# ADVANCED CONTROL APPLIED TO A GAS COMPRESSION STATION OF A PRODUCTION PLATFORM

Agustinho Plucenio*, Carolina Vettorazo†, Mario M. Campos‡, Marcelo Lima‡

*FAPEU-R. Delfino Conti, s/n - 88040-370, Trindade, Florianópolis, SC, Brasil

†Dpto. de Automação e Sistemas
Universidade Federal de Santa Catarina,88040-900, Florianópolis, SC, Brasil

‡Petrobras-CENPES, Ilha do Fundão, Rio de Janeiro, Brasil

Emails: plucenio@firenzeconsult.com, carolvettor@gmail.com, mariocampos@petrobras.com.br, marceloll@petrobras.com.br

**Abstract**— In many oil production platforms compressed gas is vital for its operation. It is used for sales through the exportation line, for gas lift operations, for well re-injection programs, for electrical power generation by gas turbines and for water heating, etc. A common setup consists of two centrifugal compressors working in parallel. Any failure requiring stopping one or both compressors is extremely disturbing to the operations and may cause high economical losses. This paper discuss the use of a new surge indicator and the application of Practical Nonlinear MPC to complement the regulatory control of a gas compression system.

**Keywords**— Dynamic system modeling, Nonlinear Model Predictive Control, Compressor control

## 1 INTRODUCTION

Application of control to centrifugal compressors are normally developed to deal with the fast dynamics of the machine. It is known that this poses a difficulty to data acquisition and control calculations. Sampling times as low as 100 microseconds have been reported, (Cortinovis et al., 2012). Advanced control systems based in MPC (Model Predictive Control) have been applied to the control of centrifugal compressors using high sampling rate approaches as in (Budinis and Thornhill, 2015), (Overvag, 2013), (Cortinovis et al., 2012). The objective of this paper is to present the application of an advanced control, with two PI controllers and a MPC, tuned as a higher level control of a compressor station with two compressors in parallel which was modeled using a large set of algebraic and differential equations. The advanced controller uses a low sampling rate (like 5 seconds) but its prediction ability can help the overall system performance by anticipating actions to avoid surge and improve energetic efficiency. The main difficulty is to obtain simple nonlinear models based in difference equations to be used for prediction. Apart from that a new surge indicator is proposed which is easy to compute and takes into account not only the flow rates but also suction pressure and angular speed of the compressor. The idea is to have a surge indicator that can be used for prediction and is affected by the manipulated variables.

In (Smeulers et al., 1999) a brief report of a similar application is presented. This paper is organized as follows: In section 2 the compressor station model is briefly discussed, in 3 a new surge indicator is presented, in 5 the MPC algorithm is developed and results are presented and section 6 concludes the paper.

## 2 Compressor station

A compressor station shown schematically on figure 1 was modeled as part of a research project with the participation of Petrobras-CENPES and UFSC. The model is described in a separated paper submitted to this congress. Is is based in the work of Greitzer which has been slightly modified (Gravdahl and Egeland, 1998), (Gravdahl, 1998) and (Boinov et al., 2006). It does not deal with a single stage only but with the whole system, from the separator vessel to the gas distribution header.

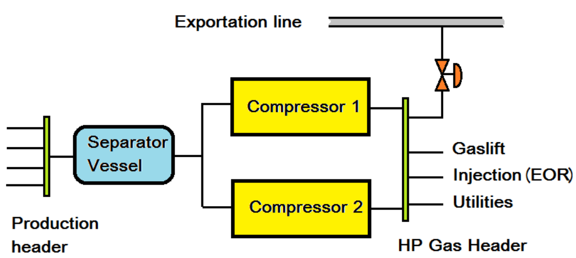


Figure 1: Gas compressor station block diagram

The simulation of the compressor station includes all the typical control loops like the pressure vessel control, anti-surge control and gas header pressure control manipulating the exportation flow-rate.

## 3 Surge indicators

It is a well known fact that centrifugal compressors work in a limited range of operating conditions. A compressor is designed to work with a nominal suction gas flow-rate, suction pressure and angular velocity. Too low suction flow-rate for a certain suction pressure and angular velocity may drive the compressor to surge. Surge is an unstable operating condition of a compressor resulting in flow reversal and pressures fluctuations (M.P.Boyce et al., 1983), (Hansen, 2008). Extremely high flow-rate, on the other hand, may take the compressor to a stonewall behavior characterized by gas sonic velocity flows. The metric frequently used to describe a centrifugal compressor operation is heading. Heading accounts for the energy per gas mass unit delivered by the compressor. Figure 2 shows a typical manufacturer plot of a compressor stage relating head ($H$), volumetric flow-rate ($Q$) and angular velocity ($w$).

