Development of a Fault Tolerant Actuation System-Modelling and Validation

K.S.Grewal [A,], R. Dixon [A], J. Pearson [B]
[A] Control Systems Group, Loughborough University. Loughborough. Leicestershire. UK.
[B] SEIC, BAE Systems, Honeywell Park, Loughborough. Leicestershire. UK.

Abstract:
It is generally accepted that incorporating so-called 'smart' control and monitoring technologies can improve the reliability and availability of industrial systems. 'Smart' control can be defined as making full use of all the measured, inferred and a priori information that is available from a system. In general terms, the idea is that system level knowledge can be developed and used to check sensors for problems, to detect and identify faults as they develop and, where appropriate, to re-configure the controller(s) to accommodate plant or sensor faults until repair can be effected. To-date success, in terms of real industrial applications of the more advanced techniques, has been limited. Hence, demonstrators are needed. The work described in this paper is part of an on going project aimed at demonstrating these "smart" concepts on a Stewart-Gough platform comprising six pneumatic actuators. To-date the research has focussed on specifying the demonstrator system and developing and validating models of the pneumatic system. This is probably the most important step in designing a fault tolerant actuation system – as the model is the foundation of the other algorithms.

Keywords: pneumatic, modelling, validation

Introduction

Pneumatic actuators are often used in industrial applications. Such applications include robots and manipulators, welding and riveting machines, pick-and-place devices, vehicles, and in many other types of equipment. The reasons associated for their use are good power/weight ratio, ease of maintenance, cleanliness, and having a readily available and cheap power source [1].

The first attempts to analyse pneumatic control systems was reported by Shearer (1956) [2]. This was further extended by Burrows (1969) [3], and Scavarda *et al* (1987) [4]. Who proposed two linearized state space models of a non-linear pneumatic system: One describes the behaviour of the system about a constant speed steady state and the other is valid around the equilibrium position rather than only at the central position. Using approximations of the model, allows the use of a restricted range of the optimum parameters that are selected with classical methods (Chillari *et al,* 2001) [5]. Also see for example (Kaitwanidvilai and Parnichkun, 2005 [6]; Lee *et al,* 2002 [7]; Hamiti *et al* 1996 [8]). In this paper, a model is derived based on a single pneumatic actuator set-up. The derived model is then validated against the actual system and the results are compared. In the following sections, the experimental set-up is described. After this, the model of the pneumatic system is formulated. Then the derived model is validated against the actual pneumatic system.

Experimental set-up

The experimental set-up is illustrated in Figure 1 and 2. The set-up shows the xPC Target coupled with Matlab/Simulink®, which provides a real-time environment. A host and a target computer are connected using a TCP/IP network**.** Matlab/Simulink® is run on the host computer, this is where the system is designed using xPC target I/O blocks. Using external mode the system file is built and compiled within the host computer. Then downloaded to the target computer where it is executed using the real-time kernel. PCI cards are used to send and receive signals between the target and the system. For this work, a Bimba double acting pneumatic cylinder and a Festo five port proportional valve is used. The position signal is measured via a Linear Resistive Transducer (LRT) mounted in the cylinder rear section. The acceleration signal is acquired using an iMEMS® accelerometer mounted on the end of the piston rod.

