Practical issues in model-based surge control for centrifugal compressors

Practical issues in model-based surge control for centrifugal compressors

Jan van Helvoirt[♦], Bram de Jager, Maarten Steinbuch
Technische Universiteit Eindhoven
Department of Mechanical Engineering
Control Systems Technology Group
P.O. Box 513, 5600 MB Eindhoven, The Netherlands

Jan Smeulers
TNO Science and Industry
Department Flow and Structural Dynamics
P.O. Box 155, 2600 AD Delft, The Netherlands

ABSTRACT

The performance and operating range of centrifugal compressors is limited by the occurrence of an aerodynamic instability called surge. This paper deals with the technological barriers for feedback stabilization of full-scale centrifugal compression systems. A dynamic model for a single stage compressor test rig is presented and a surge control strategy is discussed. Then the critical problems of model accuracy and sensor and actuator limitations are discussed. These problems are illustrated with simulation results and experimental data. We conclude by suggesting directions for further work in order to successfully implement active surge control in full-scale applications.

1 INTRODUCTION

The operating range of turbo compressors is limited towards low flows by the occurrence of surge and rotating stall. Although both aerodynamic instabilities are related, we will focus on compressor surge in this paper.

Surge is an unstable operating mode of a compression system, characterized by large oscillations in compressor flow and pressure rise. Surge reduces compressor performance and the resulting thermal and mechanical loads can cause structural damage. Hence, suppression of this unstable behaviour is of great value from a practical perspective.

[♦] E-mail address j.v.helvoirt@tue.nl. The corresponding author is also partially employed by TNO Science and Industry, Department of Flow and Structural Dynamics, P.O. Box 155, 2600 AD Delft, The Netherlands.

The dynamic behaviour of compression systems has been studied extensively over the past sixty years. In particular we mention the relevant work on analysis and modelling of compressor surge. Surveys in this field are given in (1-4).

One of the first nonlinear models of transient compressor behaviour was proposed in (5). This model exploits the analogy between surge oscillations and a Helmholtz resonator, an idea first suggested in the linearized analysis from (6). Although developed for axial compressors, the authors of (7) showed that this nonlinear model was also applicable to centrifugal compressors. To this day, it is the most widely used dynamic model in the field. Many modifications and extensions were suggested, for example in (8-10), but the basis principles of the so-called Greitzer model remained unaltered.

The idea to actively suppress compressor instabilities initiated from (11). The general goal is to achieve stabilization of the compression system by modifying its dynamics through the use of active feedback. This approach has been studied extensively over the last two decades; see for example (12-15).

Despite the large research efforts and the potential to impact industrial compressor operability, full-scale applications of active surge control have not been realized yet. Promising experimental results are obtained with laboratory-scale setups, see for example (16-18), but implementation of active surge control is still in an early stage of development. In our current work we aim to take a next step towards full-scale implementation by demonstrate the principle of active stabilization. For this purpose we use a dedicated test setup that is normally used in the development of full-scale compression systems.

In this paper we will address some of the critical technology barriers for full-scale application of surge control. We will start with a short introduction of the investigated compression system and give some remarks on the used simulation model and control strategy. Then we discuss how model accuracy and sensor and actuator limitations complicate the implementation of active surge control. We will use a mix of theory, simulation results and experimental data to illustrate how critical these problems are. We finish with conclusions and suggestions on how these challenging problems can be solved.

2 CENTRIFUGAL COMPRESSION SYSTEM

Our research aims to implement active surge control on a centrifugal compression system. In this section we introduce the experimental setup and we briefly discuss the developed model for the dynamics of the system. Finally, we give some comments on the chosen control strategy.

2.1 Experimental setup

Our experiments were conducted on a single stage centrifugal compressor test rig that is normally used to test industrial compressors for the oil and gas industry. The whole installation is schematized in Figure 1. The compressor is driven by a 1.7 MW electric motor that is connected to the shaft through a gearbox. The rotational speed of the compressor can be varied between 6,000 and 16,000 rpm. The compressor operates in a closed circuit that makes it possible to use different pure gases or gas mixtures.

