System for recovering energy in hydraulic circuit by using a small Pelton turbine

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Abstract.

Energy recovery and energy dissipation reduction are nowadays very important topics of academic and industrial research in the field of fluid power systems. In this paper, the authors wanted to study the possibility to recover the hydraulic energy that would be otherwise wasted through a pressure relief valve in a typical industrial hydraulic system, using a Pelton turbine connected to the shaft of the pump. The hydraulic system is a power transmission application in which a fixed displacement pump is applied. The energy usage on the actuator side is discontinuous and thus creates high energy waste through the pressure relief valve. For this application, the use of a variable displacement pump is considered to increase too much the cost of the system, so an alternative solution is proposed, which is based to the recovery of the wasted energy. The operating conditions considered are in the range of high pressure and low flow rate, so the hydraulic system here examined has been compared to small hydraulic power plant that uses Pelton turbines, where the installation faces watercourses for energy transformation. In this article the design of a Pelton turbine is then examined for the application previously introduced, by means of standard design methods; the goal of the design is to obtain the best efficiency by appropriate sizing. Successively, a lumped parameter model of the entire system, created to validate the concept when working in the real operating conditions, is presented and discussed.

Keywords. Hydraulic system, energy recovery, Pelton turbine, design.

1. Introduction

Fluid power systems are widely used in industrial and mobile applications thanks to some clear advantages in the power transmission, which are: the high power-to-weight ratio, capability of being integrated in complex systems – in relatively small spaces–, the capability to be quickly reversed, or operated intermittently, the fast response and acceleration, and the reliable operation and long service life. However, as well known, they are characterized for having some critical issues, one of the more concerning nowadays is the low efficiency. Since the sensibility to the energy waste and pollution is now increasing more than ever, forcing each technology to improve, also in the fluid power field different solutions have been introduced to compensate the gaps, in some cases with installation of electrical equipment, in other with different solutions that comprise the recovery/regeneration of otherwise wasted energy. Researchers, for example, address this topic with some alternative solutions for standard fluid power architectures [1, 2]. An entire special issue has been dedicated recently to energy efficiency in fluid power systems [3].

In this paper an industrial fluid power system is considered and a possible way to recover some energy is studied, with reference to the use of the system for a specific characteristic working cycle. As it is typical for some industrial applications, a single fluid power generator group, made of a fixed displacement pump driven by an electric motor and a pressure relief valve, is used for multiple purposes, and hence it's not optimized for a specific application. Here, this generator group is used to replicate a working cycle characterized by the use of the fluid power for a short time and then during the rest of the time the hydraulic power produced is wasted through the pressure relief valve.

Several possible more efficient alternatives to this simple system already exist, which may be more suitable for the working cycle, i.e. variable displacement pumps with different displacement controls. However, with the idea of improving an existing system that already employs a fixed displacement pump which, for various reasons, the user doesn't want to modify, an alternative idea is to recover the energy that would otherwise be wasted. The reasons for not replacing the fixed displacement pump with a variable one can be different, for example: initial cost, issue in choosing a more delicate pump in case the system is working with an average level of contamination of the fluid, critical response time of the pump to possible rapid requests of flow rate by the actuation.

The concept explored in this paper is to re-transform in mechanical energy that fraction of hydraulic energy that would be wasted by the system through the pressure relief valve. In this way, we can generate a closed loop of power transformation that would help to decrease the absorption at the source. Because of the discontinuous working phase – that leads to drag losses during inactivity – and the atypical configuration, it was decided to explore the chance to use a Pelton turbine instead of using a positive displacement motor to recover the energy. If a hydraulic motor is used, which would be operated frequently without receiving flow rate from the system, the two main concerns are the losses and hence the power adsorption at the pump shaft and the rotation of the motor without enough flow rate even to lubricate the gaps, resulting in overheating and possible wear. Of course, the first point concern also the use of the Pelton turbine and was taken into account in the analysis. To minimize resistances and losses in this closed loop of energy recovery the initial intention was to design the machine to be directly connected to the pump shaft, but after first calculations (discussed in chapter III) it was evident that this solution was not possible. Anyhow the loss introduced by the presence of a reduction drive is quite low and the solution of a Pelton turbine plus reducer may be appealing. In the next sections we describe the modified layout of the system, discuss the Pelton turbine design according to two practical methods and present the modelling of the complete system, to simulate the duty cycle and verify if the solution is convenient from the energy recovery point of view.

