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Fault Simulation for Hydraulic Cylinder Drive

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Summary

Computer based simulation is being used more and more in the design phase of machinery. The same computer model can be used to design appropriate means for machine condition monitoring. Information on the modeled faults will help to find useful methods for classifying and describing different faults. Sometimes simulation can be the only feasible approach, since certain faults have low probability for occurrence or testing is impossible in real-world pilots. This report discusses faults in electrical signal, control valve jamming, and leaks in feedback-valve-controlled hydraulic cylinder. Faults are examined in terms of position, pressure, and flow effects during the different work phases of the cylinder.

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Foreword

This report was prepared as part of the project Monitoring and Diagnostics - Lifetime Management of Mobile Machinery (Liikkudia). The project is part of VTT' activities focused on safety and reliability. The authors would like to thank Metso Paper Oy for providing the original model for fault simulation.

Espoo

Authors

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1 Introduction

Computer based simulation is a useful tool in situations where testing of machinery is expensive or dangerous. Computer model in the design phase helps to optimize the product's characteristics before a real prototype is built. The model can also be used to examine failure conditions, with the information acquired proving valuable for fault detection and diagnosis. As it is possible to obtain easily simulated data from different points in the model, the number of condition monitoring measuring points (mainly position, velocity, acceleration, force, pressure, torque and flow but also temperature and electric signals of control system) can be selected economically. Testing different measurements and their location in a real machine or laboratory prototype is an expensive task. Also, limit values for alarms are easy to set.

If the computer model is updated to match real production machine, the model can be used as a condition monitoring method. The measurements are compared with simulation outputs. Deviations indicate that there is something wrong in the machine. Usually, the amount of deviation is connected to the fault severity. The number of sensors can be reduced when measurements are linked to the model. This results in lower total data output, a situation which almost always creates problems for industrial condition monitoring. The direct deviation is the simplest method of model based fault detection, also more complex methods can be implemented [Isermann & Balle, 1997; Isermann 1997]. At the same time measurement parameters (e.g., sampling and A/D conversion values) can be tested virtually.

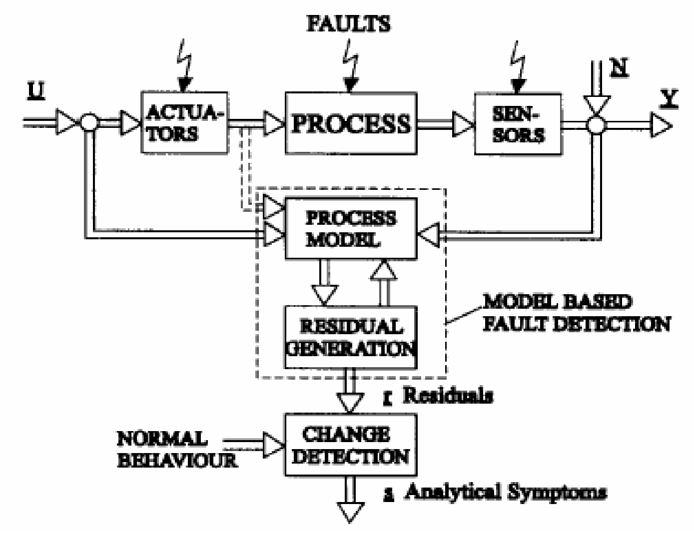

Figure 1. Scheme for model-based fault detection. Machinery can be described as a mechanical process [Isermann & Balle, 1997].

Computer model also provides a favorable environment for fault diagnosis method design [Isermann & Balle, 1997; Isermann 1997; Leonhardt & Ayobi, 1997]. A simulation environment may also be the only environment for testing of methods because extreme conditions usually do not appear frequently in real machines during normal operation.

This report has been made in the project Monitoring and Diagnostics - Lifetime Management of Mobile Machinery (Liikkudia). The goal of this project is to endow machinery with the ability to fault detection and self-diagnosis by the help of remote data handling. Generating fault data in pilots is usually difficult in relatively short term projects. Laboratory testing, on the other hand, is also time consuming and expensive. Therefore it was agreed, that one of the project goals is to study the utilization of simulation in fault data production. The specific model was adapted by the permission of Metso Järvenpää.

2 Goals

The goal of this research is to produce more faults for a hydraulic cylinder drive simulation model than in the previous work [Mustonen & Tervo, 2001] and to supply fault data to Liikkudia project needs. This data is used for research on model based fault detection and diagnosis system.

3 Description of the Target and the Problems Simulated

The target for simulation is a force-feedback proportional valve and cylinder (Figure 2) from the reeling section of a paper machine.

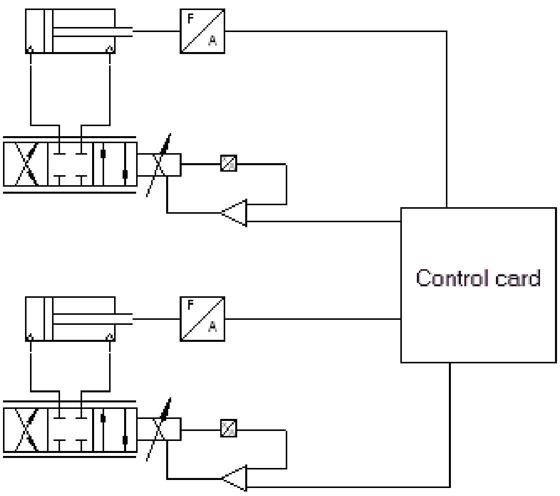

Figure 2. Control system of cylinder [Palokangas, 1999].

The purpose of this cylinder drive is to maintain constant line load between the reeling drum (the smaller roll in Figure 3 in contact with the paper web) and reeling cylinder (the larger roll in Figure 3 in contact with the paper web) during reeling. The reeling process is affected by various harmful vibrations. Vibrations are especially detrimental to the reeling process when the drum is vibrating with respect to the cylinder. This kind of vibration is the sum of many different vibration types. Paper characteristics (e.g., quality and weight), mechanical structures (e.g., natural frequencies and their multiples or roll eccentricity), hydraulics (e.g., oil elasticity), and the control system (e.g., A/D conversion delay) affect reeling dynamics.

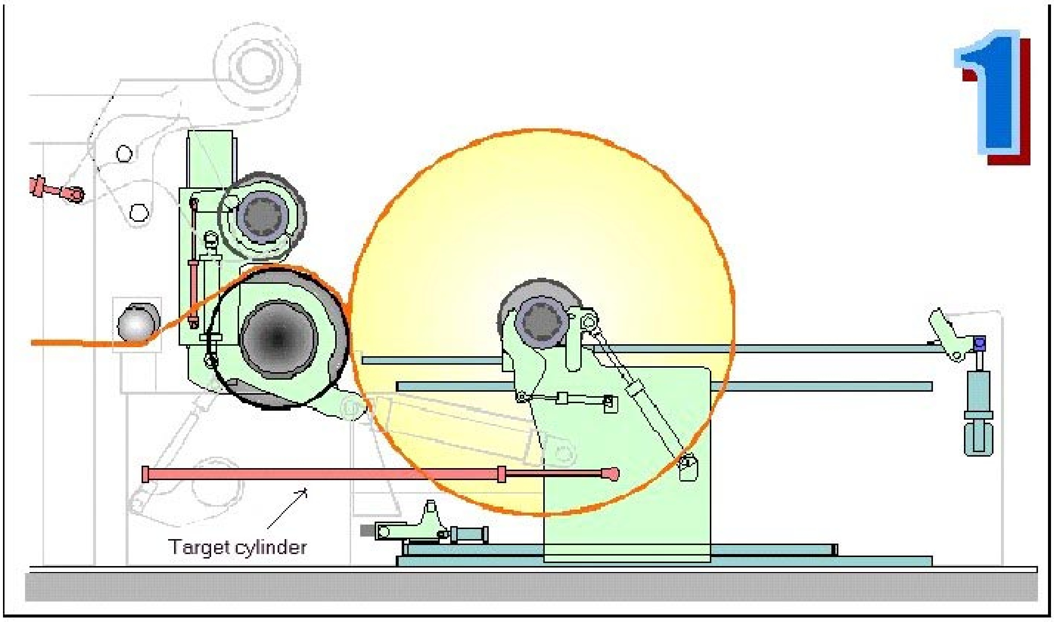

Figure 3. Reeling section [Palokangas, 1999].

Also, the reeling process itself causes vibration in the system. One example is shown in Figure 4. The surface is deformed due to load, and the contact between rolls (nip) is altered. This phenomenon causes a greater disturbance if the deformation does not cease within a revolution. Also, the paper web thickness (profile) changes and the possible wrinkling of bottom layers due to roll change operation impart vibrations to the reeling process [Palokangas, 1999]. As testing is nearly impossible for product loss reasons, the paper machine environment is a target that can benefit greatly from this kind of fault simulation.

