Distributed Model Predictive Control of Centrifugal Compressor Systems

K. Jones * **A. Cortinovis** * **M. Mercangoez** *

* *ABB Switzerland Ltd., Corporate Research, Baden-Dättwil, Switzerland (e-mail: katie.jones@reactive-robotics.com, andrea.cortinovis@ch.abb.com, mehmet.mercangoez@ch.abb.com).*

Abstract: The performance and computational cost of distributed MPC for the control of compressor networks is investigated in simulation. Both cooperative and non-cooperative approaches are considered and compared to the performance achieved with centralized control in the presence of a discharge-side disturbance. Two systems, each with two compressors, are studied: one is arranged in parallel configuration and the other in series. Due to the high degree of non-linearity of both systems, the models are re-linearized at each time step and a linear, delta MPC formulation is used. The controller is implemented using the quadratic program solver `qpOASES`, and a C++ implementation is used to evaluate the computational cost of each control approach. The non-cooperative controller exhibited a significantly reduced computation time relative to the centralized controller (25% and 40% lower for the parallel and serial configurations, respectively), while the cooperative controller did not significantly reduce the computation time. For the parallel configuration, both distributed and centralized controllers had identical control performance. For the serial configuration, only the cooperative controller achieved similar performance to the centralized approach; the non-cooperative controller demonstrated a 9% reduction in the minimum surge control distance reached in the downstream compressor.

Keywords: Distributed mpc, anti-surge control, nonlinear mpc, compressor modeling, variable speed drive

1. INTRODUCTION

Centrifugal gas compressors are employed in a wide range of industrial applications, particularly for gas transportation, extraction and processing. Compression is an inherently energy-intensive process, with well over 90% of operating costs spent on energy; small improvements in efficiency can therefore have a significant impact on the operating costs. At the same time, compressors are critical components in natural gas installations, meaning even short downtimes can also have a large economic impact. Compressor control systems must therefore maintain the compressor's operating point within its safe operating regime, avoiding instabilities that may damage its machinery and lead to such downtimes. The most relevant of these instabilities for control is surge, a phenomenon that occurs when the pressure ratio observed by the compressor is too high for the given mass flow, leading to oscillations in both the mass flow and output pressure, as well as vibrations and an increase in temperature. Surge can cause permanent damage to machinery in a relatively short time span and it is imperative for industrial compressor control systems to avoid it.

Compressor control thus consists of two sometimes competing goals: process control, wherein a given process variable (e.g. output pressure) is maintained at a given reference value, and anti-surge control (ASC), which keeps the compressor out of the surge regime. The performance of a compressor's anti-surge controller (ASC) is relevant for its efficiency: the most efficient operating points often lie on or near the surge line bounding the unstable surge regime. An increase in the performance of the ASC thus leads directly to an increase in the attainable efficiency for a given system as the compressor can be operated nearer to the surge line.

The current state-of-the-art in compressor control uses two independent controllers for process control and ASC. The process controller tracks a setpoint of a process variable (e.g. output pressure) by manipulating the speed of the gas turbine or electric driver driving the compressor, while the ASC keeps the compressor away from surge by manipulating the position of a recycle valve. This valve can be opened to allow flow from the outlet back to the inlet, effectively increasing the mass flow of the compressor and decreasing its pressure ratio, moving the system away from surge. Current industrial practice is to implement these controllers using simple PID controllers, with added loop decoupling and hand-tuned open-loop control responses near boundaries to address nonlinearities and coupling. Constraints are generally treated using clipping and anti-windup logic, which require further tuning.

Such a decoupled approach is necessary when considering gas turbine-powered compressors, as the dynamics of the

⋆ Sponsor and financial support acknowledgment goes here. Paper titles should be written in uppercase and lowercase letters, not all uppercase.

turbine – and thus the process control loop – are much slower than those that lead to compressor surge. The transition of compressor control from gas turbines to electric drivers with much faster dynamics has, however, opened the door for new, multivariable control algorithms that combine process control and ASC into a single controller. Such a multivariable controller can take advantage of the quick response of electric driver torque compared to the recycle valve opening to decrease the response time of the ASC to disturbances, thereby increasing its performance. In recent years, model predictive control (MPC) has been proposed as an alternative to frequency-domain approaches as it can explicitly consider both the coupling and physical constraints that make compressor control so challenging. *** Add references *** The advantages of such an approach were shown in Cortinovis et al. (2015) for a single compressor.

In industrial applications, compressors are often combined, either in parallel to increase the mass flow at a particular point, or in series to increase the overall pressure ratio achieved. The major disadvantage of MPC compared to conventional control approaches is the computational complexity inherent in the approach, and the resulting difficulty of executing the controllers at a sampling rate fast enough to handle the relatively fast dynamics observed in compressors. In particular, as the number of states and inputs increase – as for systems of multiple, coupled compressors – so does the required computation time of an MPC controller, making traditional MPC impractical in many situations.