2. BRIEF DESCRIPTION OF THE CONCEPT

As announced in the introduction, we want to explore a design solution where the recovery of energy is performed by using a Pelton turbine, that should collect the high-pressure fluid otherwise wasted through the relief valve of the system during passive intervals of the duty cycle considered. For the practical application, we are referring to a test rig that is replicating the duty cycle of interest: the Pelton turbine will be coupled to the pump-motor shaft via a reduction drive, (the design of the Pelton turbine excludes the chance to connect the turbine directly to the pump as it is explained later), the fluid sent to the turbine is first collected in an accumulator with a chosen pre-charged pressure. This was a mandatory choice due to the discontinuous characteristics of the flow rate discharged by the relief valve. The accumulator allows to reduce the irregularities in the flow rate and pressure. An opportune control logic, based on the accumulator pressure, decides when opening the on/off valve that faces the turbine runner. The working position of this valve is the nozzle of the turbine. A layout of the test rig is shown in Figure 2.1.

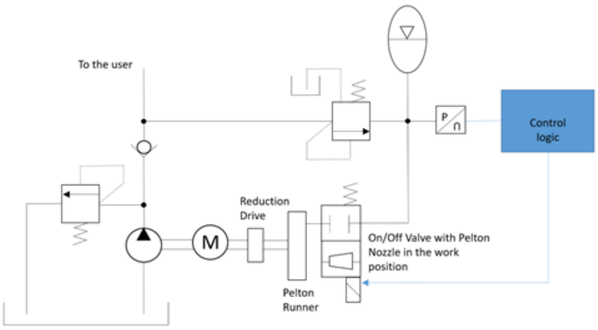

Figure 2.1. Hydraulic diagram of the fluid power generator group of the test rig with the integration of the Pelton turbine

The next step is to design a Pelton turbine that can be used in the application referring to the target operating conditions. The size of the turbine starts from the estimation of the mean value of power lost that is around some kW.

3. DESIGN OF THE PELTON TURBINE: DISCUSSION

3.1. Preliminary observations

In turbomachinery different dimensionless parameters classify the operative conditions and are used, thanks to the Similitude Theory, to choose the type of turbine, as a function of the machine size and target flow [4, 5]. In this study these parameters are also used to estimate how far the actual operating condition is from the standard ones.

A turbine preliminary design is being considered with some input parameters that circumscribe the boundary condition of the design [6, 7, 8]:

- the net head available H, a characteristic of the system, equivalent to $\frac{p}{g\rho}$ where g is the gravitational acceleration, p the fluid pressure and ρ is the fluid density;
- volumetric flow rate $\dot{Q}_{jet}$, another characteristic of the system, even if sometimes it is substituted by the power at the shaft P;
- the rotational speed N, a mechanical characteristic of the system; in the first step of the design process we are considering that the turbine is directly jointed to the pump-motor shaft that rotates at 1500 [rpm].

The first important design factor for turbines is the *specific speed* N_s, that is derived from the previous parameters listed. The value of this factor is a guideline that helps to select the type of turbine that is more efficient at the nominal operating conditions considered for the design. In this work the definition of the specific speed considered is expressed by (1):

$$N_s = N\frac{\sqrt{\dot{Q}_{jet}}}{H^{3/4}} \tag{1}$$

In Eqiation (1) N is expressed in revolution per unit of time, so N_s is not dimensionless; it can be transformed in a dimensionless parameter (the speed number) using the angular speed instead of N and dividing for $g^{3/4}$, but it is often used as shown in (1) in diagrams and tables. Alternatively, the *power specific speed* can be defined as in (2):

$$\Omega_{sp} = \Omega\frac{\sqrt{P/\rho}}{H^{5/4}} \tag{2}$$

In (2) Ω is the *angular speed* expressed in rad/s, P is the power and ρ is the fluid density. In general, low specific speed machines correspond to low volumetric flow rates and high heads, whereas high specific speed machines correspond to high volumetric flow rates and low heads [9].

