CONTROL OPTIMIZATION OF A LHC 18 KW CRYOPLANT WARM COMPRESSION STATION USING DYNAMIC SIMULATIONS

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ABSTRACT

This paper addresses the control optimization of a 4.5 K refrigerator used in the cryogenic system of the Large Hadron Collider (LHC) at CERN. First, the compressor station with the cold-box have been modeled and simulated under PROCOS (Process and Control Simulator), a simulation environment developed at CERN. Next, an appropriate parameter identification has been performed on the simulator to obtain a simplified model of the system in order to design an Internal Model Control (IMC) enhancing the regulation of the high pressure. Finally, a floating high pressure control is proposed using a cascade control to reduce operational costs.

KEYWORDS: dynamic simulation, helium refrigerator, IMC, floating pressure control.

INTRODUCTION

During the last years, a PROcess and COntrol Simulator (PROCOS) has been developed at the European Organization for Nuclear Research (CERN) to perform dynamic simulations of large-scale cryogenic systems [1-2]. This simulator can be used for various tasks as, for example, operator training, virtual commissioning of control programs or optimization. This paper presents an approach for the optimization of the high pressure control in a 4.5 K refrigerator used for the particle accelerator LHC (Large Hadron Collider). It is worth to mention that dynamic simulators have been already used to optimize high pressure control [3] but in a different way.

First, the modeling of the warm compression station and of the cold-box is described. Then, the identification of the process and the development of an Internal Model Control

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(IMC) are detailed and some results are presented. Finally, in a third part, a floating high pressure control is presented and simulation results are discussed to conclude.

MODELING OF A 4.5 K REFRIGERATOR FOR THE LHC

Eight 4.5 K helium refrigerators are used to cool-down the LHC superconducting magnets. Each refrigerator is composed of a warm compression station and an 18 kW @ 4.5 K cold-box. A complete LHC refrigerator has been modeled under the modeling and simulation software EcosimPro©, which is able to describe industrial process behaviors by using an object oriented approach where each equipment (compressor, valve, etc.) is represented by a set of Differential and Algebraic Equations (DAE). The corresponding model makes use of the library for cryogenic equipments that has been previously developed at CERN in the framework of CERN cryogenic system simulations [1-2].

Component models

Helium properties are computed from linear interpolations performed in 2-D tables that have been done offline with HEPAK© [4]. Warm Screw Compressor models consider ideal isotherm compressions. Compressors have a volumetric flow that only depends on slide valve positions. For a given volumetric flow $\dot{F}$, the mass-flow is simply calculated as $\dot{m} = \rho_{in} \cdot \dot{F}$ where ρ_{in} is the input density of helium.

Valve models perform an isenthalpic expansion and mass-flows are computed using a classical CV formulation. Note that model takes into account sonic and subsonic flows.

Turbine mass-flows are calculated considering an ideal isentropic flow through a nozzle and an isentropic efficiency is then computed dynamically.

The models embed also the different volumes involved in the system (i.e. piping and helium tanks). Each volume is considered isochor, isotherm and isobar but singular pressure drops can be included. The thermodynamic states of volumes are computed from a mass and an energy balance taking into account the convection heat transfer between the gas and the enclosure.

Heat exchanger models are based on a spatial discretization of each stream where heat transfer coefficients are computed from a Logarithmic Mean Temperature Difference method (LMTD) and friction equations are included to compute pressure drops.

Warm compression station model

Warm compressor stations for the LHC are composed of two stages of compressors, creating a low, a medium and a high pressure, see FIGURE 1. The low stage compresses helium from 0.1 MPa to 0.4 MPa by using three compressors in parallel providing 300 g/s each. The high stage compresses helium from 0.4 MPa to 1.8 MPa with two compressors in parallel providing 820 g/s each. A compressor station has a total electrical power of 4.5 MW, therefore 36 MW are needed for the eight LHC compression stations, which constitute the main cost of cryoplants.

The high pressure (*HP*) is controlled by two antagonist valves: the discharge valve (*CV180*) that discharges the high pressure into the buffer and a charge valve (*CV189*) that charges the low pressure from the buffer. The low pressure (*LP*) is controlled by two by-pass valves between the HP and the LP (a large valve *CV175* and a small valve *CV176*). The same technique is used to control the medium pressure (*MP*) with the large valve *CV177* and the small valve *CV178*.

