Proceedings of
ASME TURBO EXPO 2001
June 4-7, 2001, New Orleans, Louisiana

2001-GT-0570

EXPERIMENTAL EVALUATION OF A HIGH-GAIN CONTROL FOR COMPRESSOR SURGE SUPPRESSION

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ABSTRACT

The present paper considers the suppression of surge instability in compression systems by means of active control strategies based on a high-gain approach. A proper sensor-actuator pair and a proportional controller are selected which, in theory, guarantee system stabilization in any operating condition for a sufficiently high value of the gain. Furthermore, an adaptive control strategy is introduced which allows the system to automatically detect a suitable value of the gain needed for stabilization, without requiring any knowledge of the compressor and plant characteristics. The control device is employed to suppress surge in an industrial compression system based on a four-stage centrifugal blower. An extensive experimental investigation has been performed in order to test the control effectiveness in various operating points on the stalled branch of the compressor characteristic and at different compressor speeds. On one hand the experimental results confirm the good performance of the proposed control strategy, on the other they show some inherent difficulties in stabilizing the system at high compressor speeds due to the measurement disturbances and to the limited operation speed of the actuator.

INTRODUCTION

Surge instability strongly limits the operating range and the performance of compression systems. As it is known, surge occurs at low compressor flow rates, causing highly undesirable oscillations in the system. By means of a control system it is possible to attenuate or eliminate the phenomenon, so allowing the plant to operate in naturally unstable points. In particular, the active control techniques are based on the use of a suitable sensor/actuator pair in a closed loop control device. The control is effective if the actuation is capable of dissipating the unsteady energy surplus introduced in the system by the compressor when it operates in the stalled region.

In the last decade much work has been devoted to the study of active suppression of compressor surge. A large part of the literature is based on the work of Greitzer (1976) and Moore and Greitzer (1986) who proposed dynamical models of the compressor instability that have been deeply exploited for the analysis and the design of control systems. Epstein et al. (1989) firstly suggested that surge can be prevented by actively damping the small disturbances which originate the instability while their amplitude is low. Experimental demonstrations of active stabilization of surge have been given by several investigators, e.g., Ffowcs Williams and Huang (1989) and Pinsley et al. (1991). An extensive study was carried out by Simon et al. (1993), who analyzed the performance of several sensor/actuator configurations, together with a proportional compensator, by means of a local stability analysis based on a linearized model.

After these pioneering works, the compressor surge control has attracted many researchers, as it appears from several recent contributions which suggest more sophisticated techniques to face the problem. A Lyapunov approach has been proposed by Behnken and Murray (1997) and by Gravdal and Egeland (1999). A nonlinear approach based on backstepping has been suggested by Gravdal and Egeland (1997) and by Banaszuk and Krener (1999). The problem of bifurcation control, based on the Moore-Greitzer model, is addressed by McCaughan (1990) and Kang et al. (1997). A feedback linearization method is presented by Badmus et al. (1996). Extensive surveys are provided by Gu, Sparks and Banda (1999) and by Willems and de Jager (1999).

The aim of the present work is to investigate the application of a high-gain type control for the suppression of surge instability in an industrial size compression system. With “high-gain control” we mean an approach based on a proper selection of sensor, actuator and control law which, at least in theory, guarantees system stabilization in any operating condition for a sufficiently high value of the gain. Furthermore, surge suppression is here intended as the capability of removing the system from a surge limit cycle, rather than the easier task to avoid instability by damping the small disturbances which originate it. Consequently, any local stability analysis or linearized model of the controlled compression system is here avoided for purposes of control design and performance prediction. Indeed, the dynamics of a compression system under unstable operation is strongly nonlinear, and the assumption of small perturbations of the steady equilibrium point is far from being fulfilled, especially in the case of compressors which exhibit abrupt stall (Giannattasio et al., 2000).

The compression system considered in the present work is based on a four-stage centrifugal compressor, which has been previously used by the authors for various experimental investigations in both stable and unstable operating conditions (Arnulfi et al., 1995, 1996, 1999a). A nonlinear lumped-parameter model of the compression system was also worked out, which proved to be capable of correctly predicting the system dynamics at different compressor speeds and throttle valve settings (Arnulfi et al., 1999b). Finally, this compression system was employed to investigate the effectiveness of an innovative device for the passive control of surge based on the use of an oscillating water column (Arnulfi et al., 2000).

The present active control device includes a sensor of differential pressure between the plenum and the compressor outlet, a proportional controller and an actuation valve at the plenum exit. The sensor/actuator pair has been selected on the basis of a theoretical analysis performed by Giannattasio (1999), who critically revised the work of Simon et al. (1993). Furthermore, Blanchini and Giannattasio (2000) demonstrated, by using a nonlinear approach, that such a control has the “high-gain” property and they also discussed the limitations introduced by the occurrence of control saturation. All these results are synthetically reported in the present paper, together with further considerations about the influence of measurement disturbances, unmodelled fast dynamics and low-pass filter on the effectiveness of the proposed control system. In particular, a simple linear model is employed to show that, in practice, the gain value cannot be chosen arbitrarily large without the risk that the disturbances compromise the local stability. This latter argument suggested the idea of using an adaptive control strategy, which allows the controller to automatically increase the gain until the system is removed from surge, so avoiding the overtuning of the gain. Furthermore, the adaptive control has the remarkable property that it does not require any knowledge of the compressor and valve characteristics.

After introducing some theoretical arguments concerning the above mentioned topics, the present paper reports the results of an esperimental analysis of the actively controlled compression system. Unsteady measurements of compressor and plenum pressures, mass flow rate and actuator position have been performed in different naturally unstable operating conditions, by varying both steady flow rate and compressor speed. Tests have been carried out by using both the classic proportional control and the adaptive strategy. The experimental results provide a rather complete information about the effectiveness and the limits of the proposed control system.

SYSTEM DESCRIPTION AND MODEL

Figure 1 shows a sketch of a compression system with some of the sensors and actuators proposed in the literature for the control of surge instability. The basic system is formed of a compressor, a volume (plenum) and a throttle valve, while the control device consists of a sensor of a proper system output, a controller where the required control law is applied, and an actuator which introduces the feedback signal into the system. The control effectiveness strongly depends on the appropriate selection of the sensor/actuator pair, which has to be based on the knowledge of the compression system dynamics under unstable operation and on the specific technical features of the considered plant.

