Hydraulic Servo Modelling

Digital Twin Lab

September 1, 2020

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1.1 Introduction

This section describes a simplified version of the *servo linear actuator* presented in the *digital twin* models of Prinoth and Liebherr vehicles. The aim of this analysis is to identify the main physical quantities which affect the dynamic and steady state behavior of the servo. Moreover, a kind of normalization will be evaluated in order to build a model which considers the main variables which characterize the dynamical behavior.

1.2 Mathematical model

The system investigated, consists of the valve-controlled linear actuator shown in Figure 1 and Figure 2

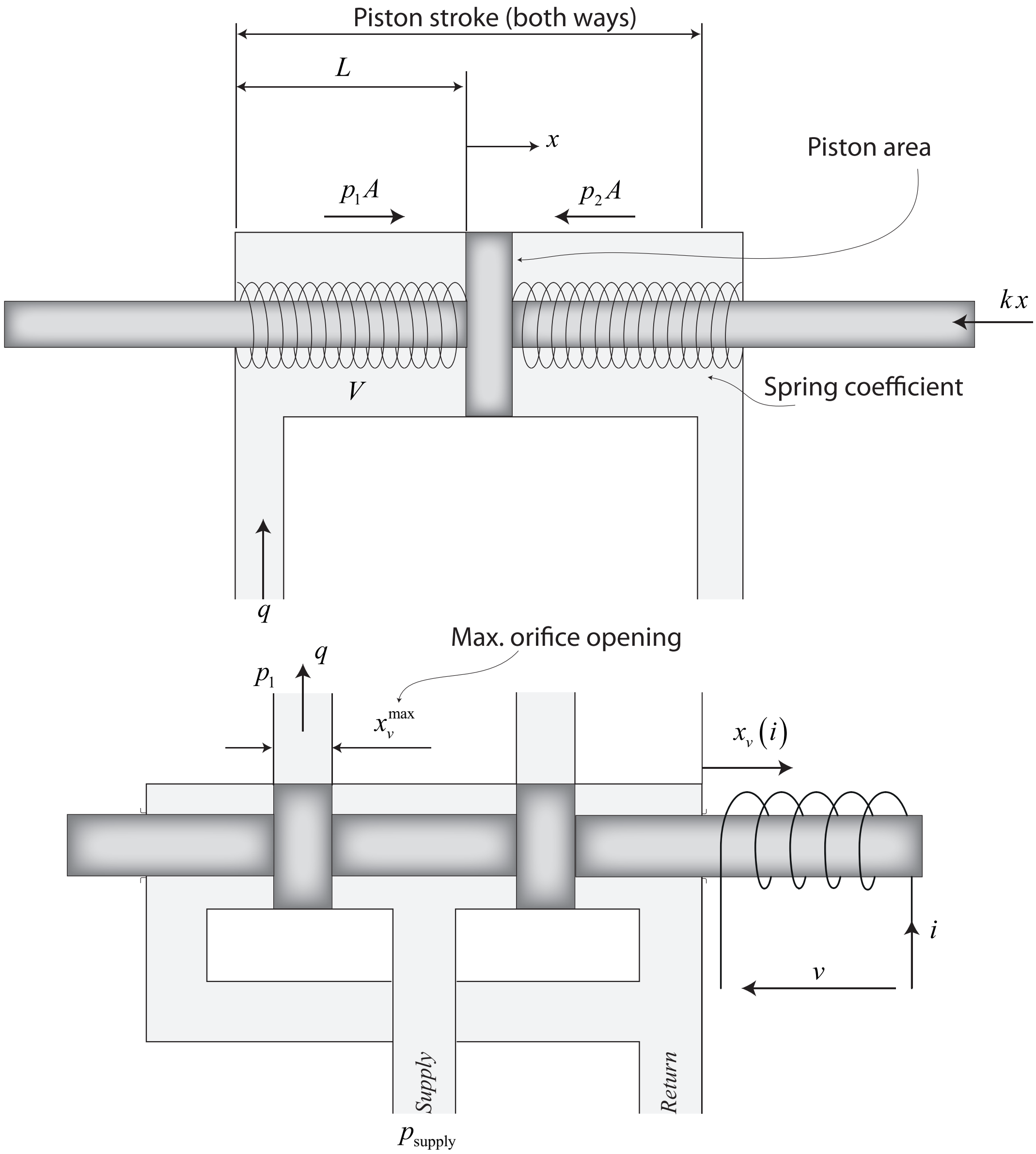

Figure 1: Schematic of the four-way valve-controlled linear actuator.

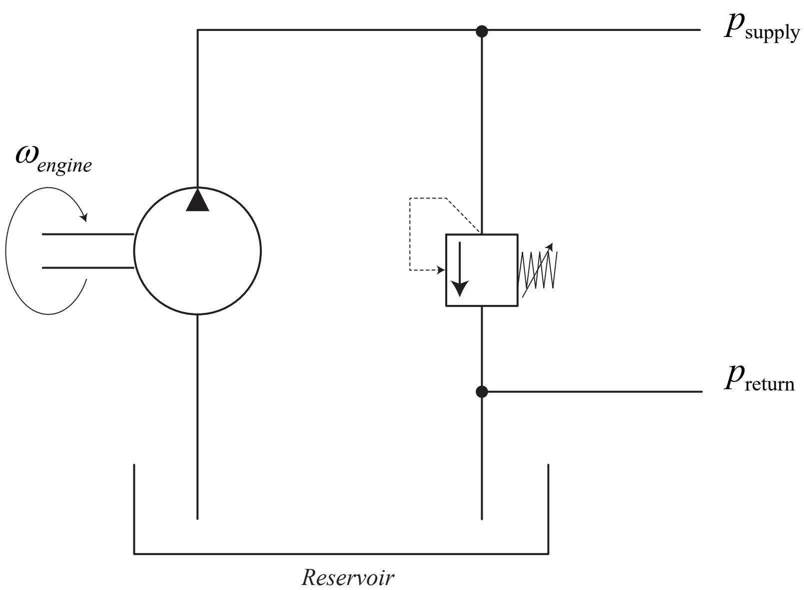

Figure 2: Pressure supply circuit.

Suppose the system is symmetrical and without leakages, we may write the following set of equations

$$\begin{cases} q(t) = C_v x_v^2 \sqrt{\dfrac{2}{\rho}} \sqrt{|p_{suply} - p_1|}\,\mathrm{sign}\big(p_{suply} - p_1\big) \\ \dfrac{dp_1(t)}{dt} = \beta \left(\dfrac{q(t) - Av(t)}{V + Ax(t)} \right) \\ \dfrac{dp_2(t)}{dt} = \beta \left(\dfrac{Av(t) - q(t)}{V + A\big[L - x(t)\big]} \right) \\ \dfrac{dv(t)}{dt} = \dfrac{1}{m} \Big[A\big(p_1 - p_2\big) - bv(t) - kx(t) \Big] \\ \dfrac{dx(t)}{dt} = v(t) \end{cases} \tag{1.1}$$

where $\rho = 850kg/m^3$ is the density of the fluid, x_v is the valve orifice where we suppose the orifice area is x_v^2, k is the spring coefficient, b the damping coefficient and m the mass of the piston. As is common in most of the hydraulic systems, the ratio between force and mass is very high rendering the inertia effects less significant.

