively high storage efficiencies.

Our project aims at analysing the possibility of using a rotodynamic pump turbine to drive the liquid. More precisely the project purposes are to characterise experimentally and to model the behaviour of a

closed cycle using no separation between air and water. This will be started by the proposal of a dynamic model of a basic system and experimental validation of it. This will result in the analysis of the services this type of storage can bring to the electrical network.

1-Siemens bipolar 70kW electric motor
2-KSB Multitec A50/12A pump/turbine
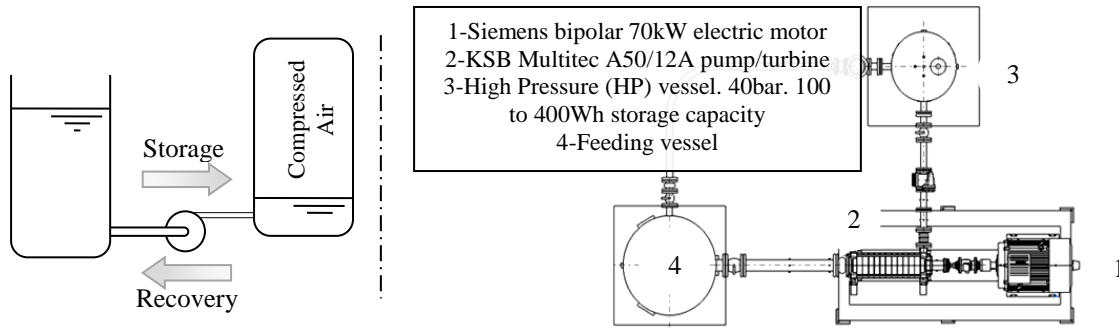3-High Pressure (HP) vessel. 40bar. 100 to 400Wh storage capacity
4-Feeding vessel

Figure 1: schematic description of the system & some details on the test rig

The test rig (Figure 1) is composed of an accumulation steel vessel, a feeding vessel, a multistage rotodynamic reversible pump-turbine and an electric motor/generator. This project is carried out thanks to the financial support of the French agency for environment and energy management (ADEME), which actively induces research on smart grids and storage solutions.

The present paper introduces the models used for the simulation of HYPES test rig dynamic behaviour, some considerations on the functional issues to be solved and a comparison between a low pressure storage test results and functional modelling simulation results.

2   Modelling

The current model, used to simulate the behaviour of the actual test rig configuration should be useful for the proposal and experimental validation of a reliable model that could be used for other HYPES storage structures and this will be the first step of a smart grid configuration analysis.

The model can be divided into the different physical domains involved in the process. Analysed domains are: mechanical, hydraulic and thermal. Figure 2 presents an approximated electrical analogy for these sub-models.
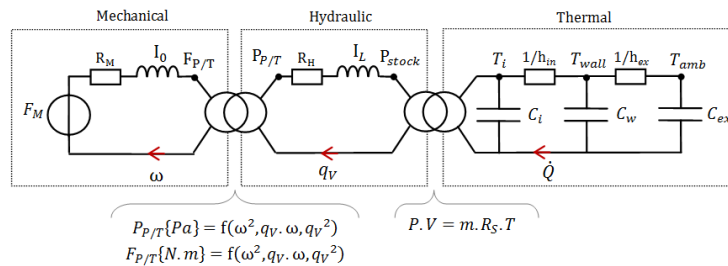

$$P_{P/T}\{Pa\} = f(\omega^2, q_V \cdot \omega, q_V^2)$$
$$F_{P/T}\{N.m\} = f(\omega^2, q_V \cdot \omega, q_V^2)$$
$$P.V = m.R_S.T$$

Figure 2: sub-models of the physical sub-domains

The left hand side torque source (Figure 2, F_M) is the motor. A controller is used to calculate in function of the reference speed the necessary torque to be applied by the motor in the model. Inductive elements (I_0, I_L) represent the mechanical and hydraulic inertias. Capacitances in the thermal sub-model represent internal energy accumulation in internal air (C_i), vessel wall or other intermediate heat storage element (C_w) and external air (C_{ex}). Resistive elements ($R_M, R_H, ...$) correspond to various energy losses. Water temperature increase is neglected since the thermal capacity of the water is high. The thermal equivalent resistance between air and water and the subsequent thermal capacitance of the water are not illustrated for legibility reasons. Since thermal capacitances (m. $C_p\{J/K\}$) of metal, water and external air are high relatively to the expected thermal energies, the capacitive elements C_w and C_{ex} can also be replaced by perfect temperature sources.