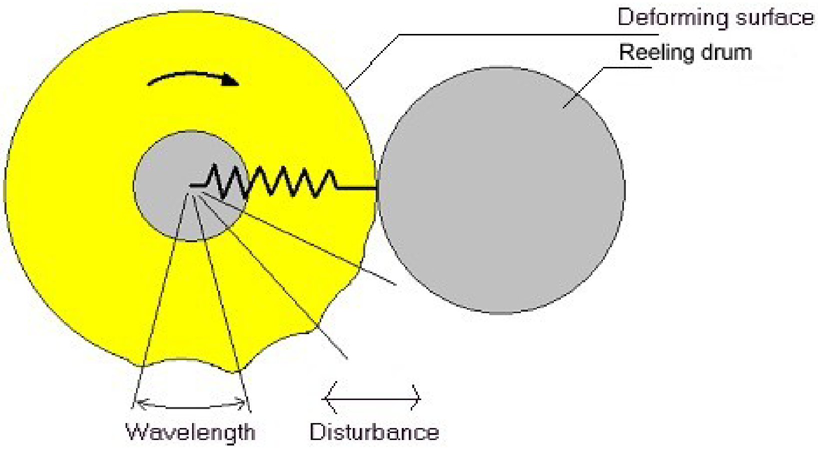

Figure 4. Surface deformation affected by damping of reeled paper [Palokangas, 1999].

3.1 Simulation Model

The hydraulic cylinder and valve model applies the principle of centralized pressures. The hydraulic circuit is divided into volumes in which the pressure is assumed to be uniform. Valves and pipelines are assumed to be flow limiters and, as the pressures are known from volumes and damping, flows passing limiters can be calculated [Mikkola, 2002]. Modeled flows and pressures are shown in Figure 5.

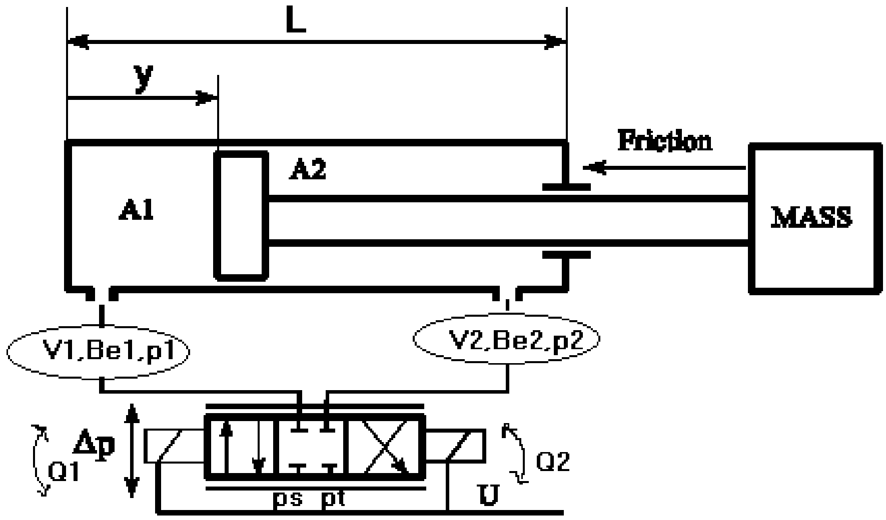

Figure 5. Valve-controlled hydraulic cylinder.

The following parameters are used in the model:

p_s = supply-side pressure [Pa]
p_t = tank pressure [Pa]
p_1 = cylinder-side pressure [Pa]
p_2 = piston-side pressure [Pa]
Q_1 = cylinder-side flow [m^3/s]
Q_2 = piston-side flow [m^3/s]
B_{e1} = effective bulk modulus of cylinder side [Pa]
B_{e2} = effective bulk modulus of piston side [Pa]
V_1 = volume of cylinder side [m^3]
V_2 = volume of piston side [m^3]
A_1 = cylinder-side area of piston
A_2 = piston-side area of piston
y = position [m]
L = length of cylinder [m]
U = voltage input [V]
Δp = pressure difference [Pa]

Flows are defined using the pressure differences as follows [Palokangas, 1999]:

$$\begin{cases} Q1_{p1pt} = av_1 Q_N \sqrt{\dfrac{p_1 - p_t}{\left(\dfrac{\Delta p_N}{2}\right)}} \quad as\ \ p_1 \geq pt \\ Q1_{ptp1} = av_1 Q_N \sqrt{\dfrac{p_t - p_1}{\left(\dfrac{\Delta p_N}{2}\right)}} \quad as\ \ p_1 \leq pt \\ Q1_{le} = \dfrac{v_{alle}}{200} \dfrac{Q_N}{P_{Ns}} (p_1 - p_t) \end{cases} \tag{1}$$

where
$Q1_{p1pt}$, $Q1_{ptp1}$ = flow along valve control edge [m^3/s]
av_1 = spindle opening on control edge 1
Q_N = nominal flow through valve [m^3/s]
Δp_n = nominal input pressure difference [Pa]
$Q1_{le}$ = leak flow along closed control edge [m^3/s]
v_{alle} = total leak flow in % of valve's max. flow
P_{Ns} = nominal input pressure of valve [Pa]

In Figure 6. Spindle openings and related pressures are presented schematically. Pressures are defined in equation 2 [Mikkola, 2002; Palokangas 1999].

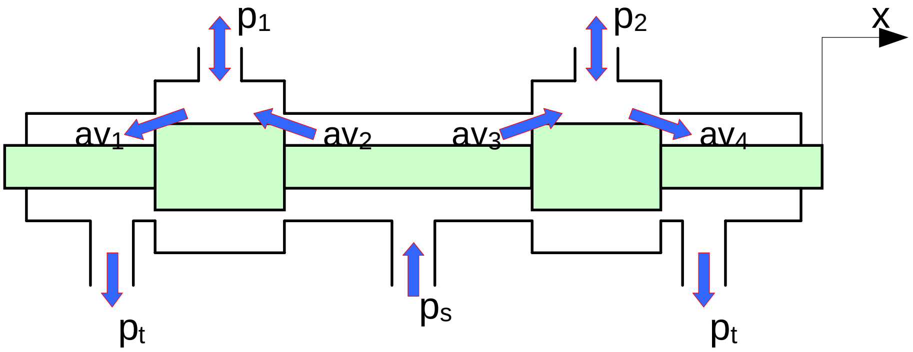

Figure 6. Flows and control edges of proportional valve spindle [Palokangas, 1999].

$$\begin{cases} \dot{p}_1 = \dfrac{B_{e1}}{V_1 + A_1 y}(Q_1 - A_1 \dot{y}) \\ \dot{p}_2 = \dfrac{B_{e2}}{V_2 + A_2 (L - y)}(Q_2 + A_2 \dot{y}) \end{cases} \tag{2}$$

Effective bulk modulus can be calculated as follows:

$$\frac{1}{B_e} = \frac{1}{B_o} + \frac{1}{B_c}\frac{V_c}{V_t} + \frac{1}{B_p}\frac{V_p}{V_t} + \frac{1}{B_h}\frac{V_h}{V_t} + \frac{1}{B_a}\frac{V_a}{V_t} \tag{3}$$

where
B_e = effective bulk modulus
B_o = bulk modulus of oil
B_c = bulk modulus of cylinder
B_p = bulk modulus of pipe
B_h = bulk modulus of hose
B_a = bulk modulus of air
V_t = total volume
V_c = volume of cylinder
V_p = volume of pipe
V_h = volume of hose
V_a = volume of air

Valve spindle position is calculated from the voltage input at second-order transfer function [Palokangas, 1999; Virvalo, 1999]:

$$\ddot{x} = u(t)\omega_n{}^2 - x\omega_n{}^2 - 2\dot{x}\zeta\omega_n \tag{4}$$

where
x = spool position [m]
$\dot{x}$ = spool velocity [m/s]
$\ddot{x}$ = spool acceleration [m/s^2]
$u(t)$ = voltage input [V]
Σ_n = nominal angular velocity of vibration [rad/s]
Σ= cancellation ratio

There are also other valve characteristics that have an effect on spindle movement. Feedback sensor, controller, valve dead zone, valve hysteresis, A/D conversion, and control delay factors are included in the model. The force equation for the cylinder is as follows [Mikkola, 2002]:

$$F_s = A_1 p_1 - A_2 p_2 - \int F_\mu \tag{5}$$

where
F_s = supply force
ΣF_μ = sum of friction forces

Pressure loss is used to model friction forces in the cylinder. With these equations, Q_1, Q_2, p_1, and p_2 can be examined for the duration of the simulation.

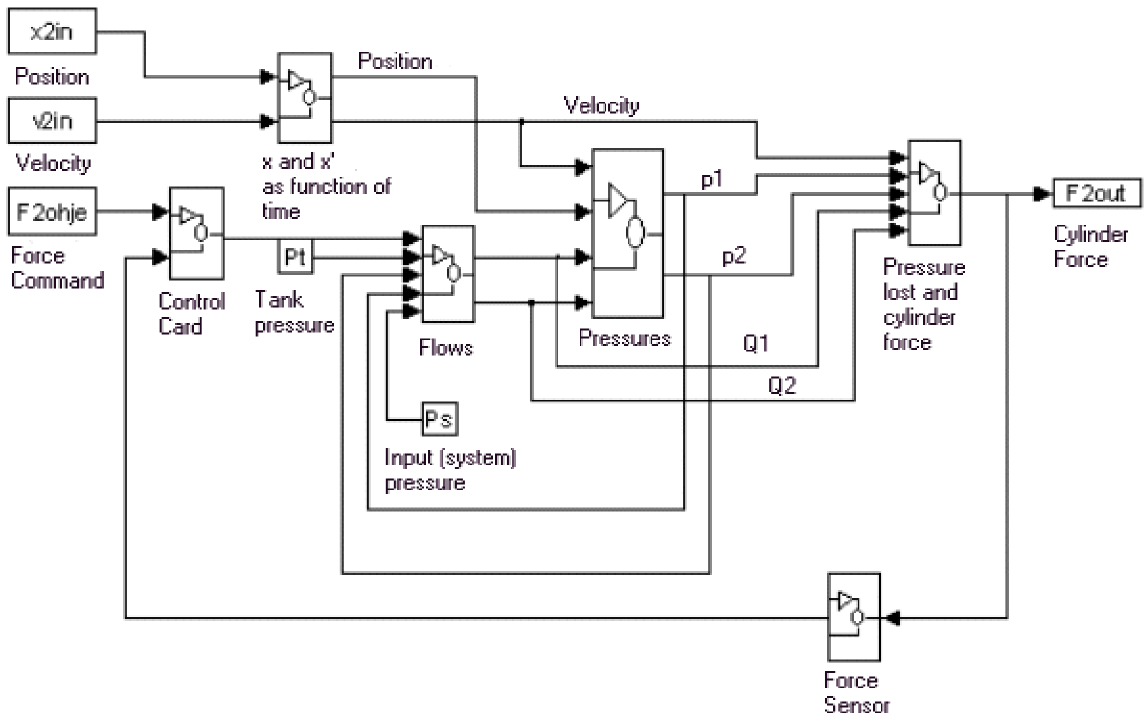

Figure 7. Simulink model with force-feedback-controlled cylinder [Palokangas, 1999].