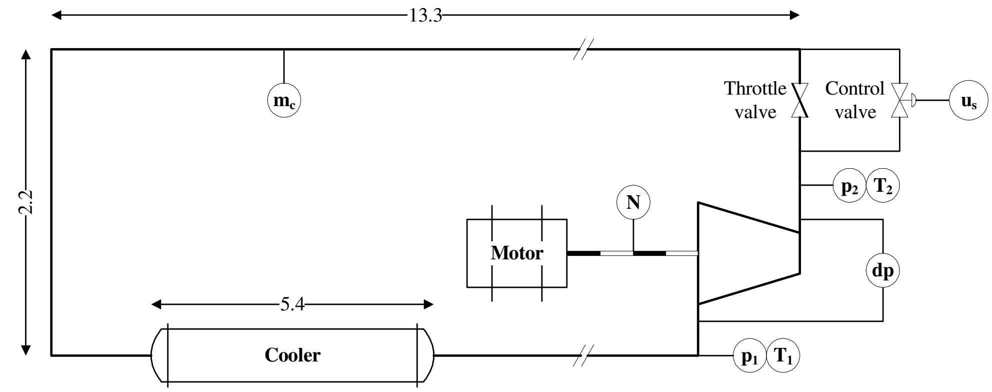

Figure 1 Schematic representation of centrifugal compressor test rig.

Furthermore, this configuration allows for varying the average pressure in the system between 1 and 15 bar. All results presented in this paper were obtained from experiments with N_2 gas (28.0134·10−3 kg/mol, R = 296.8 J/kg·K), an average suction pressure of 5 bar and at rotational speeds of 7,000 and 9,000 rpm. Throttling of the compressor is done by means of a butterfly valve (WouterWitzel DN 300). The parallel control valve (Schubert & Salzer 8021, DN 20) is used for more precise adjustments of the mass flow rate and it also serves as an actuator for the system. The return piping contains a measurement section with a flow straightener, an orifice flow meter with a diameter ratio of 0.533, and a gas cooler. With the combination of the throttle, return piping, and cooler, the conditions of the gas at the compressor inlet were kept approximately constant.

On the compressor shaft a shrouded impeller is mounted that contains 17 blades and has hub and exit diameters of 126.5 mm and 332.5 mm, respectively. The constant width diffuser has inlet and exit diameters of 354.3 mm and 490 mm, respectively. The complete aerodynamic package, consisting of the impeller, diffuser, inlet and return channels, is installed in a wide casing that offers the flexibility to test a variety of assemblies and to install additional measurement equipment, see also Figure 2.

The compression installation is equipped with numerous temperature probes (J-type thermocouple) and static pressure transducers (Rosemount) to determine the steady-state performance of the compressor, see also Figure 1. A dedicated data acquisition and control system is used for operating the installation, converting and recording sensor outputs, and for online monitoring. The sample time for all static measurements is 6 s.

Additional dynamic pressure transducers (Kulite) were installed in the suction and discharge pipes to measure the pressure rise fluctuations during experiments. Control valve opening was measured via an internal position sensor. For control implementation we used a TUeDACS/1 QAD module connected to a computer with Matlab/Simulink software. This system was also used to operate the control valve. The sampling rate of all dynamic I/O channels was set to 500 Hz.

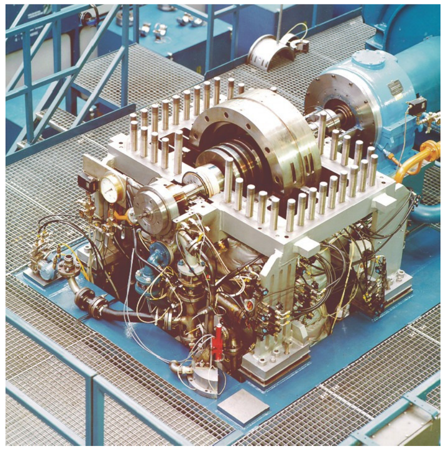

Figure 2 Picture of the centrifugal compressor with top casing removed.

