

Simulation for rapid prototyping: Development of control system for a pneumatic servo actuator using Simulink

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Abstract

Research and development of control systems for servo applications are a wide area of interest. Controller design tends more and more towards the use of advanced, nonlinear control technics. This paper highlights the advantages and benefits of doing a thorough simulation study and of using rapid prototyping tools as an integrated part in the design of servo applications. The mathematical modelling of an pneumatic clutch actuation servo is presented, together with illustrations of implementation of the model in the Matlab/Simulink simulation software. Some characteristic experimental results are shown, verifying the validity of the mathematical model.

Keywords: servo system design, pneumatic servo actuator, modelling, simulation, control, automatic code generation

1 Introduction

In industry there are a variety of areas where mechanical servo systems¹ are extensively used, e.g. mining, aircraft and spacecraft control, automotive, cargo handling and offshore to mention some major industries. Hence, research and development of control systems for servo applications are a wide area of interest. During the past few years there has been an increased focus on time and cost effectiveness in the development of control systems. The advantages of doing a thorough simulation study when designing a control system, has been increasingly appreciated by industry since it reduces both development time and cost when exploited properly.

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¹The term *servo system* is in this paper used merely as a general name of a mechanical positioning actuator.

Especially the introduction of automatic code generation software—also known as *rapid prototyping*—dramatically reduce time from when a simulation model is available, to the implementation of the developed control system. Figure 1 illustrates the automatic code generation principle, e.g. code generation from the Simulink implemented controller and downloading of the code to the controller.

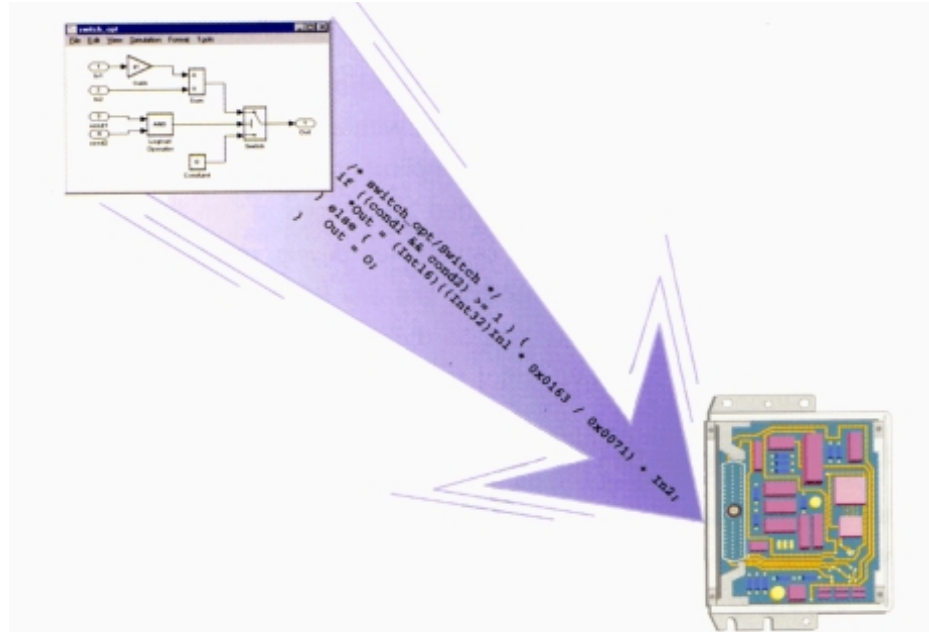


Figure 1: Illustration of automatic code generation and downloading from the controller implemented in Simulink, to the electronic control unit which is the actual controller of the system.

Traditionally, linear control theory and frequency domain methods are used in the design of control systems for servo applications, and has a long history of successful industrial applications. However, the development of servo systems are tending more and more towards the use of advanced, nonlinear control techniques. The increasing access to low-cost, high-speed computation, makes it cheap to implement advanced control algorithms. Physical systems are inherently nonlinear and for many systems—among them the pneumatic clutch actuator discussed in this paper—the desired performance may only be possible using nonlinear control techniques. Nonlinear control might also make it possible to replace expensive linear actuators² with low-cost, nonlinear actuators etc.