To identify the parameters which affect the dynamics of the system we consider the step response case as shown in Figure 3

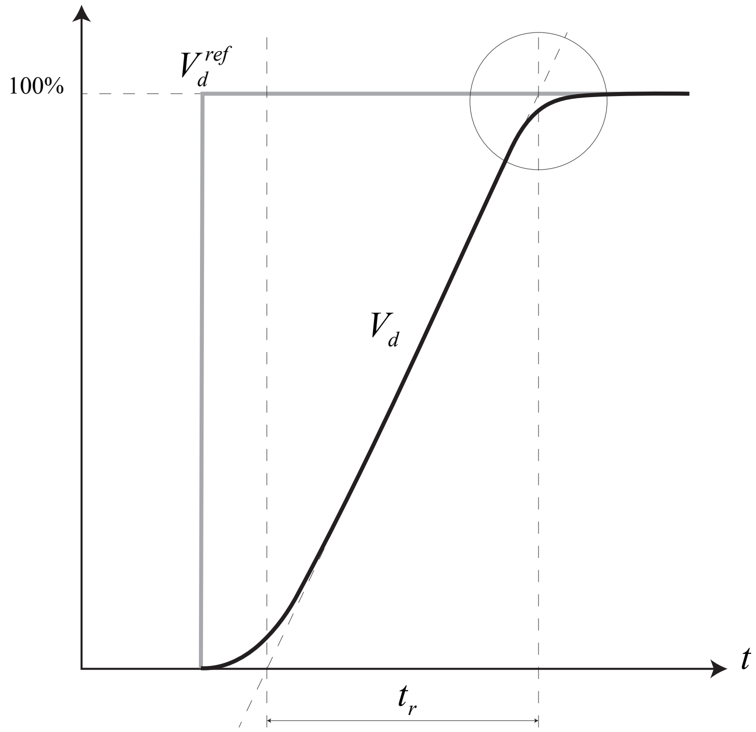

Figure 3: Example of real step response.

where the orifice valve is opened suddenly at maximum area and the piston moves at maximum speed taking the time t_r to cover the overall length L. In this condition we may consider the continuity equation which, neglecting any compressibility effects and leakages, results in the following balance equation

$$q^{max} = Av^{max} \tag{1.2}$$

or

$$C_v \left(x_v^{max} \right)^2 \sqrt{\frac{2}{\rho}} \sqrt{|p_{supply} - p_1|} = A \frac{L}{t_r} \tag{1.3}$$

the second equation which must hold is the equilibrium force at steady state condition, which reads

$$kL = |p_1 - p_2| A \tag{1.4}$$

supposing $p_{return} = 0$ and $p_{supply} = p_{nom}$, we can assume, at nominal condition, $p_1 = \frac{p_{nom}}{2}$ and $p_1 - p_2 = \frac{p_{nom}}{2}$ resulting in

$$\begin{cases} C_v \left(x_v^{max} \right)^2 \sqrt{\frac{2}{\rho}} \sqrt{\frac{p_{nom}}{2}} = A \frac{L}{t_r} \\ kL = \frac{p_{nom}}{2} A \end{cases} \tag{1.5}$$

where t_r is measured from step a response. For the purpose of the parameter fitting the following constraints should be taken into account

$$\begin{cases} k \leq \frac{p_{nom} A}{2L} \\ q(x_v^{max}) \geq A \frac{L}{t_r} \end{cases} \tag{1.6}$$

which means the selected spring coefficient k should not exceed a certain value but can be also selected lower and the selected maximum valve orifice x_v^{max} should ensure enough flow rate to meet the piston rise time requirement.

The control of the linear actuator includes, in general, a position controller of the swash-plate. The control of the swash plate is implemented by a feedback controller ch actuates the orifice of the valve. The control of the valve orifice is actuated by a coil which is managed by an internal current control. For a given coil current i corresponds to a given value of the orifice position x_v (we assume the area of the orifice is x_v^2). The dynamic of the current control (which is not instantaneous) can be approximated by a PT1 transfer function with a time constant of

$$\tau = \frac{L_{coil}}{R_{coil}}$$

Figure 4 shows the implemented control with modelling of the coil transfer function and saturation (with antiwindup) of the valve orifice.