In this article a distributed MPC (dMPC) control approach is proposed, which overcomes this limitation by dividing the optimization problem posed by a multi-compressor system into sub-problems to be solved at the individual compressor level. These sub-problems can be solved in parallel, reducing the required sample time as compared to a centralized MPC solution. Some information exchange between the compressors occurs and the sub-problems are solved interatively to allow the system to converge towards an optimal solution, though global optimality is not guaranteed. Two variants of dMPC are examined: a cooperative scheme where each compressor optimizes a single cost function using its own inputs, and a non-cooperative scheme where each compressor optimizes a cost function based only on its own inputs. The use of dMPC implies a loss of performance compared to a traditional, centralized MPC controller, as the solution obtained is no longer guaranteed to be globally optimal. The consequences of this loss of optimality on control performance – in particular ASC performance – are examined for both the cooperative and non-cooperative control approaches. Two compressor systems are considered as test cases to evaluate the performance of the dMPC approach in simulation: a parallel and a serial configuration, each with two compressors. In addition to the controller performance, the computational efficiency of each control approach is evaluated.

2. MODELLING & SIMULATION

The controllers presented in this work were evaluated in simulation, using a model based on a two-compressor system at an ABB Research Facility. The physical setup is described in Cortinovis et al. (2015) for a single compressor. Two different configurations were tested: a parallel configuration and a serial configuration, each consisting of two compressors which, for simplicity, were assumed to be identical.

The single compressor unit used in this work, depicted in Figure **??**, refers to a centrifugal compressor driven by a variable-speed electric driver, with a tank at both the inlet (suction) and outlet (discharge). Both the suction and discharge tanks are connected to atmospheric pressure by a valve. These suction and discharge valves represent the flow conditions upstream and downstream of the compressor, respectively, such that various flow conditions and disturbances can be simulated by simply changing the position of the valves. The artificial disturbances applied using these valves are designed to mimic situations frequently encountered in industrial applications (e.g. shutdown or startup of a downstream compressor). The compressor model used is based on the Gravdahl-Greitzer model described in Gravdahl and Egeland (1999). It is defined by five states: the suction (p_s) and discharge (p_d) pressures, the mass flow through the compressor (q_c), the rotational speed of the compressor (ω_c), and the mass flow rate through the recycle valve (q_r). The two inputs to the system are the torque applied to the electric driver (T_d) and the position of the recycle valve (u_r). The controlled outputs are the output pressure (p_out) and the surge distance (SD), a measure of how far the compressor is from surge. The interested reader may refer to Cortinovis et al. (2015) for a more detailed description of the model and its identification.

The parallel and serial compressor systems studied in this work are shown in Figure **??**. In the parallel case, the two compressors' outlet tanks both discharge into a single discharge tank and the pressure in this discharge tank (p_t) is the primary process variable. This pressure also serves as an additional state. For the serial configuration, the upstream compressor discharges directly into the inlet of the downstream compressor. The states, inputs and outputs are the same as those of the individual compressors.

3. MODEL PREDICTIVE CONTROL FORMULATION

Two variants of MPC are considered in this work: centralized and distributed control. Centralized control is considered too computationally expensive to implement in practice, necessitating the development of more efficient distributed controllers, but it is used as a benchmark to evaluate the performance of the distributed controllers. The system considered is non-linear, however, in order to take advantage of efficient quadratic program (QP) solvers, a linearized approach is taken in this work whereby the non-linear system is re-linearized at each time step and the resulting linear model used to generate the optimization. In order to control the relatively fast dynamics leading to compressor surge, the controllers operate at a sampling rate of $t_\mathrm{s} = 50\,\mathrm{ms}$. The model used for the MPC controller formulation is the same as the one used in Section **??** to simulate the compressor. No model mismatch was considered in order to compare the controllers' performance in an ideal scenario.

The nonlinear model is linearized at each sampling instant about the current states ($\hat{\boldsymbol{x}}_k$) and inputs ($\boldsymbol{u}_{k-1}$), and discretized using the 4th order Runge-Kutta method. At sampling instant k it is thus given by:

$$\begin{aligned}\Delta\hat{\boldsymbol{x}}_{k+i+1} &= A_k \Delta\hat{\boldsymbol{x}}_{k+i} + B_k \Delta\boldsymbol{u}_{k+i} + \boldsymbol{f}_{\mathrm{d},k}\left(\hat{\boldsymbol{x}}_k, \boldsymbol{u}_{k-1}\right) \\ \Delta\boldsymbol{y}_{k+i} &= C_k \Delta\hat{\boldsymbol{x}}_{k+i}, \qquad i \geq 0\end{aligned} \tag{1}$$

where $\boldsymbol{f}_{\mathrm{d},k}$ is the derivative of the system evaluated at the linearization point, $\hat{\boldsymbol{x}}_k$ is the state estimate at time instant k, $A_k = A_k\left(\hat{\boldsymbol{x}}_k, \boldsymbol{u}_{k-1}\right)$, $B_k = B_k\left(\hat{\boldsymbol{x}}_k, \boldsymbol{u}_{k-1}\right)$ and $C_k = C_k\left(\hat{\boldsymbol{x}}_k, \boldsymbol{u}_{k-1}\right)$ give the model linearized about the current operating point, and $\Delta\left(\cdot\right)$ refers to the difference in $\left(\cdot\right)$ relative to the linearization point.