The model was created using Matlab R11 Simulink building blocks. The original model [Palokangas, 1999] was connected to the Working Model software. The mechanics were simulated in that environment. Because the Working Model software and mechanics information were unavailable, the characteristics from mechanics were modeled simply with one rigid component. Also, the force feedback was changed to position feedback with acceleration derived from force signal. This helped to define the desired input more easily as position. A constant force was set for the model, and two opposite steps were defined as the input signal.

3.2 Electrical Signal Failure

Electrical signal failure suggests that the grounding has failed in some part of the system. Electrical signal failure was modeled as a zero-volt input to the valve for a certain time period. This kind of fault could be the result of a cable cover breakage and cable movement or connector oxidation and material expansion affected by temperature change.

3.3 Leaks

Four types of leaks were modeled. The first is an air leak caused by a hole in the p_1-side pipe, and the second is an air leak caused by a hole in the p_2-side pipe. The third is a hole between the p_1 and p_2 pressure volumes; fourth is an air leak from both the p_1 and p_2 volumes. The third type of leak could be caused by seal failure inside a cylinder. The modeling included turbulent and laminar flow losses. Both must be included because the length/-diameter ratio is under 100. Air leaks could result from pipe corrosion or seal wear in the cylinder or control valve. Flows were modeled as follows [Backé, 1986; Merrit, 1967]:

$$\begin{cases} Q_{out} = Q_{in} - Q_{leak} \\ Q_{leak} = sign(p_x - p_y)\dfrac{-K_L\dfrac{128\nu\rho}{\pi} + \sqrt{\left(K_L\dfrac{128\nu\rho}{\pi}\right)^2 + 4\left(K_T\dfrac{8\rho}{\pi^2}(p_x - p_y)\right)}}{2K_T\dfrac{8\rho}{\pi^2}} \\ K_L = \dfrac{l_{hole}}{d_{hole}^{\;4}} \\ K_T = \dfrac{\zeta_{hole}}{d_{hole}^{\;4}} \end{cases} \tag{6}$$

where
Q_{in} = flow without leak [m^3/s]
Q_{leak} = leak flow [m^3/s]
Q_{out} = flow with leak flow [m^3/s]
p_x, p_y = pressures [Pa] (p_1, p_2 and p_{air})
l_{hole} = length of hole [m]
d_{hole} = diameter of hole [m]
ξ_{hole} = flow resistance coefficient of hole

3.4 Valve Spindle Jamming

The situation of valve spindle jamming was modeled previously [Mustonen & Tervo, 2001]. In this work, the position of the spindle during the stoppage and its time in that position were taken as initial values. The model was changed so that the start time and the duration of jamming were set in the model. The benefit of this change is that now the time values were sufficient to describe a jam and the spindle position could remain unknown. A spindle jam could result from large particles from the oil or accumulation of large particles in the valve.

4 Limitations and constraints of the Study

Dynamics of mechanical components (rigid components) and the reeling process were not included in this research. These affect the total dynamics of the modeled system. Therefore the modeled work cycle has no relationship to reeling. The dimensions of the leak were kept constant in all situations. The duration of control valve jamming as well as the zero-volt input was 0.5 seconds.

5 Results

The simulation time was 20 s. First, the model seeks static equilibrium, which is found at 9 s. At 10 s, an input step signal from 0 to 1 m is generated, and at 15 s, an input step signal from 1 to 0 m is generated. Simulink uses variable size for each iteration step in these simulations. Eleven simulations were completed. These simulations are summarized in Table 1. Input signal, actual position, actual position minus input signal, p_1, p_2, Q_1, and Q_2 are plotted in Appendix 1.

Table 1. Summary of simulations in Appendix 1

No.	Description of simulation
1	Normal situation
2	Zero-volt input from 12 to 12.5 s
3	Zero-volt input from 14 to 14.5 s
4	Zero-volt input from 16 to 16.5 s
5	Pressure leak in cylinder-side pipe (p1) to air; hole length: 5e-3 m, hole diameter: 5e-5 m
6	Pressure leak in piston-side pipe (p2) to air; hole length: 5e-3 m, hole diameter: 5e-5 m
7	Pressure leak between cylinder- and piston-side pressure volumes (p1 to p2 and p2 to p1); hole length: 5e-3 m, hole diameter. 5e-5 m
8	Pressure leak in cylinder side and piston side to air (p1 and p2); hole length: 5e-3 m, hole diameter: 5e-5 m
9	Control valve jam from 12 to 12.5 s
10	Control valve jam from 14 to 14.5 s
11	Control valve jam from 16 to 16.5 s

6 Discussion

The simulation model used in this work was a modification of that used in a previous study [Mustonen & Tervo, 2001], which in turn was modified from the original work [Palokangas, 1999]. The model here was created using manual switches for each type of fault in different positions in the simulation chain. Using this method, it is possible to change faults rather easily with the model.

Controller operation is rather slow. The negation of input signal and actual position is quite different (appendix 1 up right figures). One reason for this is that the model had restrictions on the maximum piston speed. Also, when 0 m position is wanted for the cylinder, 0.1 m is the actual position. The controller is handled using position/velocity/time lookup tables [Mustonen & Tervo, 2001] instead of Simulink's own PID block. The controller was left without modification. The original work [Palokangas, 1999] was done with a PID block.

Reality of the simulated faults has not been verified. The effects of process and mechanical vibrations to hydraulic system probably cause substantial disturbance to measurement signals. Also these faults may not ever be present in a real system. This work serves as a reference for general examination of simulated hydraulic cylinder drive faults. Verification is needed for faults at least in laboratory level. There exist several un-modeled parameters that may have an effect on measurement signals.

6.1 Electrical Signal Failure

A zero-volt-input fault was simulated in different phases of the work cycle. The first simulation was performed when the piston was moving toward the 1-m position (simulation 2). The second was when the piston was at the 1-m position (simulation 3), and the third was when the piston was moving back toward its starting position (simulation 4). In simulations 2 and 4, failure can be clearly seen from the position signal. The zero-volt control signal changes spool position very rapidly, and this can be seen directly from the flows. This affects the pressures as well, which fluctuate quite strongly. The pressure vibration can be seen also from the piston position. This kind of variation in hydraulic system pressure is undesirable because it results in wear affecting the system. In simulation 3, only minor changes in flows could be seen. This is a result of a little spindle position change. The controlling input signal was already near zero volts.

6.2 Pressure Leaks

A pressure leak in p_1 to air (simulation 5) results to little fluctuation in the position signal when the piston is at the 1 m position. There is a steady pressure increase in both pressures as the controller tries to compensate for the lost flow with added pressure between 13 and 15 s. This affects the position, too. Flow variation diminishes during that time period as well. A pressure leak in p_2 to air (simulation 6) leads to less variation in pressures and position than seen in simulation 5 between 13 and 15 s. The pressure amplitude is higher in this case. This is a result of the smaller volume under piston-side pressure. In the flow, there is a sudden negative change in Q_2 at 15 s. This is a result of the slower response of pressure on the time axis. In simulation 8, both pressure volumes experienced leakage to air. Very small fluctuation could be seen in the position. The manifestations were in general a sum of those found in simulations 5 and 6. Pressures were quite high during the 13-to-15-s interval but vibrations minor. Simulation 7 dealt with connected flows. Heavy vibration could be seen in both pressure and flows during 1-m steady input. Also, the position fluctuated greatly. The flows between pressure zones contributed to fluctuation in all parameters. It is difficult to model a leak between seal surfaces or pipe holes. The shape was here modeled as a circle, whereas the right shape should probably be something closer to an oval or semicircle. Also, the leak was modeled for the entire duration of the simulation. It should be useful to model leaks for a smaller period of time, as was done here with the valve jamming or zero-volt input.

6.3 Control Valve Jamming

A control valve jamming (simulations 9, 10, and 11) is simulated in the same position as zero-volt input. Nothing different from what is encountered in normal situations can be seen from the position signal. The phenomenon is too fast, and damping is too high. These faults can be seen best from the flow diagrams. Pressures also have some small impact on valve jams. Simulation 11 is an exception: no pressure or flow impact could be seen in this simulation. The pressure and the flow were too stable at the time of this jam.