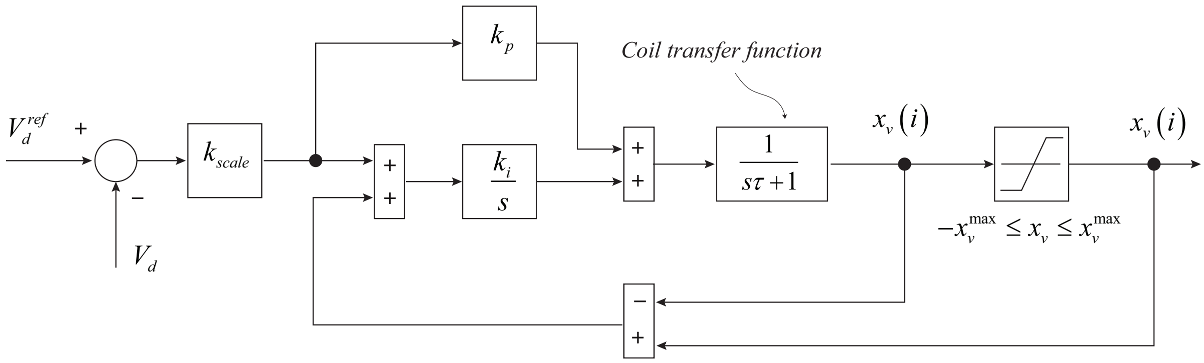

Figure 4: Control servo architecture.

1.3 Case study - Liebherr Danfoss H1P147

Consider the nomalized step response, shown in Figure 6 evaluated with the H1P147 simulink model released by Danfoss. The step response is evaluated applying a voltage step to the valve coil and measuring the flow rate in the circuit supposing the engine speed is constant at 2000 rpm and no-load, see Figure 5.

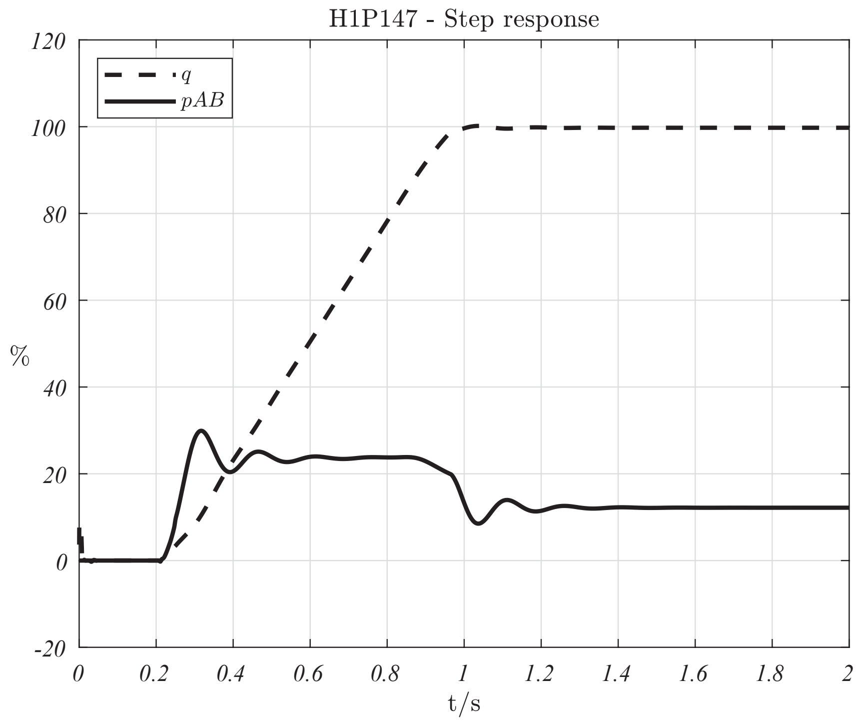

Figure 5: H1P147 step response - drive line differential pressure (solid) and flow (dashed).

