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An integrated control technique for compressor operation

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***Abstract*—In the gas industry the control of the compression system has two primary objectives: the operation of the machine and also its protection against damage such as that following a surge cycle. The interaction between these two control actions can be strong, causing instability and oscillations. Moreover whenever the gas is recycled for surge protection, it is not delivered as final product and energy is wasted during its compression [1]. These observations provide the incentive for better integrated control.**

While many researchers have been working on the control of surge, not much has been done to improve the control of the performance or the integration between the two control objectives, even though this need has already been highlighted in the past [2].

This paper proposes a control scheme based on the characteristic map of the compressor, taking into account inlet disturbances. The proposed control solution is comparable to the state-of-the-art control solution under slow disturbance dynamics and allows a tighter pressure control during fast changing dynamics, reducing at the same time the proximity to the surge region. For this reason it represents a step forward in the direction of process control integration for compressor applications.

***Keywords*—process control; integrated control; compressor; surge**

I. INTRODUCTION

The control of a compression system has two primary objectives. The first objective is to operate the machine in order to deliver the desired amount of gas at a predefined pressure. The second objective is to protect the machine against damage such as that following a surge cycle. Surge is a dynamic instability of the flow characterized by fast axisymmetric oscillations of the gas [3]. The main consequence of surge is vibrations that can seriously damage components of the machine such as blades, bearings and casing.

The interaction between the two control actions can be strong and the controllers may end up pushing the compressor into opposite directions.

In the literature and also in industrial practice the integration between the performance controller and the antisurge controller is usually limited to decoupling and detuning. In order to prevent inverse response and oscillations, one of the two controllers is usually slowed down. However this will cause less responsive overall control [1]. Moreover transportation of gas has high operative cost strictly related to the compression step so it is important to consider the energy consumption during normal operation but also during surge prevention.

The survey in [4] showed many authors have been working on the control of surge, adopting different approaches such as passive or active control. However not much has been done to improve the control of the performance of the compressor or the integration between the two control objectives. References [5] and [6] deal with the manipulation of the drive torque in order to prevent surge and therefore they represent an initial effort in the direction of integration. However they were not focused on the improvement of the performance of the machine during normal operation.

This paper represents the initial effort in the direction of compressor control integration in order to obtain overall control stability. It proposes a control scheme based on the characteristic map of the compressor, taking into account inlet disturbances. The shaft speed is manipulated in order to meet the process demand. This decision has already been proven to be energy efficient [7]. The proposed control solution meets the process demand during inlet disturbances while reducing at the same time the proximity of the operating point to surge. This allows a tighter performance control without the need of gas recycle for mild inlet disturbances.

The structure of the paper is the following. In Section 2 the mathematical model of the compressor is presented, together with its implementation in MATLAB Simulink and its validation against industrial data. In Section 3 the paper gives a brief overview on the state-of-the-art control of compressors and presents the new control solution. In Section 4 the disturbances used to test and compare the standard solution and the new solution are listed and motivated and the simulation results are presented. Finally in Section 5 the conclusions of the work are drawn, together with some insight on future research directions.

II. COMPRESSOR MODEL

A. *Mathematical model*

A well-established compressor model [8] has been used together with industrial data coming from the reference case

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study. Both performance map and efficiency map have been built using invariant coordinates [9] and implemented into the model.

The compressor unit is composed of four main parts: the compressor itself, the duct, the plenum and the throttle valve (Fig. 1). The compressor increases the gas pressure from p_{01} to p_{02}. The duct connects the compressor with the plenum, where the mass accumulates in the volume V_p. Finally the throttle valve regulates the flow m_t that leaves the system. This throttle valve represents the resistance that the compressor requires to overcome the pressure drop along the downstream section of the plant.