Transformers in Figure 2 are used to illustrate the relationship between the sub-models associated to the pump/turbine's characteristics (left) and the storage gas behaviour. The equations describing the dynamic behaviour of these three domains were written in integral causality form in order to solve them by numerical integration in a Matlab-Simulink environment. The following sub-sections focus on the two critical aspects of the system behaviour related to storage vessel and the pump-turbine.

2.1   Vessel modelling and efficiency considerations

Energy is stored in form of elastic potential energy in the accumulation vessel. This energy is related to the pressure and volume at which a given mass of air will be stored. In the compression chamber, pressure varies in a range limited by isothermal (constant temperature) and adiabatic (perfectly isolated and potentially very high temperature variations) volume transformations. Since the polytropic coefficients (relation between pressure and volume) of the compression and expansion are not known, the first principle equation for a closed volume filled by a perfect gas is used to describe the behaviour of air:

$$K.(T_{ext} - T_{air}) - P_{stock}.\frac{dV}{dt} \approx m_{air}.C_v.\frac{dT}{dt} \tag{1}$$

The right hand side term is corresponding to the internal energy variation. The first term of the left hand side is representing the heat flow. It is written, in a simplified form, as the product of a global heat transfer coefficient, mostly related to natural convection($K\ \{W/K\}$), and a temperature difference. The other term is the mechanical work rate i.e. the product of the air pressure (P_{stock}), deduced from perfect gas equation (equation 2), and the gas volume (V) variation which is the opposite of water volume flow rate introduced in the vessel. This flow rate is deduced from the hydraulic sub-model.

$$P_{stock}.V = m_{air}.R_s.T_{air} \tag{2}$$

The time integration of equation 1 allows estimating the air mean temperature (T_{air}), and thus the pressure, variations. Then it becomes obvious how the balance between the mechanical power and thermal power will govern the pressure variations. The only ''external'' variable on equation 1 is the global exchange coefficient K. The air mass (m_{air}) is considered constant and valued at initial state.

Some compression/expansion cycles, separated by a storage time (see pressure evolution in Figure 3), were simulated in order to evaluate the influence of K and the storage time duration on the accumulation efficiency (η) i.e. the output/input energy ratio. The storage time ($\Delta t2$, Figure 3) separating the end of the compression and the beginning of the expansion was normalized by the duration of the compression duration ($\Delta t1$); see definition of D in Figure 3. The pressure drop during $\Delta t2$ is due to thermal exchanges between air and its environment.
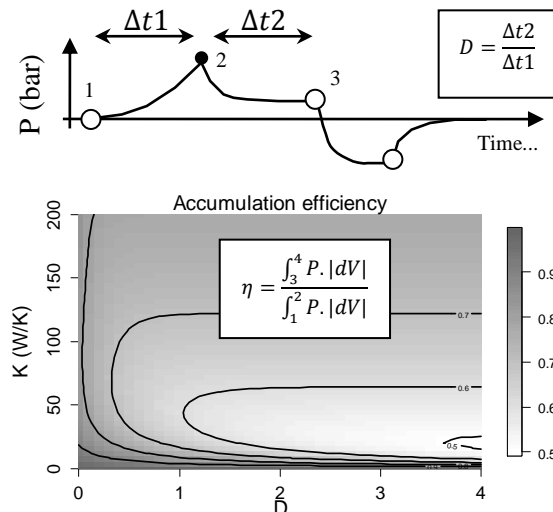