Selected references on the modeling and conventional, linear control of dynamic systems are given in [Seborg et al. 1989], [Haugen 1994] and [Egeland 1993]. Recent work on advanced controller design for pneumatic robot manipulators are presented

²The word *actuator* is in this context referring to the general name of the control admission of a system, i.e. thrusters on a rocket, valves on a hydraulic motor etc.

in [Pandian et al. 1997] and [Bouri et al. 1994]. [Drakunov et al. 1995] is an interesting paper on the design of an automotive Anti-Break System illustrating implementation of a sliding mode controller with friction force observer on a hydraulic system. References to the use of several different simulation software are given in [International Workshop on Computer Software for Design, Analysis and Control of Fluid Power Systems 1999].

The contribution of this paper is a case study that illustrates the use of simulation and rapid prototyping in the development of control systems for servo applications. The application concerned is a pneumatic clutch actuation servo, designed to be used in an automotive gear shift system on heavy duty trucks. Emphasis are made on the modeling part of the design process and the implementation of the simulation model in Simulink, and the control algorithm is left out. Results from experimental tests are included to show how well the model represents the physical system. More thorough documentation of the system and controller design of the pneumatic servo application are given in [Kaasa 1999a], [Kaasa 1999b] and [Kaasa 2000].

An introduction to control system design for servo systems in general are given in section 2 *Control system design*. The case study application is described in section 3 *Description of the system*. The mathematical model is described in section 4 *Mathematical model*, which is divided into the subsections *Dynamic force balance*, *Pressure dynamics*, *Valve flow characteristics and actuator leakage* and *Valve dynamics*. Section 5 *Implementation of model in Simulink* gives a brief description of the implementation in Simulink. Finally, the validation of the model is presented in section 6 *Experimental results*, and conclusions in 7 *Conclusions*.

2 Control system design

A typical development sequence in servo system design consists of the following phases:

- Model development.
- Implementation of the model in a simulation tool.
- Development of the control system.
- Simulating the system.
- Deploying the control algorithm (tested by simulations) to a prototype system consisting of servo hardware and micro-processor controller.
- Development of final control algorithm in e.g. C, and putting the system together in a production process.

2.1 Controller design

In the cases where the system includes nonlinear elements, a linear approximation of the system may not be a sufficient representation of the actual system. This is the case if the system for instance includes so-called hard nonlinearities³ which often are found in control engineering, or when the system is subjected to nonlinear control. In most cases, analytic solutions of system responses do not exist, hence in order to verify robustness, effects of disturbances and the general performance of the system, simulations are required.

The control algorithm used on the clutch actuator is a third order sliding mode controller, hence, the controller is depending on feedback of position, velocity and acceleration. However, only a position measurement is available on the system, and the other two states have to be estimated by a state observer⁴.

2.2 Advantages of a thorough simulation study

The implementation of a well defined simulation model usually requires in-depth knowledge of modeling and simulation. For many companies—especially smaller companies with few research resources available—this represents a relatively high threshold to overcome in order to take simulations in use in their design process. However, there are several advantages that follow from doing a thorough simulation study.

The number of experimental tests could be minimized, and the process of "trial and error" on prototypes are avoided. Besides, having a well defined simulation model makes the testing of different design alternatives less expensive and time-consuming, and helps decide on which design alternative to produce.

2.3 Rapid software prototyping

Conventional software development is done manually from the specifications of the control system and involves a time-consuming process of programming, testing of code and code review. Using the programming environment *dSpace* and the simulation software *Matlab/Simulink* together with a specialized electronic control unit (ECU), it is possible to automatically generate and download the controller algorithm to a prototype servo to be installed for testing in the truck.

³Hard-nonlinearities denotes effects that are discontinuous of nature such as Coulomb friction, saturation, dead-zones, backlash and hysteresis.

⁴Obtaining the acceleration by double numerical differentiation of the position signal require filtering and introduces a time-delay, which degrades the tracking performance considerably.

