

# NUMERICAL SIMULATION OF THE ALBA SYNCHROTRON LIGHT SOURCE COOLING SYSTEM RESPONSE TO PUMP START-UP AND SHUT-DOWN

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## Abstract

The ALBA Synchrotron Light Source cooling system is submitted to regular pump start-ups and shut-downs. Moreover, pumps can trip due to motor power failures. As a result, the piping system can be subjected to surges and pressure oscillations. The 1D thermo-fluid simulation software Flowmaster® has been used to predict these transient conditions taking into account the fluid compressibility, the pipe elasticity, the characteristic time response of the check valves and the pump/motors moments of inertia. During pump start-ups, significant pressure rises are detected that can be reduced by readjusting the PID controller parameters. Unexpected pump shut-downs do not appear to provoke significant water hammer conditions. However, pressure fluctuations are generated mainly in the same pumping line but also in the rest of the system due to the particular common return configuration. In all the cases the pressure regulation mechanisms acting on the pump rotating speeds serve to attenuate the consequences of these transients. Finally, the feasibility of the model to simulate the effect on the system response of trapped air inside the pipes has also been evaluated.

## INTRODUCTION

The Consortium for the Exploitation of the Synchrotron Light Laboratory (CELLS) is in implementation state of a transversal project whose objective is to develop and improve the ALBA's synchrotron cooling system. It is aimed at improving the system reliability and stability faced to flow changes induced during normal operation and occasional failures.

One particular activity consists of understanding the cooling system response to start-up and shut-down of any pump device in the cooling system and detecting if pressure variations created in transient events may lead to failures and/or piping components' deterioration.

It is well known that transient flow conditions in piping systems with large flow rates can cause unwanted water hammer problems. Water hammer creates high amplitude pressure waves that travel back and forth inside the pipe network until they are dissipated by friction [1-2]. Excessive high pressures can lead to damage or rupture of

pumps and valves. Excessive low pressures can lead to air entrance, cavitation and release of large amounts of dissolved air.

Faced with the difficulty and risk of carrying out experimental tests on the actual pipe network, it has been decided to use the existing Flowmaster® model [3] to simulate numerically the system response. For that, the model has been modified to take into account the inertial and compressibility effects in pumps and pipes by replacing and adding the adequate components. In addition, it has also been necessary to program new scripts and to make changes in the PID controllers in order to simulate the real pumping system start-up/shut-down procedures.

## NUMERICAL MODEL

A key parameter to calculate water hammer phenomenon in pipes is the pressure wave propagation speed,  $a$ , that can be estimated with Eq. (1) where  $B$  is the bulk modulus of elasticity of the liquid,  $\rho$  is the fluid density,  $D$  is pipe diameter,  $e$  is the pipe wall thickness and  $E$  is the modulus of elasticity of the pipe wall material.

$$a = \sqrt{\frac{B/\rho}{1 + \frac{DB}{eE}}} \quad (1)$$

For example, the wave speed  $a$  is around 1200 m/s for water at 23 °C flowing inside a stainless steel pipe of size DN 150 Sch 10. Accordingly, the adequate wave speed for all the different pipe sizes of the network have been calculated and used in the model.

Most common causes of transient problems in pipe systems are the pump start-up and pump shut-down due to a power failure (pump trip). To correctly simulate these effects the pump and the motor moments of inertia must be included in the model. For that, the Thorley empirical equations have been used [4]. The motor inertia,  $I_m$ , and the pump inertia,  $I_p$ , have been calculated with Eq. 2 and Eq. 3, respectively, where  $P$  is the brake horsepower in kW and  $N$  is the rotational speed in rpm.

$$I_m = 118 \left(\frac{P}{N}\right)^{1,48} \quad (2)$$

$$I_p = 1,5 \cdot 10^7 \left(\frac{P}{N}\right)^{0,9556} \quad (3)$$

The generation side of the cooling system is comprised of four pumping stations named P07, P08, P09 and P10 for the Experimental Area (EA), the Storage (SR), the Booster (BO) and the Service Area (SA) consumption rings, respectively. In P08 and P10 two pumps operate simultaneously in parallel, meanwhile in P07 and P09 only one pump is working. Another pump, P11, is in

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charge of delivering the hot water to the chillers that cool it down again. Accordingly, the adequate moments of inertia for all the different pump sizes have been calculated and used in the model.

Finally, the non-return valves mounted at the discharge pipe lines have also been modelled. They are tilting disk check valves that only allow the passage of the fluid in a single direction shown in Fig. 1. If the flow tries to return, the valve will prevent this by shutting under its' own weigh. To model them, a characteristic operating time of 1 s and a minimum flow velocity of 0.6 m/s were assumed.