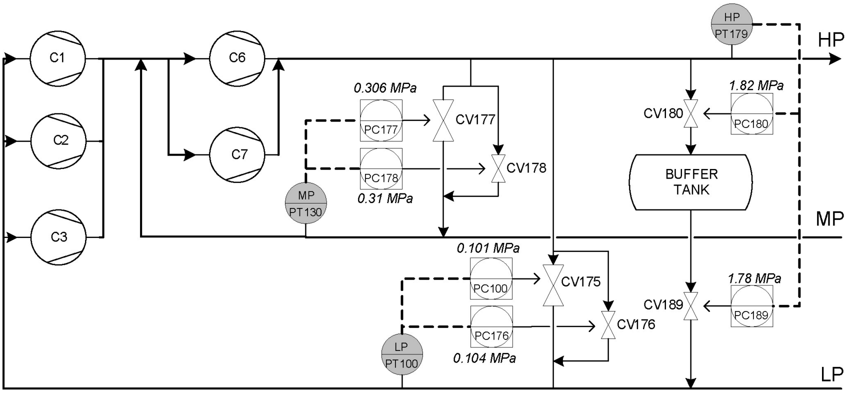

FIGURE 1. General scheme of the warm compressor station model for the LHC with controllers.

As valves are not identical, a classical split-range control is not really adapted because the dynamics on each valve have to be different. Hence, there are two independent PI controllers (Proportional Integral controllers) with 2 different set-points to maintain pressures between 2 values and avoiding valve oscillations (some example of set-points is given in the FIGURE 1).

The compressor station model is composed of volumetric compressors, valves and different helium volumes. The model contains *560 algebraic equations* and *74 differential algebraic equations*.

Cold-box model

The model is composed of 10 heat exchangers, 10 turbines, different control valves and a phase separator embedding an electrical heater and a heat exchanger inside, see FIGURE 2. The helium distribution in the LHC and the thermal shields are modeled by large helium volumes where constant thermal loads are applied as boundary conditions. The model contains *4000 algebraic equations* and *346 differential algebraic equations*.

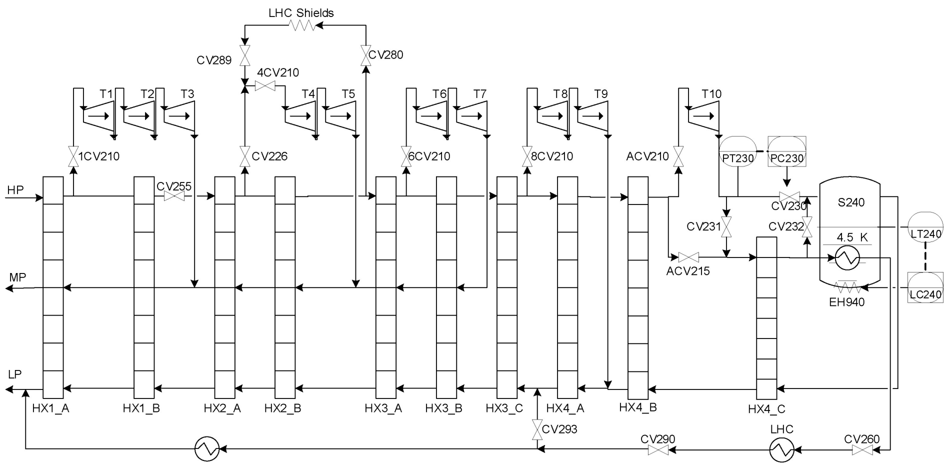

FIGURE 2. Model of the 18 kW @ 4.5 K cold-box for the LHC.

OPTIMIZATION OF THE HIGH PRESSURE CONTROL

Some minor control problems have been observed on the high pressure control as oscillations and a slow recovery time in case of "strong"-impacting events as, for example, a turbine stop. To enhance this control, several methodologies have been taken into account. Only the control of the high pressure will be treated as the low and the medium pressure control work satisfactory with classical PI controllers.

Parameter identification from simulations

The first step to design a new control law is deriving an appropriate model of the process. The use of the first principles model leads to non-linear complex equations, useless to develop control laws. The best way in finding a simple model is to perform parameter identification. In our case, standard identification procedures can destabilize systems and provoke failures on cryoplants. By using this observation, the idea considered here is to perform the identification on the complex theoretical model of the plant to obtain in a second time a simple linear transfer function around the operating point.