The equations of the model are the mass balance for the plenum (1), the momentum balance for the duct (2), the moment of momentum balance for the rotating shaft (3), the flow through the throttle valve (4), the equation of the torque of the compressor (5) and the equation of the compressor characteristic (6):

$$\frac{dp}{dt} = \frac{a^2{}_{01}}{V_p}(m - m_t) \quad (1)$$

$$\frac{dm}{dt} = \frac{A_1}{L_c}(\Psi_c p_{01} - p) \quad (2)$$

$$\frac{d\omega}{dt} = \frac{1}{J}(\tau_d - \tau_c) \quad (3)$$

$$m_t = k_t\sqrt{p - p_{out}} \quad (4)$$

$$\tau_c = \mu r_2^2 \omega m \quad (5)$$

$$\Psi_c(\omega, m) = \frac{p_{02}}{p_{01}} \quad (6)$$

where a_{01} is the sonic velocity at ambient condition, A_1 is the duct throughflow area, L_c is the duct length, J is the total inertia of the system, τ_d is the driver torque, τ_c is the compressor load torque, k_t is the throttle parameter, μ is the slip factor and r_2 is the impeller radius (a list of definitions of the variables is at the end of the paper). In an open cycle system p_{out} is equal to p_{01} while in a closed system it is the downstream back pressure.

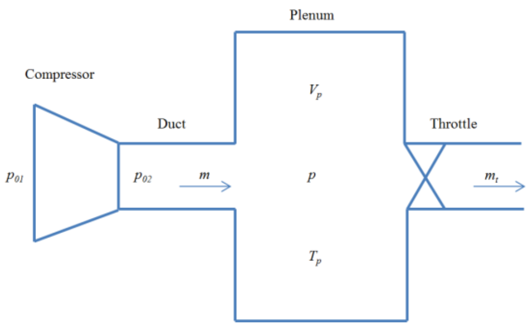

Fig. 1. Model of the compressor (modified from [5])

B. Compressor maps

The compressor characteristic Ψ_c and also the polytropic efficiency η_{poly} are non-linear functions of the mass flowrate m and also of the rotational velocity ω. Fig. 2 shows an example of compressor characteristic in order to illustrate the degree of non-linearity of this function. In this work non-dimensional maps have been used, in order to take into account the effect of inlet disturbances on the steady-state maps [10]. Therefore the new axes of the compressor maps are:

$$m_{nd} = \frac{m\sqrt{T_{01}}}{p_{01}} \quad (7)$$

$$N_{nd} = \frac{N}{\sqrt{T_{01}}} \quad (8)$$

where m_{nd} and N_{nd} are respectively the non-dimensional mass flow rate and the non-dimensional rotational shaft speed while T_{01} is the inlet temperature. Ψ_c and η_{poly} are already non-dimensional quantities.

C. Implementation in MATLAB Simulink

The model of the compressor has been implemented in MATLAB Simulink. The block diagram has two input variables: the inlet pressure p_{01} and the torque provided by the driver τ_d. The main output is the controlled variable i.e. the outlet pressure p.

The polytropic efficiency η_{poly} is used to estimate the outlet temperature of the gas T_{02} according to the following equation:

$$T_{02} = T_{01}\left[1 + \frac{1}{\eta_{poly}}\left(\Psi_c{}^{(k-1)/k} - 1\right)\right] \quad (9)$$

where k is the specific heat ratio evaluated at the temperature:

$$T_m = \frac{T_{01} + T_{02}}{2} \quad (10)$$

The mechanical power of the compressor has been estimated according to:

$$P_m = m(h_{02} - h_{01}) \quad (11)$$

where the enthalpies h_{02} and h_{01} are respectively evaluated at the temperatures T_{02} and T_{01} [11].

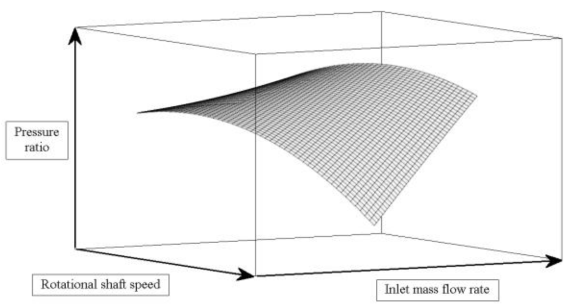

Fig. 2. Characteristic map of a compressor

D. Model validation

The model of the compressor was validated against data coming from an industrial case study. The maps and data sheets of the multistage machine were used to validate the model during steady state simulations while an industrial simulator was used to validate the dynamic behaviour during transients between steady states. The industrial simulator was provided by ESD Simulation Training and it is currently used in their training course on ‘Carbon Dioxide Compression’ [12]. It is a fully validated industry-standard model. The model presented in this paper is a simplified model for the purpose of process control and it includes the first stage of compression. The ESD simulator was run in open loop (i.e. with the speed controller in manual mode and constant output) and tested via inlet step disturbances and the agreement between this industrial model and the model presented in the paper was satisfactory.