6.4 Pressure Relief Valve Fault

The intention was to simulate a pressure relief valve jamming with this model, too. The problem was that stability was not achieved in the model when the pressure relief valve was included. The valve was left out of the final model. A reduced, semi-empirical model of a single-stage pressure relief valve was used. The flow through this pressure relief valve is [Handroos, 1990; Mikkola, 2002]:

$$\begin{cases} Q_r = K_r \sqrt{p_r} \\ \dot{K}_r = \dfrac{\left(p_r - p_{ref}\right) - \left(C_1 + C_2 p_r\right) \cdot K_r}{C_{dyna}} \end{cases} \tag{6}$$

where
Q_r = flow through pressure reduction valve
K_r = valve throttle coefficient
p_r = pressure in valve
p_{ref} = reference pressure in valve
C_1 , C_2 = empirical coefficients
C_{dyna} = empirical dynamic coefficient

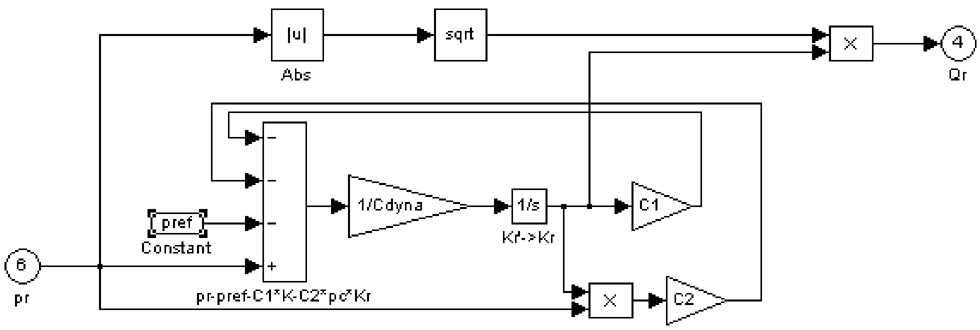

Figure 8. Simulink model of a reduced semi-empirical model involving a single stage pressure relief valve.

6.5 Follow-up Work

The next logical step after simulation work would be to verify measurement results experimentally. As previously mentioned, it might be difficult to verify that these situations occur as modeled because they don't appear often in the production environment and testing could be expensive. Still, it is possible to use the model for diagnostics development. Analysis and testing of fault diagnostic methods (e.g. [Leonhardt & Ayobi, 1997; Isermann, 1997]) for detecting these different faults has been started. The methods are to be connected with work phase or time window somehow. A special problem exists in zero-volt input and valve jamming faults, where the fault is not visible during all work phases. Statistical methods could be used to detect failures from data gained over a time window encompassing the necessary work phases. One possible and probably necessary continuation would be to add mechanical vibration to the model via simulation software such as ADAMS or Working Model.

7 Conclusions

In this paper, a simulation model of a hydraulic cylinder drive from the reeling section of a paper machine was used to produce data related to leaks, control valve jamming, and electrical signal failures. Fault models were not verified for cost reasons. The mechanical and reeling process-properties were not modeled and so the effect of faults might be seen quite differently from real production system measurements. The work serves as a reference for general examination of simulated hydraulic cylinder drive faults and as the groundwork for model based fault detection and diagnosis for similar machinery. Generated fault data serves the project needs, but verification is suitable at least in laboratory level.

Electrical signal failure was modeled as a zero volt input for a certain time period. The output signals plotted were pressure, flow and position. The pressure relief valve jamming fault was modeled, but the modeled system was not stable enough to perform the simulation. The zero-volt input fault was manifested as heavy pressure vibration, and the control valve jamming was clearly visible as spikes in the flows. These faults were not visible in every simulation. In the situation where the piston was not moving, no difference to normal situation was found. Fluid leaking (atmospheric pressure) was detected as a pressure increase in different parts of work cycle. Leak inside a hydraulic system caused heavy vibration to all output signals.