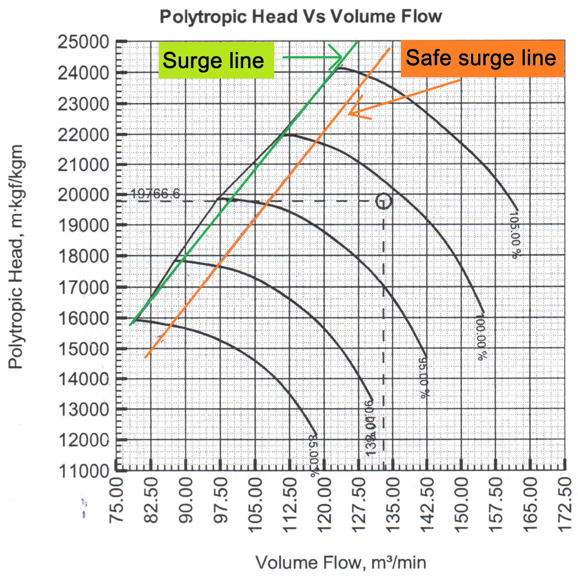


Figure 2: Head vs. flow-rate

On the graph it is shown two lines. The surge line and the safe surge line. The surge line are the union of all the points $(H, Q)$ where $\frac{dH}{dQ} \geq 0$. Operating points on this line or at its left will take the compressor to operate in surge. A common practice is to draw a line parallel to the surge line, the safe surge line with an offset to limit the range of operation of the compressor and thus to ensure a safe distance from surge. Good explanation on this technique can be found in (Mario Cesar M. Massa Campos and Perez, n.d.). The anti-surge control of regulatory systems consists in computing the distance to the safe surge line. For a given compressor shaft angular speed and a value of head, the value of the flow-rate on the safe surge line is computed, $Q_{ssl}^{sp}$. If the actual flow-rate $Q$ is higher than $Q_{ssl}^{sp}$ no action is done. If $Q$ is smaller than $Q_{ssl}^{sp}$ the recycling valve is manipulated to implement $Q_{ssl}^{sp} - Q$ as shown in figure 3.
Several efforts have been done to define a surge indicator as in (Batson, June, 1996), (Ahmed Hafaifaa and Mouloud, 2014) and (Mirsky et al., 2012). In (Mirsky et al., 2012)

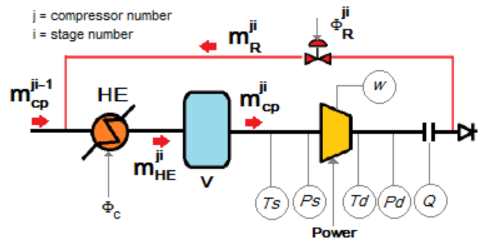


Figure 3: Typical compressor stage

a reduced variable is obtained starting with the conventional head and volumetric flow-rate considered as measured by an orifice plate. The surge indicator obtained is supposed to be immune to the effects of variations in the gas characteristics as compressibility, heat ratio or molecular weight. It is a very interesting indicator but since it depends on the pressure drop on the orifice plate, writing its prediction for MPC usage becomes too complicate. In (Smeulers et al., 1999) another surge indicator was defined. For a given head and angular speed of the compressor shaft,

$$I_{surge}^{ji} = \frac{m_{ji}^{sl}}{m_{ji}} \quad \text{, where} \tag{1}$$

$j$ is the compressor number, $i$ the compressor stage, $m_{ji}^{sl}$ is the mass flow-rate on the surge line and $m_{ji}$ is the actual mass flow-rate. To be away from surge $d_{surge}^{ji}$ should be smaller than 1. The variables $m_{ji}$ and $m_{ji}^{sl}$ can be computed with polynomials derived from manufacturer data as the one shown in figure 2. A typical centrifugal compressor polynomial shows a dependency of the head to compressor shaft angular velocity and volumetric flow-rate of order two. Both, $Q_{surge}^{ji}$ and $Q^{ji}$ can be obtained given a compressor shaft angular velocity and the ratio between discharge and suction pressure. Thus, the surge index can alternatively be written as

$$I_{surge}^{ji} = \frac{Q_{ji}^{sl}}{Q_{ji}} \tag{2}$$

It is expected that the ratio will take into account part of the uncertainty due to changes on the compressor behavior when operating with gas with different properties.

