FAST RESPONSE PNEUMATIC SERVOSYSTEMS WITH DIGITAL VALVES

G. Belforte, G. Mattiazzo, S. Mauro
Department of Mechanics - Politecnico di Torino
C.so Duca degli Abruzzi 24

ABSTRACT
The use of digital modulating valves for the development of pneumatic servosystems is becoming a widely employed technique. The main requirement for the valves is low response time, and typically it involves the size to be relatively small. The paper deals about the possibility to use small, fast response valves coupled with larger, and slower, valves in order to obtain fast response and wide bandwidth in large size servosystems. The faster valves provide the system with the capability to obtain high precision, while the larger valves allow high flows in order to increase the fastness of the system.

KEYWORDS
Pneumatic servosystems, digital valves, control

NOMENCLATURE

- $C$: global valves medium conductance, as computed by the controller
- $C_i$: controller computed medium conductance for $i^{th}$ valve
- $C_{i,min}$: minimum medium conductance for $i^{th}$ valve
- $C_{i,max}$: maximum medium conductance for $i^{th}$ valve, coincides with valve conductance
- $d_{ci}$: duty cycle of command signal for $i^{th}$ valve
- $d_{b,0}$: upper limit of dead band around zero in the medium conductance vs. duty-cycle characteristics for $i^{th}$ valve
- $d_{b,100}$: lower limit of dead band around 100% in the medium conductance vs. duty-cycle characteristics for $i^{th}$ valve

INTRODUCTION
Pneumatic actuation can be a good solution wherever a need for low cost, clean and reliable power transmission is felt and system stiffness is not a major requirement. Latest developments in mechatronics led to the possibility of applying pneumatic technology even for the realisation of servosystems. In this field, the use of modulating digital valves is a technique which provide a valuable alternative to the use of flow or pressure proportional valves. The main advantage of this choice is the low cost of digital valves, which are usually much cheaper then proportional devices. On the other side the realisation of effective servosystems can require computationally expensive algorithms to compensate non lineairities, and hence the use of a digital microcontroller is strictly necessary. The disposability of a good computing power for the controller can permit the application of advanced algorithms with the goal of increasing the system precision and bandwidth. Several authors faced the problem in the last years: Mennetjel\textsuperscript{1} proposed the use of a special designed valve; Ferraresi\textsuperscript{12}, Bobrow and Jabbar\textsuperscript{13} McDonnel and Bobrow\textsuperscript{4,5,6} Gorce and Guillard\textsuperscript{7,8} Richard and Scavarda\textsuperscript{9}, Tang\textsuperscript{10}, Pandian\textsuperscript{11}, Nortsugu\textsuperscript{12} and Richeri\textsuperscript{13,14} investigated the capability of expanding the bandwith of pneumatic force and position controlled servoaaxis applying different kind of control technique. They all agree that to obtain good performance sophisticated algorithm are required.

The use of digital valves for servosystems is analysed in \cite{15,16,17,19}, while \cite{19} and \cite{20} show that fuzzy logic can be the optimal instrument to manage the nonlinearity of such systems.

This work tackles the problem of broadening the bandwith of pneumatic servosystems with modulating digital valves. In general, good operation of these systems is determined by the high speed of the valves used. This requirement means that the mobile mass of the poppet is low and therefore makes size limitation compulsory. The solution proposed envisages adjusting the air flow by a pair of valves, one specially designed for this type of application and therefore distinguished by high speed and small size, and the other of a standard type with a response time of around 10 ms and large size. The latter valve can make it possible to manage high flows when necessary and can therefore allow high performance to be obtained. This paper describes results obtained in the pressure control of a tank and a possible algorithm for valve coordination.
SYSTEM DESCRIPTION

Figure 1 shows the layout of the experimental system.

![Figure 1 - System layout](image)

Flow directed from air supply to capacity (1) is controlled by a small size, high frequency digital valve (2) and by a large size, low frequency digital valve (3). Valves (4) and (5) are identical to valves (2) and (3) respectively and they control exhaust flow from the capacity. A transducer (6) measures pressure in the tank. Finally a PC (7) equipped with suitable I/O boards perform the control algorithm receiving the pressure value and providing the command signal for all the valves. An amplifier (8) converts logical signal from the PC into power signal. Table 1 provides a description of the main characteristics of all the components.