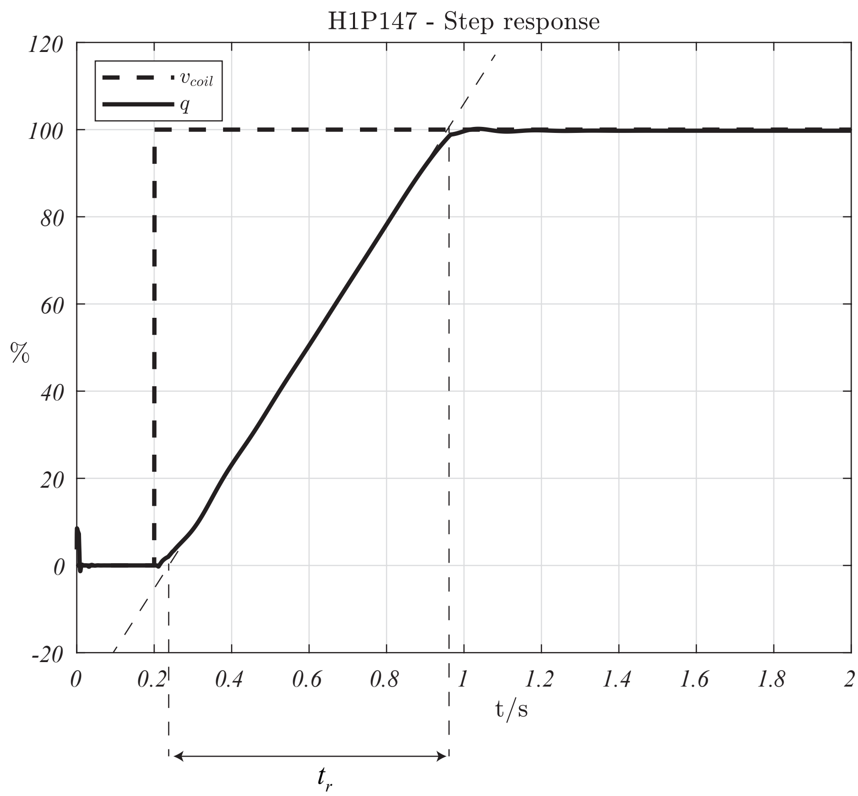

Figure 6: H1P147 step response - spool voltage (dashed) and flow (solid).

By a simple comparison we can evaluate that $t_r = 0.7\,\mathrm{s}$.

Using the digital twin model *EDC Danfoss H1P147* the parameters which affect the dynamics of the servo are identified as

- `Servo-actuator parameters`
  - `Piston area`
  - `Piston stroke (both ways)`
  - `Spring coefficient`
- `Valve parameters`
  - `Max. orifice area = (Max. orifice opening)`^{2}
  - `Max. orifice opening`
- `Spool control`
  - `Proportional gain`
  - `Integral gain`
  - `Lag time constant (coil):` this data is derivable from datasheet of the pump.

The list of parameters above shall be tuned according an experimental test (or from the supplier model). The rest of parameters of the digital twin model can be leaved constant or, where is possible, taken from a data-sheet, e.g. coil parameters for the spool valve.

In the next section, we will treat two methods for dimensioning the above list of parameters:

- In the first method, using preliminary assumptions and some hypothetical data such as `Piston stroke` or `Piston area`, we manually derive the missing of parameters of our simulation model.
- In the second method we are showing how to use a simulink tool, called `Parameter Estimator` to derive the tunable parameters. This method could also be useful to bound some parameters in order to obtain results in accordence with some physical aspects.

1.3.1 Preliminary estimation by hand

In the firs method we assume the following operating condition

- the boost pump has a nominal flow rate of $q_{nom} = 52\,\mathrm{L\,min^{-1}}$ at $2000\,\mathrm{min^{-1}}$ of engine speed.
- the nominal working pressure of the linear actuator is about
$$\Delta p = \frac{p_{nom}}{2} \approx 12\,\mathrm{bar}$$
- the nominal flow rate (at maximum piston speed) is about
$$q = \frac{q_{nom}}{2} \approx 26\,\mathrm{L\,min^{-1}}$$
.