## 4 Prediction model

Prediction models should be simple and fast to compute. They must relate the controlled variables to the manipulated variables of the control system. The dynamic of the variables along the prediction horizon will be affected mainly by the compressors shaft inertia and the volume of the pressure vessels. The models are built using Euler approximations of the differential equations of interest. The main controlled variables will be the surge index $I_{surge}^{ji}$ while the manipulated variables

ISSN 2525-8311

will be the set-point of the pressure on the separator vessel, $P_E^{sp}$, the set-point of the gas header pressure, $P_H^{sp}$, the recycling mass flow-rates converted to valve openings, $\Phi^{ji}$.

$$
\begin{aligned}
&m^E_{in}(k) = m_E(k) \qquad (3)\\
&m^E_{out}(k) = m^{11}(k) - m^{11}_R(k) + m^{21}(k) - m^{21}_R(k) + m^{flare}_{(k)}\\
&P^E_{(k)} = P^E_{(k-1)} + T_s K_V^{PE}\left(m^E_{in}(k-1) - m^E_{out}(k-1)\right)\\
&e^{PE}(k) = -P^{sp}_E(k) + P^E(k)\\
&u^1_P(k) = u^1_P(k-1) + \gamma_{11} e^{PE}(k) + \gamma_{12} e^{PE}(k-1) + Bias^1_{(k)}\\
&u^2_P(k) = u^2_P(k-1) + \gamma_{21} e^{PE}(k) + \gamma_{22} e^{PE}(k-1) + Bias^2_{(k)}
\end{aligned}
$$

In equation 4 $m^E$, $m^{11}$, $m^{21}$, $m^{11}_R$ and $m^{21}_R$ are respectively, mass flow-rate on input of separator vessel, compressor 1 stage 1, compressor 2 stage 1, compressor 1 stage 1 recycling, compressor 2 stage 1 recycling. $T_s$ is the sampling time and $K_V^{PE} = ZRT_E/(MWV_{sep.})$. $u^1_P \in (0, 1.1)$ and $u^2_P \in (0, 1.1)$ are the power control to the compressors drive. The variables $Bias^1$ and $Bias^2$ are auxiliary inputs to implement a load sharing strategy. They will be modeled to the same heuristic as implemented in the regulatory control system which corresponds to driving the compressors to achieve the same distance to surge. The distances to surge can be written as

$$
\begin{aligned}
&d^{11}_{sur} = m^{11}(1 - I^{11}_{surge}) \qquad (4)\\
&d^{12}_{sur} = m^{12}(1 - I^{12}_{surge})\\
&d^{13}_{sur} = m^{13}(1 - I^{13}_{surge})\\
&d^1_{sur} = \frac{1}{3}\sum_{i=1}^{3} d^{1i}_{sur}\\
&d^2_{sur} = \frac{1}{3}\sum_{i=1}^{3} d^{2i}_{sur}\\
&d^{mean}_{sur} = \frac{1}{2}\left(d^1_{sur} + d^2_{sur}\right)\\
&e^1_{sur} = d^{mean}_{sur} - d^1_{sur}\\
&e^2_{sur} = d^{mean}_{sur} - d^2_{sur}\\
&Bias^1(k) = Bias^1(k-1) + \gamma^{11}_{su} e^1_{sur}(k) + \gamma^{12}_{su} e^1_{sur}(k-1)\\
&Bias^2(k) = Bias^2(k-1) + \gamma^{21}_{su} e^2_{sur}(k) + \gamma^{22}_{su} e^1_{sur}(k-1)
\end{aligned}
$$

The suction pressures of the intermediate stages

$$
\begin{aligned}
P^{ji}_s(k) = P^{ji}_s(k-1) + T_s K_V^{ji}\big(m^{ji-1}(k-1)\\
- m^{ji-1}_R(k-1) + m^{ji}_R(k-1) - m^{ji}(k-1)\big). \qquad (5)
\end{aligned}
$$