III. CONTROLLER DESIGN

A. Overview on the control of industrial compressors

In the industrial practice the control of pressure and surge is usually pursued via two separate controllers: the performance (or capacity) controller and the antisurge controller. Although they usually act on different manipulated variables, the final control result may change substantially depending on the interaction between the two control actions. Usually their integration is limited to detuning and/or decoupling [1]. A more integrated solution allows a faster response of the system, overall control stability and finally also energy saving. An advanced control system can reduce the energy consumption [2] and this is one of the goals of this work.

In the classic approach to the control of the performance of a compressor, a cascade controller is used, as represented in Fig. 3. In the cascade structure, the pressure controller is the master controller while the speed controller is the slave controller. The controlled variable is the outlet pressure p and the manipulated variable is the rotational shaft speed N. As the cascade structure involves an actuator with its own local controller (in this case the speed controller), it compensates for disturbances that can affect N. As this variable cannot be directly manipulated, the torque of the driver τ_d has to change in order to keep the controlled variable at its set point.

B. Integrated approach to compressor control

This work represents an initial effort in the direction of control integration for industrial multistage compressors. The predictive controller presented in the paper is able to improve the overall control performance because it can (i) control the outlet pressure and (ii) reduce the risk of surge by reducing the proximity to it during inlet disturbances. Therefore this control solution goes into the direction of control integration.

The control approach proposed in this paper is based on the characteristic of the compressor. This characteristic is commonly used to define the operating envelope of the machine (especially in regards to surge location) and also for performance estimation but it is not usually adopted for control design. In this work it has been used for control purposes as well and this is one of the contributions of the paper. It has been implemented in the performance controller in order to estimate the correct speed depending on the inlet conditions of the process fluid.

C. Predictive controller

The predictive controller predicts the rotational shaft speed rather than calculating it via feedback loop only. In its cascade control structure, the master controller is composed of two blocks: the feedback control block and the predictive control block. These two blocks represent the predictive controller in Fig. 4. At any time t^* the predictive control block determines the rotational shaft speed N^* according to the instant value of inlet pressure p_{01}^* and flow rate m^* [13]. The inputs to this block are the inlet disturbances and the set point of the outlet pressure. Inside this controller the inverse characteristic of the compressor is modelled. Therefore the predictive controller can estimate the speed controller set point according to the ideal pressure ratio of the machine, defined as:

$$\Psi_{c,id} = \frac{p_{SP}}{p_{01}^*} = f(m^*, N^*) \qquad (12)$$

The predictive controller block differs from a model predictive controller as it does not optimize an objective function. The reason is that there is only one controlled variable (the outlet pressure) and therefore there is no trade-off between two or more control objectives.

The compressor characteristic is a non-linear function of mass flow rate and rotational shaft speed and it was fitted using a second degree polynomial function. The discrepancy between the model of the compressor characteristic and the model of the inverse characteristic causes a small deviation between the optimal N and N^*. This is due to the fitting process itself. In