2.2 Compressor model

In order to describe the dynamic behaviour of the compression system we developed a 2nd order nonlinear model, analogous to (5). The derivation of the model was previously discussed in (19, 20). The key elements of the model are repeated here since they are used later on in this paper.

The model consists of the following impulse and mass balance

$$\begin{aligned} \frac{dm_c}{dt} &= \frac{A_c}{L_c}\left(\Delta p_c(m_c, N) - \Delta p\right) \\ \frac{d\Delta p}{dt} &= \left(\frac{a_1^2}{V_1} + \frac{a_2^2}{V_2}\right)\left(m_c - m_l(\Delta p, u_l) - m_s(\Delta p, u_s)\right) \end{aligned}$$

where m_c represents the compressor mass flow and $\Delta p = p_2 - p_1$ denotes the pressure difference between the volumes V_1 and V_2 (with associated sonic velocities a_1 and a_2). The mass flows m_l and m_s through the throttle and control valve are both functions of the pressure difference and the valve opening u_l and u_s respectively. The term Δp_c denotes the quasi-stationary compressor characteristic at a rotational compressor speed N, see Figure 3. Determination of this curve and its role in the implementation of active surge control will be discussed later on. Simulations of stable and unstable behaviour of the system, identification results and validation data for two test rig configurations can be found in (19). In this paper the following values were used: $A_c = 0.0186$ m^2 and $L_c = 1.73$ m, and for the volumes $V_1 = 3.28$ m^3 and $V_2 = 0.270$ m^3.

2.3 Model-based control design

Over the years numerous different sensor and actuator configurations have been proposed for compressor surge stabilization, see (12, 20). The general requirements for a surge control system are that it stabilizes the system in an efficient way and that the control system is suitable for the harsh environment common for industrial compressor applications.

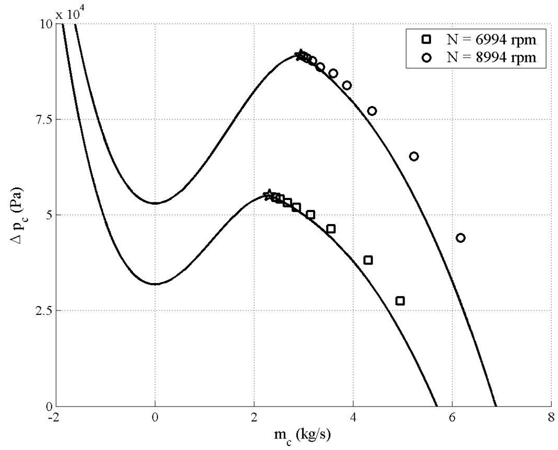

Figure 3 Quasi-stationary compressor map Δp_c with measured operating points. The stars are points on the approximated surge line.

A control approach that answers both requirements is the so-called positive (one-sided) surge controller proposed in (14). This controller uses a total plenum pressure sensor in combination with a control valve connected to the plenum. We point out that a control valve is better suited for harsh environments in comparison with for example air injectors and loudspeakers. With this controller losses that result from control actions after stabilization are reduced since the valve is closed when stabilization is achieved. A detailed discussion of this approach is given in (17) and similar results were obtained from closed-loop simulations for our test rig.

Drawbacks of the one-sided feedback are that stability can only be proven when the eigenvalues of the linearized system matrix are complex. In practice this means that stabilization can only be achieved in a limited portion of the unstable region. Furthermore, due to the use of a pressure sensor only, the linearized system has a non-minimum phase (NMP) zero, limiting the achievable bandwidth of the closed-loop system. Finally, successful stabilization with the one-sided control strategy is, just as with other control methods, hampered by the trade-off between bandwidth and capacity for commercially available control valves and by the sensitivity to model inaccuracies. This will be discussed in more detail below.

3 ACTIVE SURGE CONTROL

This section deals with the steps we took after the dynamic model development and validation and the model-based control design. We will first discuss the actual implementation of the surge controller on the centrifugal compressor test rig and the experimental results that we have obtained. We will then focus on the problem of actuator limitations and model uncertainties and how they form barriers for achieving stabilization of surge during closed-loop experiments. We will illustrate these problems with results from additional closed-loop simulations.