The pressure on the gas header

$$
\begin{aligned}
&P_H(k) = P_H(k-1) + T_s K_V^{PH}(m^{13}_{(k-1)} + m^{23}_{(k-1)} - \qquad (6)\\
&m^{13}_R(k-1) - m^{23}_R(k-1) - m^{exp}_{(k-1)} - \sum_{s=1}^{np} m^s_{(k-1)}
\end{aligned}
$$

The gas header pressure is controlled by manipulating the exportation flow-rate.

$$
\begin{aligned}
&e^{PH}_{(k)} = P^{sp}_H(k) - P_H(k) \qquad (7)\\
&m^{exp}_{(k)} = m^{exp}_{(k-1)} + \gamma_{H1} e^{PH}_{(k)} + \gamma_{H2} e^{PH}_{(k-1)}
\end{aligned}
$$

The compressors shaft angular velocities are

$$
\begin{aligned}
w^j(k) = w^j(k-1) + \frac{T_s K^{jia}_{Power} u^{ji}_P(k-1)}{w^j(k-1)} -\\
T_s K^{jb}_{Power}\left(\sum_{i=1}^{3} m^{ji}(k-1)\right) w^j(k-1). \qquad (8)
\end{aligned}
$$

The surge index $I^{ji}_{surge}$ are defined in equation ??. The compressor flow-rate $m^{ji}$ could be kept constant during prediction equal to the last measured value. But they are certainly affected by the dynamics of $w^j$ and $P^{ji}$. One possibility is to use the compressor polynomial derived from the manufacturer data to each compressor stage to derive a prediction.

$$
\begin{aligned}
&y_j = \frac{w_1}{w^j_N} \qquad (9)\\
&x_{ji} = \frac{Q_{ji}}{y^j}\\
&r_{ji} = \frac{P^{ji+1}}{P^{ji}},\ P^{j4} = P_H,\ P^{j1} = P_E\\
&r_{ji} = a^{ji} + b^{ji} x_{ji} + c^{ji} y_j + d^{ji} x^2_{ji} + e^{ji} x_{ji} y_j + f^{ji} y^2_j.
\end{aligned}
$$

For a given pair $(w^j, r^{ji})$ the stage flow-rate $Q^{ji}$ and its surge volumetric flow-rate $Q^{ji}_{sur}$ can be computed using the compressor polynomials.

$$
\begin{aligned}
&Q^{ji}_{sur} = k^{ji}_1 w_j + k^{ji}_2 w^2_j\\
&k^{ji}_1 = \frac{-b^{ji}}{2d^{ji} w^j_N}\\
&k^{ji}_2 = \frac{e^{ji}}{2d^{ji}(w^j_N)^2}\\
&Q^{ji} = Q^{ji}_{sur} \pm \sqrt{(Q^{ji}_{sur})^2 - C^{ji}}\\
&C^{ji} = (f^{ji}/d^{ji}) y^4_j + (c^{ji}/d^{ji}) y^3_j +\\
&((a^{ji}/d^{ji}) - (r^{ji}/d^{ji})) y^2_j \qquad (10)
\end{aligned}
$$

## 5 Application of advanced control

The objectives of the advanced control layer are:

- avoid surge,
- minimize power consumption,
- minimize flaring.

If the power compression system operator is asked about his best expectation his answer might be: To operate with minimum power consumption avoiding surge. The following manipulated variables are selected:i) Separator pressure set-point ($P^{sp}_s$), Gas Header pressure set-point ($P^{sp}_H$) and recycling valves opening ($\phi^{ji}$). For the sake of minimizing power consumption the smaller the header pressure the better. On the other hand there is no sense on manipulating the header pressure to avoid surge. The sensibility of the surge index to change of header pressure is small and rapid changes in the header pressure would propagate disturbances. For that reason an heuristic was adopted to update the header pressure set-point. The heuristic is based on the maximum pressure ($P^i_{valve}$) measured on the downstream side of all control valves connected to the header and the minimum pressure considering the compressor data sheet, ($P^{min}_H$).

$$
P^{sp-min}_H = max(P^{min}_H, (1+\epsilon) max(P^i_{valve})). \qquad (11)
$$

The value of $\epsilon$ is chosen to provide enough play to the valve to act without saturating. With this definitions, the header pressure set-point is updated using a filter to smooth the changes,

$$P_H^{sp}(k) = \alpha P_H^{sp}(k-1) + (1-\alpha) P_H^{sp-min} \quad (12)$$

The pressure of the separator vessel is basically the suction pressure of the first stage of both compressors. In case of flow-rates close to the compressor nominal value, the separator pressure should stay at its nominal value.