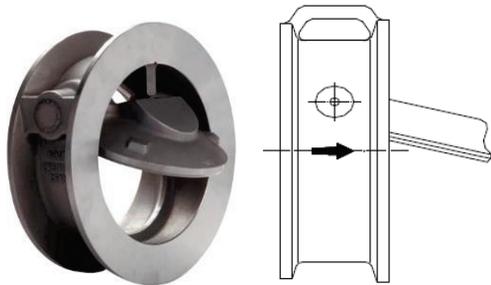


Figure 1: Tilting disc check valve mounted at pump discharge pipeline (arrow indicates permitted flow sense).

It must be noted that the cooling system presents several regulation mechanisms that are automatically applied to guarantee hydraulic and thermal stability. All of them have been considered in the current model. In terms of pump pressures, a series of proportional–integral–derivative controllers (PID's) try to maintain the maximum values below a fixed set-point by regulating the pump's rotational speed. In terms of water temperature, a series of mixing valves and chillers ensure a temperature of 23 °C at the consumption rings also by means of PID's controllers acting on three-way valves and heat exchangers.

### SIMULATED RESPONSE

#### Normal Pumping System Start-Up

The typical start-up procedure is carried out sequentially starting from pump P11 and finishing with the two pumps in P10 as indicated in Fig. 2. The total time takes around 100 s.

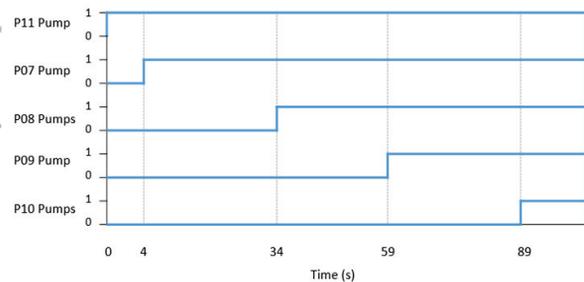


Figure 2: Selected instants of time for each pumping station start-up based on current practice by ALBA operators.

As it can be seen in Fig. 3, for each pump start-up the corresponding delivery pressure rises above the pressure

set-point during a certain period of time. The highest pressure increase is found in P09, reaching the 54% of its set point value, which is  $1.02 \cdot 10^7$  Pa. Meanwhile, the lowest overpressure is produced by pump P11. The reason for that is the observed delay time between PID controller and pump.

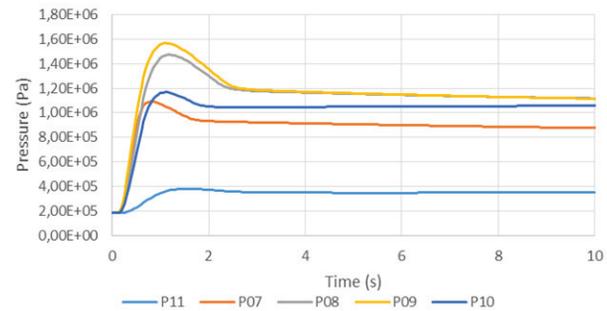


Figure 3: Pump delivery pressures during the initial seconds of sequential start-up.

In order to reduce such overpressures that might damage the system components, an adequate modification of the PID controller's characteristic values has been preliminary investigated with the model. Some results are shown in Fig. 4 for P09. It has been found that adjusting the proportional constant,  $K_p$ , around a value of 1, the pressure increase can be reduced to 32% of the set point. Regarding the integral and derivative constants, the simulations did not show any significant improvement in overpressure levels.

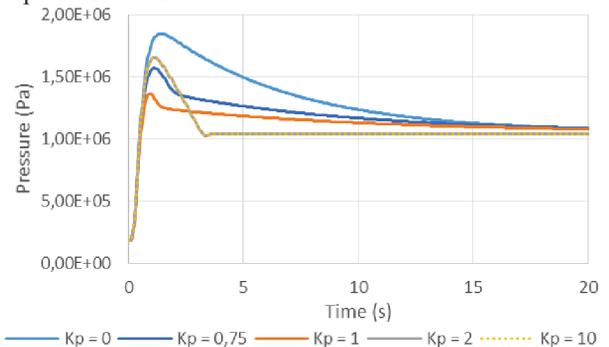


Figure 4: Effect of adjusting the  $K_p$  PID parameter on the pressure evolution during P09 start-up.

#### Pump Trip Without Thermal Load

The effects of the unexpected shut-down of only one of the pumps in P08 station have been simulated without thermal demand and the mixing valves at 50% opening ratio. As it can be observed in Fig. 5, the closure of the check valve happens around 0.8 s after the pump trip. As a result, there is a reverse flow during 0.3 s until the valve closes completely. At this point, a typical water hammer disturbance is generated through all the pumping lines. Fortunately the pressure wave is of low amplitude and short duration.

It must be noted that the expansion wave at the pump inlet seems to reach the vapour pressure, which could induce cavitation. After the water hammer, a low frequency fluctuation is generated in the system that disappears after 20 s.

Another point of interest is the significant effect observed in P11 rotational speed and the common return pipe during the transient.

At the end, the second pump is capable of compensating the lack of flow from the tipped pump and the original pressure set-point is achieved. However, this is not the case in P10 pumping line and at the end of the transient the system cannot maintain the nominal flow rate and delivery pressure.