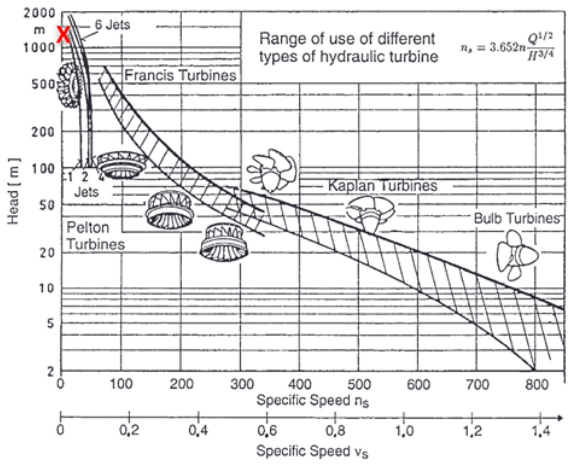

Figure 3.1.2. Operative setting point of the discussed system, red cross, with application ranges of various types of turbines [10]

In Figure 3.1.1 it is possible to compare the operative conditions of the considered system (red cross in figure) to the most common operating range of hydraulic turbines. The common values for Ω_s are also shown in Table I.

TABLE I. OPERATING RANGES OF HYDRAULIC TURBINES [9]

	Pelton turbine	Francis turbine	Kaplan turbine
Specific speed Ω_s [rad]	0.05 – 0.4	0.4 – 2.2	1.8 – 5.0
Head H [m]	100 – 1770	20 – 900	6 – 70
Optimum efficiency [%]	90	95	94

The high delivery pressure value on the delivery line of the test rig is the high and the volumetric flow rate is the small, leading to a critic low value of *specific speed* and *power specific speed* (3):

$$\Omega_s = 2\pi \cdot N_s = 0.01 \tag{3}$$

To have an idea of the *rotational speed* that should be used as reference operating condition for the turbine we can refer to diagrams such as the one in Figure 3.1.2, where *typical efficiencies* of turbines are related to the *specific speed value*. Again, we can see that our operating condition is at the limit also for a Pelton turbine and that, even if it may be possible to design the turbine, the efficiency would be too low. The idea to directly connect the turbine to the pump-motor shaft that rotates at 1500 rpm failed since this would lead totally out of good efficiency range, hence a reduction drive between the turbine and the pump is necessary.

To obtain a total energy transformation, diameter and velocity of jet flow through the nozzle that faces the turbine runner are defined: the velocity is function of the pressure – defined by the equation of energy transformation – and following, the diameter depends on the volumetric flow rate value (at given pressure). In this paper the transformation efficiency of the nozzle is not considered for simplicity. It is also considered to insert an anti-shock relief valve on the pump delivery line for discharging undesired peaks of pressure and for safety reason. Finally placing an accumulator just before the logic element, allow to control the on-off valve with a simple logic: when the accumulator is full and the pressure reaches the maximum value, an electrical signal opens the logic valve and the flow starts to reach the turbine; the valve remains open till the minimum pressure set is reached, under the minimum pressure the electrical signal is off, the valve closes and the accumulator restarts to fill. These modifications of the standard fluid power generator group allow to overcome the problem of the small and irregular flow coming from the circuit. Without these actions, the behaviour of the turbine would be too irregular and, as shown in Figure 3.1.2, the efficiency of the Pelton turbines drops to low values when working far from the nominal condition.