In agreement with process identification methodologies [5], two uncorrelated Pseudo Random Binary Sequences (PRBS) are applied to the 2 inputs (charge and discharge valves) to excite the process on an adequate bandwidth while other controllers are working normally in closed-loop, see FIGURE 3. Moreover, the sum of the turbine and by-pass valve mass-flows (*CV175*, *CV176*, *CV177*, and *CV178*) has been taken as measurable disturbance. The MATLAB® identification toolbox '*ident'* [6] has been used to perform continuous parameter identification in the Laplace formalism. The following first-order model has been obtained by identification:

$$
\begin{aligned}
y(s) &= P_1(s)\cdot u_1(s) + P_2(s)\cdot u_2(s) + P_d(s)\cdot d(s) \\
&= \frac{0.0219}{296\,s\ +\ 1}u_1(s) + \frac{0.0268}{292\,s\ +\ 1}u_2(\mathrm{s})\ +\ 0.014\frac{481s\ +\ 1}{799\,s\ +\ 1}e^{-8s}\ d(s)
\end{aligned}
\tag{1}
$$

where "*s*" denotes the Laplace variable; P_1, P_2 and P_d are transfer functions; y, u_1, u_2 and d are respectively Laplace transforms of the high pressure, the discharge valve, the charge valve and the sum of turbine and bypass mass-flows (disturbance).

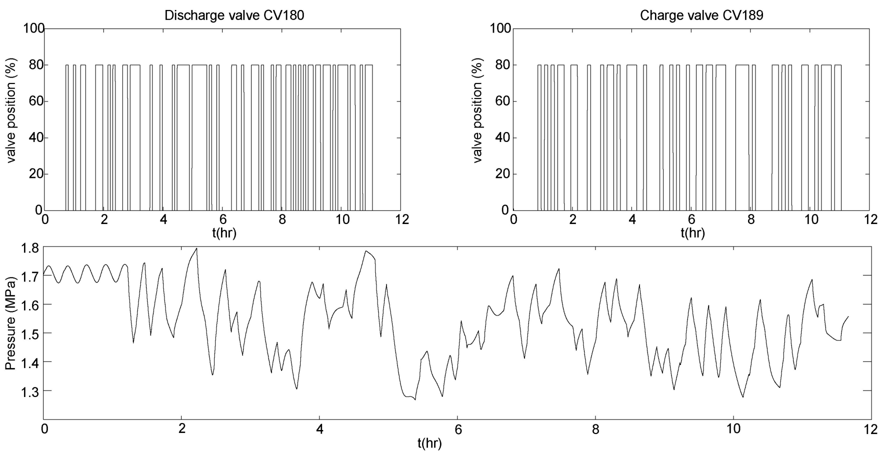

FIGURE 3. Valve positions and high pressure during parameter identification (simulation).

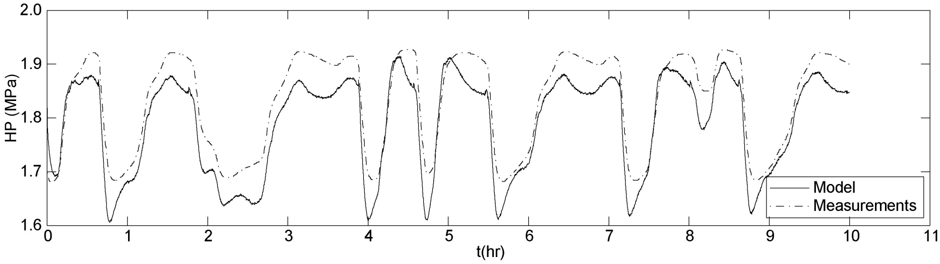

FIGURE 4. Validation of the high pressure model with experimental measurements.

The model has been first validated by statistical tests to ensure the absence of identification bias (input signals are close to white noises and not correlated together; residue is close to a white noise and not correlated to the input signals). Then, the model has been visually validated by some open- and closed-loop simulations and also by measurements made on the real plant, see, for instance, FIGURE 4.

Internal Model Control

After having obtained the model, the conventional PI controller has been replaced by a model-based controller in order to ameliorate the control of the high pressure and reject disturbances. An Internal Model Control (IMC) has been chosen for its robustness and its easy implementation. The controller embeds the model of the process defined in equation (1) and it is then designed according to the user's requirements [7]. In the case of a first-order model, the controller is defined as:

$$Q = \frac{\tilde{P}^{-1}}{\lambda \cdot s + 1}, \tag{2}$$

where $\tilde{P}$ is the process model; and λ is a tuning parameter which corresponds to the desired closed-loop time constant. For this class of systems, it is important to integrate an anti-windup to take into account valve saturations (valves are opened between 0% and 100%). The Internal Model Control with an anti-windup proposed in [8] has been used to design the controller, see FIGURE 5. The anti-windup splits the controller Q of equation (2) into two sub-controllers $Q1$ and $Q2$:

$$\begin{cases} Q_1 = F_A \cdot \tilde{P} \cdot Q \\ Q_2 = F_A \cdot \tilde{P} - 1 \end{cases}, \tag{3}$$

$$F_A = \frac{T}{K \cdot \lambda}(\lambda \cdot s + 1), \tag{4}$$

where F_A is the anti-windup filter designed to obtain a causal and stable system in closed-loop, K is the process gain; and T is the process time constant.