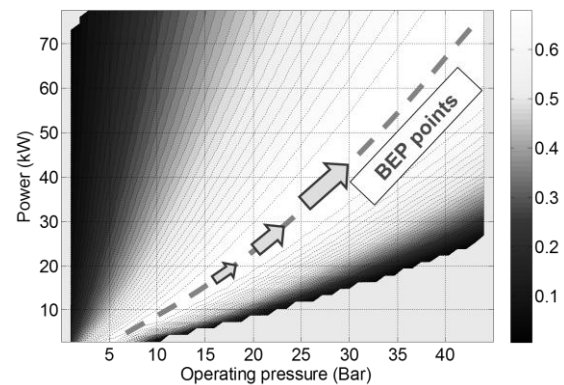Figure 3: a cycle example and efficiency

Figure 4: pump efficiency as function of the operating pressure and power demand

In the example of Figure 3, the initial pressure for each calculation was 5bar, the initial volume 1m³ and the final volume was 0.1 m³.A constant 20 m³/h flow rate is imposed alternatively positive and negative for the different combinations of K and D. Four situations appear in Figure 3:

- K close to 0: the high efficiency zone corresponds to a purely adiabatic cycle; whatever the storage time is, if the extra-pressure related to temperature increase can be kept during storage time, the efficiency will be 1.

- D close to 0: efficiency remains high because in this case losses during storage time are avoided.

- At increasing values of D and for "low" K values efficiency decreases. This is a classical polytropic cycle with long storage time; this could be the more realistic since storage time is not a controlled parameter.

- For higher values of K, efficiency increases because even if heat is "given" to the ambient (or another heat storage device) during compression, with lower temperature differences, heat is correctly recovered during expansion, and thus a relatively high pressure is maintained (isothermal path compared to the adiabatic one).

Consequently, two possibilities appear: try to minimize K or to maximize it. The first one implies large heat storage systems and extremely effective heat insulation capacities that call for potentially expensive technology and is obviously not possible or whished in current direct ''air-water'' interface configuration. The second solution needs great heat exchange capacities that increase with the power rate of the storage device. Estimations done on the magnitude of heat fluxes by numerical and experimental methods for our particular case of storage vessel configuration show relatively low values of K. Increase of these fluxes needs design of new vessel geometries or use of heat exchangers.

2.2 Pump/Turbine and considerations on power flexibility

The behaviour of Pump-turbine was modelled using the steady operating charts. Dimension-less parameters of the machines were used to estimate its behaviour in the various operating points.

Energy accumulation by air compression is not a natural application for rotodynamic pumps. The main difference with its usual applications is the continuous and potentially rapid variation of the operating conditions. A natural answer to this is the use of a pressure dependent reference angular speed imposed to the machine in order to maintain the best efficiency point (BEP, see Figure 4). This becomes a problem when trying to store or recover a given power rate depending on the network manager requirements because the BEP power depends on the pressure at which the air is contained in the vessel (Figure 4). Turbine operation has the same problem for an even narrower ''good'' efficiency hill around BEP.

Solutions to this problem are already known for PSH plants that manage lower pressure variations: use of mobile guide vanes for turbines regulation or bypass operation using separated machines for pumping and recovering simultaneously in order to obtain the desired power input/output [5]. Also, we can find in Sylvain Lemofouet PhD thesis [2] an interesting solution that combines a hydro-pneumatic system to a secondary more flexible storage device, super-capacitors, in order to maintain the BEP operation for the hydraulic system.

Another power flexibility solution was analysed and simulated during the present project, based on the combination of several pre-charged closed cycle parallel vessels operating at different pressures [6]. The idea was to increase the degrees of freedom by making possible the choice of better operating pressure depending on the demanded power rate. The energy density is not optimized but the operation at relatively high efficiencies can be maintained for a large power range.

3 Comparisons between model and experimental results

In this section an experimental test example is presented; it corresponds to storage/recovery cycles for which maximal operating pressure was 5bar. Some comparisons with the model are also proposed.

3.1 Test rig instrumentation and control

The instrumentation of the test rig is composed by:
- Two Keller 33X piezoresistive pressure transmitters of 10 and 45 bar measurement range with a 0,01% FS uncertainty located upstream and downstream of the pump/turbine and a third one installed on the HP vessel.
- Endress & Hauser 0,2 % FS uncertainty electromagnetic flow meter.
- 1024 pulse per rotation encoder for the motor rotational speed the measurement
- HBM torque transducer
- Krohne contact radar level transducer is inserted in the vessel for the air volume estimation

Mean temperature of the air in the vessel is estimated by the perfect gas equation.