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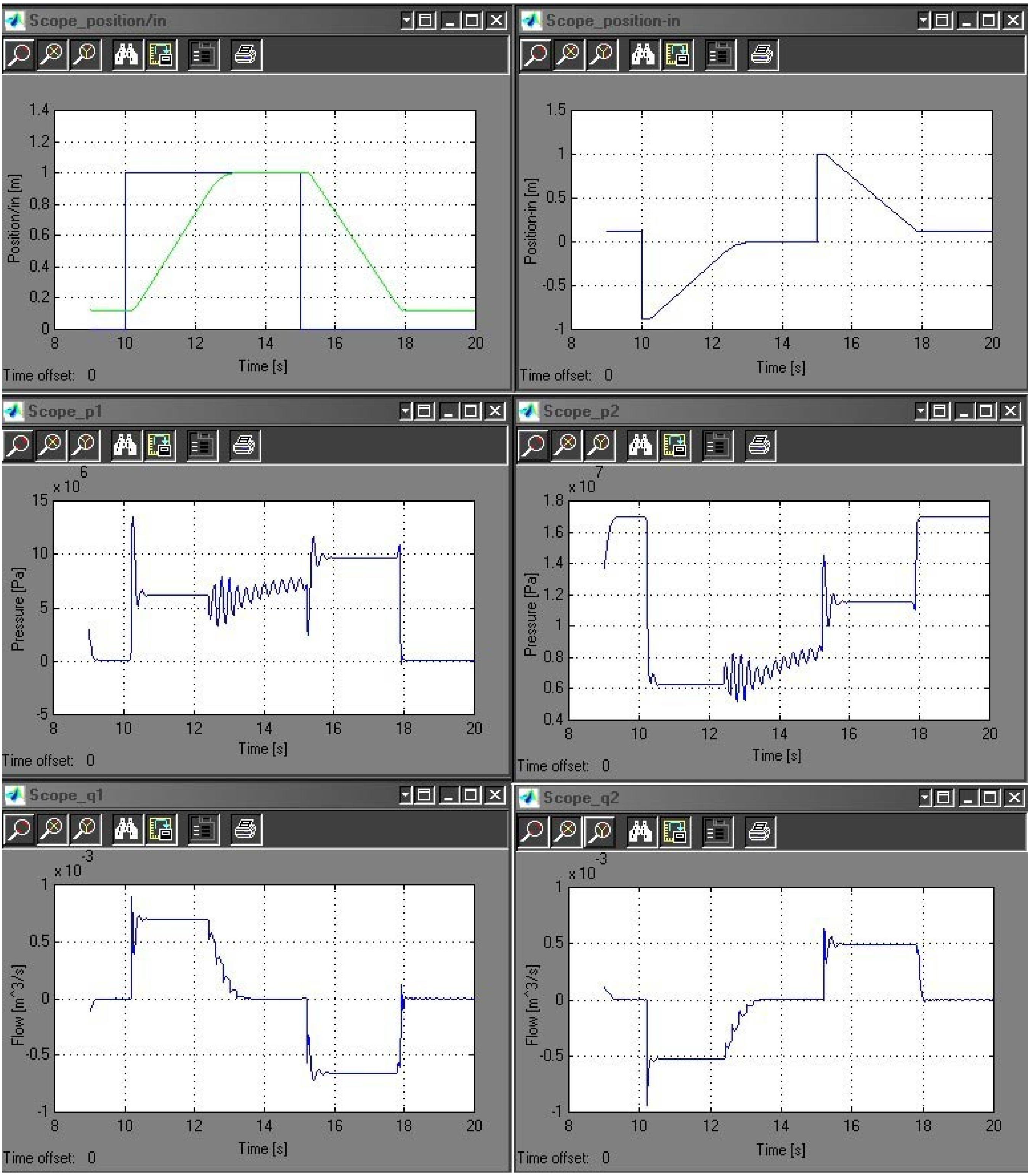
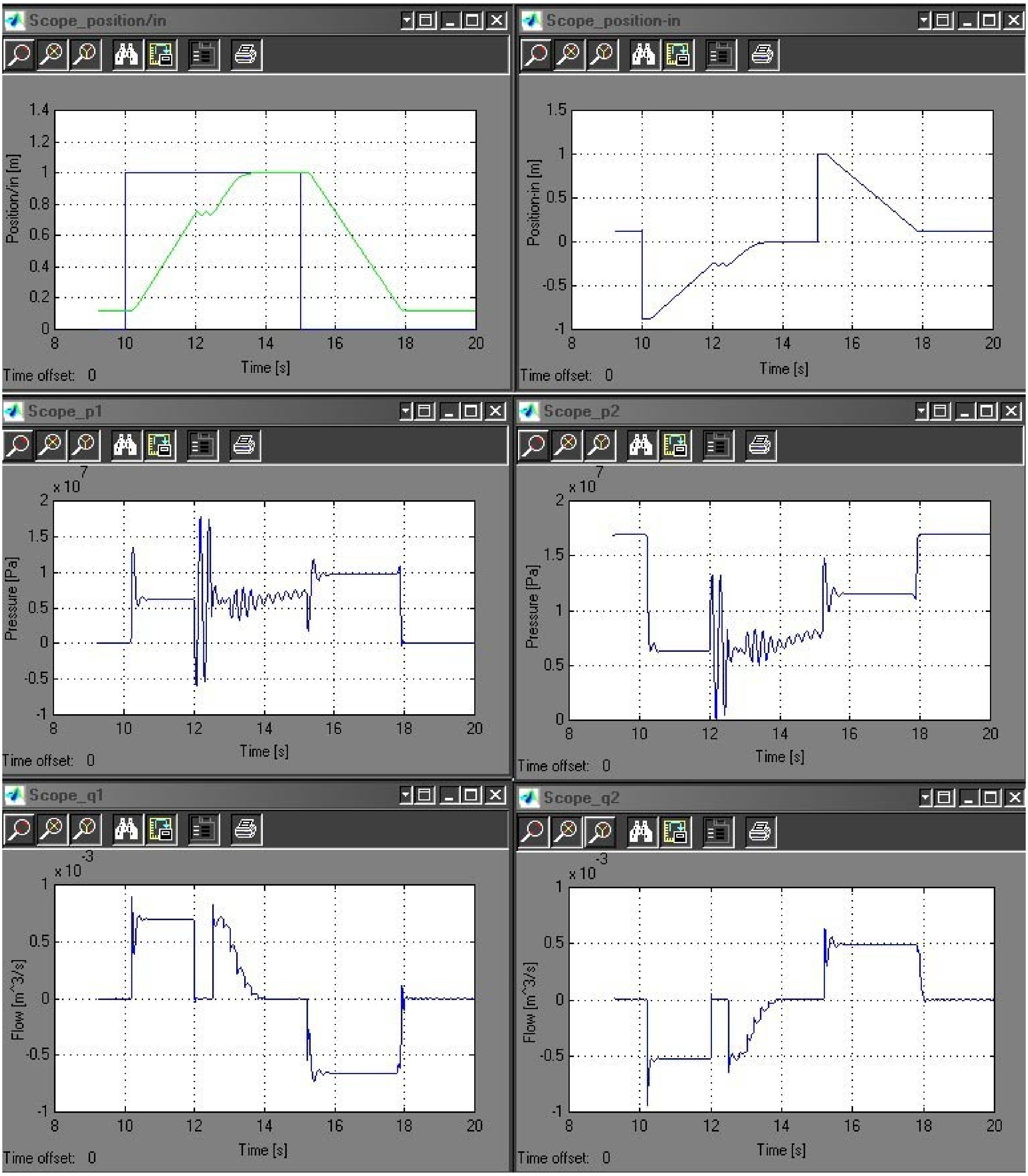
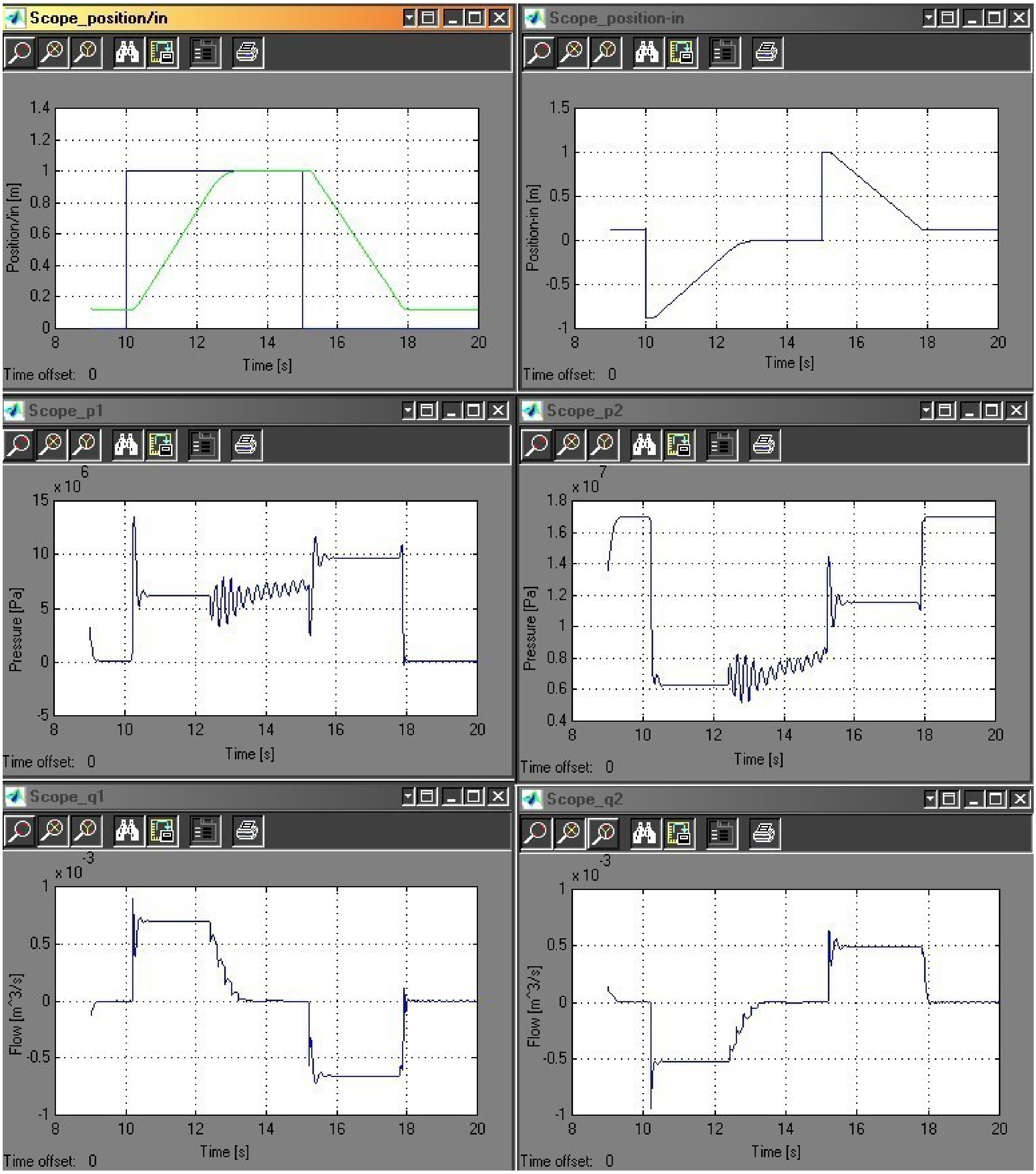


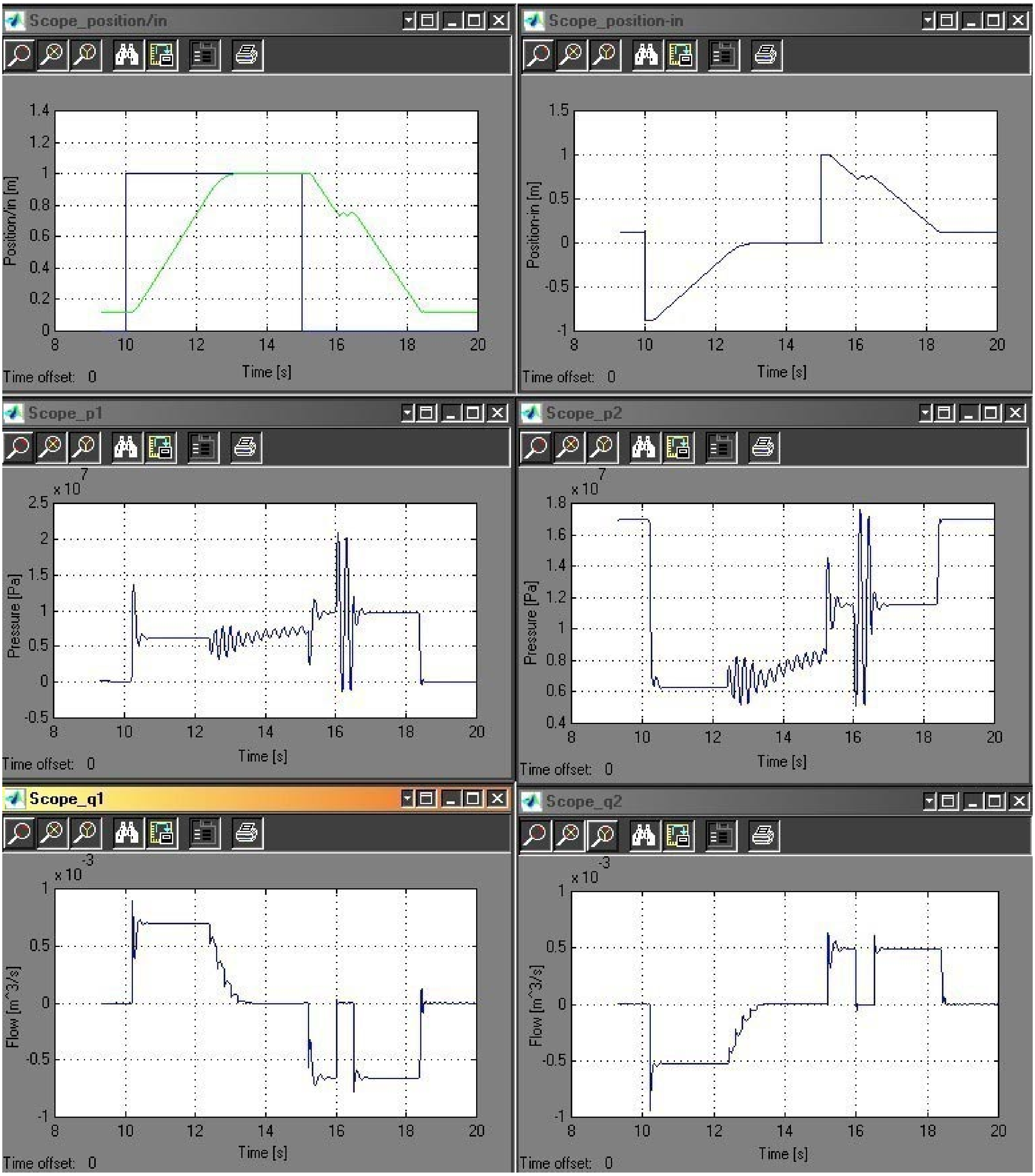
Simulation 1. Normal situation; input signal and actual position, input signal minus actual position, p1 = cylinder-side pressure, q1 = cylinder-side flow, p2 = piston-side pressure and q2 = piston-side flow.