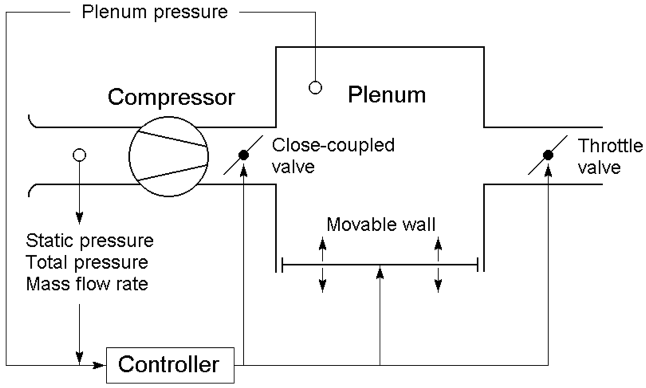

Fig. 1 - Controlled compression system

Model of the basic compression system

A simple and useful model of the basic compression system is the one proposed by Greitzer (1976) and successfully employed by the authors for simulating the system dynamics in both uncontrolled and controlled conditions (Arnulfi et al., 1999b, 2000; Giannattasio et al., 2000). It is a nonlinear lumped-parameter model and results in the following dimensionless equations:

$$\frac{d\varphi_c}{d\tau} = B\left(\psi_c - \psi_p\right), \quad (1)$$

$$\frac{d\varphi_t}{d\tau} = \frac{B}{G}\left(\psi_p - \psi_t\right), \quad (2)$$

$$\frac{d\psi_p}{d\tau} = \frac{1}{B}\left(\varphi_c - \varphi_t\right), \quad (3)$$

$$\frac{d\psi_c}{d\tau} = \frac{1}{\tau_c}\left[\psi_{cs}\left(\varphi_c\right) - \psi_c\right], \quad (4)$$

$$\psi_t = \left(A_c/A_t\right)^2 \varphi_t^2 . \quad (5)$$

Equations (1) and (2) are the momentum conservation equations in the compressor and throttle pipes, respectively; Eq. (3) expresses mass conservation in the plenum; Eq. (4) is a first-order model of the compressor dynamics; Eq. (5) represents the throttle characteristic.

Greitzer parameter *B*, which appears in Eqs (1)-(3), is defined as $B = U/2\omega_H L_c = \left(U/2a_p\right)\sqrt{V_p/A_c L_c}$ and can be interpreted as the ratio of pressure and inertial forces acting in the compressor pipe (Greitzer, 1976). The value of this parameter strongly affects the system stability and the control effectiveness. Valve parameter *G* in Eq. (2) is defined as $G = L_t A_c / L_c A_t$ and exerts a minor influence on the system dynamics (Greitzer, 1976). Term $\psi_{cs}(\varphi_c)$ in Eq. (4) refers to the steady-state compressor characteristic, while the time constant of the first-order compressor dynamics, τ_c, can be related to the time needed for the complete development of a stall cell (Greitzer, 1976; Arnulfi et al., 1999b).

Sensor-actuator selection and model

The sensor/actuator pair considered in the present work has been selected from the twelve different options suggested by Simon et al. (1993). Those authors considered a standard proportional control and all the combinations of four sensors (compressor flow rate, plenum pressure, compressor face total and static pressure) and three actuators (close-coupled valve in the compressor delivery pipe, plenum bleed valve, movable plenum wall), see Fig. 1. On the basis of a linear stability analysis, Simon et al. attained to the conclusion that the combination of a sensor of compressor mass flow rate with a close-coupled valve is far the best choice for a control device of maximum effectiveness. However, such a result was obtained for fixed values of some operating parameters (steady equilibrium positions of throttle valve and close-coupled valve) which, moreover, were not reported in the paper. On the contrary, the variation of the steady operating point of an actual compression system was duly considered by Giannattasio (1999), who revised the comparative analysis of Simon et al. by using the same linear approach. The results of that study show that pair "compressor mass flow sensor/close-coupled valve" is clearly superior to the other ones only for comparatively small values of the close-coupled valve fraction open, which imply large pressure losses in the compressor delivery pipe. Moreover, it turns out that five of the twelve sensor/actuator pairs have to be discarded because of severe gain-independent stability constraints, which cannot be removed by the control. Finally, some other combinations require extremely large values of the gain for an effective system stabilization (compressor face total or static pressure/close-coupled valve) or involve technical complications (control of both speed and position of the movable plenum wall). In conclusion, the best compromise between control performance and technical requirements appears to be attained by the use of a sensor of total or static pressure at the compressor inlet and of an actuation valve at the plenum exit. These controls turn out to be high-gain ones, in the sense that both allow system stabilization to be attained in any operating condition of the compressor for a sufficiently high value of the gain. In theory, the solution with the static pressure sensor appears to be superior to the other one because it does not imply any gain-independent constraint to system stability, while the total pressure sensor is effective only when the static stability condition is satisfied (the slope of the compressor characteristic must be less than the slope of the valve characteristic in the steady equilibrium point) (Simon et al., 1993; Giannattasio, 1999). However, such a condition is commonly satisfied in most of the actual compression systems, while a very simpler implementation of the sensing device is allowed by the use of the compressor face total pressure as system output. In fact, a simple application of the one-dimensional momentum equation shows that the compressor face total pressure, referred to the ambient pressure, is proportional to the mass flow acceleration in the compressor duct, i.e.,

$$\psi_{01} = \frac{p_{01} - p_0}{\frac{1}{2}\rho_0 U^2} \propto -\frac{1}{B}\frac{d\varphi_c}{d\tau} = \psi_p - \psi_c ,$$

where the last equality descends from Eq. (1). This means that ψ_{01} is also proportional to the pressure difference between any two points of the compressor duct, so that a representative sensor signal can be drawn by simply using a differential pressure trasducer between plenum and compressor outlet. For this reason, the following system output is defined and used in the present work:

$$y = \frac{p_p - p_c}{\frac{1}{2}\rho_0 U^2} = \frac{L_d}{L_c}\left(\psi_p - \psi_c\right), \qquad (6)$$

where L_d / L_c is the ratio of the equivalent lengths of the compressor delivery pipe and the whole compressor duct. This ratio is introduced in order to reduce pressure difference $\psi_p - \psi_c$, which is responsible for the flow acceleration in the whole compressor duct, to the pressure difference which causes the acceleration of the only mass contained in the duct portion between compressor outlet and plenum. With definition (6) of the system output, the present control law is written as:

$$u_r = \left(\frac{A_t - A_{ts}}{A_{ts}}\right)_r = K\, y , \qquad (7)$$

where K is the gain and the system input required by the controller, u_r, is defined as the dimensionless difference between the throttle flow area, A_t, and its steady equilibrium value, A_{ts}. In the present work, the valve at the plenum outlet is assumed to perform both functions of throttling device and actuator. These functions can be kept apart by simply introducing a bleed valve, as the actuator, in parallel with the throttle valve (Simon et al., 1993). The two options are quite equivalent from a conceptual point of view, the best choice depending only on technical considerations.