<table>
<thead>
<tr>
<th>Component</th>
<th>Main characteristics</th>
</tr>
</thead>
<tbody>
<tr>
<td>Valves 2,3 (type 1)</td>
<td>Conductance 1 $10^{-8}$ m$^3$/sPa ANR</td>
</tr>
<tr>
<td></td>
<td>Response time 1 ms</td>
</tr>
<tr>
<td></td>
<td>PWM carrier freq. 50 Hz</td>
</tr>
<tr>
<td>Valves 4,5 (type 2)</td>
<td>Conductance 2.3 $10^{-8}$ m$^3$/sPa</td>
</tr>
<tr>
<td></td>
<td>ANR</td>
</tr>
<tr>
<td></td>
<td>Response time 20 ms</td>
</tr>
<tr>
<td></td>
<td>PWM carrier freq. 10 Hz</td>
</tr>
<tr>
<td>Pressure transducer</td>
<td>Range 0 - 1 MPa</td>
</tr>
<tr>
<td></td>
<td>Cut-off freq. 2500 Hz</td>
</tr>
<tr>
<td>Tank</td>
<td>Volume: from 1.8 $10^{-3}$ to 8 $10^{-3}$ m$^3$</td>
</tr>
<tr>
<td>PC and I/O boards</td>
<td>Pentium 2</td>
</tr>
<tr>
<td></td>
<td>N.I. AT-MIO-16E I/O board</td>
</tr>
<tr>
<td></td>
<td>N.I. PC-TIO-10 output board</td>
</tr>
</tbody>
</table>

Table 1 - system components

CONTROL ALGORITHM

Figure 2 shows a block diagram of the controlled system.

![Figure 2 - Block diagram of the system](image)

The applied control algorithm can be divided up in two parts: the first one computes a desired global conductance value for the valve set (1,2,3,4), while the second one computes actual command signal to realise the desired conductance value.

The high level controller is composed by a simple proportional controller whose gain $K_p$ is continuously changed by a fuzzy supervisor. The supervisor computes the optimal proportional gain according to the pressure error $P_{set}-P$ and to the value of reference pressure. The rules and the membership functions for inputs "error" and "$P_{set}$" were defined according to optimal values experimentally obtained in a series of tests with different constant values of the proportional gain $K_p$.

The low level controller must generate the pulse signal for the valves in order to realise the conductance value computed by the high level controller. PWM modulation technique was applied as it allows continuous conductance management. Then the conductance is the sum of the mean conductance of each valve.

It must be stated that the low level controller operates in open loop and that it must face the presence of dead bands in the duty-cycle vs. conductance relationship. The actual flow rate depends on the upstream and downstream pressures, and this is the main reason for which a continuous proportional gain adjusting is needed.

The simultaneous use of valves with different dynamic characteristics causes the need of using different PWM carrier frequencies for them. This involves that the signal update frequency must be equal to the lowest PWM carrier frequency; in order to obtain good system performance it is necessary to update the command signal for the faster valve with the frequency of its carrier frequency. This means that at each update of the faster valve command signal the global valves conductance shows a variable saturation value which is determined by the command signal for the slower valve. Particularly the lower saturation value coincides with the mean conductance of the larger valve, while
the upper limit is defined by the sum of the lower one and the maximum conductance of the faster valve. In order to maximise the system performance it is than necessary to optimise the capability of updating the mean conductance of the valve set. The proposed algorithm computes the desired medium conductance for each valve according to the following:

\[
\begin{align*}
C_i &= \begin{cases} 
0 & C \leq C_{i,\min} \\
C & C_{i,\min} < C < C_{i,\min} + C_{j,\max} \\
\frac{2}{C_{i,\max}} & C_{i,\min} + C_{j,\max} < C < C_{i,\max} \\
\frac{2}{C_{i,\min}} & \end{cases} \\
\end{align*}
\]

The algorithm must change according to the actual value of the desired total conductance in order to accomplish the presence of dead bands in the duty cycle - medium conductance characteristic of each valve. The duty signal for the command signal of each valve is then computed considering a linear characteristic defined as

\[
dc_i = \frac{C_{i,\max} - C_{i,\min}}{db_{i,100} - db_{i,0}} dc_{i,0} + C_{i,\min}
\]

This expressions show how important is the precise knowledge of the valve characteristics. The width of dead-bands depends on the ratio between the valve response time and the period of the duty cycle modulation; usually it is possible to define this last parameter in order to obtain a dead-band not larger than 20%. The response time is even the most important parameter influencing the value of the minimum medium conductance of the valve.