Figure 1*: The experimental test rig*

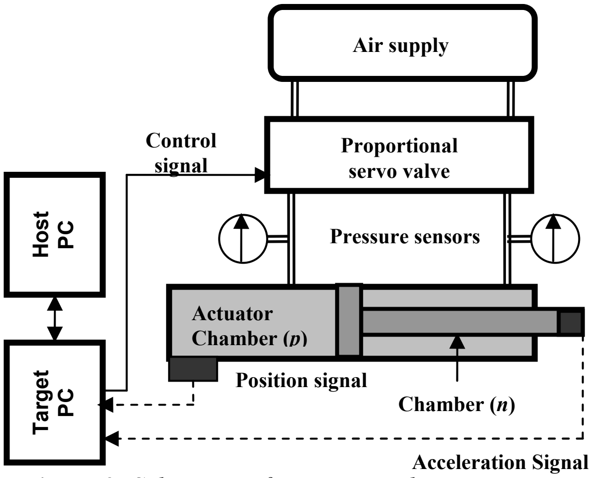

***Figure 2**: Schematic of experimental set-up*

Modelling pneumatic actuator system

In order to model an approximate linear transfer function, describing the dynamics of the pneumatic system shown in Figure 2. The thermodynamic analysis of the system is initially presented. The subsequent description model is comparable to that which is presented in (Kaitwanidvilai and Parnichkun, 2005 [6]; Lee *et al,* 2002 [7]; Hamiti *et al* 1996 [8]). It is assumed that the system undergoes an adiabatic process (the rate of heat exchange through the system boundary is ignored).

The dynamic model derived is developed based on the relationship between (i) the air mass flow rate and the pressure changes in the cylinder chambers, and (ii) the equilibrium of the forces acting at the piston, including the friction forces. A block representation of the pneumatic model is shown in Figure 3. Certain assumptions are considered for the construction of the model these include:

- The air is a perfect gas.
- Homogeneous (uniform) pressure and temperature in both chambers.
- Supply pressure variation not considered.
- Temperature variation not considered.
- Air loss is not considered.
- The length and dimensions of the feeding pipes are neglected.

Valve model

From Lee *et al,* 2002 [7]; the following equation can express the mass flow rate through an orifice

$$\dot{m} = A_c \lambda_2 \frac{P_u}{\sqrt{RT_s}} f\left(\frac{P_d}{P_u}\right) \qquad (1)$$

Where $\dot{m}$, P_u, P_d, R and T_s are the mass flow rate, pressures at the input and output ports (upstream and down stream), the gas constant and the absolute temperature respectively. A_c is the effective area of the valve orifice, which changes according to spool position. In Equation (1) the flow function f has the following expression:

$$f\left(\frac{P_d}{P_u}\right) = \begin{cases} \frac{\lambda_1}{\lambda_2}\sqrt{(P_r)^{2/\gamma} - (P_r)^{(\gamma+1)/\gamma}}, & P_r > P_{Crit,} \\ 1, & P_r < P_{Crit,} \end{cases} \qquad (2)$$

With
$P_r = P_d / P_u$

Where γ is the ratio of specific heat (air: 1.4) and P_{Crit} is the critical pressure ratio having the following expression:

$$P_{Crit} = \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}} = 0.528 \qquad (3)$$

For sonic and subsonic cases, where λ_1 and λ_2 are the constants are given by

$$\lambda_1 = \sqrt{\frac{2\gamma}{\gamma-1}} = 2.645\,, \qquad (4)$$

$$\lambda_2 = \sqrt{\gamma\left(\frac{2}{\gamma+1}\right)^{(\gamma+1)/(\gamma-1)}} = 0.684\,, \qquad (5)$$

Cylinder model

The following equation is applicable to each of the cylinder chambers, assuming isentropic (without change in entropy) behaviour of air.

$$P\left[\frac{V}{m}\right]^{\gamma} = \text{Constant} \qquad (6)$$

Where P, V and m are pressure, volume, and mass of air in cylinder. Differentiating equation (6) with respect to time gives:

$$\dot{P}V + \gamma P\dot{V} = \left[\frac{\dot{m}}{m}\right]PV \qquad (7)$$

Using equation (7) and the ideal gas law

$$PV = mRT_s \qquad (8)$$

A relationship between cylinder pressure and mass flow rate into the cylinder is obtained

$$\dot{P}V + \gamma P\dot{V} = \gamma \dot{m} RT_s \quad (9)$$

Then the relationship between the mass flow rate of air and the change of both pressure and volume in chambers be written as:

$$\dot{m}_p = \frac{V_p}{\gamma RT_s}\frac{dP_p}{dt} + \frac{P_p}{RT_s}\frac{dV_p}{dt} \quad (10)$$

$$\dot{m}_n = \frac{V_n}{\gamma RT_n}\frac{dP_n}{dt} + \frac{P_n}{RT_s}\frac{dV_n}{dt} \quad (11)$$