To complete the model of the control device, a simple first order model of the actuator dynamics is introduced in order to account for the time lag between the command output of the control law, u_r, and the response of the actuator, u:

$$\frac{du}{d\tau} = \frac{1}{\tau_a}\left(u_r - u\right). \qquad (8)$$

With the introduction of system input u, Eq. (5) can be rewritten as:

$$\psi_t = \left(A_c / A_t\right)^2 \varphi_t^2 = \left(A_c / A_{ts}\right)^2 \left(\frac{\varphi_t}{1+u}\right)^2 . \qquad (9)$$

If ψ_t is eliminated from Eq. (2) by using Eq. (9) and Eqs (6)-(7) are introduced in Eq. (8), the model of the controlled compression system results in five ordinary differential equations in the unknowns φ_c, φ_t, ψ_p, ψ_c and u. They can be solved numerically for given values of parameters B, G, τ_c, τ_a, L_d / L_c and gain K, if the steady-state characteristics of the compressor, $\psi_{cs}(\varphi_c)$, and of the throttle valve, A_{ts}, are known. Numerical simulations performed by Giannattasio et al. (2000) showed that the proposed control device is capable of suppressing surge within almost the whole unstable operating range of the compressor with reasonable values of the gain. Furthermore, they showed that the predictions of the non-linear model can be substantially different from the ones of a linear stability analysis, especially in the case of compressors which exhibit abrupt stall.

SUPPRESSION OF SURGE CYCLES

A considerable simplification of the model described in the previous section can be obtained by neglecting the flow inertia in the throttle duct ($G = 0$) and the time-lags in the transient responses of compressor and actuator ($\tau_c = \tau_a = 0$). Such approximations are well accepted in the literature since the simplified model has been shown to capture the fundamental dynamics of the compression system (Greitzer, 1976; Simon et al., 1993). In this case, the system of equations reduces to:

$$\frac{d\varphi_c}{d\tau} = -B\left[\psi_p - \psi_{cs}(\varphi_c)\right], \tag{10}$$

$$\frac{d\psi_p}{d\tau} = \frac{1}{B}\left[\varphi_c - (1+u)\,\varphi_{ts}(\psi_p)\right], \tag{11}$$

where $\varphi_{ts}(\psi_p) = A_{ts}/A_c\sqrt{\psi_p}$ is the steady characteristic of the throttle and $u = A_t/A_{ts} - 1$ represents the control action. This system is associated with the output:

$$y = \frac{L_d}{L_c}\left[\psi_p - \psi_{cs}(\varphi_c)\right]. \tag{12}$$

For a more compact notation, denote by $\mathbf{x}(\tau) = (\varphi_c(\tau), \psi_p(\tau))^{\mathrm{T}}$ the system state at time τ, and by $\mathbf{x}_s = (\varphi_s, \psi_s)^{\mathrm{T}}$ the steady equilibrium state which is the solution of the equations

$$-B\left[\psi_p - \psi_{cs}(\varphi_c)\right] = 0\,, \tag{13}$$

$$\frac{1}{B}\left[\varphi_c - \Gamma(\psi_p)\right] = 0\,. \tag{14}$$

As mentioned in the introduction, the main goal of the present control strategy is to remove the system from surge limit cycles driving the state to the target equilibrium point, $\mathbf{x}_s$. When the system is under deep surge, linearization methods cannot be applied to the stabilization problem, since surge regime is characterized by a strongly non-linear behaviour (Greitzer, 1976; Giannattasio et al., 2000). Therefore, we consider the non-linear stabilization approach adopted by Blanchini and Giannattasio (2000). Let us consider the proportional control

$$u(\tau) = K\,y(\tau) \tag{15}$$

where K is a real constant. This simple control has been investigated by several authors (see, for example, Ffowcs Williams and Huang, 1989; Simon et al., 1993) and it has the following property if applied to system (10)-(11) (Blanchini and Giannattasio, 2000).

Proposition (high gain stabilizability): there exists $\overline{K} > 0$ such that for each $K \geq \overline{K}$ and for any initial condition $\mathbf{x}(0)$ which belongs to (or which is inside) a limit cycle, the convergence condition, $\mathbf{x}(\tau) \to \mathbf{x}_s$ as $\tau \to \infty$, is guaranteed.

This theoretical result, which is based on model (10)-(11), states that control (15) with a sufficiently large value of $K > 0$ is a suitable stabilizing control for the present class of systems.

However, from a practical standpoint, the following considerations can be done which partially invalidate this result.

i) The control is subject to saturation constraints of the form $-1 \leq u \leq M = A_{t,max}/A_{ts} - 1$, which means that the actual control action will be

$$u = \mathrm{sat}_{[-1,M]}(K\,y)$$

where

$$\mathrm{sat}_{[\alpha,\beta]}(q) = \begin{cases} \alpha & \text{if} \quad q < \alpha \\ q & \text{if} \quad \alpha \leq q \leq \beta \\ \beta & \text{if} \quad q > \beta \end{cases}.$$

Having a too high value of the gain is useless in the presence of such strict saturation constraints. This problem has been theoretically investigated by Blanchini and Giannattasio (2000), who showed that, even for the simplified model (10)-(11), whenever the high gain saturated controller fails there will be no controller that removes the system from limit cycles.

ii) The real system is affected by disturbances. In particular, the presence of sensor noise can compromise the system stabilization. To eliminate such a noise the sensor signal must be processed by a low-pass filter, the cut-off frequency of which has to be suitably chosen. Sensor noise affects the control output, so producing actuator oscillations the amplitude of which increases with the value of the gain. High permanent oscillations are not desirable because they causes actuator mechanical and electrical stress.

iii) The second order system (10)-(11) does not consider part of the system dynamics such as the actuator dynamics, the compressor dynamics, and the filter dynamics. As previously mentioned, these dynamics can be reasonably neglected in the simulation of an open loop system. Unfortunately, it is known that a high gain feedback can excite such dynamics at the point that they may compromise stability.

The three points above show that a too high value of gain K may be not practically useful or even dangerous. To better explain this fact one can use a linearized model of the system.