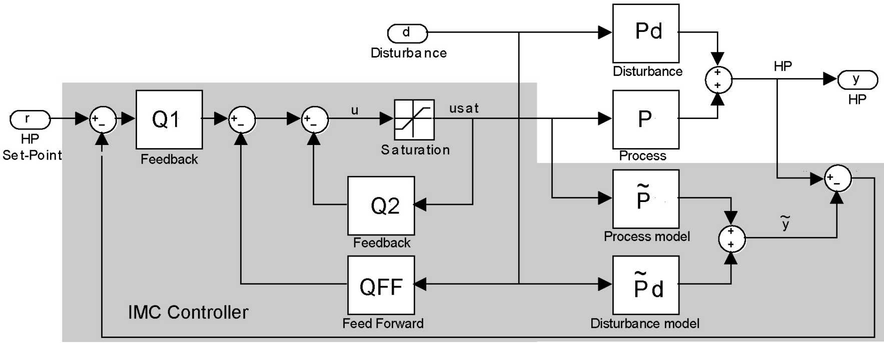

FIGURE 5. Internal Model Controller including an anti-windup system and a feed-forward action.

Moreover, a feed-forward action has been added into the controller to compensate the measurable disturbance (i.e. the sum of turbine and by-pass mass-flows). For an Internal Model Control, the feed-forward controller is defined as:

$$Q_{FF} = F_A \frac{\tilde{P}_d}{\varepsilon \cdot s + 1}, \tag{5}$$

where $\tilde{P}_d$ is the disturbance model; and ε is a filter time, necessary in our case to obtain a causal system as $\tilde{P}_d$ has a zero in the numerator, see equation (1).

Simulations have been performed during a turbine stop to compare PI and IMC controllers, see, for example, FIGURE 6. The IMC controller allows the removal of steady-state oscillations and disturbances are well rejected with a faster recovery time of the system: the control is improved and actuator moving is satisfactory.

FLOATING HIGH PRESSURE CONTROL

The value of the high pressure influences the refrigeration power available in the cold-box. At a constant high pressure, the thermal load is adjusted by an electrical heater (*EH*) that controls the liquid helium level in the phase separator S240, see FIGURE 2. The power

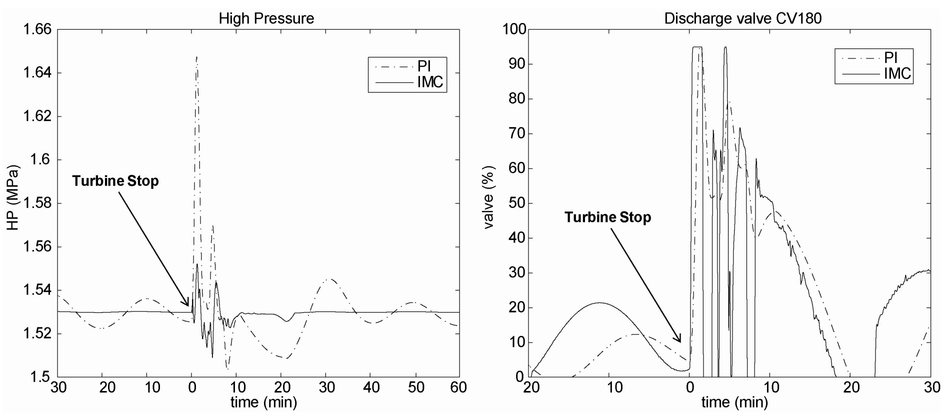

FIGURE 6. Comparison of the pressure control between the PI and the IMC controller during a turbine stop.

delivered by the heater represents the refrigeration power available in the case of some fast load disturbance and it must be sufficient. Nevertheless, heating power is generally too high because systems are over-dimensioned and compression power can be then decreased if high pressure is lower; this can lead to a large saving of energy and of operational costs.