Subscripts p and n are the actuator chambers, respectively. $\dot{m}_p$ is the mass flow rate into chamber p, and $\dot{m}_n$ is the mass flow rate into chamber n. V_p is the air volume in chamber p, V_n is the air volume in chamber in n, P_p is the pressure in chamber p, P_n is the pressure in chamber n. T_s is the temperature.

The dynamics of the cylinder motion can be described by:

$$M\ddot{x} + F_f\dot{x} = A(P_p - P_n) = A\Delta P \quad (12)$$

Where M is the piston mass, x is the position of the piston, A is the bore area, F_f represents the viscous friction coefficient and coulomb friction force.

To make the system linear, a small deviation from an initial equilibrium point is considered. Equation (10)-(12) can be written in linearized form as:

$$\Delta\dot{m} = \frac{V_{p0,n0}}{\gamma RT_s}\Delta\dot{P}_{p,n} + \frac{P_{p0,n0}}{RT_s}\Delta\dot{V}_{p,n} \quad (13)$$

$$M\Delta\ddot{x} + F_f\Delta\dot{x} = A(\Delta P_p - \Delta P_n) = A\Delta P \quad (14)$$

Where Δ denotes a perturbation from the operating point. The values of the state variables can be defined by ($x=0$, $P_p=P_{p0}$, $P_n=P_{n0}$, $V_p=V_{p0}$ and $V_n=V_{n0}$).

The mass flow rate is identical (in magnitude) for both chambers and is proportional to the valve input voltage. Hence

$$\Delta\dot{m}_p = K\Delta v \quad and \quad \Delta\dot{m}_n = -K\Delta v \quad (15)$$

Where K is the servo valve constant (Kg.s^{-1}.v^{-1}) determined from the valve's data-sheet.

With the assumption of incompressibility the rate of change of volumes can be written as

$$\Delta\dot{V}_p = A\Delta\dot{x} \quad and \quad \Delta\dot{V}_n = -A\Delta\dot{x} \quad (16)$$

Substituting equation (15) and (16) into equation (13), then rearranging the equations for chambers p and n gives:

$$\Delta\dot{P}_p = -\frac{\gamma AP_{p0}}{V_{p0}}\Delta\dot{x} + K\frac{\gamma RT_s}{V_{p0}}\Delta v \quad (17)$$

$$\Delta\dot{P}_n = \frac{\gamma AP_{n0}}{V_{n0}}\Delta\dot{x} - K\frac{\gamma RT_s}{V_{n0}}\Delta v \quad (18)$$

Then rearranging equation (14) gives:

$$\Delta\ddot{x} = \frac{A}{M}(\Delta P_p - \Delta P_n) - \frac{F_f}{M}\Delta\dot{x} \quad (19)$$

Equations (17), (18) and (19) can be represented in state space (see equation 20) or block diagram (see Figure 3) form.

$$\dot{X}\begin{bmatrix}\Delta\dot{P}_p\\ \Delta\dot{P}_n\\ \Delta\dot{x}\\ \Delta\ddot{x}\end{bmatrix} = \begin{bmatrix}0 & 0 & 0 & -\frac{\gamma AP_{p0}}{V_{p0}}\\ 0 & 0 & 0 & \frac{\gamma AP_{n0}}{V_{n0}}\\ 0 & 0 & 0 & 1\\ \frac{A}{M} & \frac{-A}{M} & 0 & \frac{-F_f}{M}\end{bmatrix}\begin{bmatrix}\Delta P_p\\ \Delta P_n\\ \Delta x\\ \Delta\dot{x}\end{bmatrix} + \begin{bmatrix}K\frac{\gamma RT_s}{V_{p0}}\\ -K\frac{\gamma RT_s}{V_{n0}}\\ 0\\ 0\end{bmatrix}\Delta v$$

$$y = [0 \quad 0 \quad 0 \quad 1]\begin{bmatrix}\Delta P_p\\ \Delta P_n\\ \Delta x\\ \Delta\dot{x}\end{bmatrix} + [0]\Delta v \quad (20)$$