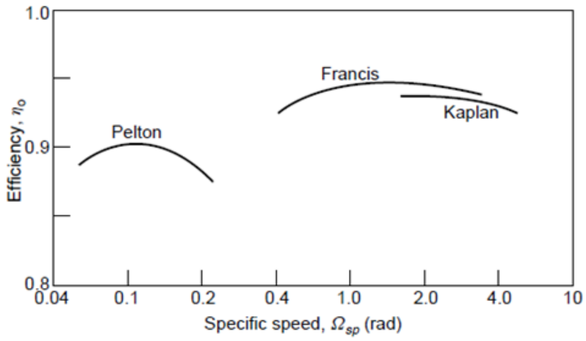

Figure 3.1.2. Typical design point efficiencies of Pelton, Francis and Kaplan [9]

3.2. Design of the Pelton turbine

It is decided to proceed with the design of a Pelton turbine with 1 jet (usually more than 1 jet is applied when the net head is medium-small and the flow is high – the opposite situation of the actual operating condition in our system). Two different theories are used for the machine design: the design A is taken from The Micro-hydro Pelton Turbine Manual by J. Thake [11] and the design B from Pelton Turbines by Z. Zhang [12]. The design A approach [11] is thought for small Pelton turbine applications and it uses some simplified equations based on the specified installation case. Encountering some difficulties in adapting the present case to standard procedures, it is decided to explore a second method [12], which is more recent and theoretically more adaptable to general cases. The two different theories of design are briefly described in the following.

The Thake's theory [11] dimensions the entire turbine on the jet diameter that, as explained before, is depending on the inlet flow rate. The entire geometry is hence determined by this value and the designer has only to scale the design to the application. Equation (4) is used to calculate the *pitch circle diameter* PCD of the turbine, an extremely important characteristic. A parameter δ for the scaling is suggested by common practices and it is normally taken between 0.1 and 0.12 even if in some cases of high head it can reach lower values, such as 0.06÷0.07 [13].

$$\delta = \frac{d_{jet}}{PCD} \tag{4}$$

The common approach of reducing the δ value in cases of high head is justified by the need to avoid high stresses, fatigue problems and cavitation damage caused by too small runner diameter [13, 14]. As already explained, the present application is characterized by an important disproportion between the diameter and velocity of the fluid jet (caused by high pressure and low flow rate operating conditions). Applying a standard value of δ would generate a very small runner diameter that rotates at very high speed. After some investigation, it is decided to proceed assuming a value of $\delta = 0.02$. Once decided the value of the PCD, the minimum number of buckets and all the other dimensions are determined consequently [11]. The output from this theory is a Pelton turbine with a diameter designed on the value of flow rate, but the geometry obtained is less suitable to manage the velocity vector that, in the present case, is extremely high. In Figure 3.2.1 the output turbine CAD model from the first design is shown on the left.

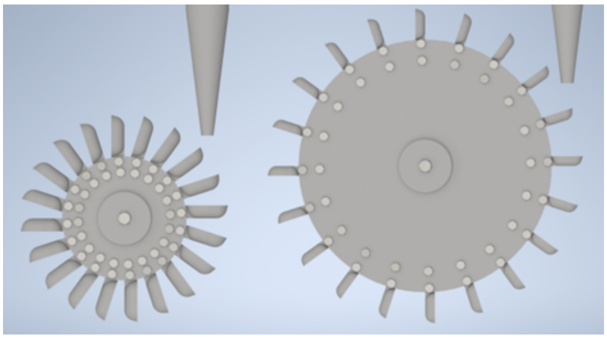

Figure 3.2.1. Design A (left) and design B (right)

The Zhang's method [12] uses both jet velocity and jet diameter to design the turbine. The procedure starts with the definition of coefficients that describe the turbine and correlate the jet properties to it. They are the *peripheral speed coefficient* k_m, which defines the ratio of the peripheral speed of the Pelton wheel on the jet speed (5) and the *bucket volumetric load* φ_B, that correlates the flow rate to the bucket dimension (the flow rate is proportional to the square of the jet diameter) (6), where h is the bucket width.