A floating pressure approach has been already tested successfully at CERN in 1994 with the LEP 12 kW refrigerators [9] and other techniques appear elsewhere to perform cost saving [10]. For these reasons, a new PI controller (called *HC240*) has been designed to control the heating power in the phase separator, manipulating the high pressure set-point to make a cascade control, see FIGURE 7. Hence, the high pressure is not controlled anymore at a constant value but at a floating value depending on the thermal load and on the cold-box state. Note that, in this case, the medium pressure is also floating and the low pressure remains controlled at 0.1 MPa. If the load decreases, the HP and the MP go down, and some compressors can be stopped, decreasing the electrical costs.

The PI controller *HC240* has been set-up on the real plant (with the former PI control on the HP) and a lot of oscillations appeared on the heater and on the high pressure because of interactions between the high pressure, the heater and the level control in the phase separator. First, observed oscillations have been reproduced in simulation and then, the former HP controller has been replaced by the IMC controller developed in the previous section: oscillations were attenuated, but with the overall system still unstable.

To understand instabilities and tune efficiently all the controllers, identification has been performed as previously to obtain a simple model of the phase separator level (*L*) as a function of the high pressure (*HP*) and of the electrical heater (*EH*). The following transfer functions have been obtained:

$$L = G_1 \cdot HP + G_2 \cdot EH = \frac{0.0226}{s} HP - \frac{0.00244}{s} e^{-40s} EH \tag{6}$$

According to this model, controllers have been tuned and different parameters have been tested in simulations on the following sequence: once, the steady-state reached for a high pressure of *1.75 MPa*, the floating pressure control is activated at $t=0$ with a heater set-point of *1 kW*. Then, at $t=9h$ a step disturbance of 1.5 *kW* is applied on the phase separator. More than 30 simulations have been performed with various types of controllers and different parameters to compare them; the FIGURE 8 shows the two best simulations:

1. the best PI parameters derived using only PI controllers : the system is stabilized but it is slow and there are still fast oscillations on actuators,
2. the best PI parameters derived using the IMC for the HP control : the system is more stable and more reactive, moreover all fast dynamics on actuators are removed, allowing a smoother control.

It has been established that the IMC control of the high pressure allows a faster and a more stable floating pressure control and that the parameters of the level controller are very important to ensure the overall stability of the system.

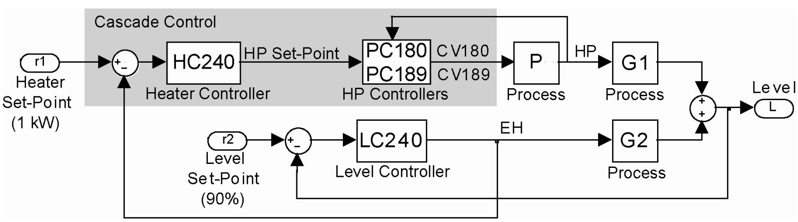

FIGURE 7. Cascade control architecture for floating high pressure control.

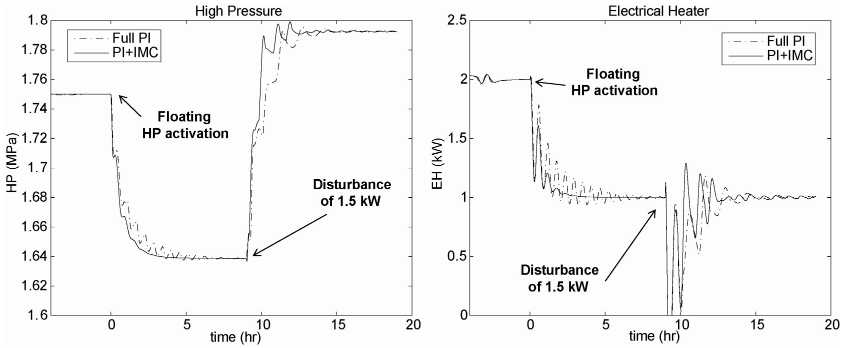

FIGURE 8. Comparisons of the floating high pressure between the full PI control and the PI/IMC control.

CONCLUSION

Control improvements have been developed and tested in simulation on the high pressure control in a LHC 18 kW @ 4.5 K refrigerator. First, the PI high pressure controller has been replaced by an Internal Model Controller (IMC) which shows a better stability on pressures and a better disturbance rejection using a feed-forward action.

A floating pressure control has been also designed by using a cascade control, and the different PI controllers have been tuned from dynamic simulations which show that the use of the IMC controller for the HP allows a faster and a smoother control.

All these results have been obtained in simulation using a physical model of the process which has been first validated with the real plant and theses control improvements will be tested in the future on the real LHC cryoplants for a definitive validation.

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