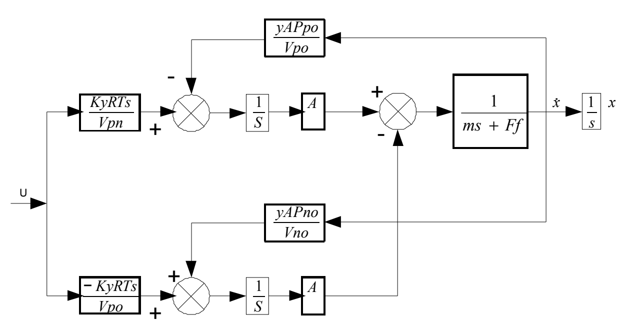

***Figure 3**: Block representation of the pneumatic System*

Model validation

In order to validate the model a number of experiments have been carried out on the open-loop actuator. The results have then been compared with those from simulation. Test inputs have included square and sine wave. A typical set of results for a square wave input is shown in Figure 4. Here, the square wave input is set at 0.6 volts and the frequency set at 0.5Hz, and the position and the pressure output responses are plotted alongside those predicted by the model. The simulation results show reasonable agreement with those from the experiment. The position results show particularly a good match, whilst those for the two cylinders pressures capture the dominant response, though there is clearly some longer term mode that is not represented in the model. These may well be due to non-linearities associated with pneumatic systems that are not captured in the model. It should be noted that as position control is the overall objective this is the key response which needs to be correct.

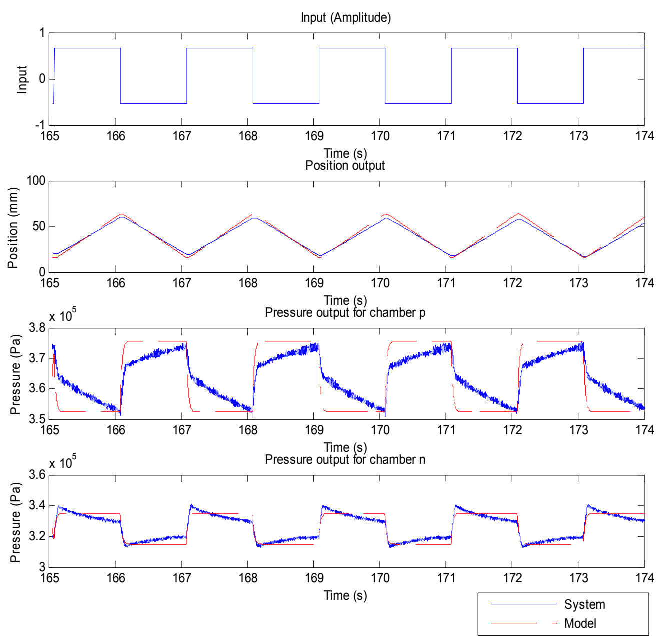

Figure 4: *The comparisons between the system and model outputs for a square wave input.*

Conclusion

The paper has described a model of the pneumatic actuation system. The model consists of two main sections, namely, the valve model and the cylinder model.

The model was configured to represent a real actuator and experiments were performed in order to validate the model against the actual system. Comparison of the simulation outputs revealed the model is a valid representation of the actual pneumatic system. The derived model is intended to be used as the foundation for future work. This will include design and synthesis of a control strategy. Using 'smart' control incorporated within a fault tolerant control system.

Acknowledgements

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