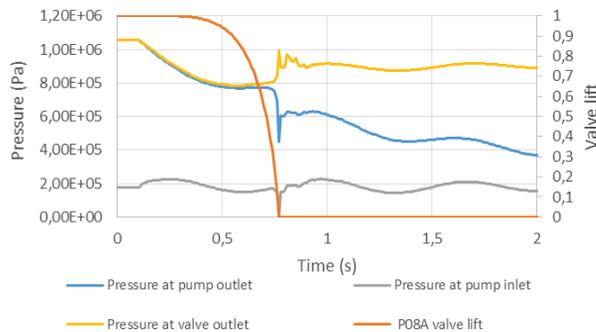


Figure 5: Evolutions in P08A pumping line of the pressures and the check valve position after a pump trip.

### Pump Trip with Thermal Load

In the case that the synchrotron light source is at full operation, the most critical case corresponds to a pump trip effecting P11 which is responsible of feeding the chillers to reduce the temperature of the cooling system water.

The simulated evolution of the mixed water temperature at each of the consumption rings after a P11 trip is shown in Fig. 6. It is significant to note that, due to the length of the pipe network, the consequences take around 600 s to reach the rings. At this end point, the system is obviously unable to cool the water anymore if P11 is not started again.

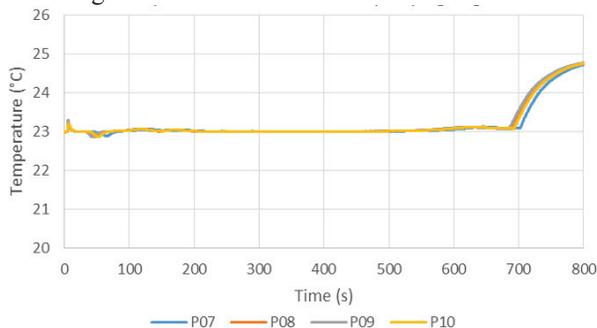


Figure 6: Temperature evolution at ring's inlets after a P11 pump trip occurring at time 0 s.

### Normal Pumping System Shut-Down

The system pressure response to a simultaneous stoppage of all the pumps once the thermal demand is already closed is presented in Fig. 7. Some oscillations are detected in all the pumping lines but they are rapidly attenuated. This procedure is not currently being applied in ALBA, which is actually a sequential shut-down of the pumping lines what makes the consequences even softer.

### Effect of Trapped Air

In Fig. 8 it can be seen that the presence of an accumulator of air at the exit of the ring when the flow in this ring is stopped due to a pump shut-down damps the amplitude of the pressure oscillations and increases the period.

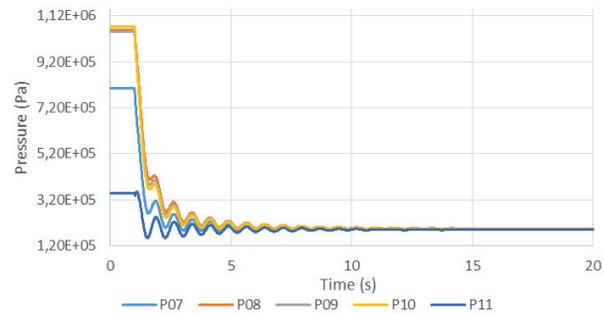


Figure 7: Pressure evolution at pump outputs during a simultaneous pump shut-down.

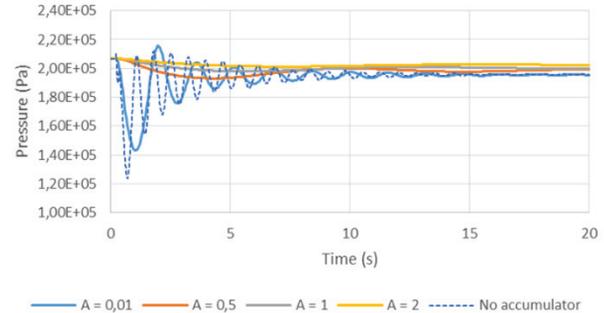


Figure 8: Effect on pressure oscillation due to P08 pumping station shut-down for different air volumes.

## CONCLUSION

The normal start-up manoeuvre of the pumping system has been simulated and the fluid dynamic response has been analysed. The results indicate the generation of significant pressure rises. To mitigate them, changes in the PID controller parameters have been proposed, improving the transient behaviour reducing such peaks.

The simulation and analysis of sudden pumps' shut-downs due to system unexpected failures has served to identify the consequences on the system behaviour. The calculations have been carried out without and with simultaneous thermal regulation. The results serve to identify the most dangerous situations and the time spans of the transient events. This information can be used to predict undesired behaviours.

Finally, the effect of air in the pipes has been analysed during the pump shut-down and it has been confirmed that the transient pressure fluctuations predicted in the system are modified.

## REFERENCES

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