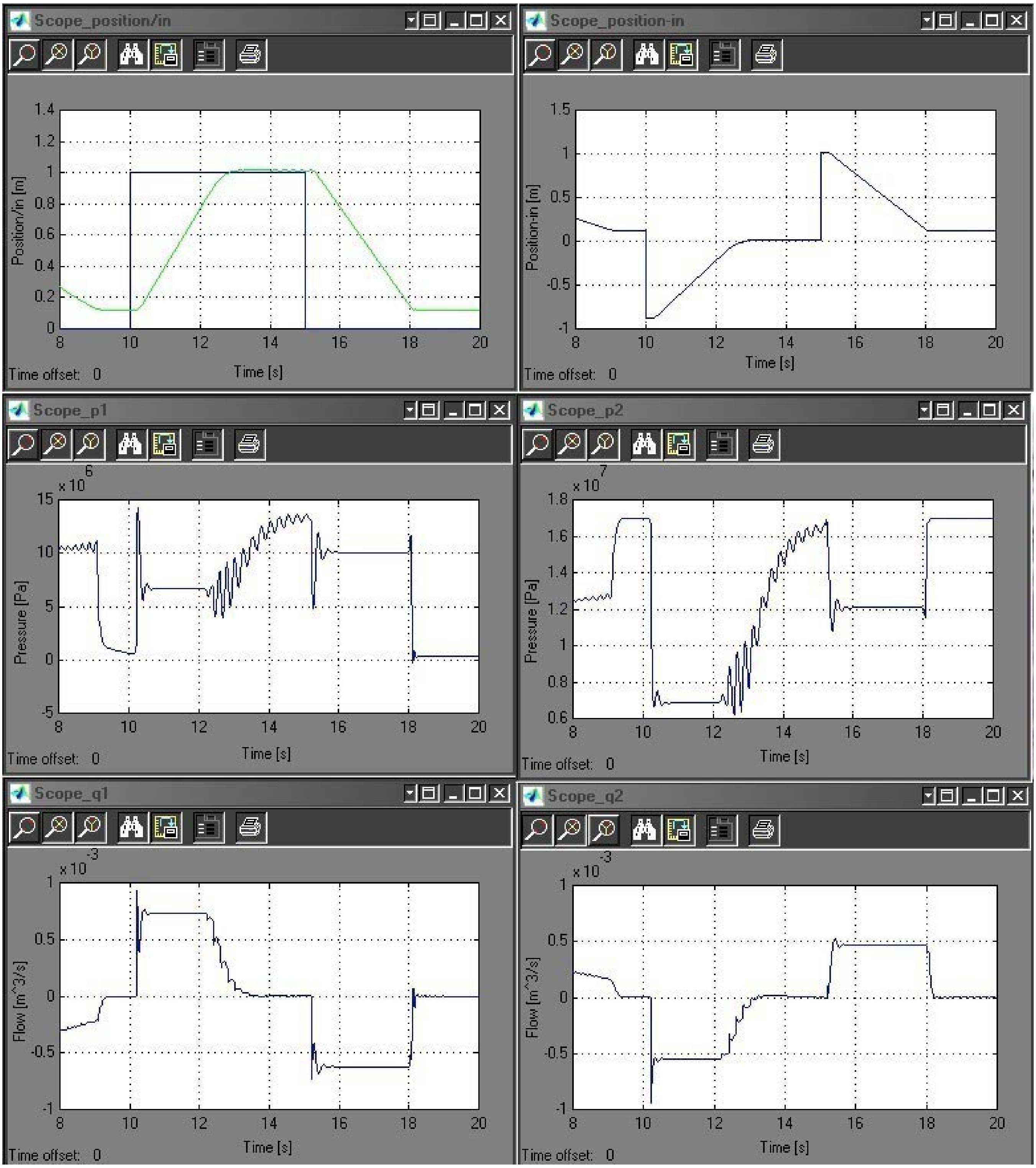
Simulation 2. Zero-volt input from 12 to 12.5 s.



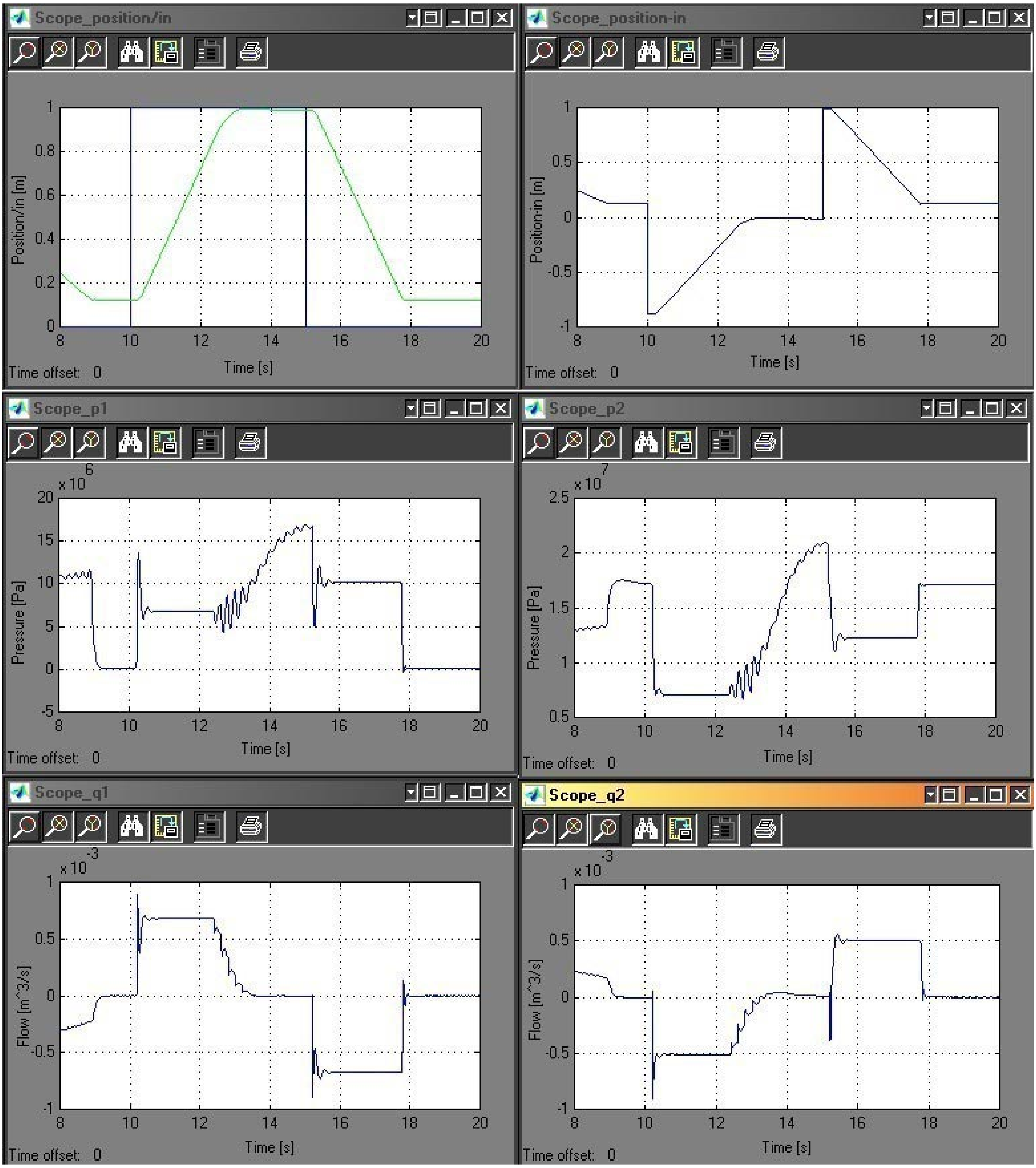
Simulation 3. Zero-volt input from 14 to 14.5 s.