Linear analysis

Linear analysis is not sufficient on its own to assure global stability, namely, in the present case, to assure surge suppression. Nevertheless, it turns out to be very useful to point out some limitations. Indeed, the stabilization problem requires, as a necessary condition, that the considered equilibrium point is locally stable, because as the system approaches the equilibrium the linear model provides a faithful description of its behaviour. System (10)(11)(12) admits the following linear approximate representation:

$$\frac{d\,\delta\varphi_c}{d\tau} = -B\left(\delta\psi_p - m_c\delta\varphi_c\right), \tag{16}$$

$$\frac{d\,\delta\psi_p}{d\tau} = \frac{1}{B}\left(\delta\varphi_c - \frac{1}{m_t}\delta\psi_p - \varphi_s\,\delta u\right), \tag{17}$$

$$\delta y = \delta\psi_p - m_c\,\delta\varphi_c\,, \tag{18}$$

where $m_c = (d\psi_{cs}/d\varphi_c)_s$ and $m_t = (d\psi_t/d\varphi_t)_s = 2(A_c/A_{ts})^2\varphi_s$ denote the slopes of the compressor and valve characteristics, respectively, in the steady equilibrium point, and term L_d/L_c has been omitted in Eq. (18) since it can be thought to be included in the gain. The transfer function of the open loop system is

$$F(s) = \frac{-c\,s}{s^2 + b\,s + a}\,, \tag{19}$$

where $a = 1 - m_c/m_t$, $b = 1/Bm_t - Bm_c$ and $c = \varphi_s/B$. The denominator of the closed-loop transfer function is

$$p(s,K) = s^2 + (b + cK)s + a \ , \tag{20}$$

which turns out to have roots with negative real part (linear stability condition) for a sufficiently large value of *K*, i.e., for

$$K > -b/c = \frac{B}{\varphi_s}\left(Bm_c - \frac{1}{Bm_t}\right) = B^2\frac{m_c}{\varphi_s} - \frac{1}{2\psi_s} . \tag{21}$$

However, the situation is slightly different if fast dynamics or a low-pass filter are considered. For the simple exposition, let us consider the case in which only the filter is taken into account (its use is necessary in practice) and let us consider, for instance, a filter having the following transfer function:

$$F_{filter}(s) = \frac{1}{1 + 2(\xi/\omega_0)s + s^2/\omega_0^2} \tag{22}$$

where $\omega_0/2\pi$ is the cut-off frequency and $\xi < 1$ is a positive parameter which depends on the particular filter. The closed-loop characteristic polynomial turns out to be

$$p_{filter}(s,K) = \left(s^2 + bs + a\right)\left(1 + 2\frac{\xi}{\omega_0}s + \frac{s^2}{\omega_0^2}\right) + cKs \tag{23}$$

and, for any value of a, b, c, ξ, ω_0 , there exists a limit value, $K_{INST} > 0$, such that for $K \geq K_{INST}$ local stability is lost. This property can be proved by plotting the positive root locus or just using the parameterized Routh-Hurvitz table; the formal proof is skipped for brevity. Furthermore, even under condition $K < K_{INST}$ there may be some problems. Indeed, for high values of *K* the closed-loop linearized system may have (stable) modes which are poorly damped. These modes are extremely sensitive to disturbances, so that their effect can result in permanent oscillations around the equilibrium point. In a nonlinear context, these oscillations may have a destabilizing effect and they can prevent the surge to be completely suppressed.

A trade-off choice of k: the adaptive control

The previous considerations show that, practically speaking, *K* has to be chosen large, but not so much as to compromise local stability. This choice is hard to be made by computation due to the nonlinear nature of the system behaviour during surge.

An efficient way to tune *K* is to do it adaptively. The basic idea behind this approach is that the controller automatically increases the gain until the system is removed from surge. This goal can be accomplished by using, instead of a constant *K*, a time varying non-decreasing gain, $K(\tau)$, as long as system output *y* is outside a tolerance interval having an amplitude, ε, fixed by the user. Such a gain can be computed by the adaptive control law

$$\frac{d\,K(\tau)}{d\tau} = \mu\,\sigma(|y|) , \tag{24}$$

where $\mu = const. > 0$ is an adaptation parameter and

$$\sigma(|y|) = \begin{cases} 0 & \text{if } |y| \leq \varepsilon \\ |y| - \varepsilon & \text{if } |y| > \varepsilon \end{cases}$$

represents the distance of output *y* from threshold interval [-ε,ε]. The convergence of this control law has been theoretically investigated and numerically validated by Blanchini and Giannattasio (2000). Note that, if convergence occurs and $y(\tau)$ enters interval [-ε,ε], then condition $dK/d\tau = 0$ holds and the adaptation stops; from that moment we have $K(\tau) = const. = K_\infty$. A great advantage of this procedure is that it does not require the knowledge of the compressor and valve characteristics. Furthermore, by its nature, *K* is increased as far as it is necessary for stabilization, so that the practical problem of having a too high value of the gain can be solved.

In practice, the adaptive procedure can be used in two steps:

1) *Training session*: the adaptive control is applied and the limit value of the gain, K_∞ , is detected.
2) *Working session*: the controller $u = K\,y$ is applied with $K = const. = K_\infty$.

EXPERIMENTAL TESTS

In order to verify the effectiveness of the proposed control strategy, an industrial compression plant has been coupled to a properly designed control device and a comprehensive set of measurements has been performed.

Test plant and instrumentation

The compression system, shown in Fig. 2, is based on a low pressure multi-stage centrifugal compressor driven by a DC motor through a speed increasing gear. The blower includes four impellers, with 16 backswept blades and 465 *mm* outer diameter, and vaned diffusers. The compressor inlet consists of a radial bellmouth duct with a 125 *mm* inner diameter ($A_c = 122.7$ cm^2), while the delivery pipe is connected to a cylindrical plenum of large volume ($V_p = 3.1332$ m^3). The equivalent length of the compressor ducting turns out to be $L_c = 13.5$ *m*, while the equivalent length of the compressor delivery pipe is $L_d = 7$ *m*.

The normal operation speed of the considered blower ranges from 2000 to 4000 *rpm*. The minimum value of the system stability parameter *B*, corresponding to the compressor speed of 2000 *rpm*, is

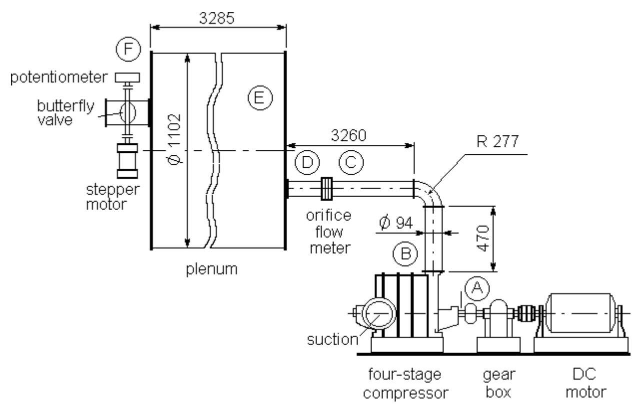

Fig. 2 - Experimental compression plant.