$$k_m = \frac{PCD \cdot \Omega}{2 \cdot v_{jet}} \tag{5}$$

$$\varphi_B = \left(\frac{d_{jet}}{h}\right)^2 \tag{6}$$

Practical applications establish the range in which these coefficients can vary. Then the wheel size, the number of injectors and the rotational speed of the turbine can be chosen respecting these constraints. The standard ranges that normally are adopted are (7) and (8)

$$\varphi_B \cong 0.11 \tag{7}$$

$$k_m \cong 0.45 \div 0.48 \tag{8}$$

This study does not analyse the effect of the oil viscosity in the bucket and therefore (7) is accepted. Instead (8) must be analysed widely. The range derives from the theoretical efficiency equation of the energy conversion between the jet and a straight-moving bucket expressed in (9) where β_2 is the angle of the outgoing flow from the bucket [12].

$$\eta_{th} = 2k_m(1 - k_m)(1 - cos\,\beta_2) \tag{9}$$

From (9) the highest value of efficiency corresponds to $k_m = 0.5$, then the real value considered is lower, (8), because of the losses in the transformation process. Different contributions of losses must be estimated:

- Missing flow interception: when the velocity of the fluid and the velocity of the bucket differs too much from the theoretical ideal value of 0.5, it can happen that part of the flow is not intercepted by the bucket for the conversion of energy. In this case a coefficient R_Q is introduced to consider the percentage of fluid reaching the bucket, the rest of the fluid – and respectively its energy – is considered wasted [12].

- Windage and friction: the resistance of the rotation in mixture of air and liquid and the mechanical resistances are estimated on the base of simplified models [12, 13]. The shape of the Pelton wheel, together with air that particularly resists against the buckets, complicates the estimation of this friction. Considering the high specific density of the air-oil mixture that is present due to the very high-speed of the wheel, the number of buckets and the orientation of the runner – horizontal or vertical – it is possible to estimate the losses with an accuracy of ±50% [12]. To have higher precision it is suggested to perform a comparison with experimental data, when available. This loss term is addressed as $\eta_{f\&w}$ in the following.

Normally these total losses represent a small percentage on the total efficiency of the turbine (< 1% [12]). Unluckily, in our case the rotational velocity of the wheel, due to the high speed of the fluid, reaches high values, consequently the amount of these losses may have an important impact on the output efficiency. It is decided to proceed as explained in the following: the dimension of the PCD is taken the biggest possible (deemed acceptable for this application) to reduce as much as possible the rotational velocity; then the losses are estimated as function of k_m to realize a new efficiency curve, which will be different from the theoretical efficiency defined in (9). From it, a new value for k_m is taken considering the best condition on the curve of the real efficiency and finally the *rotational speed velocity* and the *reduction drive ratio* are calculated. Differently to design A in this methodology the number of buckets is a variable value that depends on k_m and Ω_s [12]. The calculations suggest a very high number of buckets. By practice this value is limited to 20÷22 due to the narrow space between adjacent buckets. In this application the big PCD and the small bucket would allow more space to adjust more elements, but the cutout edge shape machined at the bucket tips would result deformed by the velocity vectors applied in the shape definition. To avoid an upset of the entire geometry it is decided to accept more space between elements and the limit of 20 buckets is kept.

In Figure 3.2.2 is possible to see the calculation of the new efficiency curve referred to design B. The final curve of η_r is obtained from the theoretical η_{th} multiplied by the missing flow factor R_Q and the friction and windage efficiency $\eta_{f\&w}$. Grey bars represent the inaccuracy in the estimation of $\eta_{f\&w}$. Higher resistances of the model moved the ideal *peripheral speed coefficient* from the ideal value of 0.5 to 0.42. After choosing the number of buckets, it was verified that the full factor R_Q begins decreasing once the maximum efficiency has been reached.

The output of the second design approach is shown on the right in Figure 3.2.1. Finally, a re-calculation of the real efficiency as function of the rotational speed has been performed for both designs A and B and the nominal operative velocity of each turbine can be estimated referring to the point of maximum efficiency, as shown in Figure 3.2.3.