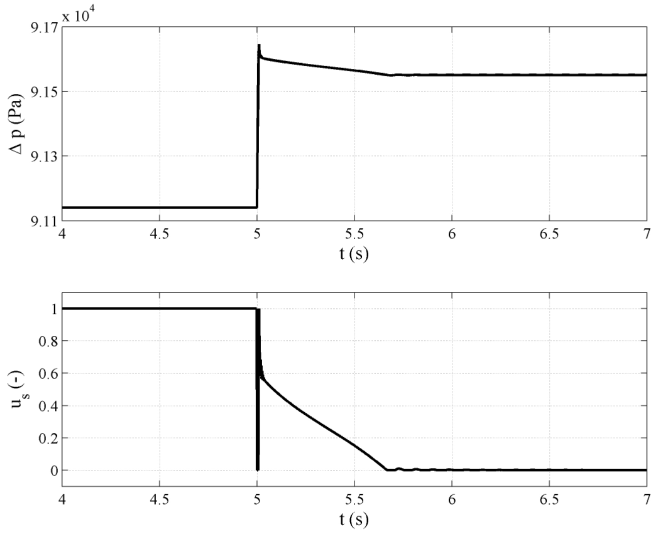

Figure 4 Closed-loop simulation result.

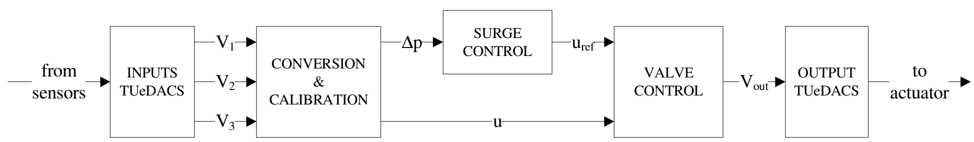

Figure 5 Schematic representation of surge control implementation.

3.1 Control implementation

For the synthesis of the one-sided surge controller we used the linearization of the developed model around a desired operating point (m_{c0}, Δp_0, u_{l0}, u_{s0}) where the subscript $_0$ indicates an equilibrium. We then applied the root-locus design method as in (17) to calculate the control gain **K** in the feedback law

$$u_s(t) = \max\left(0, \mathbf{K}\left(\Delta p(t) - \Delta p_0\right)\right)$$

In Figure 4 the result of a closed-loop simulation is shown for the intended operating point during experiments. This result shows that the designed controller stabilizes the nonlinear simulation model of the centrifugal compressor test rig in the desired operating point.

In the actual implementation we used a TUeDACS/1 QAD module to link the pressure sensors to the control valve actuator via a real-time computer code of the above feedback law. A block diagram of the implementation is shown in Figure 5. The valve controller was used to accurately realize the valve opening u_{ref} required by the surge controller.

At the start of each closed-loop experiment, the compressor was operated in a stable regime with the control valve fully open. After 5 seconds the controller was turned on while at the same time the desired valve position was set to $u_{s0} = 0$. The results of an experiment with $N = 9000\ rpm$, $m_{c0} = 2.9\ kg/s$, $\Delta p_0 = 9.15 \cdot 10^4\ Pa$, and $u_{so} = 0$ are shown in Figure 6.