0.304, which turns out to be far higher than the critical value of the present compressor, $B_{crit}=0.06$ (see Abed et al., 1993, for a rigorous definition of bifurcation parameter B_{crit}). This results in a great system tendency to instability and accounts for the deep surge conditions observed by Arnulfi et al. (1999a) also at the lowest compressor speeds.

A butterfly valve at the plenum exit performs both functions of throttling device and actuator. The valve has a disc of 101 *mm* diameter and its movable part, having a moment of inertia of 1.5 $kg\ cm^2$, is mounted on ball bearings. It is driven by a stepper motor with a rotor inertia of 0.56 $kg\ cm^2$, a maximum torque of 1 *Nm* and a resolution of 200 *steps/rev*.

The instrumentation system is shown in Fig. 3, where the capital letters refer to the measurement point locations in Fig. 2. Pressures and temperatures are measured by means of inductive transducers and K-thermocouples, respectively, while a magnetic pick-up is used for the compressor rotational speed. The mass flow rate is measured by means of an orifice flow meter mounted in the compressor delivery pipe. Although this instrument is normally used for steady flow measurements, it was considered acceptable in the present case because of the slow dynamics of surge (a few cycles per second). The throttle angular position is measured by means of a precision potentiometer. The stepper motor is driven by a Power Driver Unit which allows a quarter of step resolution (0.45°) to be selected.

The acquisition of temperatures and rotational speed is performed only once at the beginning of each test, while the signals of pressure and valve angular position are acquired simultaneously at a sampling rate which must be high enough to correctly represent surge dynamics. This sampling rate is determined by an external trigger provided by a pulse generator.

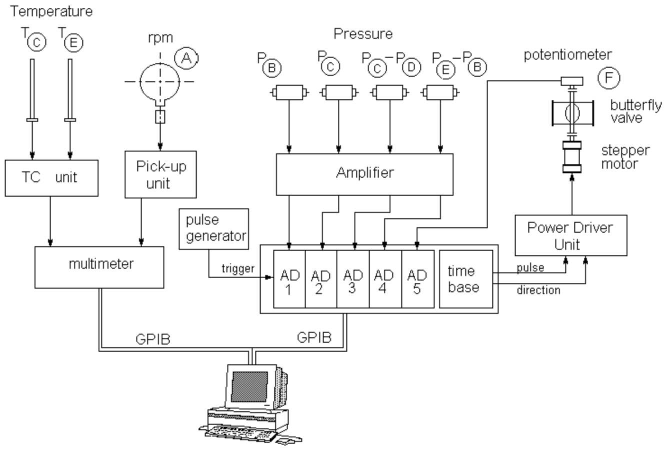

Fig. 3 - Schematic of the instrumentation system.

Experimental procedure

Previous experimental tests performed by Giannattasio et al. (2000) had shown some difficulties in stabilizing the compression system due to software limitations and to the use of a stepper motor with an excessively large rotor inertia. These limitations have been largely reduced in the present work by using a more efficient computer-based system for data acquisition and control and by selecting a lighter and faster actuation device.

A new software has been developed with the aim of maximizing the control speed. A sampling rate of 30 *Hz* has been selected as a compromise between the need of a faithful reproduction of the surge dynamics and the requirement of a sufficiently large time interval between two subsequent acquisitions which allows the desired angular displacement of the actuator to be completed. The stepper motor has been driven at the frequency of 4000 *Hz*. Over this value the strong accelerations of the actuator during the control originate inertial torques which can exceed the motor torque. On the basis of the values of sampling rate and motor frequency the code computes the maximum number of motor steps which are allowed in a sampling period. When the time needed for calculations and I/O operations is considered, the residual sampling period allows a maximum of about 110 motor steps (50°). If the controller requires a larger actuator displacement, the codes limits the number of steps at its maximum value.

As mentioned above, the acquisition of the pressure and angular displacement signals occurs simultaneously on five separate channels. The system output signal, which is acquired as the differential pressure between plenum and compressor outlet, is firstly corrected by adding the pressure losses between the two measurement sections. These losses are estimated as a fraction of the differential pressure signal from the orifice flow meter (a fraction of 60% has been considered on the basis of preliminary calibration tests). After being acquired and corrected, the system output signal is filtered by means of a low-pass 2nd-order Butterworth filter ($\xi = 1/\sqrt{2}$ in transfer function (22)) in order to eliminate the measurement disturbances. Preliminary tests suggested an optimum value for the cut-off frequency between 4 and 6 *Hz* (the maximum frequency of the surge oscillations in the present compression system is close to 1 *Hz*). At this point, the proportional or adaptive control law is applied to the filtered data, so obtaining the valve flow area required by the control. This area value is turned into valve angular position and is compared with the actual butterfly angle computed from the potentiometer signal. The difference between these two angles is converted into number of pulses to the stepper motor, which are limited in case they exceed the maximum allowable value.

The present software for data acquisition and control allows the throttle valve to be moved to the desired steady position and the control device to be enabled or disabled when required, without interrupting the processes of data acquisition and recording.

Experimental results

Tests have been carried out at the compressor speeds of 2000, 2500, 3000 and 3500 *rpm* and for 9 different angular positions of the throttle valve (from 5° to 25° with a step of 2.5°) corresponding to operating points on the unstable branch of the compressor characteristic curves, see Fig. 4. In each of these test conditions, the two-stage procedure described previously (training session and working session) has been applied, by always starting the control after a fully developed surge had been obtained. At first, the adaptive control has been performed by assuming the value of $50/\omega_H$ for the nondimensional adaptation parameter, μ. However, a different choice of this parameter within a very large range does not change significantly either the limit gain, K_∞, or the stabilization time, as shown by Blanchini and Giannattasio (2000). In the cases of successful adaptive stabilization, the system has been taken back to unstable operation and the control test has been repeated, by using the

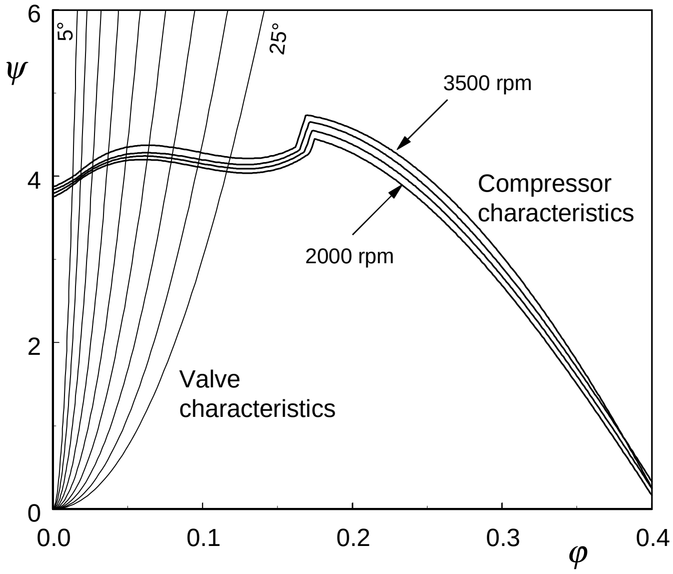

Fig. 4 - Test points as intersections of valve and compressor characteristics.

