Control Engineering Practice 17 (2009) 815–833

Contents lists available at ScienceDirect

Control Engineering Practice

journal homepage: www.elsevier.com/locate/conengprac

Variable structure surge control for constant speed centrifugal compressors

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ARTICLE INFO

Article history:
Received 8 May 2008
Accepted 9 February 2009
Available online 20 March 2009

Keywords:
Centrifugal compressors
Surge protection
Fuzzy control
Sliding mode control
Gain scheduling
Nonlinear systems

ABSTRACT

This paper presents the application of active control of surge in constant speed centrifugal compressors based on the Moore–Greitzer (MG) model. Different controllers are developed for a compression system equipped with a close-coupled valve (CCV) and a throttle control valve (TCV). The combination of the two valves helps suppress surge and assists in overcoming the drawbacks of each valve when it is used individually. The presented controllers are evaluated based on many performance indices. Accordingly, a case-based variable structure controller is developed that succeeds to stabilize the system for various operating conditions and to extend the stable range well beyond the surge line. A flow extension of 76% and a stable throttle level of 0.1, with substantially decreased pressure settling times, are reported. The developed controller performs significantly better compared to other nonlinear controllers, such as the backstepping controller.

1. Introduction

Surge is an unstable operating mode of compression systems which occurs at low mass flows where the pressure delivered by the compressor is less than the plenum pressure. This instability is global, one-dimensional, and nonlinear. It is characterized by a limit-cycle oscillation in mass flow and pressure rise. According to Cumpsty (1989), surge may be so violent that the mass flow is reversed and the compressed gas emerges out of the inlet; this is sometimes called deep surge. On the other hand, it may be very mild so that the operating point orbits around a mean value with no flow reversal, and the main evidence for the surge is an audible burble.

Deep surge limits the operating range at low mass flow rates and degrades the system performance and efficiency. If left unattended, a destructive flow reversal can occur, causing catastrophic damage to the system. Several measures have been introduced to suppress surge and to increase the stable operating range of the compression system. These measures can be categorized into techniques that are based on better compressor interior design or variable geometry, and techniques that attempt to suppress surge by control (Willems, 2000).

Over the past few decades, surge control techniques have been extensively explored. The main advantages of these techniques are their applicability to a wide range of machines and their considerable performance improvement compared to other techniques. Moreover, these techniques can be easily added to existing machines. Traditionally, the surge problem was tackled by using surge avoidance techniques (Gravdahl & Egeland, 1999). However, these techniques limit the operational range and reduce the efficiency of the compression system. Therefore, active surge control has been introduced as an alternative approach to suppressing surge rather than avoiding it.

Active surge control was first introduced in the literature by Epstein, Williams, and Greitzer (1989). It aims at overcoming the drawbacks of surge avoidance by stabilizing some part of the unstable region in the compressor map. Applicability of active surge control to experimental setups was also first demonstrated by Williams and Huang (1989). In the past two decades, active control has developed into a mature research field.

Past research indicates various methods for designing surge controllers. A proportional feedback controller is proposed in Epstein et al. (1989). Its main drawback is the limited operating region due to linearity. Given that the problem is nonlinear, the use of nonlinear techniques seems more promising, since the nonlinearities are dealt with directly. Typical examples are gain scheduling and Lyapunov-based controllers (Simon & Valavani, 1991), structural feedback (Gysling, Dugundji, Greitzer, & Epstein, 1991), adaptive control (Billoud, Gallard, Huu, & Candel, 1991), backstepping (BS; Krstic, Protz, Paduano, & Kokotovic, 1995), bifurcation (Liaw & Abed, 1996), H_∞ (Weigl & Paduano, 1997), sliding mode control (SMC; Sanadgol & Maslen, 2004), and fuzzy logic control (FLC; Laderman, Greatrix, & Liu, 2003; Shehata, Abdullah, & Areed, 2008).

Among the several actuators used for stabilizing compression systems, the most common are the throttle valves

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doi:10.1016/j.conengprac.2009.02.002

(Willems, Heemels, de Jager, & Stoorvogel, 2002) and the bleed valves (Fontaine, Liao, Paduano, & Kokotovic, 2004). Other examples include variable inlet guide vanes (Camp & Day, 1998), loudspeakers (Williams & Huang, 1989), tailored structures (Gysling et al., 1991), recirculation (Balchen & Mumme, 1988), movable plenum walls (Epstein et al., 1989), air injection (Behnken & Murray, 1997), synthetic jets (Breuer, Schmidt, & Epstein, 1998), tip clearance (Sanadgol & Maslen, 2005), and close-coupled valves (CCV; Gravdahl & Egeland, 1999). Several studies and experimental results have been performed on different types of sensors and actuators for surge control. It is concluded that among the most promising methods is to actuate the system with a CCV or an injector with the use of mass flow feedback (Simon, Valavani, Epstein, & Greitzer, 1993).

This paper addresses the use of a CCV and a throttle control valve (TCV) for active surge control. The objectives are to stabilize the compressor under various operating conditions and to extend the compressor's stable range well beyond the surge line. Different controllers are presented, from which a case-based variable structure controller is developed that optimizes pre-defined performance indices. The reminder of this paper is organized as follows. The mathematical model of the centrifugal compressor is presented in Section 2 along with the model validation. Section 3 discusses the system's response using different control techniques. The variable structure controller is presented in Section 4. Conclusions are summarized in Section 5.