and the following physical data

- The rise time (as shown in Figure 6) $t_r = 0.7\,\mathrm{s}$.
- `Piston stroke (both ways)` $= 0.04\,\mathrm{m} = 2L$
- $\tau = \frac{L_{coil}}{R_{coil}} = \frac{140\,\mathrm{mH}}{14.20\,\Omega}$

Hence the `Piston area` and `Spring coefficient` are calculated as follows

$$\begin{cases} \texttt{Piston area} = A = \dfrac{q_{nom} t_r}{2L} = 0.0152\,\mathrm{m}^2 \\ \texttt{Spring coefficient} = k = \dfrac{p_{nom} A}{2L} = 910 \times 10^3\,\mathrm{N\,m^{-1}} \end{cases} \tag{1.7}$$

The dimensioning of the valve parameters `Max. orifice opening` and `Max. orifice area` are selected to guarantee the maximum speed of the servo piston which results in the following flow requirement

$$\begin{cases} \texttt{Max. orifice opening} = x_v^{max} = \sqrt{\dfrac{q_{nom}}{2} \dfrac{1}{0.7\sqrt{\frac{p_{nom}}{2}}\sqrt{\frac{2}{\rho}}}} = 0.003\,41\,\mathrm{m} \\ \texttt{Max. orifice area} = (x_v^{max})^2 = 1.165 \times 10^{-5}\,\mathrm{m}^2 \end{cases} \tag{1.8}$$

Proportional and integral gain of the valve position control can be selected manually by the simulation or automatically by the application `Parameter Estimator` available in Simulink.

- $k_p = 1$
- $k_i = 0.01$

1.3.2 Refining by iterative method

The second method is based on the automatic estimation of the parameters. Starting from the previous calculation it is possible, by the `Parameter Estimator` tool, to refine the parametrization fitting.

Using the `Parameter Estimator` we have obtained the following data

- `Servo-actuator parameters`
  - `Piston area` $= 0.019\,49\,\mathrm{m}^2$
  - `Piston stroke (both ways)` $= 0.036\,47\,\mathrm{m}$
  - `Spring coefficient` $= 809 \times 10^3\,\mathrm{N\,m^{-1}}$
- `Valve parameters`
  - `Max. orifice area = (Max. orifice opening)`^{2} $= 1.255 \times 10^{-5}\,\mathrm{m}^2$
  - `Max. orifice opening` $= 0.003\,54\,\mathrm{m}$
- `Spool control`
  - `Proportional gain` $= 1.014$
  - `Integral gain` $= 0.001$