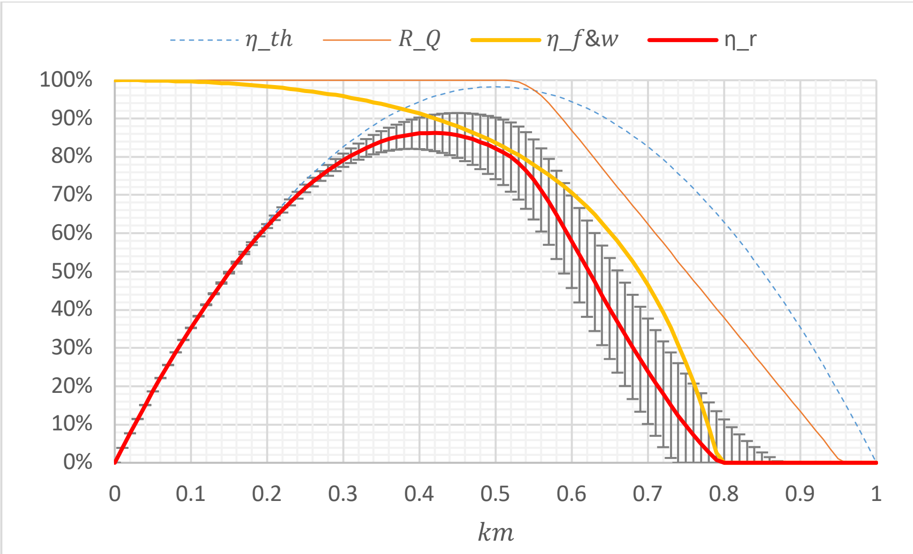

Figure 3.2.2. Contribution of loss in efficiency curve of design B. Black bars represent the effect of resistances inaccuracy

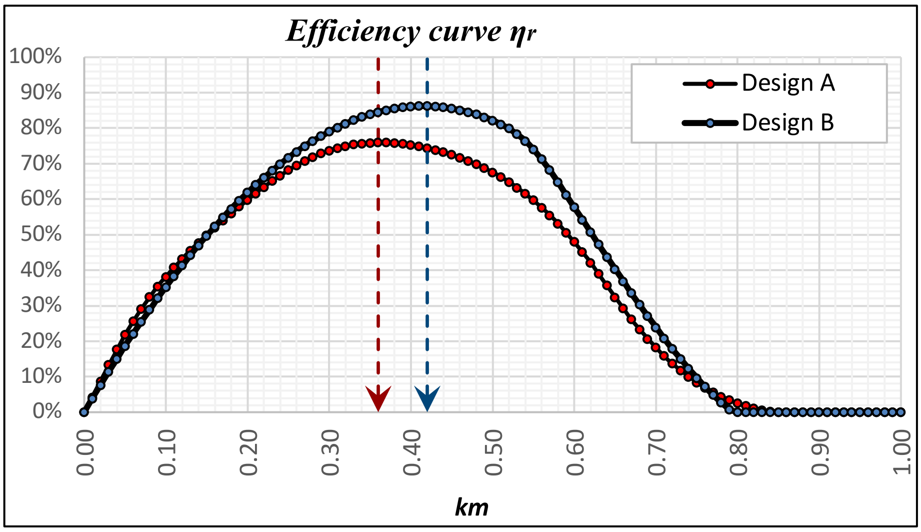

Figure 3.2.3. Characteristic curves of the turbines

4. SYSTEM SIMULATION

The entire system is then modelled with a lumped parameter approach developed in OpenModelica Environment [15] making use of the elements of the hydraulic library developed by SmartFluidPower [16] and presented in Figure 4.1.

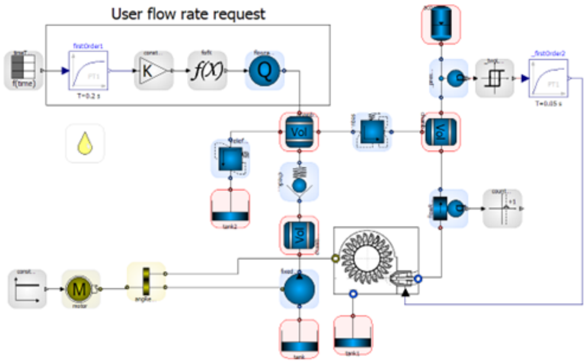

Figure 4.3. Lumped parameter model in OpenModelica environment [15]

In the model it is possible to identify:

- the generation of power composed by pump and electrical motor connected with the reduction drive to the turbine;
- the user flow rate request, numerically controlled by an external table in the model;
- the system of accumulator and activation valve already discussed in Chapter 3. Here the pressure sensor directly controls the opening of the nozzle.