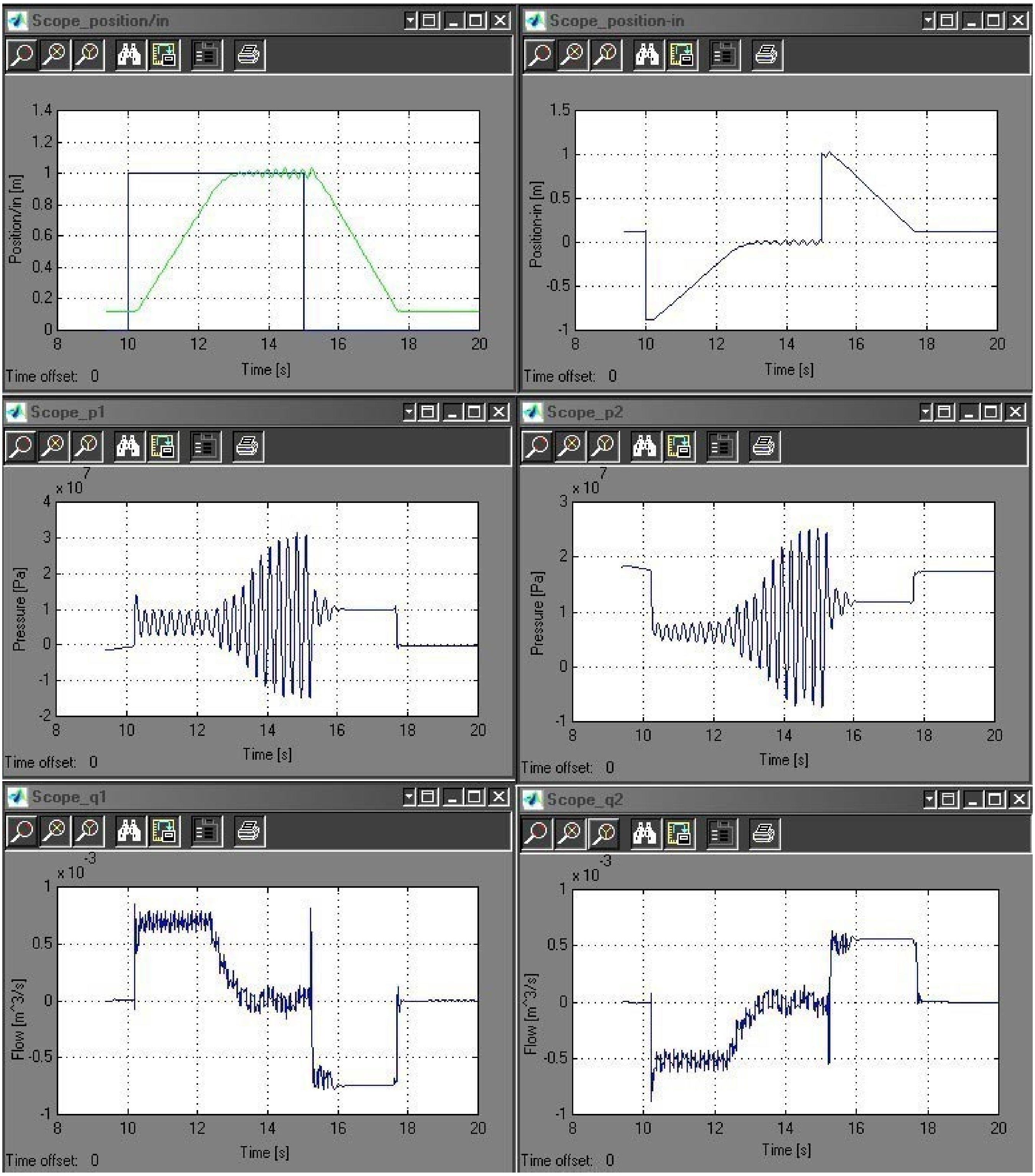
Simulation 4. Zero-volt input from 16 to 16.5 s.



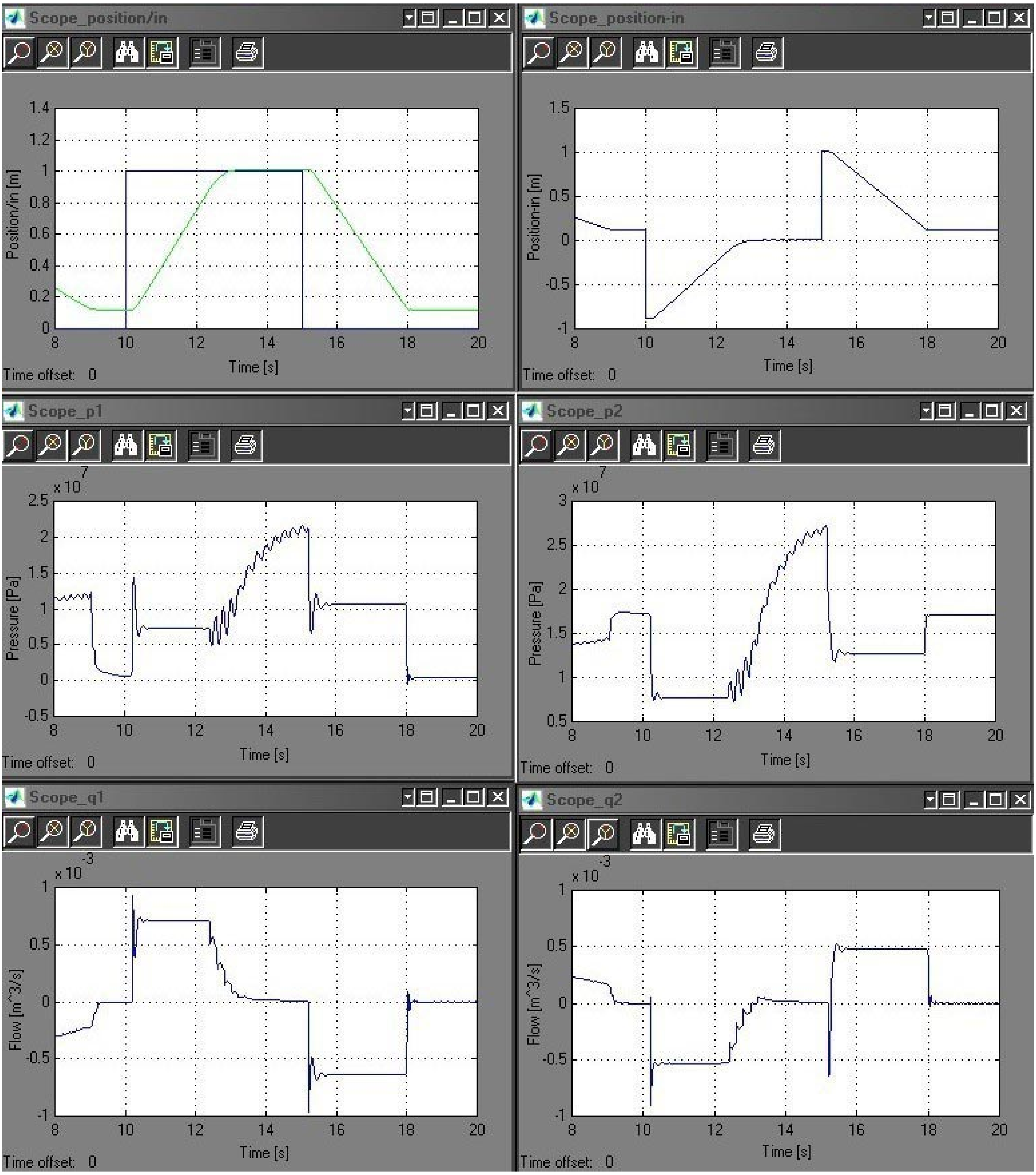
Simulation 5. Pressure leak in cylinder-side pipe to air; hole length: 5e-3 m, hole diameter: 5e-5 m.



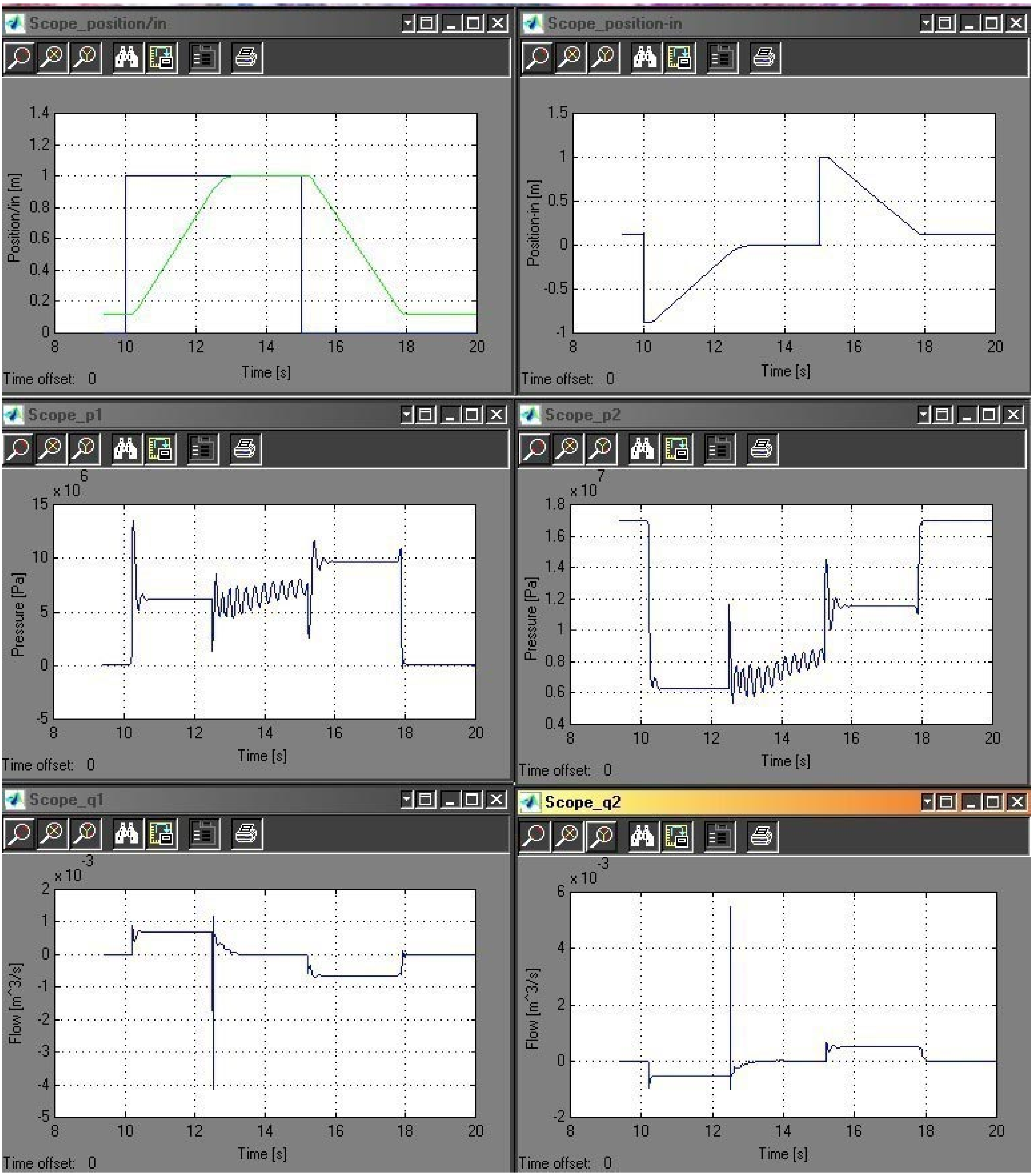
Simulation 6. Pressure leak in piston-side pipe to air; hole length: 5e-3 m, hole diameter: 5e-5 m.