During a decrease of the input flow-rate, decreasing the separator pressure affects both, the surge index and the power consumption.

Smaller pressure at the compressor suction decreases the gas density and consequently increases the volumetric flow-rate for the same mass flow-rate. The consequence is that less recycling is needed to obtain a surge index smaller than one. So, decreasing the separator pressure during a decrease in the input flow-rate is beneficial to the surge control and to minimize power consumption. In order to implement a quick response to changes in the input flow-rate and to help on the surge control, the following heuristic is used.

$$\begin{aligned}
&e(k) = m^{nom} - m^{in}(k) \\
&P_s^{sp}(k) = P_s^{sp}(k-1) + \gamma_1 e(k) + \gamma_2 e(k-1) + \\
&\beta\left(P_s^{nom} - P_s^{sp}(k-1)\right) \text{ with} \\
&\gamma_1 = K_c\left(1 + \frac{T_s}{T_i}\right) \\
&\gamma_2 = -K_c \\
&P_s^{min} \le P_s^{sp}(k) \le P_s^{nom}. \qquad (13)
\end{aligned}$$

In equation 13

- $m^{nom}$ is the sum of the nominal mass flow-rate of each compressor. It can also be obtained with a recursive filter over the measured input flow-rate in order to account for slow changes related to the designed values.
- $m^{in}(k)$ is the actual pressure vessel input flow-rate,
- $\beta$ is the gain which restore the pressure vessel set-point to its nominal value,
- $K_c$ and $T_i$ are respectively the gain and the integration time of the controller,
- $P_s^{min}$ is the minimum value for the separator vessel pressure.
- $P_s^{nom}$ is the nominal value of the separator vessel pressure. This is the pressure for normal operations.

### 5.1 *Simulated results with the heuristic control proposed*

In this subsection the simulated results with the application of the heuristic control is presented. For comparison, the simulation using fixed set-points to the separator and gas header pressure is presented first.

Figure 4 shows the results obtained with the simulation of the compressor system model. The simulation starts with the flow-rate entering the system ($m_{in}$) equal to the compressors nominal value. At time equal to 2 minutes the input flow-rate starts to decrease, reaching 50% of its original value. This causes the regulatory control system to act trying to keep the pressures set-point of the pressure vessel ($P_s$) and gas header ($P_H$) and to avoid surge by opening the recycling valves to implement recycling flow-rates ($m_R^{ji}$) as shown in the figure. On figure 4 $u_{power}^1$ and $u_{power}^2$ are respectively the power control of compressor 1 and 2.$RPM1$ and $RPM2$ are the compressors shaft angular velocity measured in turns per minute. $P_{exp}$ is the back-pressure of the exportation line and $\Phi_{exp}$, $\Phi_{l1}$ and $\Phi_{l2}$ are respectively the valve openings of the exportation line and line for process 1 and 2.

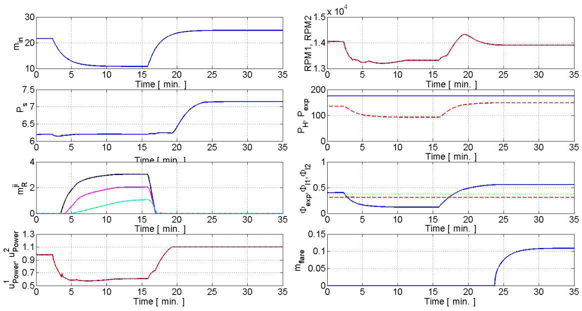

Figure 4: Gas compressor simulation with regulatory control

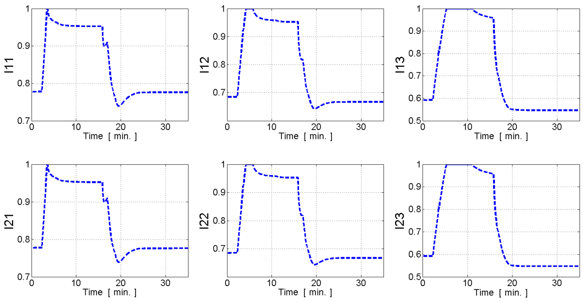


Figure 5: Gas compressor simulation with regulatory control-surge index

The regulatory control is using a surge index setpoint of 0.95.