The resulting discrete-time, linear system is then augmented to include both error states as integrators for offset-free control, and delayed input states for the recycle valve. One integrator for each output is added, as well as 40 delayed states per compressor (multiplied by a sampling rate of 50 ms to give a total delay of 2 s). With these additions, the augmented state of the system ($\Delta\hat{\boldsymbol{x}}_k^a$) is given by:

$$\Delta\hat{\boldsymbol{x}}_k^a = \begin{bmatrix}\Delta\hat{\boldsymbol{x}}_k \\ \boldsymbol{u}_k^{\mathrm{del}} \\ \boldsymbol{e}_k\end{bmatrix}, \qquad \boldsymbol{u}_k^{\mathrm{del}} = \left[u_{\mathrm{r},k-n_{\mathrm{del}}+1} \ \cdots \ u_{\mathrm{r},k-1} \ u_{\mathrm{r},k}\right]^{\intercal} \tag{2}$$

where $\hat{\boldsymbol{x}}_k$ contains the original states of the system, $\boldsymbol{u}_k^{\mathrm{del}}$ contains the delayed recycle valve inputs from the earliest to most recent, n_{del} is the total number of delayed states and $\boldsymbol{e}_k$ contains the integrator states.

The augmented system dynamics are then as follows:

$$\begin{aligned}\Delta\hat{\boldsymbol{x}}_{k+i+1}^a = &\underbrace{\begin{bmatrix}A_k & \left[B_k^{\mathrm{del}} \ 0\right] & 0 \\ 0 & \left[0 \ A^{\mathrm{del}}\right] & 0 \\ 0 & 0 & I_{n_{\mathrm{e}}\times n_{\mathrm{e}}}\end{bmatrix}}_{A_k^a}\left(\Delta\hat{\boldsymbol{x}}_{k+i}^a - \begin{bmatrix}0 \\ \begin{bmatrix}u_{\mathrm{r},k-1} \\ 0\end{bmatrix} \\ 0\end{bmatrix}\right) \\ &+ \underbrace{\begin{bmatrix}B_k^{\mathrm{nodel}} \\ B^{\mathrm{aug}} \\ 0\end{bmatrix}}_{B_k^a}\underbrace{\begin{bmatrix}u_{\mathrm{r},k+i} \\ \Delta T_{\mathrm{d},k+i}\end{bmatrix}}_{\Delta\boldsymbol{u}_{k+i}^{\mathrm{a}}} + \begin{bmatrix}\boldsymbol{f}_{\mathrm{d},k} \\ 0 \\ 0\end{bmatrix}\end{aligned} \tag{3}$$

$$\Delta\boldsymbol{y}_k = \underbrace{\left[C_k \ 0 \ I_{n_{\mathrm{e}}\times n_{\mathrm{e}}}\right]}_{C_k^a}\Delta\hat{\boldsymbol{x}}_k^a \tag{4}$$

where n_{e} gives the number of integrator states, and $A_k^a = A_k^a\left(\hat{\boldsymbol{x}}_k, \boldsymbol{u}_{k-1}\right)$, $B_k^a = B_k^a\left(\hat{\boldsymbol{x}}_k, \boldsymbol{u}_{k-1}\right)$ and $C_k^a = C_k^a\left(\hat{\boldsymbol{x}}_k, \boldsymbol{u}_{k-1}\right)$ give the augmented, linearized model of the system.

The first component of $\boldsymbol{u}_{k+i}^{\mathrm{del}}$ is multiplied by B_k^{del}, and must therefore be corrected by a factor $u_{\mathrm{r},k-1}$, the value about which the system was linearized. The other components of $\boldsymbol{u}_{k+i}^{\mathrm{del}}$ are not corrected because they are only multiplied by A^{del} which simply shifts them up a row.

4. SIMULATION RESULTS

5. CONCLUSION

REFERENCES

Cortinovis, A., Ferreau, H., Lewandowski, D., and Mercangöz, M. (2015). Experimental evaluation of MPC-based anti-surge and process control for electric driven centrifugal gas compressors. *Journal of Process Control*, 34, 13–25. doi:10.1016/j.jprocont.2015.07.001. URL `http://www.sciencedirect.com/science/article/pii/S095`

Gravdahl, J.T. and Egeland, O. (1999). Compressor Surge and Stall: An Introduction. In *Compressor Surge and Rotating Stall*, 1–62. Springer London.