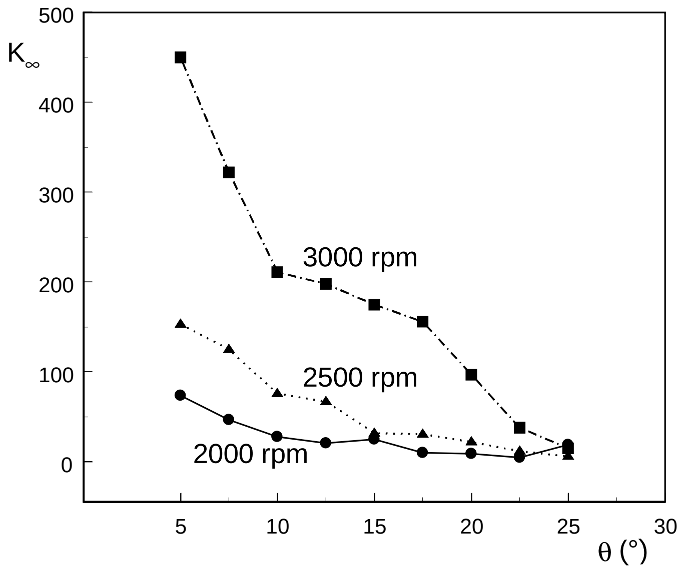

Fig. 5 - Limit gain values of the adaptive control.

value of K_∞ obtained in the training session as the constant gain of a standard proportional control.

The suppression of surge has been obtained in all the test conditions, except the ones at the highest compressor speed, 3500 *rpm*. The summary of the present results is shown in Fig. 5, which reports the values of K_∞ obtained in the stabilized conditions. The limit gain turns out to strongly increase with compressor speed and throttle closing. These expected trends can be justified even by a linear stability analysis. In fact, Eq. (21) shows that, if m_c>0, the gain grows with B^2, and hence with U^2, while it varies as the inverse of the steady equilibrium flow coefficient, φ_s.

Detailed representations of the adaptive control action are provided in Figs. 6*a-d*, which show the time traces of flow coefficient, plenum pressure coefficient, filtered system output and valve angle for the steady equilibrium condition corresponding to θ_s=20° and for all the considered compressor speeds. The time required for the system stabilization is observed to increase with the compressor speed, i.e., with *B*: at 2000 *rpm* 3 Helmoltz periods are sufficient while about 50 periods are required at 3000 *rpm.* The achievement of system stabilization is shown by the disappearance of the low-frequency surge oscillations in the signals of flow rate, plenum pressure and system output. After the system stabilization has been reached, the flow rate appears to be almost constant, while the plenum pressure signal is affected by high-frequency oscillations the amplitude of which increases with the compressor speed. These oscillations might result from the fact that the plenum pressure has been obtained by summing up the signals of compressor delivery pressure (which is highly disturbed) and differential pressure between plenum and compressor outlet, rather than from direct measurements.

The behaviour of the adaptive control is quite evident in the plots of the valve angle. After the control has been started, the valve moves with oscillations of growing amplitude, due to the increasing gain, until the system stabilization is attained. However, high-frequency oscillations remain in the angle signal after the surge has been suppressed. They are due to residual disturbances in the filtered output signal which are amplified by the gain, so resulting in oscillations whose amplitude increases with the compressor speed.

The operating condition at 3500 *rpm* (Fig. 6*d*) shows the limits of the performance of the present control device. In fact, the actuator is not capable of performing the large displacements required by the controller (frequent saturated valve openings should occur in this operating condition), due to valve accelerations which exceed the stepper motor capability.

The harmful effects of disturbances and actuator speed limitations on system stabilization have been fully confirmed by numerical simulations of the controlled system dynamics. The model of Eqs. (1)-(8), together with adaptive control law (24), has been employed to numerically test the operating condition of Fig. 6*d*, by introducing either a limitation of the maximum angular velocity of the valve or a sinusoidal disturbance of system output *y*. The simulations have been performed with the following values of the parameters: *B*=0.53, *G*=0.025, τ_c=4, τ_a=0.1, $L_d/L_c = 0.52$, $K(0)$=0, μ=8.4, ε=0.1. The amplitude and the frequency of the disturbance have been set equal to 0.05 and 4 *Hz*, respectively, which correspond to the leading component of the measured residual noise in the filtered *y* signal. In ideal conditions (absence of disturbances and actuator speed limitations) the simulation predicted a complete suppression of surge. On the contrary, when varying the maximum allowable actuator speed (without disturbances), system stabilization was observed only for values of $(d\theta/dt)_{\max}$ greater than about 1500 °/*s*, which is close to the estimated limit performance of the actual control device. Furthermore, stability was never reached when the sinusoidal disturbance was added to the system output, not even in conditions of no actuator speed limitation.