The storage and recovery test is done in a hydraulic machine best operating point configuration. In this purpose, the input/output pressure difference ($\Delta P_{P/T}$) is measured in order to estimate the BEP speed (ω_{BEP}) using the BEP value of the dimensionless pressure of the machine (ψ_{BEP}). The accumulation vessel opening and isolation is done by a pneumatic driven ball valve. The management of the valve and the motor speed are illustrated in Figure 5.

Vessel was pre-charged to 2,4bar in order to achieve a realistic storage cycle for which operating at low pressures is not useful in terms of conversion efficiency and power; this pre-charge is followed by

a wait time in order to start the cycle at ambient temperature. Maximal pressure of compression was set to 5bar and initial volume of air was around $0,17 \ m^3$. The pumping and turbining periods were launched manually. The data acquisition and control frequency was 60Hz.
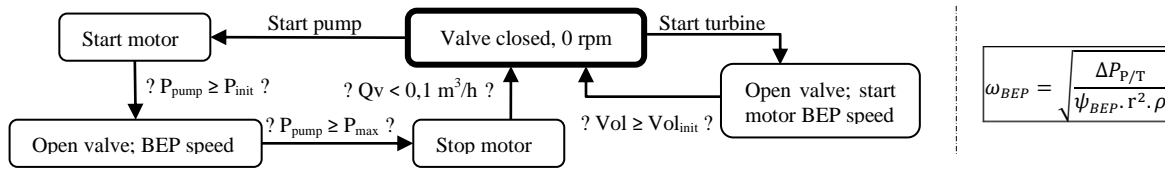
Figure 5: schematic description of valve and motor management and the calculation of BEP speed

3.2 Experimental results and comparison to model predictions

The model used for these comparisons is based on the elements illustrated in the Figure 2. Like in the experience, the only driving physical variable is the motor torque calculated by a PI speed controller used in the model. The reference speed and the valve are managed as it is done for the test rig and described in Figure 5. For this simulation a global heat transfer coefficient and a constant and uniform ''air-surrounding'' interface temperature were implemented. This global transfer coefficient empirical expression was obtained based on various compressions/expansions analyse using the method proposed by [7].

Figure 6 shows the evolution of the air pressure inside the storage vessel and the ''high pressure'' of the pump/turbine i.e. the outlet pressure for the pump and inlet pressure for the turbine. Difference between air and machine pressure in Figure 6 is related to water height difference and line pressure loses. Machine pressure difference is relatively well predicted except during start-ups and shut downs of the motor and valve opening and closing periods during which the transient phenomena have to be better modelled.
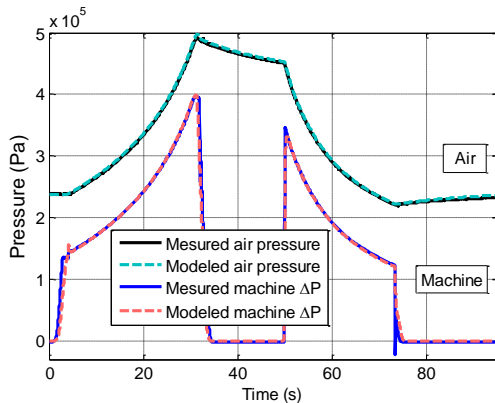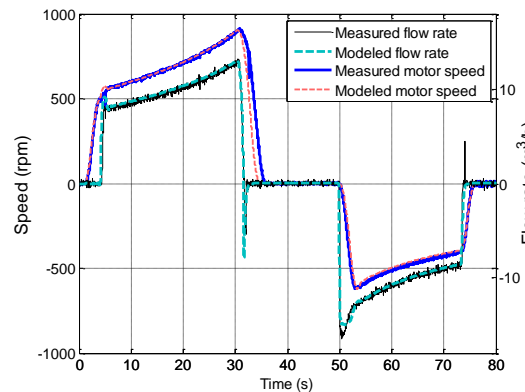
Figure 6: machine high pressure and air pressure