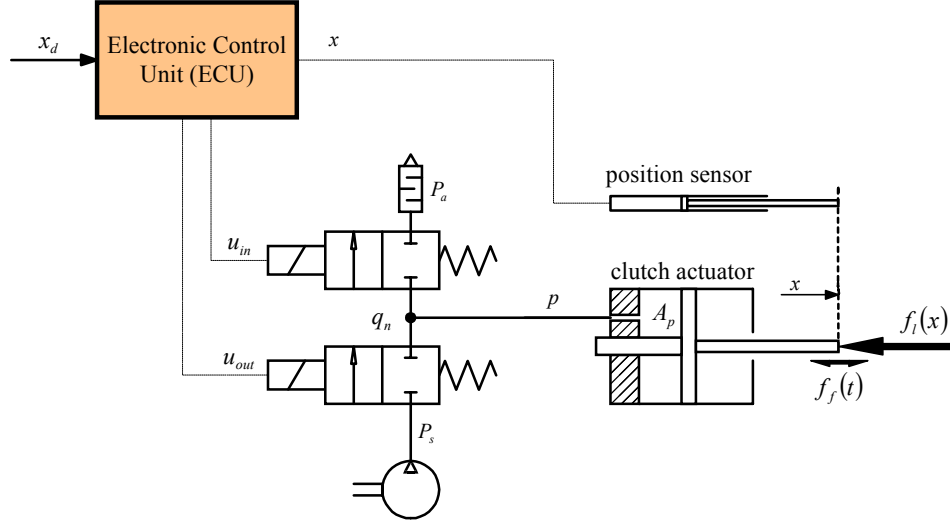


Figure 2: Simplified schematic of the clutch actuator servo system.

3 Description of the system

A simplified schematic of the actuator servo system is given in Figure 2. The actuator position x is controlled by operating supply and exhaust valves to give the necessary pressure p acting against the piston to balance the load force f_l , and compensate for the friction forces f_f .

The operating requirements for the clutch servo yields extreme specifications for the tracking capabilities and robustness of the controller. The pneumatic clutch servo is to substitute the conventional clutch actuation system in the truck where the clutch disc position x is mechanically operated⁵ by the clutch pedal position x_d . The servo has to perform the tracking of the demand signal x_d without any significant reduction in performance and comfort for the driver, compared to conventional mechanical clutch operation.

The tracking performance must be robust to a wide range of variations in supply pressure P_s , temperature T , and changes in clutch release spring force characteristics $f_l(x)$ due to wear of the clutch. And the controller must eliminate the effect of the relatively high friction forces in the system.

4 Mathematical model

The modeling is naturally divided into four steps:

- First the dynamic force balance of the actuator piston is derived by applying

⁵Using either a wire or a hydraulic transmission line

the linear momentum law. The forces that act on the piston are identified, while the pressure p is regarded as control input.

- The pressure dynamics is derived from the mass balance of the pressurized chamber, and the proper thermodynamic process and properties are determined with mass flow $\dot{m}$ as control input.
- The valve flow characteristics and actuator leakage model constitute the net mass flow $\dot{m}$, and is derived from compressible fluid dynamics.
- Finally a model of the valve dynamics is needed.

4.1 Dynamic force balance

The resulting force acting on the actuator piston is composed of the load force f_l^6 , the total friction force f_f and the actuator force $A_p \cdot (p - P_a)$ due to the pressure difference over the piston. The *dynamic force balance* is then expressed as

$$M_e \ddot{x} = A_p \cdot (p - P_a) - f_l(x) - f_f(t) \quad (1)$$

where M_e is the equivalent mass of moving parts, p the pneumatic chamber pressure, P_a the atmospheric pressure, f_l the static load characteristics given in Figure 3, and f_f the total friction force given in Equation 2 below.

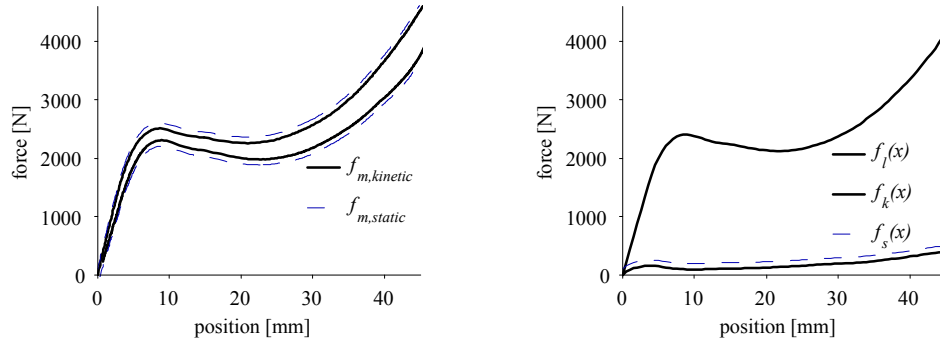


Figure 3: The left plot shows the measured *static* and *kinetic* load forces during one extension and subsequent retraction of the actuator. The right plot shows the mean load characteristics $f_l(x)$, and the resulting static and kinetic dry friction force characteristics, $f_s(x)$ and $f_k(x)$ respectively.

⁶The load force f_l is a highly nonlinear spring which is always opposing actuator extension.