The effects of the proportional control are shown in Figs. 7*a-c*, which report the time traces of φ_c, ψ_p, *y* and θ, together with the corresponding system trajectories in plane (φ_c, ψ_p), in three different operating points (θ_s=7.5°, 15°, 22.5°) at the compressor speed of 2500 *rpm*. It is observed that, when using the limit gain values of the

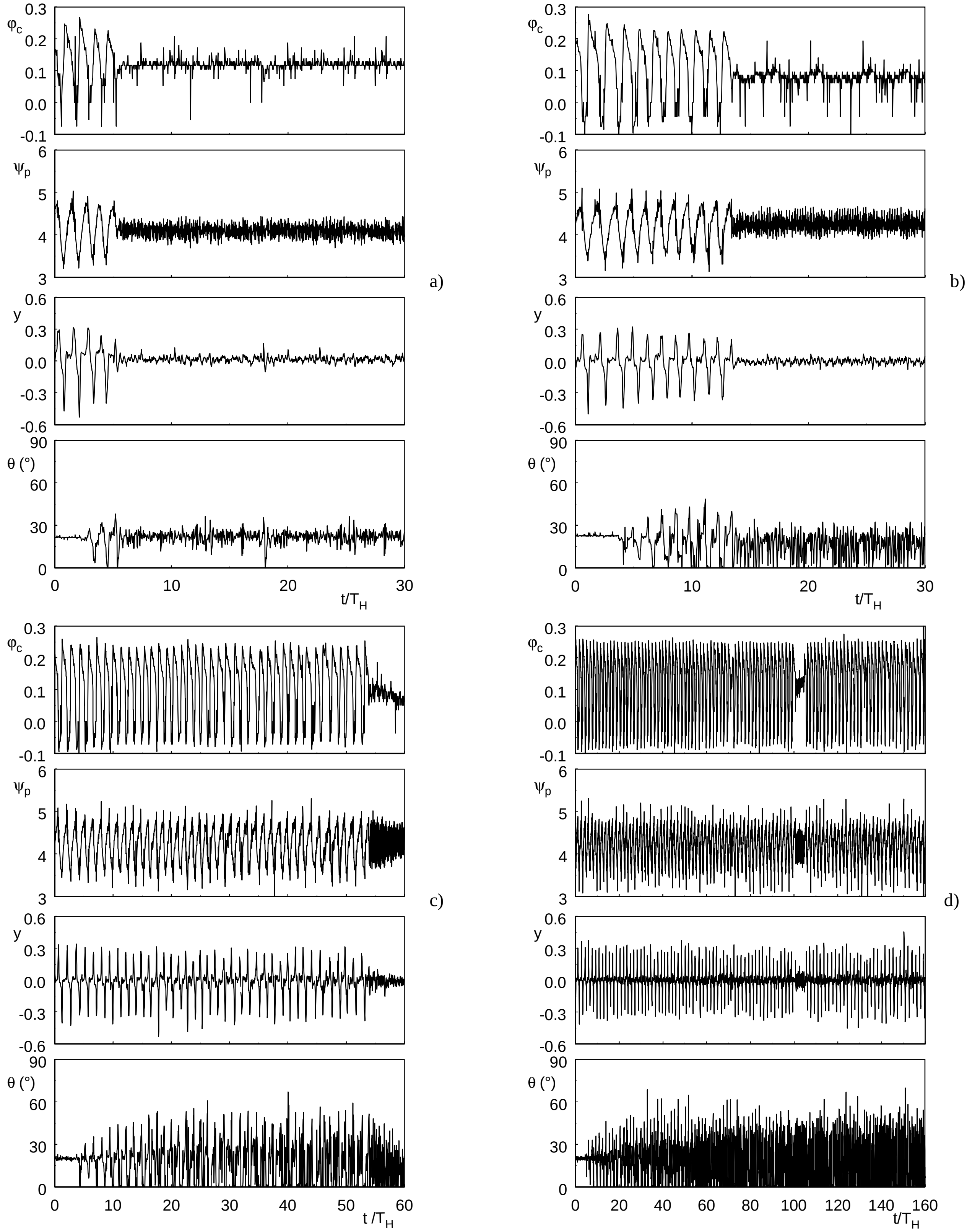

Fig. 6 - Time traces of the system parameters during the adaptive control for θ_s=20°: a) 2000 rpm; b) 2500 rpm; c) 3000 rpm; d) 3500 rpm.

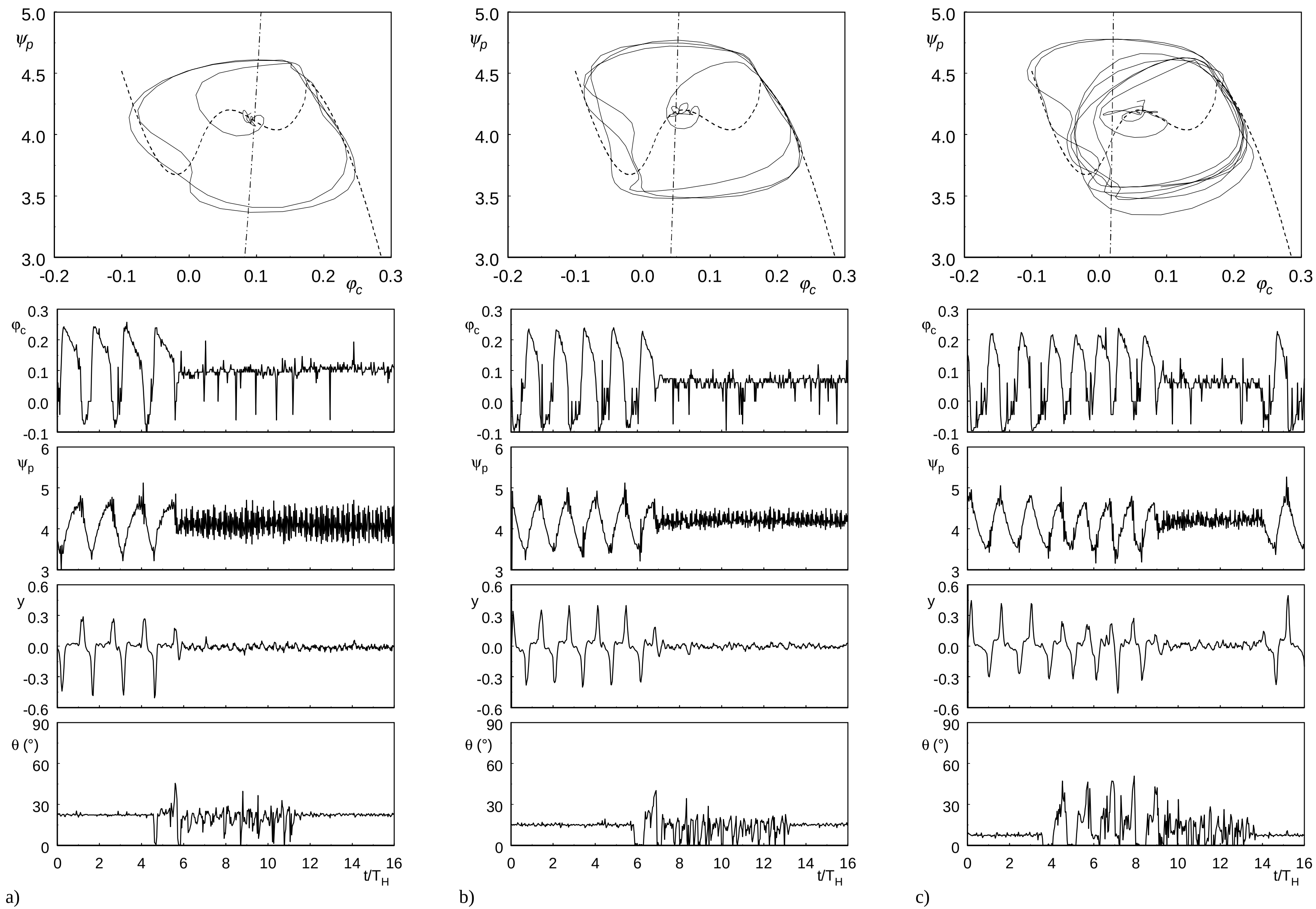

Fig. 7 - System trajectories in plane (φ_c, ψ_p) and time histories produced by the proportional control at 2500 rpm: a) θ_s=22.5°; b) θ_s=15°; c) θ_s=7.5°.