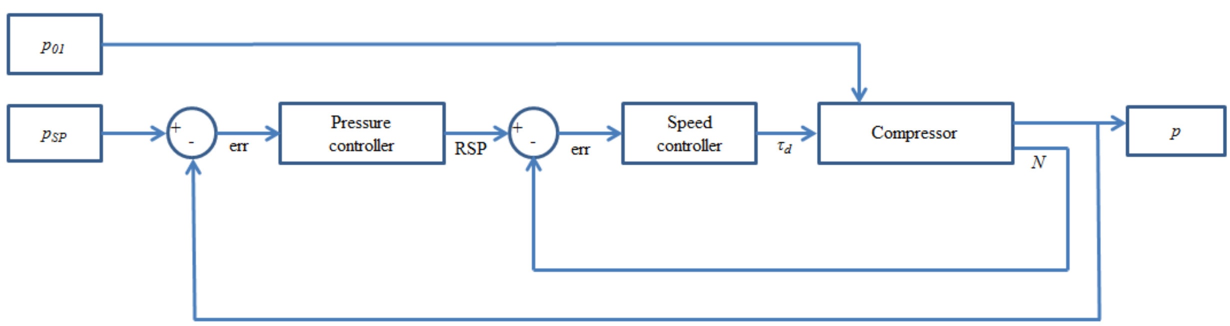

Fig. 3. Block diagram of the cascade controller

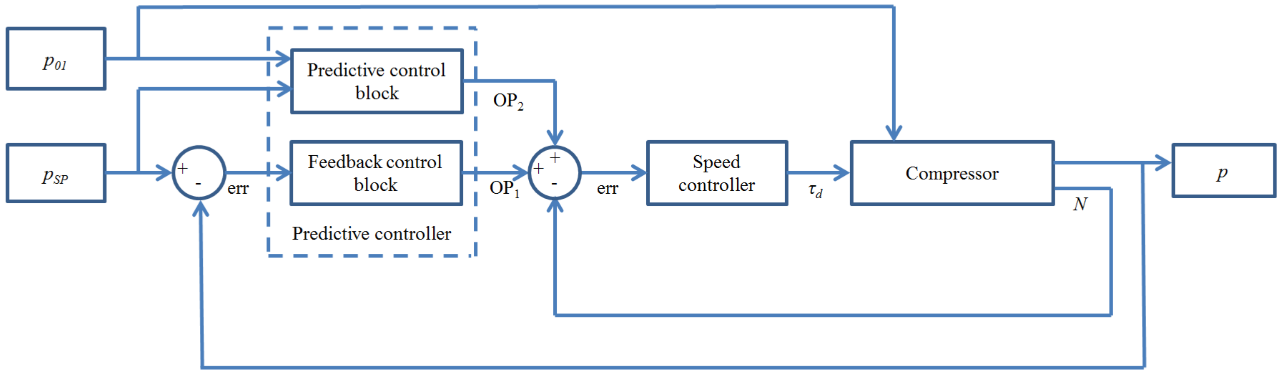

Fig. 4. Block diagram of the predictive controller

fact in the compressor model the pressure ratio Ψ_c is the dependent variable while in the controller model the rotational shaft speed N is the dependent variable and also the remote set point of the speed controller. Therefore the presence of the feedback control block can compensate this discrepancy and close the control loop. The remote set point of the speed controller is the sum of the outputs of the two controllers, OP_1 and OP_2.

IV. SIMULATIONS

A. *Case study*

The case study is based on a multistage centrifugal compressor composed of four stages mounted on a single shaft. The driver is an asynchronous electric motor. The process fluid is a mixture of carbon dioxide and water with small percentages of light hydrocarbons. This composition is typical of a flue gas coming from amine absorption of CO_2 from natural gas [14].

The target pressure ratio for the first stage of compression is 3.79. All the other process variables reported in the paper were scaled to be 1 at their design point due to non-disclosure agreement with the industrial partner of the project.

B. *Process disturbances*

Dynamic simulations were run in order to compare the time response of the two control systems (traditional cascade controller and predictive controller). The two different solutions were tested using a range of disturbance magnitudes and dynamics. The inlet pressure was changed and the outlet pressure time responses were recorded, analysed and compared.

The disturbances used for testing are considered realistic for a CO_2 post-combustion separation plant [15-17]. This separation solution using MEA solvent is the most used for chemical and natural gas industries [15] within CCS (carbon capture and storage) technology.

The disturbance scenarios tested were the following:

- Set point step changes (±5%) in the pressure controller;
- Inlet pressure step changes (±10%), pulse changes (±5% and ±10%) and stair changes (three steps of ±5%);
- Inlet pressure ramp changes (±10% during a period of 10 minutes/30 minutes/3 hours);
- Inlet pressure sinusoidal changes (amplitude 15% over a period of 24 hours, 6 hours, 1 hour, 30 minutes, 10 minute).

C. *Simulation results*

Simulation results are presented and commented for pulse changes and sinusoidal changes of the inlet pressure p_{01}.