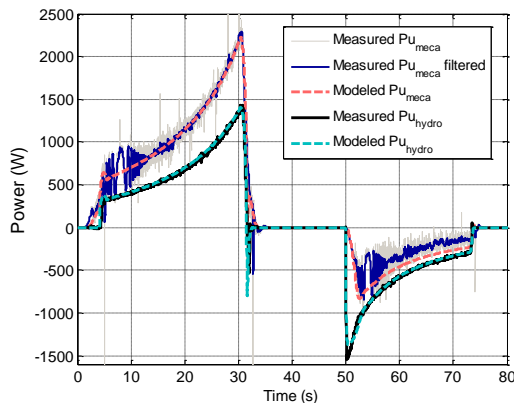Figure 7: rotational speed and flow rate
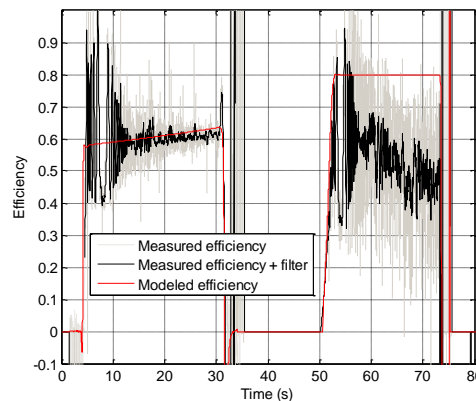
Figure 8: mechanical and hydraulic powers

Figure 9: pump turbine efficiency

The air pressure is close to the predictions; this means that temperature, air volume, mechanical work rate and heat flux applied to air are quite correctly estimated. The estimations are not so good during

storage time in particular the last one (t>73s); this is can be related to and error on air volume estimation or to the heat transfer coefficient expression that could not be adapted for convective flow configuration at the end of expansion period when temperature differences sign changes. This last aspect is not so critical during compression/expansion periods because thermal power rates are comparatively lower than mechanical ones. The correct air volume estimation depends on the model prediction of the water flow rate entering in the vessel (Figure 7); in this case the compression volume variation prediction was 2,2% different from experimental estimations.

As pressure changes during compression or expansion, the reference speed applied to the motor drive is adapted and the torque, thus the mechanical power changes (Figure 8). This adaptation makes the efficiency remain close to its maximum (Figure 9). Efficiency was defined as the hydraulic machine output/input power ratio; some inconsistent points (efficiency above 1 and lower than 0) appear during valve opening and closing due to an incorrect data treatment in these transitory periods.

The mechanical power during compression is correctly predicted contrary to the expansion period because an over estimation of the turbine torque. This is also illustrated by the over estimation of the turbine efficiency (Figure 9). Steady operating tests should be done to obtain the correct behaviour of the turbine as it was done for the pump operating mode. Also, for a given speed range, a kind of disturbance on the torque (see mechanical power in Figure 8) appears. This can be due to a resonance phenomenon not correctly modelled.

4   Conclusion

Hydro-pneumatic storage systems solutions can be supplemented with the concept proposed here, using a rotodynamic pump turbine and a free surface configuration. The modelling of such a system is introduced and the key issues of such a storage system have been illustrated: storage efficiency and power control. The increase of efficiency should certainly need the increase of heat transfer coefficient. Some power management solutions should be evaluated such as the use of a secondary power flexible storage system [2], pump and turbine simultaneous operation for power compensation (hydraulic short circuit) [5] or the use or pre-charged parallel vessels [6].

The experimental validation of a closed cycle structure began and it seems that the model predicts correctly the global observations done on a low pressure storage/recovery test. Supplementary experiments have to be done in order to test the model in a larger operating range.

In the next months, the test rig should be electrically connected to the ENSAM Lille electronics laboratory experimental micro grid in order to begin the storage system grid integration tests.

Nomenclature

$C_v =$	Specific heat at constant volume of gas		$Q_v =$	Flow rate
$F_i =$	Considered torque		$r =$	Pump/turbine wheel radius
$h =$	Considered convective heat transfer coefficient		$Rs_i =$	Specific gas constant
$\dot{Q} =$	Considered heat flow		$\Delta P_{P/T} =$	Pump/turbine's pressure difference

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