After the decrease of the input flow-rate there is an increase to 15% above its nominal value. The compressor power reach their limit and the suction flow rate can not cope with the input flow-rate on the pressure vessel. This causes the pressure to rise and trigger the relief valve causing the flaring flow-rate shown.

Figure 5 shows the surge index $I^{ji}$ for each stage $i$ of compressor $j$. It is evident that all stages reach surge condition with the surge control tuning used in the regulatory control system.

Figures 6 and 7 show the simulated results using the advanced control with the heuristic con-

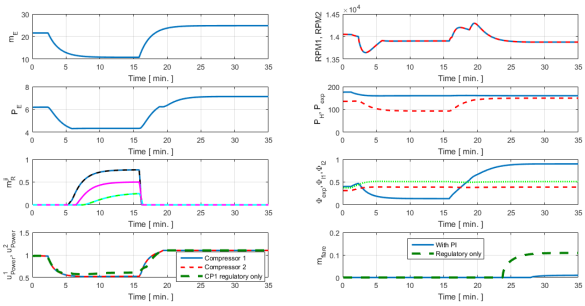


Figure 6: Gas compressor simulation with the two PI layer

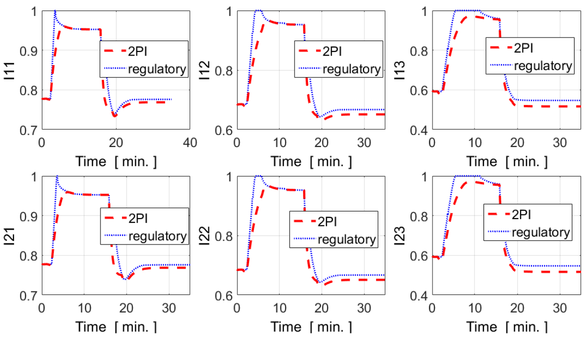


Figure 7: Gas compressor simulation with the two PI layer-surge index

trol. It is easy to note that the power applied to each compressor is lower and the surge index are kept below 1 as needed. The flaring is reduced due to the power saving applied to suck more gas. The power saving is due to the manipulation of the gas header pressure and the pressure on the separator vessel which induces less recycling flow-rate. Simulation using advanced optimization algorithms resulted in the same results.

### *5.2 Application of MPC*

The additional heuristic advanced control was able to improve the compressor performance avoiding surge, saving power, and reducing gas flaring in the tested scenario. However, the surge indexes are too close to 1 and more drastic situations could still lead to surge. To control the surge indexes and keep them below a satisfactory value a model predictive controller (MPC) was added. The MPC is a Dynamic Matrix Controller (DMC) and it is implemented as a layer above the regulatory control and the heuristic control presented in section 5.1. The DMC algorithm uses the step response of the system to compute the dynamic matrix ($G$) and the free response ($f$). Both ($G$) and ($f$) are necessary to predict the systems outputs. Since the step response is a linear representation of the system, a linearization of the non linear prediction model described at section 4 was necessary. The linearization technique applied to the original prediction model is the same used in the PNMPC algorithm, presented in (Plucenio, 2010).

In this case study the MPC calculates the recycling valves opening $\phi_{ji}^{MPC}$. that will keep surge indexes $I_{surge}^{ji}$, defined by equation 1, at a desired point. If $\phi_{ji}^{MPC}$ is greater than $\phi_{ji}^{regulatory}$ the MPC overrides the regulatory system. The DMC manipulated variables $\mathbf{u}$ are the recycling valves opening and its controlled variables $\mathbf{y}$ are the surge indexes.

The control increments $\triangle\mathbf{u}$ are obtained through the minimization of a cost function $J$ subject to constrains. The optimization problem is defined by:

$$\min_{\triangle \mathbf{u}} \quad J = [\bar{\mathbf{Y}} - \mathbf{W}]^T \mathbf{Q}_y [\bar{\mathbf{Y}} - \mathbf{W}] + [\mathbf{u} - \mathbf{u}_{ss}]^T \mathbf{Q}_t [\mathbf{u} - \mathbf{u}_{ss}] + \triangle \mathbf{u}^T \mathbf{Q}_u \triangle \mathbf{u} \tag{14a}$$

$$\text{s. to} \quad \triangle \mathbf{u}_{min} < \triangle\mathbf{u} < \triangle\mathbf{u}_{max} \tag{14b}$$

$$\mathbf{u}_{min} < \mathbf{u} < \mathbf{u}_{max} \tag{14c}$$

$$\bar{\mathbf{Y}}_{min} < \bar{\mathbf{Y}} < \bar{\mathbf{Y}}_{max} \tag{14d}$$

$$\mathbf{u} = \mathbf{u}(k-1) + \triangle\mathbf{u} \tag{14e}$$

The cost function $J$, equation 14a, has three objectives: 1) to keep $y$ at their references, 2) to take $u$ to their target, and 3) to penalize $\triangle\mathbf{u}$.