In the pulse change disturbance represented in Fig. 5 the inlet pressure p_{01} changes by +5% at time t=300s and by -5% at time t=700s. The pulse dynamic can therefore be assimilated to two following step changes with same magnitude and opposite sign. Although the inlet disturbances are symmetric, it is possible to see that the pressure time response is not symmetric. This is due to the non-linearity of the compressor characteristic. The response of the cascade controller has a larger deviation and is not oscillatory, while the response of the predictive controller swings first in one direction then the other. However, for this type of fast disturbance the response of the two controllers is comparable in terms of settling time and peak deviation.

Two different sinusoidal disturbances are represented in Fig. 6. In the first case (left) the amplitude of the disturbance is 15% and the period is 30 minutes while in the second case (right) the amplitude is still 15% while the period is 10 minutes. In both cases the predictive controller shows a tighter

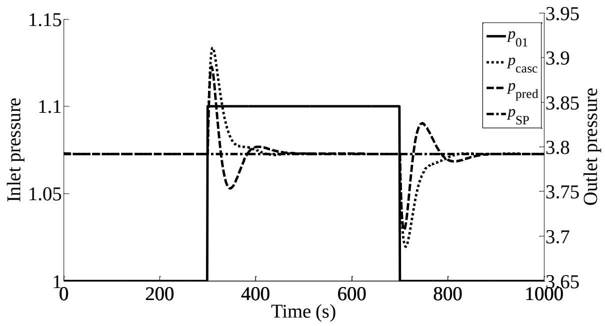

Fig. 5. Time response of the two control systems for positive pulse change of p_{01}

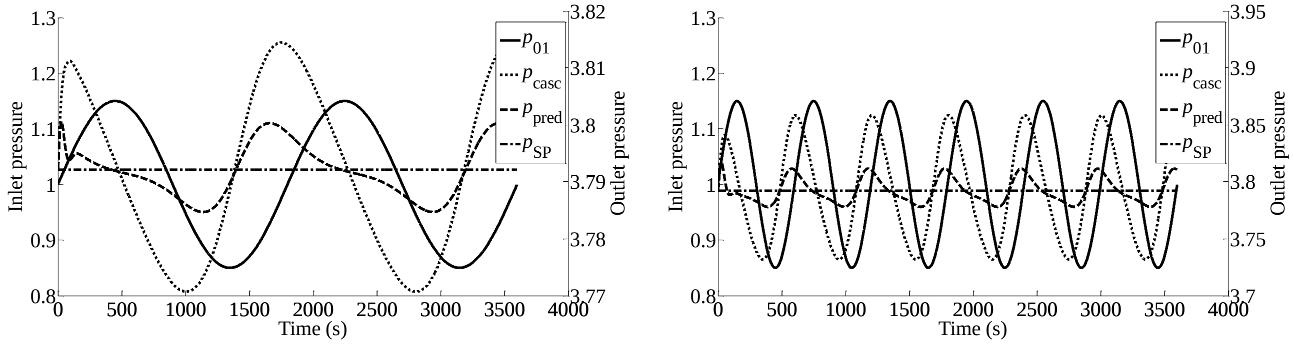

Fig. 6. Time response of the two control systems for slower (left) and faster (right) sinusoidal changes of p_{01}

control on p compared to the traditional cascade controller, and this is more evident in the second case. This is due to the fact that the predictive controller reacts faster than the traditional pressure controller to an inlet disturbance. Therefore it performs better in the second case where the period of the disturbance is smaller.

According to the simulations results, the response of the predictive control system varies depending mainly on the dynamic of the disturbance itself. For slow dynamics (ramp dynamics and sinusoidal dynamics with low frequency) the response of the two controllers is similar. In fact with a slow dynamic there is no need for a fast-acting controller. However when the inlet disturbance is characterized by a fast dynamic (sinusoidal change with high frequency) the predictive controller response shows smaller height of peaks of the outlet pressure. Therefore in these cases the amplitude of oscillation of the controlled variable is smaller. In fact the standard cascade controller is not fast enough to counteract the effect of the disturbance on the plant itself.

Fig. 7 shows the path of the operating point on the compressor map during a step disturbance (left) and a sinusoidal disturbance (right). The operating envelop is defined by four lines: the surge line, the choke line, the minimum speed line and the maximum speed line [7]. In both cases during the transients the predictive controller is able to keep the operating point further from the surge line. Because of the magnitude of the disturbances and the initial position of the operating point there is no surge, however it may happen under stronger disturbances. Future research will address this issue.