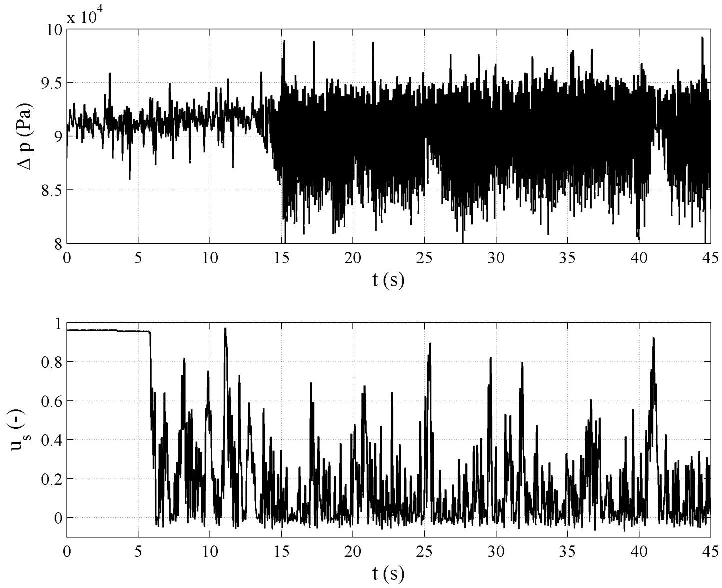

Figure 6 Measured input and output during closed-loop experiment at 9000 rpm.

From the pressure measurement in Figure 6 we note that no surge occurs between 5 and 13 seconds after which the system becomes unstable. Note that around 25 and 42 seconds the surge amplitude diminishes due to the large control valve opening. We conclude that the controller did not readily stabilize the compression system in the unstable regime while maintaining a low average control valve opening. Furthermore, we point out that, even after low-pass filtering of the measured data, noise levels were significant. This high noise resulted in large, nervous control actions. We remark that large control actions can falsify the linearity assumption that is used during the controller design. However, investigation of the effects of this falsification on the dynamic behaviour is beyond the scope of this paper.

3.2 Actuator limitations

We identified the non-ideal actuator dynamics as one of the likely causes for not succeeding to stabilize the compressor system. In order to investigate this, we first determined a model for the actuator dynamics, following the method proposed in (21). Following this procedure we obtained a 3rd order model that indicates a valve bandwidth of approximately 12 Hz. Furthermore, we found that the valve contains a time delay of 0.01 seconds.

We then proceeded by performing additional closed-loop simulations where we included the valve dynamics in the simulation model. A result of one of these simulations is presented in Figure 7. The plots in this figure show that the compression system is not stabilized by the one-sided controller when the valve has a limited bandwidth and a time delay. A more detailed analysis showed that in particular the time delay of the control valve destroys the stabilizing property of the surge controller. Unfortunately this time delay cannot be easily compensated for so implementation of the surge control algorithm requires a delay-free control valve with sufficient bandwidth and capacity.

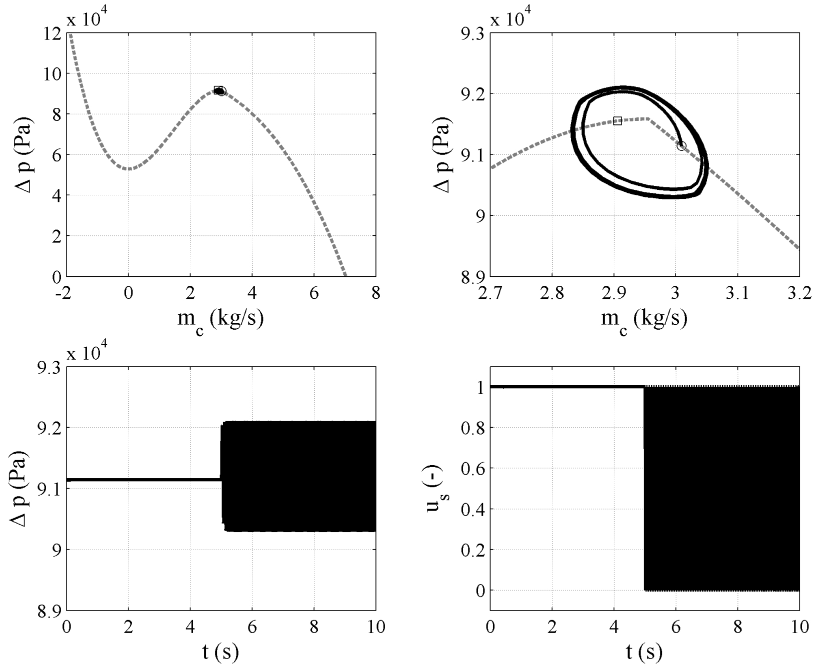

Figure 7 Closed-loop simulation result with effect of valve dynamics included.