**EXPERIMENTS**

Experiments were carried out considering a sinusoidal input reference for pressure with different values of width and mean value. Frequency was increased from 0.1 Hz up to the limit of the system bandwidth in order to measure the frequency response in different conditions. Tests were carried in two different system configuration:
- traditional, i.e. with a couple of type 1 valves, as defined in table 1
- innovative, with valves of type 1 and 2 coupled and driven as previously described.

Different capacity air tanks were considered. Table 2 summarises the exerted tests.

<table>
<thead>
<tr>
<th>Mean pressure [MPa]</th>
<th>Amplitude [MPa]</th>
<th>Tank size [m³]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.1</td>
<td>0.05</td>
<td>0.0018</td>
</tr>
<tr>
<td>0.2</td>
<td>0.05; 0.1</td>
<td>0.0044</td>
</tr>
<tr>
<td>0.3</td>
<td>0.05; 0.1; 0.15</td>
<td>0.008</td>
</tr>
</tbody>
</table>

Table 2 - exerted tests

In a first step the proportional gain was manually optimised for each test condition. The obtained value were then used to define a fuzzy logic algorithm which computes the optimal gain according to pressure error and to the actual pressure value. The need for tuning according to the actual pressure value is due to the non linearity in the relationship between conductance, flow and pressure drop.

The Bode diagram were then experimentally measured for each considered configuration to verify the obtained performance increase. The next figures show some examples of the obtained results.

Particularly, figures 3 and 4 show the time history of reference and measured pressure in test carried out with a 0.008 m³ tank, with reference pressure oscillating around 0.3 ± 0.05 Mpa. Fig. 3 refers to the use of valve of type 1 only, while figure 4 refers to the combined use of both types of valves.
Comparing figures 3 and 4 it can be noticed that the presence of valve of type 2 causes two important effects: first, response amplitude at 2 Hz is more than double than in the case in which only valve 1 is used; then, when only valve of type 1 is used, the flow shows saturation and pressure follows a triangular profile instead of the required sinusoidal one. On the contrary, the introduction of valves of type 2 avoids flow to be saturated.

The data visualised in figures 3 and 4 are summarised in figure 5 which reports the measured Bode diagram in both the configuration.

The dotted plot refers to the use of valve of type 1 only, while the continuous line refers to the combined use of both the valves. Introducing valves of type 2 the bandwidth increases from 1 Hz up to 2.2 Hz, while the phase delay is comparable in the two cases. Similar results were obtained with different value of medium pressure and for wider reference sinusoids.

The following figures show results obtained with a smaller tank (1.8 \(10^{-3}\) m\(^3\)) and with a larger one (8 \(10^{-3}\) m\(^3\)).
In the first case the presence of valves of type 2 reduces the system damping at frequencies larger than 1 Hz, but it causes the wave shape to be rather irregular because of the closeness between the frequency of the reference signal and that of the PWM modulation of type 2 valve.

Figure 8 shows the Bode diagram obtained from experimental tests.

Comparing the plots, a bandwidth increase of about 0.8 Hz is found, on a basis value of 1.5 Hz, while the phase delay plots is translated of about 0.3 Hz.

Finally figures from 9 to 11 show results of tests carried out with the 8 \(10^{-3}\) m\(^3\) tank.

As it could be expected, the figures show that when the tank size grows valves of type 1 alone can not provide satisfactory bandwidth, nor they can allow a good reference pressure wave shape reproduction. In fact the flow rate is saturated and pressure begins following a triangular wave when the reference pressure frequency is 0.5 Hz.

On the contrary, with both the valves the response is satisfactory for amplitude and shape up to 2 Hz.

As regards the bandwidth, it can be seen for the Bode diagram of figure 9 that it increases from 0.45 Hz up to 1.5 Hz.
CONCLUSIONS

The work shows that dynamic performance of pneumatic servosystems with digital valves can be increased by coupling specially designed fast and low size valves with standard slower and larger valves. An algorithm which maximise the command signal update frequency was developed and applied, and it demonstrated that satisfactory precision in reference following could be obtained together with an important increase in the bandwidth. The method can hence be applied to any kind of pneumatic servosystems.

REFERENCES