Friction forces Mechanical friction is a complicated phenomena which may be explained by three different mechanisms; static⁷ and kinetic dry friction⁸, and viscous friction⁹. The mathematical model of the *total friction force* is given by Equation 2.

$$f_f(t) = \begin{cases} f_d(x, \dot{x}) \cdot \text{sgn}(\dot{x}) + D\dot{x} & , \dot{x} \neq 0 \\ f_\Sigma(t) & , \dot{x} = 0 \text{ , } |f_\Sigma| \leq |f_d| \\ f_d(x, \dot{x}) \cdot \text{sgn}(f_\Sigma) & , \dot{x} = 0 \text{ , } |f_\Sigma| > |f_d| \end{cases} \quad (2)$$

where f_d is the dry friction characteristics of Equation 3, D the viscous damping coefficient, and f_Σ the sum of the resulting forces acting on the piston not including the dry friction force f_d .

The dry friction characteristics is expressed by

$$f_d(x, \dot{x}) = f_k(x) + [f_s(x) - f_k(x)] \cdot \exp\left(-\frac{|\dot{x}|}{V_{ref}}\right) \quad (3)$$

where f_k and f_s are the measured *kinetic* and *static* dry friction force characteristics respectively (depicted in Figure 3), and V_{ref} a reference velocity indicating the region of transition from kinetic to static dry friction. Figure 4 illustrates the principal behavior of the total friction force f_f as a function of velocity $\dot{x}$.

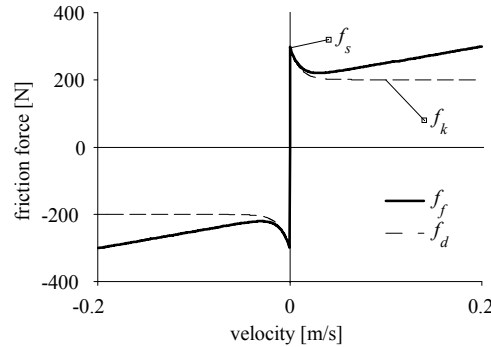


Figure 4: Principal behavior of the total friction force f_f as a function of velocity $\dot{x}$. The dashed line shows the dry friction characteristics, e.g. the viscous friction is left out.

⁷Static friction is commonly known as stiction, and is actually due to a kind of cold welding between the surfaces when there is no-or very low-relative motion between them.

⁸Kinetic dry friction is the abrasive resistance that occurs between two surfaces, also known as Coulomb friction.

⁹Viscous friction is a result of the shear stress forces of the fluid separating two surfaces.

4.2 Pressure dynamics

For the modeling of the *pressure dynamics* the following assumptions are made:

- At the concerned pressures air may be treated as an ideal gas without significant error, hence, the perfect gas relation may be used.
- The thermodynamic process of the pressurized chamber may be regarded as isentropic, which means that the losses, heat transfer and heat transport¹⁰, are small enough to be neglected.
- Dimensions of the system are relatively small, hence it is reasonable to assume that the air fluid state (p, T) is uniformly distributed throughout the system.

The pressure dynamics is derived from the mass balance of the pressurized chamber where the mass rate is expressed by

$$\frac{dm}{dt} = \dot{m} = \rho \frac{\partial v}{\partial t} + \frac{\partial \rho}{\partial t} v \quad (4)$$

where m is the total mass in the pneumatic chamber, and v and ρ the corresponding volume and density respectively. The total volume of the pneumatic chamber is given by

$$v = V_d + A_p x \quad (5)$$

where V_d is the dead volume, and A_p the effective area of actuator piston. The perfect gas relation is expressed as

$$p = \rho R T \quad (6)$$

where R is the gas constant of air and T is the system temperature. The isentropic process is governed by the relation

$$p T^{\frac{1-\kappa}{\kappa}} = P_0 T_0^{\frac{1-\kappa}{\kappa}} \quad (7)$$

where κ is the isentropic exponent¹¹ ($\kappa = 1.4$ for air), and the subindex 0 indicates initial state.

The *pressure dynamics* is then derived from the preceding equations giving

$$\frac{\partial p}{\partial t} = \dot{p} = \frac{\kappa}{V_d + A_p x} \left(R T_0 \left(\frac{P_0}{p} \right)^{\frac{1-\kappa}{\kappa}} \dot{m} - A_p \dot{x} p \right) \quad (8)$$

where $\dot{m}$ is the net mass flow into the pneumatic chamber given by Equation 9.