adaptive control, the proportional control turns out to be very effective since the system stabilization is obtained in a short time even in the more difficult case of very closed valve (the stabilization time varies from 1 Helmolt period at 22.5° to 5 periods at 7.5°). The stabilization test has been completed by inhibiting the control after the surge cycles have been completely suppressed and by observing the consequent system behaviour. As the cases in Figs. 7*a-c* are concerned, it is noticed that the system continues to be stable after the control has been removed at the steady valve angles of 15° and 22.5°, while at 7.5° it immediately returns to a surge condition. This behaviour, which has been observed also at the other compressor speeds, can be explained by observing that for large values of θ_s the equilibrium point is a locally stable one (the slope of the compressor characteristic is negative or slightly positive), while the dynamic stability condition (b>0 in Eq. (19)) is not satisfied at valve angles less than about 10°.

For a clearer representation of the system trajectories in plane (φ_c, ψ_p) the data to be plotted have been properly filtered to eliminate the high-frequency disturbances and they have been limited to few oscillations before system stabilization. Furthermore, the (φ_c, ψ_p) plots in Figs. 7*a-c* report the compressor characteristic at 2500 *rpm* (dashed line) and the steady characteristic of the valve (dashed-dotted line). These plots show that the system proceeds from a deep surge cycle to a stable condition through a short transient evolution. It is also observed that for the larger values of valve steady angle the system correctly converges to the equilibrium point (the intersection of the compressor and throttle characteristics), while for θ_s=7.5° the system moves to a flow coefficient value which is considerably larger than φ_s. This behaviour has been observed in all the other operating conditions at very low values of θ_s and it can be explained by considering that, in these cases, the possible valve motion around the equilibrium position is not symmetrical, being limited below by the saturation constraint of complete closing. Consequently, the valve oscillations, which are observed also in stabilized conditions due to the disturbances which affect the system output signal, occur around an average angular position which is larger than the desired one. Such a behaviour resulted also from numerical simulations of the controlled system dynamics, when a sinusoidal disturbance was added to output signal y.

CONCLUSIONS

A high-gain approach for the active control of compressor surge has been introduced and validated by experiments. The differential pressure between plenum and compressor outlet has been selected as the sensor signal, while the actuation is performed by means of the throttle valve at the plenum exit. Besides a standard proportional control, an adaptive strategy has been introduced in order to perform a satisfactory tuning of the gain. A computer-based control system has been coupled to an industrial compression plant based on a four-stage centrifugal blower, and an extensive experimental investigation has been performed.

The experimental results show that the proposed control strategy is capable of suppressing surge in almost the whole unstable branch of the compressor characteristic for rotational speeds up to 3000 *rpm*. In these conditions the standard proportional control turns out to be very effective if the limit gain values provided by the adaptive control are imposed. At the highest compressor speed (3500 *rpm*) the control strategy fails. This is not a conceptual limitation of the high-gain approach, which in theory assures system stabilization in any operating condition, but it mainly results from the limited actuation speed and from the interaction of high gain, sensor disturbances and actuator saturation. In fact, as the compressor speed, and hence parameter B, is increased, the system requires higher gain values to be stabilized. On one hand, such high gains cause very fast actuator displacements to be required by the controller, so that the inertial resistance of the valve can exceed the motor torque. On the other hand, the unavoidable disturbances of the system output signal are strongly amplified by the high gain, so resulting in large valve oscillations and hence in frequent conditions of actuator saturation (complete valve closing). Saturation does not necessarily cause system instability on its own (the so called "bang-bang" controls, which are based on actuator saturation, have several technical applications), but it is clear that the saturation induced by disturbances has no correlation with the system dynamics to be controlled.

Another negative effect of amplified disturbances and valve saturation is observed in the stabilized conditions at small valve steady angles: the asymmetrical valve oscillations around the equilibrium position determines an average flow rate which is significantly larger than the desired one.

In order to attenuate or possibly eliminate the limits of the proposed control strategy, further work is required which should be focused on two main topics. On one hand, an actuator of higher performance should be selected, which is not an easy task due to the difficulty of increasing both motor torque and speed (a motor with a higher torque has usually a larger rotor inertia). On the other hand, the origin of the sensor disturbances should be carefully investigated (the low frequency disturbances in the filtered signal might be due to secondary dynamics which are excited by the high gain feedback), and different filtering techniques should be possibly considered. Although the solution of both problems appears to be a rather difficult task, the authors believe that further investigations are worth pursuing, since the proposed high-gain approach has the potential for very effective applications in the field of compressor surge control.

ACKNOWLEDGMENTS

The present research has been supported by the Italian National Council of Research (*CNR*).

NOMENCLATURE

A	area
B	Greitzer parameter
G	valve parameter
K	control gain
L	equivalent length
T_H	Helmoltz period
U	impeller tip speed
V	volume
a	speed of sound
$\dot{m}$	mass flow rate
p	absolute pressure
s	Laplace transform variable
t	time
u	system input
y	system output

Greek symbols

Δp_c	compressor pressure rise

$\delta(\bullet)$	small perturbation
θ	valve angular position
ρ	density
τ	dimensionless time = $\omega_H\ t$
τ_a	actuator dynamics time constant
τ_c	compressor dynamics time constant
φ	flow coefficient = $\dot{m}/\rho_0 U A_c$
ψ	pressure coefficient = $2\,(p-p_0)/\rho_0 U^2$
ψ_c	compressor pressure coeff. = $2\,\Delta p_c/\rho_0 U^2$
ω_H	Helmoltz angular frequency = $a_p\sqrt{A_c/L_c V_p}$

Subscripts

0	ambient
1	compressor inlet
c	compressor
p	plenum
r	required by the controller
s	steady-state
t	throttle valve

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