The diagonal matrices $Q_y$, $Q_t$, and $Q_u$ are formed by the weights for reference error, target distance, and control variation.

The targets for manipulated variables, $u_{ss}$, are their optimum steady state point defined by solving a LP problem, described in 15a, above the MPC.

$$\min_{\triangle \mathbf{u}_{ss}} \quad c^T P \tag{15a}$$

$$\text{s. to} \quad I^{surge} \leq I^{surge}_{max} \tag{15b}$$

$$\triangle \mathbf{u}_{min} < \triangle\mathbf{u}_{ss} < \triangle\mathbf{u}_{max} \tag{15c}$$

$$\mathbf{u}_{min} < \mathbf{u}_{ss} < \mathbf{u}_{max} \tag{15d}$$

$$I^{surge} = F_I + G_I \triangle \mathbf{u}_{ss} \tag{15e}$$

$$P = F_P + G_P \triangle \mathbf{u}_{ss} \tag{15f}$$

$$\mathbf{u}_{ss} = \mathbf{u}(k-1) + \triangle\mathbf{u}_{ss} \tag{15g}$$

The results of this implementation are shown in figures 8 and 9. The simulation is identical to the previous one explained in section 5.1.

The MPC was able to maintain the surge indexes below the reference by implementing a recycle flow rate higher than the one obtained with the heuristic control and regulatory layer but keeping the same pressure on the gas header $P_H$ and suction pressure $P_E$. Figure 9 compares the surge indexes with and without the MPC. The surge index setpoint for the MPC is 0.93 (black dotted line in the graphs of figure 9).

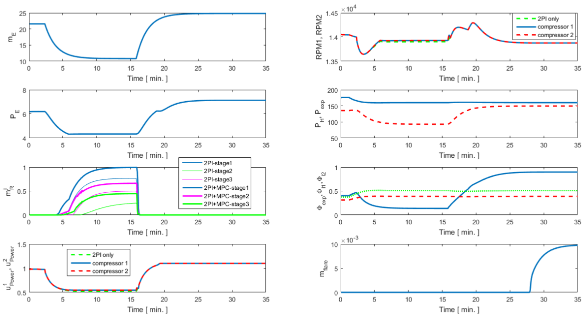


Figure 8: Gas compressor simulation with two PI layer and the MPC controller

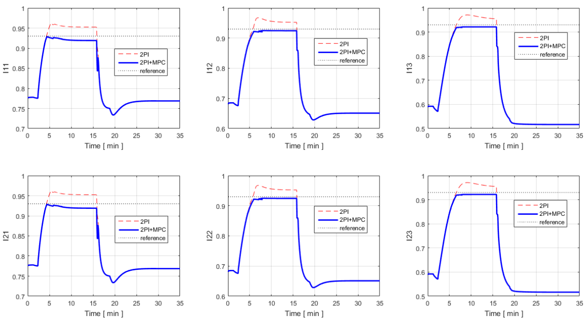


Figure 9: Gas compressor simulation with two PI layer and the MPC controller

Even though this solution consumes more power (figure 8) it guarantees that the operating point is safe and away from surge.

## 6 Conclusion

The proposed advanced control was able to reach its three objectives: avoid surge, minimize power consumption,and minimize flaring. The heuristic control alone was able to reduce the power consumption and the gas flaring and provide surge index below one. This solution might be of interest if the compressor manufacturer does not allow external inputs on its regulatory control. But there is a concern that model uncertainties, noise, and disturbances may still lead the system to surge even with the heuristic control. The MPC layer presented monitors the surge indicators, keeping them below the desired value. Because the MPC is able to predict the systems behavior it ensures that the system will operate at a point away from the surge line, even in the presence of uncertainties. It is one way to bypass the regulatory control whenever the recycling valves can be put in override control and the MPC surge index set-point are chosen smaller than those used by the regulatory control.

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