V. CONCLUSIONS

In this paper a control solution for the performance of industrial compressors was presented. The results show that the predictive controller is able to control the outlet pressure of the system while reducing the risk of compressor surge. This is valid for disturbances characterized by slow dynamics but especially when the dynamic of the disturbance is faster and the standard controller response is slow. The paper proposes a control scheme based on the characteristic map of the compressor, taking into account inlet disturbances. The proposed control solution is comparable to the state-of-the-art control solution under slow disturbance dynamics and allows a tighter pressure control during fast changing dynamics, reducing at the same time the proximity to the surge region.

The predictive controller is able to control a single stage compressor as well as multistage compressor in case of single shaft configuration, where there is only one manipulated variable. The control solution was tested on a specific case study however it can be extended to other applications as well.

The effort of this project is going in the direction of compressor control integration. The reason for that is to reduce as much as possible the recycle of gas for surge protection. In this way the energy consumption of the compression system is reduced as well. Future research work will include the

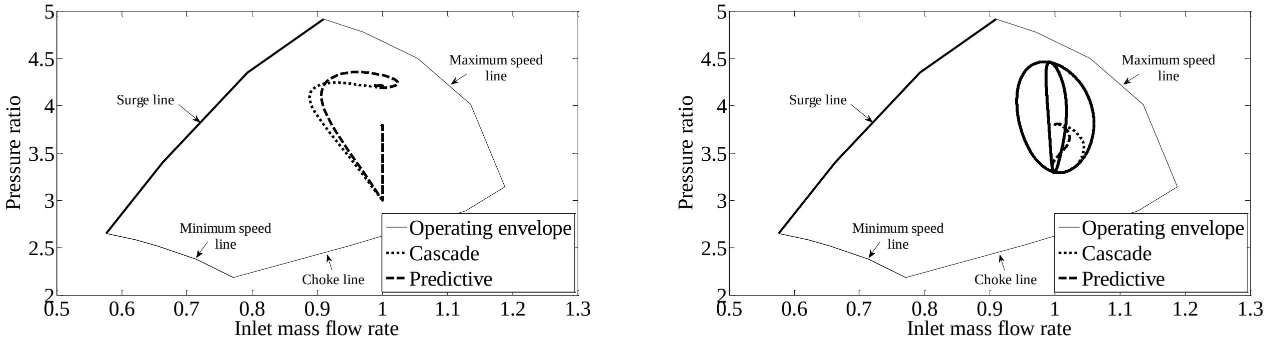

Fig. 7. Operating point path during step disturbance (left) and sinusoidal disturbance (right)

implementation of a proper antisurge control action, in order to protect the machine from more aggressive disturbances.

NOMENCLATURE

A_1	Duct throughflow area, m^2
a_{01}	Sonic velocity at ambient conditions, ms^{-1}
h_{01}	Inlet enthalpy, Jkg^{-1}
h_{02}	Outlet enthalpy, Jkg^{-1}
J	Total inertia, kgm^2
k	Specific heat ratio, adim.
k_t	Throttle parameter, $kg^{1/2}m^{1/2}$
L_c	Duct length, m
m	Compressor mass flow rate, kgs^{-1}
m_t	Throttle mass flow rate, kgs^{-1}
N	Rotational shaft speed, rpm
p	Plenum pressure, bar
p_{01}	Compressor inlet pressure, bar
p_{02}	Compressor outlet pressure, bar
P_m	Compressor mechanical power, kW
p_{out}	System outlet pressure, bar
p_{SP}	Pressure set point, bar
r_2	Impeller radius, m
T_{01}	Inlet temperature, K
T_{02}	Outlet temperature, K
T_p	Plenum temperature, K
t	Time, s
V_p	Plenum volume, m^3
η_{poly}	Compressor polytropic efficiency, adim.
μ	Slip factor, adim.
τ_c	Compressor load torque, Nm
τ_d	Driver torque, Nm
Ψ_c	Compressor characteristic, adim.
$\Psi_{c,id}$	Ideal compressor characteristic, adim.
ω	Shaft rotational velocity, $rads^{-1}$

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