¹⁰At high inlet flow rates, the enthalpy difference of the inlet flow will have a significant effect on the system chamber's air state, hence the polytropic process of equation 7 tends from isentropic (adiabatic, $n = \kappa$) towards isotropic (constant temperature, $n = 1$).

¹¹Ratio of specific heats for the fluid.

4.3 Valve flow characteristics and actuator leakage

The *net mass flow* $\dot{m}$ into the pneumatic chamber is given by

$$\dot{m} = \rho_n (q_{v,in} - q_{v,out} - q_{a,leak}) \quad (9)$$

where ρ_n is fluid density related to the normal state (for air $\rho_n = 1.205 \text{ kg/m}^3$), $q_{v,in}$ and $q_{v,out}$ are volume flow¹² through the inlet and outlet valves given by the valve characteristics of Equation 10 and $q_{a,leak}$ is the leakage flow through the actuator sealings given by Equation 12.

Valve flow characteristics The *valve flow characteristics* are given by Equation

$$q_v(p_1, p_2, T_1, a_v) = \frac{C_d K \cdot a_v}{\rho_n \cdot \sqrt{T_1}} \cdot \Omega\left(\frac{p_2}{p_1}\right) \cdot p_1 \quad (10)$$

where C_d is the flow discharge coefficient, K is a fluid characteristic constant, a_v is the valve's orifice area, and Ω is the pressure ratio flow weight function depicted in Figure 5. The subindexes 1 and 2 denotes upstream (inlet) and downstream

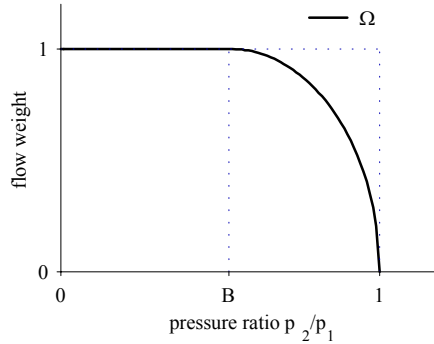


Figure 5: Graphical representation of the flow weight function Ω as a function of the pressure drop ratio $\frac{p_2}{p_1}$ over the restriction.

(outlet) of valve respectively. For the inlet flow the temperature is constant equal to the supply temperature T_s , while for the outlet flow the chamber temperature is given by Equation 7, which rearranged yields

$$T = T_0 P_0^{\frac{1-\kappa}{\kappa}} \cdot p^{\frac{\kappa-1}{\kappa}} \quad (11)$$

¹²When working with pneumatic components it is ordinary to use volume flow referred to the normal state air ($P_n = 1.013 \text{ bar}$, $T_n = 293 \text{ K}$) instead of mass flow ($\dot{m} = \rho_n \cdot q$).

Actuator leakage model The actuator leakage appear as laminar flow through the cylindrical clearance of the piston sealing. The geometry of the sealing is constructed to reduce the clearance as actuator pressure increases. Hence, the expected proportionality to the pressure drop is reduced with increase in pressure, giving the model

$$q_{a,leak} = C_a (p - P_a) \cdot \exp \left(-\frac{p}{P_{ref}} \right) \quad (12)$$

where C_a is actuator leakage coefficient, and P_{ref} is a sealing reduction parameter.

4.4 Valve dynamics

The dynamics of the on-off pilot operated valve is approximated simply by a transport time-delay from the control signal u_v to the valve opening a_v ;

$$a_v(t) = K \cdot u_v(t - T_v) \quad (13)$$

where K [m^2/V] is the orifice area constant, u_v is the on-off control signal (5 and 0 V respectively) and T_v is the transport delay (5 ms for the valves on the test rig).

5 Implementation of model in Simulink

Simulink is a block diagram based simulation tool running under Matlab, which offers a wide range of alternate methods for implementation of the system model. A major advantage is that hybrid implementations are possible, i.e. mixing discrete, continuous, transfer functions or state space subsystems in the same model.

The implementation of the clutch servo model is straightforward. Time should, however, be spent on making the model clearly set out. The possibility of using subsystems and organizing the model hierarchically should be exploited. A large model easily gets over-complex and difficult to grasp for those not currently working with the system. Grouping a model in sub-models usually also reduce the amount of redundant work since subsystems in Simulink are easily shared between models. In the clutch servo application the top level consists of two subsystems; namely the *Control system* and the *Clutch Actuator System* blocks. Figure 6 shows the opened subsystem *Clutch Actuator System* which is the Simulink implementation of the pneumatic clutch actuator model.