2. Compression system

2.1. The model

To design a surge controller, a mathematical model capable of predicting the surge and describing the compressor's flow dynamics is required. Although basic dynamic models have been available since 1955 (e.g. Emmons, Pearson, & Grant, 1955), a distinctive improvement was made in 1976 by Greitzer (1976), when a nonlinear dynamic model was presented. Major drawbacks of the earlier models were being linearized and restricted to small perturbations from equilibrium point. Greitzer's model (G model) was originally derived for axial compressors; however, Hansen, Jørgensen, and Larsen (1981) showed that it is also applicable to centrifugal compressors. Although the G model is capable of simulating surge oscillations, rotating stall is considered as a pressure drop. In 1986, Moore and Greitzer (1986) proposed a model, the Moore–Greitzer (MG) model, capable of describing transients associated with both surge and rotating stall where the latter is included as a state.

The MG model that consists of a compressor, a plenum, a throttle valve, a CCV, a TCV, and connecting ducts is shown in Fig. 1. A CCV is connected to the compressor outlet. "Close-coupled" means that the distance between the compressor outlet and the valve is assumed to be so small that no significant mass storage can occur (Gravdahl & Egeland, 1999). This assumption allows for the definition of the equivalent compressor. The pressure rise over the equivalent compressor is the sum of the pressure rise over the compressor and the pressure drop over the CCV. By varying the latter, the compressor characteristic is modified, and the system dynamics are changed. Therefore, this pressure drop can be used as the control action (Gravdahl & Egeland, 1999).

Although the CCV has the advantages of the ability to modify the compressor dynamics and the ease of installation to the system, its response is sometimes not satisfactory. A better response can be achieved through the use of a TCV installed downstream of the plenum. Nevertheless, one of the major disadvantages of using a TCV is the modifications performed on

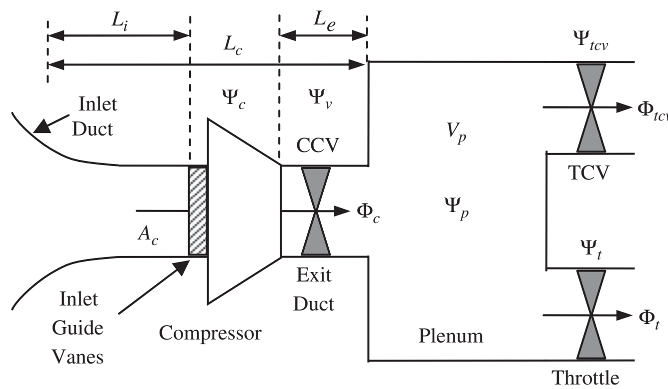

Fig. 1. The Moore–Greitzer model with a CCV and a TCV.

the controlled plenum to accommodate this valve. The concept behind the use of the TCV is to keep the valve closed, and to open it only when the plenum pressure exceeds the desired pressure.

Based on the work of Moore and Greitzer (1986), a mathematical model is derived in Gravdahl and Egeland (1999) and Willems et al. (2002) that describes the effects of both the CCV and the TCV on the compressor characteristic. The model is presented in Eqs. (1)–(3), and the description of all symbols/parameters is given in Table 1. For constant speed compressors, flow rates m, pressure differences ΔP, time t, and length L are normalized according to (4). This normalization, or non-dimensionalization, transforms the family of curves in the compressor map, one for each compressor speed, into a single characteristic. This characteristic is a nonlinear relationship between compressor pressure Ψ_c and compressor flow Φ_c. Different expressions for this characteristic have been used, but the most widely accepted one is the cubic characteristic of Moore and Greitzer (1986), which is defined in (5):

$$\frac{d\Phi_c(\xi)}{d\xi} = \frac{1}{l_c}\left(\underbrace{\Psi_c(\Phi_c) - \Psi_v(\Phi_c)}_{\Psi_e(\Phi_c)} - \Psi_p(\xi)\right), \tag{1}$$

$$\frac{d\Psi_p(\xi)}{d\xi} = \frac{1}{4B^2 l_c}(\Phi_c(\xi) - \Phi_t(\Psi_p) - \Phi_{tcv}(\Psi_p)), \tag{2}$$

where

$$B = \frac{U}{2a_s}\sqrt{\frac{V_p}{A_c L_c}} \quad \text{and} \quad l_c = l_i + \frac{1}{a} + l_e, \tag{3}$$

$$\Phi = \frac{m}{\rho A_c U}, \quad \Psi = \frac{\Delta P}{\rho U^2}, \quad \xi = \frac{U}{R}t, \quad l_i = \frac{L_i}{R}, \quad \text{and} \quad l_e = \frac{L_e}{R} \tag{4}$$

$$\Psi_c(\Phi_c) = \psi_{c0} + H\left[1 + \frac{3}{2}\left(\frac{\Phi_c}{W} - 1\right) - \frac{1}{2}\left(\frac{\Phi_c}{W} - 1\right)^3\right]. \tag{5}$$

The characteristics of the CCV, the throttle, and the TCV are given as in Gravdahl and Egeland (1999), respectively, by

$$\Psi_v(\Phi_c) = \frac{1}{k_v^2}\Phi_c^2 \quad \text{and} \quad k_v = c_v u_v > 0, \tag{6}$$

$$\Phi_t(\Psi_p) = k_t\sqrt{\Psi_p} \quad \text{and} \quad k_t = c_t u_t > 0, \tag{7}$$

$$\Phi_{tcv}(\Psi_p) = k_{tcv}\sqrt{\Psi_p} \quad \text{and} \quad k_{tcv} = c_{tcv} u_{tcv} > 0, \tag{8}$$

where c is a capacity measure of the fully opened valve, u is proportional to the valve opening and varies between 0 (fully closed, or FC) and 1 (fully open, or FO), and k is the valve gain.