The pump generates a constant amount of volumetric flow rate and, when the user is satisfied or not asking for flow rate, the pressure in the system will rise until the opening of the sequence valve that is filling the accumulator with the oil in excess. The maximum and minimum trigger pressures and accumulator capacity are designed to guarantee as much as possible regular and constant injections of flow to the turbine. The model of the turbine is built using the efficiency Equation (9) and considering an estimation of the losses already discussed inside the machine. The geometrical parameters calculated during the design are the inputs of this model in a way to perform different simulations. The estimations of friction and windage losses are calculated apart from the fluid jet velocity since they are the resistive part of the machine that is present also when the turbine is not collecting fluid from the nozzle (k_m value tending toward infinity). When opened, the pin at the nozzle lets the fluid flow towards the Pelton turbine model that elaborates it and calculates the mechanical power. The mechanical speed velocity is set constant; in the reality the system requires a sort of regulation to compensate the variation of load to the electrical motor when the turbine is switching on/off. Below the results of the turbine of design B are shown.

The simulation is performed with the user request power shown in yellow in Figure 4.2 and the excessed power produced by the pump is directed to the accumulator that opens to the turbine at the reaching of the maximum pressure value, set on the control logic. It is possible to identify the intervals in which the turbine starts to work and feed power to drive the pump.

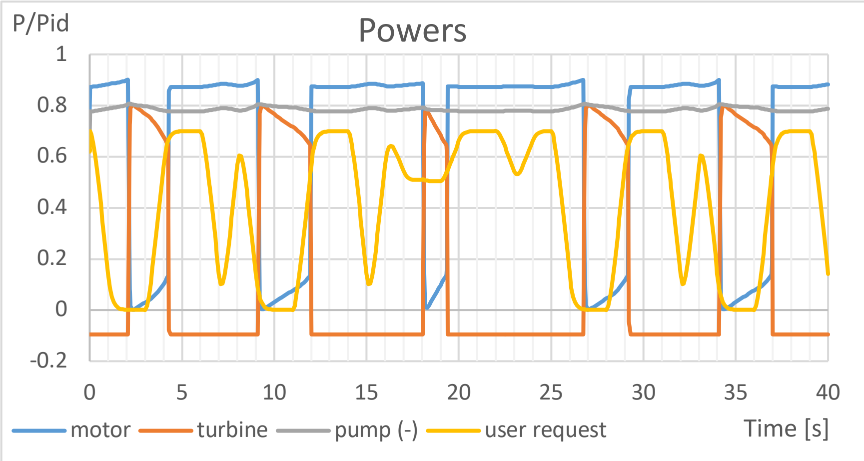

Figure 4.2. Power exchange in model simulation

During the 40 sec of simulated time, 5 injections at the turbine of different durations happen in total. In the remaining time the turbine acts as a dissipator due to frictions and windage resistance. At the accumulator ports, in Figure 4.3, it is possible to notice the action of regularization of the fluid jet properties at the turbine inlet. The fluid arrives at very irregular intensity; it is stored and lately delivered at almost constant value.