Figure 7 shows the *Dynamic Force balance* subsystem which implements Equation 1 of the clutch actuator model.

The simulations using Simulink may be run interactively from the menu bar. However, running simulations, generating inputs to the model and presenting the results are easily automated using Matlab scripts.

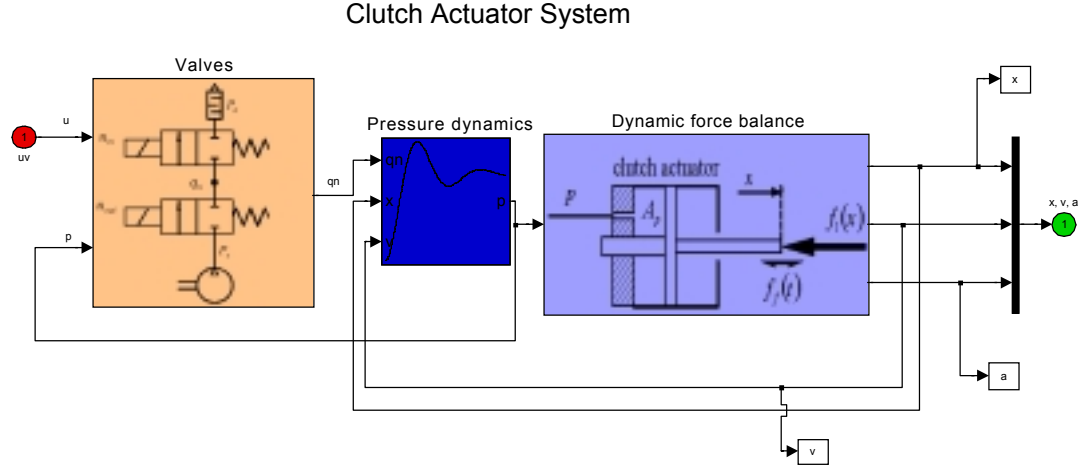


Figure 6: Simulink subsystem block diagram of the clutch actuator model. The *Valves* block includes the model of the valve dynamics and flow characteristics. The *Pressure dynamics* and *Dynamic force balance* blocks are self-explanatory.

6 Experimental results

The mathematical model of the clutch actuator is needed as a basis for the observer design, and in order to verify the clutch actuator model, it is necessary to perform open loop tests on the system.

Open loop experiments are carried out on a test rig, fitted with a position sensor, accelerometer and pressure transducer. The velocity responses, v_e and v_m , are estimated from the simulated and measured position, x and x_m , by numerical differentiation.

Several experiments have been conducted, and results from a representative experiment are briefly analyzed in this section.

The control input signal u_v – controlling the valves – are manually operated from the computer, producing an extending and retracting movement of the actuator. The right-hand side axis indicates the value of the control input signal, where +5 V operates the inlet valve, and -5 V operates the outlet valve. Figure 8, 9, 10 and 11 shows the responses in position, velocity, acceleration and pressure over a period of 1.4 s when subjected to a step input to the outlet valve. The presented experiment is not sufficient to validate the model, but reveals the highly nonlinear properties of the pneumatic system.

The shown responses indicate that the model is an adequate representation of the actual system, which is also verified by a large number of additional tests. The level of inaccuracy is within tolerable limits, i.e. the model and simulations can be used to predict the significant behavior of the physical system when subjected to