3.3 Model uncertainties

A second problem that probably affected our closed-loop experiments is that of a wrong set point value Δp_{ref} in the control algorithm. This set point is obtained from the simulation model by calculating the intersection point of the compressor and throttle characteristics.

The unstable part of the compressor curve is determined by fitting a cubic polynomial, proposed in (8), through the surge point and the point at zero mass flow. The surge point can be measured with reasonable accuracy and the point at zero flow is estimated according to (22). Due to the fact that only two points with limited accuracy are used, the resulting fit is considered to be inaccurate. Hence, the calculated value for Δp_{ref} can be wrong.

The effect of a difference between Δp_{ref} and the actual operating point of the compressor Δp_0 is schematically illustrated in Figure 8. The solid line is the resulting closed-loop throttle curve when the valve opening is set by the controller. From the figure it can be seen that two equilibriums exist when $\Delta p_{ref} \neq \Delta p_0$. The third intersection point is not stabilizable since $u_s = 0$ everywhere around this point.

The existence of a second equilibrium in the stable flow regime might explain the change in behaviour around 25 and 42 seconds that was observed during the previously discussed experiment. Finally we remark that differences between Δp_{ref} and Δp_0 can even lead to situations where only a stable equilibrium exists or no equilibrium at all. See also (17). Hence, stabilization of unstable operating points requires an accurate set point or the control algorithm should explicitly take into account possible errors in the set point.

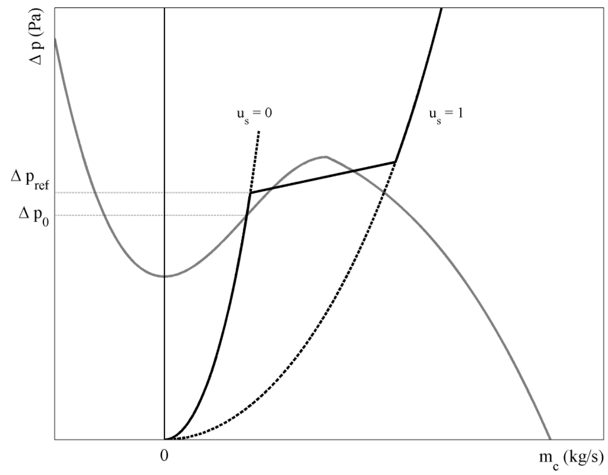

Figure 8 Scheme to illustrate the effect of a wrong controller set point Δp_{ref}.

4 CONCLUSIONS

With our research we aim to stabilize a large centrifugal compression system by implementing active surge control. In this paper we briefly reviewed our own work on dynamic modelling and control design for an industrial compressor test rig and the developments as they appear from literature.

Based on our research we identified actuator dynamics and model uncertainties as two of the major barriers for successful application of surge control for large compression systems. Development of dedicated actuators with sufficient bandwidth and capacity is therefore desirable. To improve model accuracy it seems worthwhile to further investigate open- and closed-loop identification and robust or adaptive control design.

We point out that the above recommendations are not complete since other issues like for example the linearity assumption, signal to noise ratio, and the absence of essential dynamics in the model also influence the chance for success of active surge control. In summary, successful application of surge control on full-scale systems requires an integral approach of dynamic analysis, modelling and identification, robust control design and dedicated sensor and actuator development.

REFERENCES

1. Greitzer, E.M., “The stability of pumping systems – The 1980 Freeman scholar lecture,” *ASME J. Fluids Eng.*, vol. 103, pp. 193-242, 1981.
2. Longley, J., “A review of nonsteady flow models for compressor stability,” *ASME J. Turbomach.*, vol. 116, pp. 202-215, 1994.