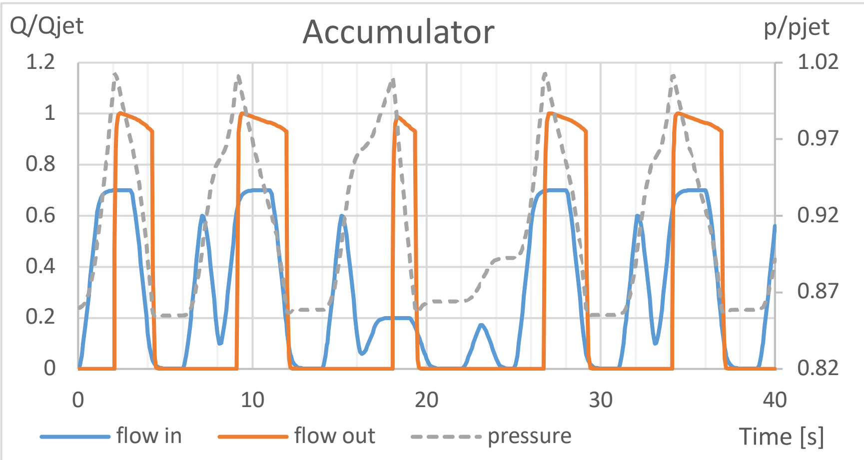

Figure 4.3. Power exchange in model simulation

The variation of volumetric flow rate at the output is due to the accumulator pressure decrease during its emptying (i.e. to the difference between min and max pressure set). Extending these trending to a long-term simulation, it is possible to estimate the recovery of energy shown in Figure 4.4. The model estimates that between the mechanical power at the shaft and the effective hydraulic power collected there is a 53% of overall efficiency, all the rest is lost. Of course, this value strongly depends on the user request. The 23% of energy is lost due to the power transmission line and the remaining 24% is the potential discharged by the relief valve. The introduction of the Pelton turbine allowed the system to collect this potential. Considering the efficiency in the collection (Figure 3.2.3) and the absorption during passive intervals the net recovery is about 11%. Even this value is user-dependent obviously, different duty cycles should be tested to have more a general idea.

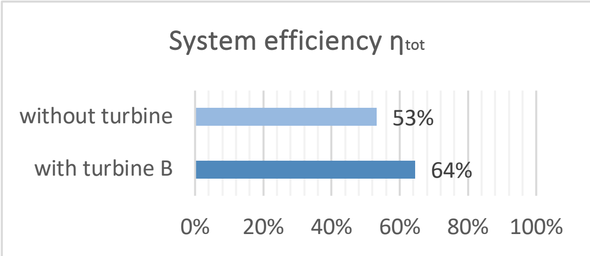

Figure 4.4. Final recover of the system efficiency loss

5. CONCLUSIONS

In this paper, the idea to use a Pelton turbine in a hydraulic industrial system to recover part of the wasted energy during a specific duty cycle has been explored. Two different Pelton turbines have been designed for the application, both requiring to introduce a reduction drive at the shaft between the pump and the turbine, introducing an additional, but typically small, loss of efficiency. The high-speed rotation of the turbine runners is a necessary condition in order to maintain the dimensions of the turbine bounded to remain under certain values. The design process of the Pelton turbine is a mixture of empirical and physical formulas and here has been developed making reference mainly to [11] and [12]. This study is done with the purpose to apply these processes in an un-experienced range of operation and verify the possibility to recover energy in the system. The two designs here experimented led to different turbine designs, with different operative points in terms of speed. It is extremely important to compare the application of these methods and the output efficiency estimations to experimental data. The two turbines have been prototyped and will be tested in the next future by ELT FLUID on the test rig taken as reference for the analysis and, successively, the experimental results will be compared to the simulation. The possibility to obtain some result from a prototype would validate the theory but, most important, it would give the possibility to focus on weak points of the application. The simulation of the lumped parameter model shows that, despite the amount of losses and resistance caused by the high rotational speed of the turbine, energy recovery is possible. Also mechanical resistance of the turbine components has been analysed, in particular the resistance of the buckets to centrifugal forces and to fatigue stress, but not reported here for brevity. A critical issue on this field may be the erosion of the edge of the bucket due to the fluid jet action. In case of splitter edge erosion, the flow detachment from the bucket surface would be more prone to happen leading to significant drop of efficiency [13, 14]. According to formulas from [12], with small jet diameters the decrease of efficiency is more influenced by the splitter rounding. Even an erosion of few decimal points of millimetre can be critic for the interface between bucket and fluid. Another point to be analysed is the air added/mixed with oil during the passage through the turbine, for which an opportune settling time in the tank should be considered.

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