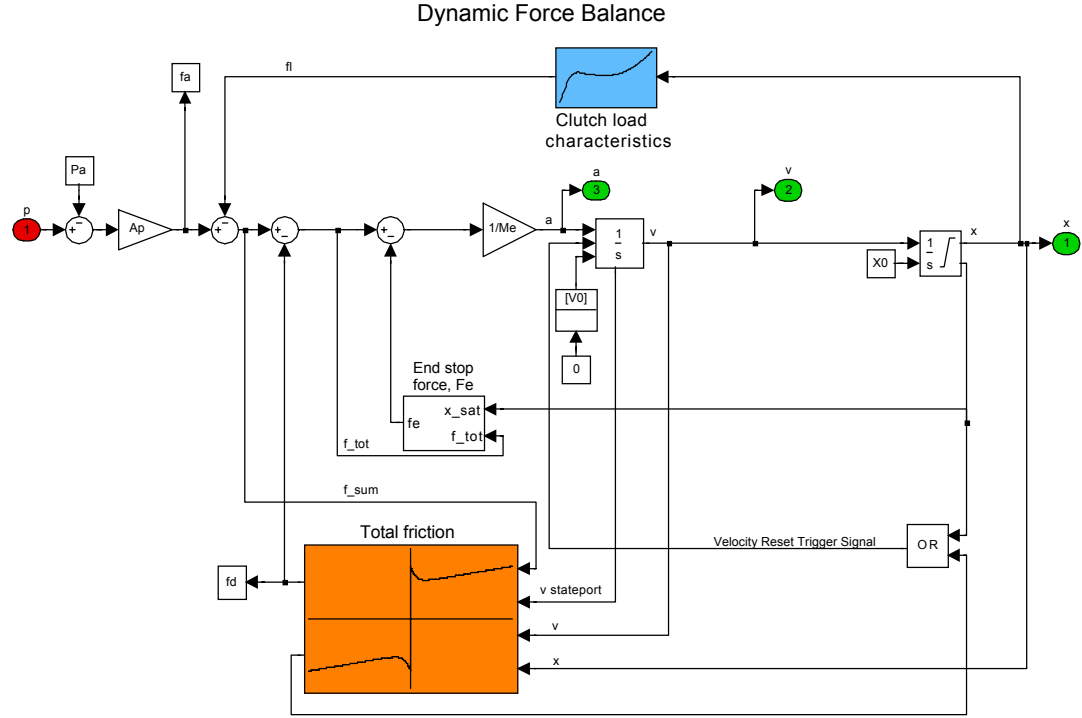


Figure 7: Simulink subsystem block of the *Dynamic force balance* model. The *Total friction* represents the mathematical model of Equation 2, while the *Clutch load characteristics* block implements the measured load characteristics using a look-up table.

closed loop control.

7 Conclusions

A main objective of the presented paper has been to highlight the advantages and benefits of doing a thorough simulation study and of using rapid prototyping tools as an integrated part in the design of servo applications. The mathematical modeling of a pneumatic clutch actuator is presented, together with illustrations of implementation of the model in the Matlab/Simulink simulation software. Some characteristic test results are shown, indicating the validity of the mathematical model and illustrating the strongly nonlinear behavior of the system.

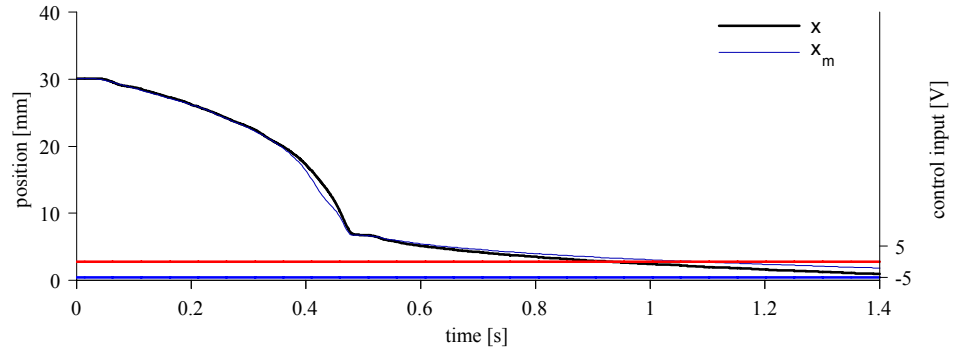


Figure 8: Responses in the simulated and measured positions, x and x_m , when subjected to a step input to the outlet valve.

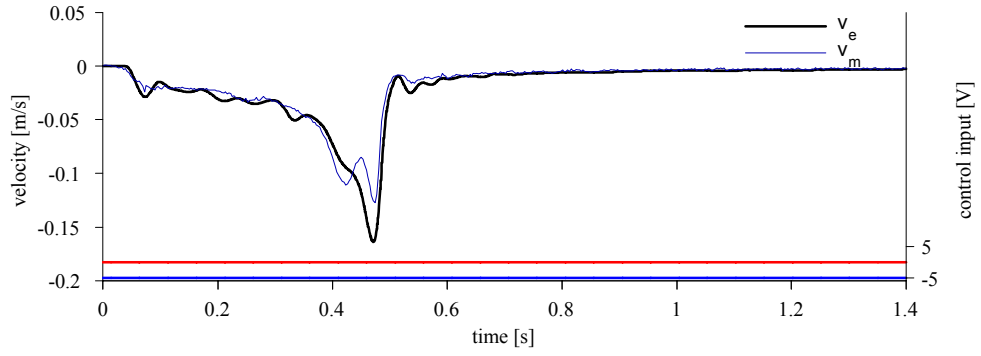


Figure 9: Simulated and measured responses in v_e and v_m when subjected to a step input to the outlet valve.