3. Willems, F., and Jager, B. de, “Modeling and control of compressor flow instabilities,” *IEEE Control Syst. Mag.*, vol. 19, pp. 8-18, 1999.
4. Paduano, J.D., Greitzer, E.M., and Epstein, A.H., “Compression system stability and active control,” *Annu. Rev. Fluid Mech.*, vol. 33, pp. 491-517, 2001.
5. Greitzer, E.M., “Surge and rotating stall in axial flow compressors: Part 1 – Theoretical compression system model,” *ASME J. Eng. Power*, vol. 98, pp. 190-198, 1976.
6. Emmons, H.W., Pearson, C.E., and Grant, H.P., “Compressor surge and stall propagation,” *Trans. ASME*, vol. 77, pp. 455-469, 1955.
7. Hansen, K.E., Jorgensen, P., and Larsen, P.S., “Experimental and theoretical study of surge in a small centrifugal compressor,” *ASME J. Fluids Eng.*, vol. 103, pp. 391-395, 1981.
8. Moore, F.K., and Greitzer, E.M., “A theory of post stall transients in axial compression systems: Part I – Development of equations,” *ASME J. Eng. Gas Turbines Power*, vol. 108, pp. 68-76, 1986.
9. Fink, D.A., Cumpsty, N.A., and Greitzer, E.M., “Surge dynamics in a free-spool centrifugal compression system,” *ASME J. Turbomach.*, vol. 114, pp. 321-332, 1992.
10. Gravdahl, J.T., and Egeland, O., “Centrifugal compressor surge and speed control,” *IEEE Trans. Contr. Syst.*, vol. 7, pp. 567-579, 1999.
11. Epstein, A.H., Ffowcs Williams, J.E., and Greitzer, E.M., “Active suppression of aero-dynamic instabilities in turbomachines,” *J. Prop. Power*, vol. 5, pp. 204-211, 1989.
12. Simon, J.S., Valavani, L., Epstein, A.H., Greitzer, E.M., “Evaluation of approaches to active compressor surge stabilization,” *ASME J. Turbomach.*, vol. 115, pp. 57-67, 1993.
13. Gravdahl, J. T., and Egeland, O., *Compressor Surge and Rotating Stall: Modeling and Control*. London: Springer-Verlag, 1999.
14. Willems, F., Heemels, W.P.M.H., Jager, B. de, and Stoorvogel, A.A., “Positive feed-back stabilization of centrifugal compressor surge,” *Automatica*, vol. 38, pp. 311-318, 2002.
15. Jager, B. de, “Rotating stall and surge control: A survey,” in *Proc. 34th IEEE Conf. Decision and Control*, New Orleans, LA, USA, 1995, pp. 1857-1862.
16. Arnulfi, G.L., Giannattasio, P., Giusto, C., Massardo, A. F., Micheli, D., and Pinamonti, P., “Multistage centrifugal compressor surge analysis: Part I – Experimental investigation,” *ASME J. Turbomach.*, vol. 121, pp. 305-311, 1999.
17. Willems, F., *Modeling and Bounded Feedback Stabilization of Centrifugal Compressor Surge*, Ph.D. thesis, Technische Universiteit Eindhoven, The Netherlands, 2000.
18. Skoch, G.J., “Experimental investigation of centrifugal compressor stabilization techniques,” in *Proc. ASME Turbo Expo 2003*, Atlanta, GA, USA, 2003.
19. Helvoirt, J. van, Jager, B. de, Steinbuch, M., and Smeulers, J., “Modeling and identi-fication of centrifugal compressor dynamics with approximate realizations,” in *Proc. 2005 IEEE Conf. on Control Applications*, Toronto, Canada, 2005, pp. 1441-1447.
20. Wal, M. van de, Willems, F., and, Jager, B. de, “Selection of actuators and sensors for surge control,” *J. Propulsion and Power*, vol. 18, pp. 84-92, 2002.
21. Helvoirt, J. van, Bosgra, O., Jager, B. de, and Steinbuch, M., “Approximate realization of valve dynamics with time delay,” in *Proc. 16th IFAC World Congress*, Prague, Czech Republic, 2005.
22. Meuleman, C., *Measurement and Unsteady Flow Modelling of Centrifugal Compressor Surge*, Ph.D. thesis, Technische Universiteit Eindhoven, The Netherlands, 2002.