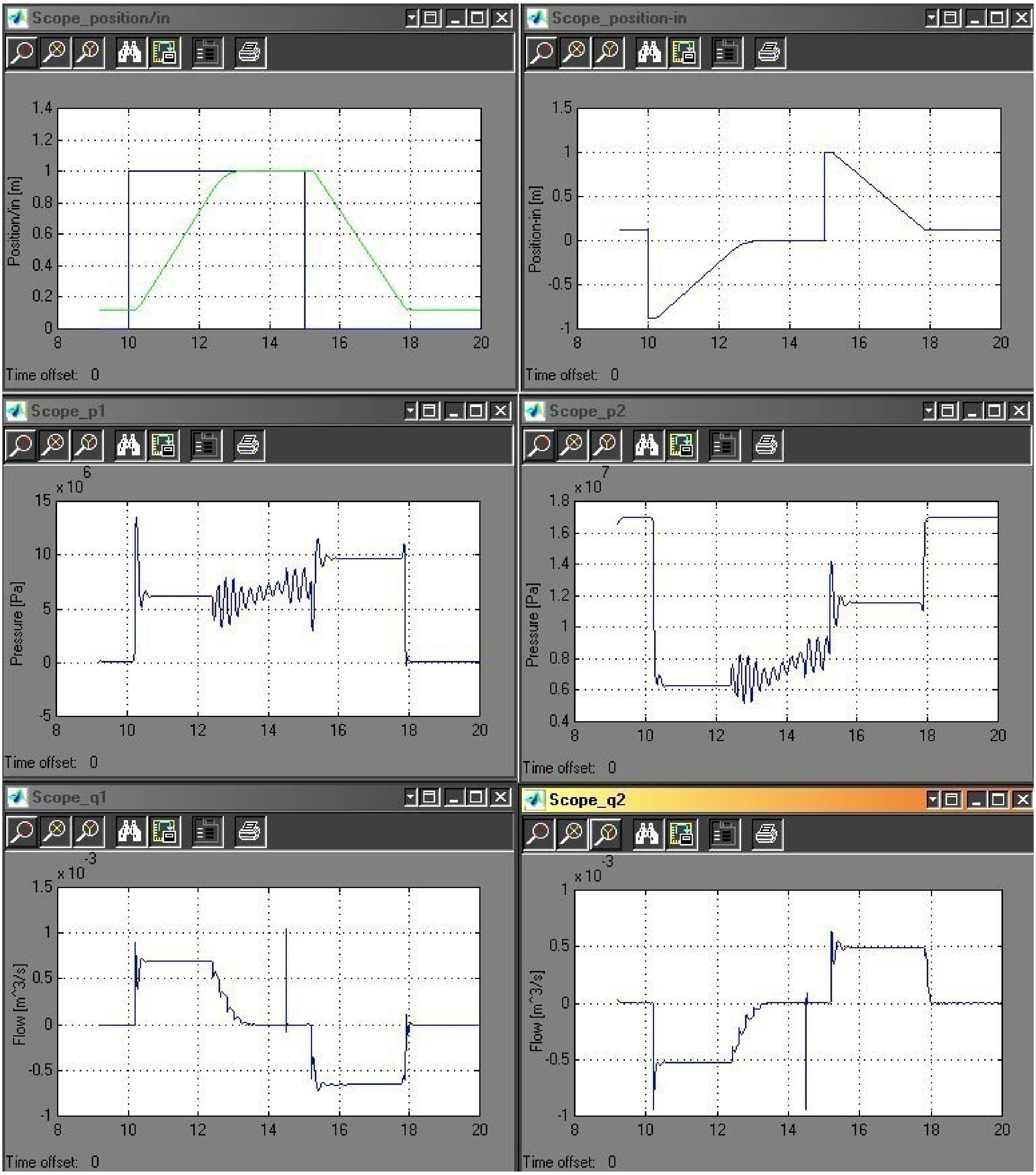
Simulation 7. Pressure leak between cylinder- and piston-side pressure volumes - e.g., failure of seal; hole length: 5e-3 m, hole diameter: 5e-5 m.



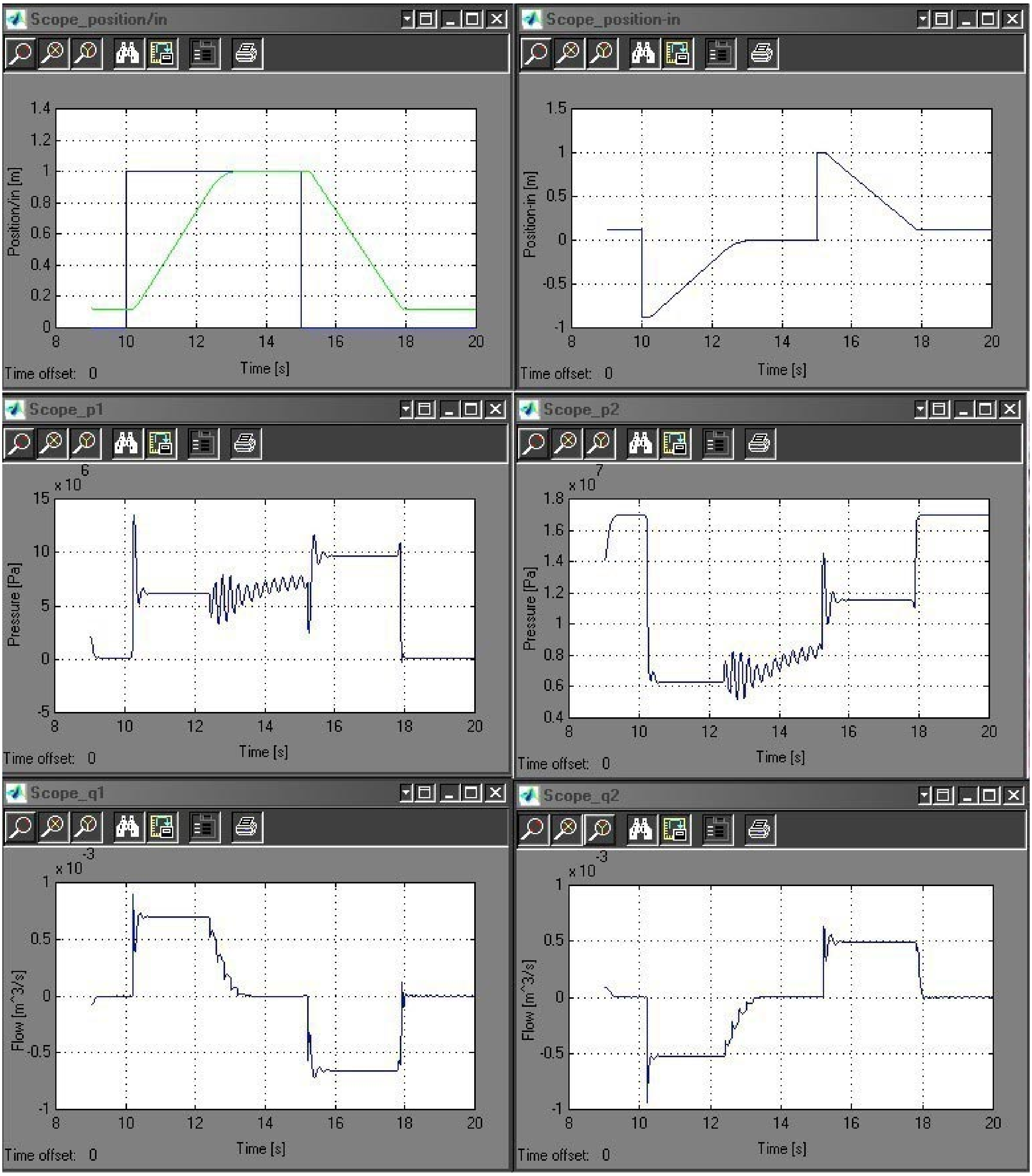
Simulation 8. Pressure leak in cylinder side and piston side to air; hole length: 5e-3 m, hole diameter: 5e-5 m.



Simulation 9. Control valve jam from 12 to 12.5 s.



Simulation 10. Control valve jam from 14 to 14.5 s.



Simulation 11. Control valve jam from 16 to 16.5 s.