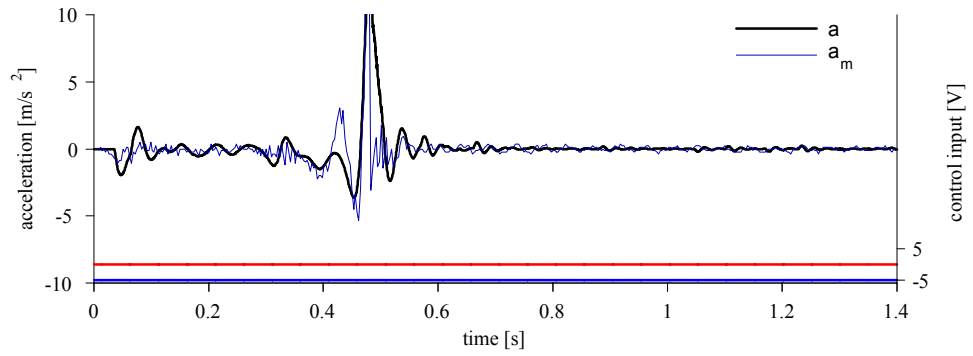


Figure 10: Responses in simulated and measured acceleration, a and a_m , when subjected to a step input to the outlet valve.

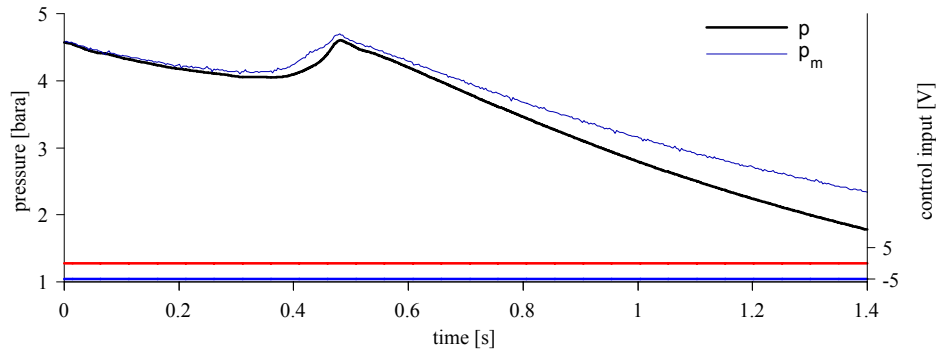


Figure 11: Simulated and measured responses in the pressures, p and p_m , when subjected to a step control input to the outlet valve.

References

- Bouri, M., Thomasset, D., Richard, E. and Scavarda, S.: 1994, Nonlinear sliding mode control of an electropneumatic servodrive, *in* 7th Bath Int Fluid Power Workshop (ed.), *Innovations in Fluid Power*, Bath, UK, pp. 201–219.
- Drakunov, S., Özgüner, Ü., Dix, P. and Ashrafi, B.: 1995, ABS control using optimum search via sliding modes, *IEEE Transactions on Control Systems Technology* **3**(1), 79–85.
- Egeland, O.: 1993, *Servo Teknikk*, Tapir forlag.
- Haugen, F.: 1994, *Regulering Av Dynamiske Systemer*, Tapir forlag.
- International Workshop on Computer Software for Design, Analysis and Control of Fluid Power Systems*: 1999, Fluid Power Net, NTNU Fluidteknikk, Division of Thermal Energy and Hydro Power, Norwegian University of Science and Technology, Trondheim, Norway.
- Kaasa, G.-O.: 1999a, The modelling and design of an electropneumatic automotive clutch control system, *Technical report*, Norwegian University of Technology and Science (NTNU).
- Kaasa, G.-O.: 1999b, *Nonlinear control of an electropneumatic automotive clutch actuation system*, Master's thesis, Norwegian University of Technology and Science, NTNU.
- Kaasa, G.-O.: 2000, Model of the eccentric clutch servo, *Technical report*, Kongsberg Automotive ASA.

- Pandian, S. R., Hayakawa, Y., Kanazawa, Y., Kamoyama, Y. and Kawamura, S.: 1997, Practical design of a sliding mode controller for pneumatic actuators, *Journal of Dynamic Systems, Measurement, and Control* **119**, 666–674.
- Seborg, D. E., Edgar, T. F. and Mellichamp, D. A.: 1989, *Process Dynamics and Control*, John Wiley & Sons, Inc.