1.3.3 Simulation results

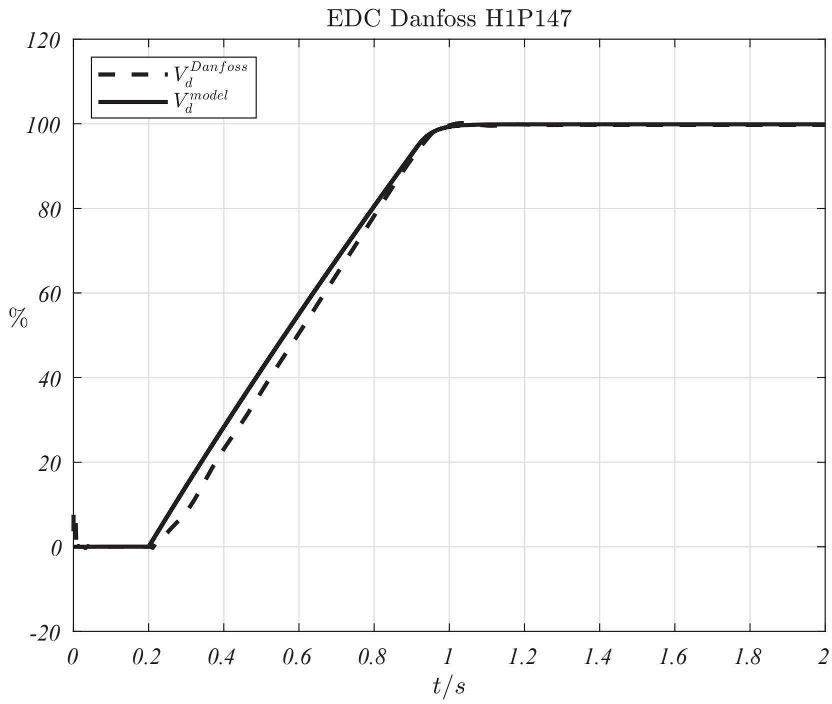

Figure 7: Step response before automatic fitting (refine).

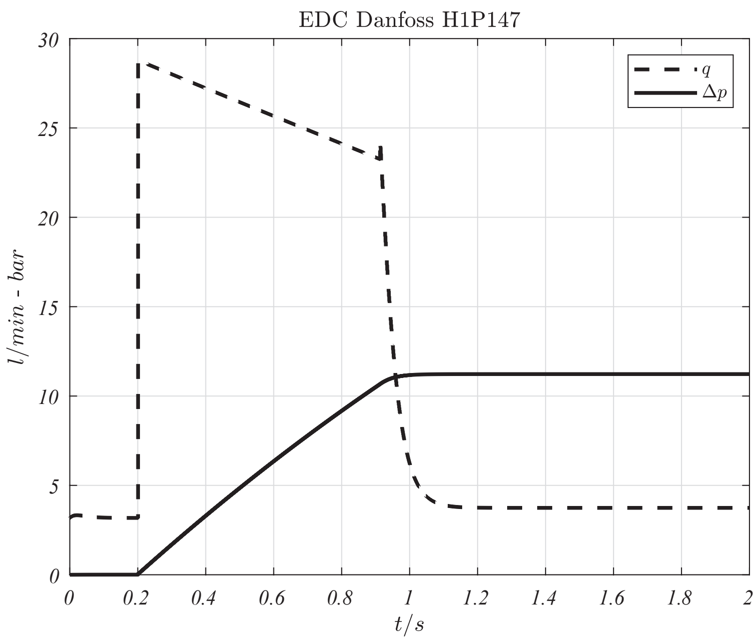

Figure 8: Step response before automatic fitting (refine) - supply flow q (dashed) and servo differential pressure (solid).

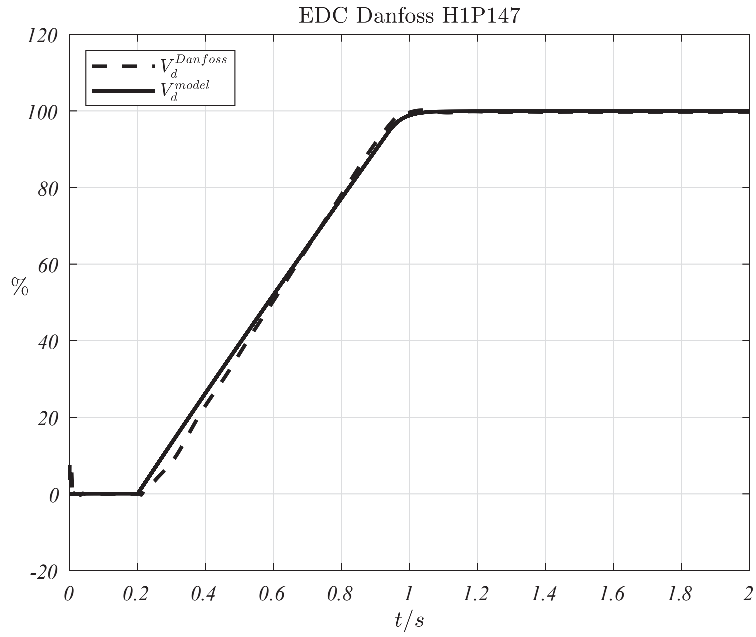

Figure 9: Step response after automatic fitting (refine).

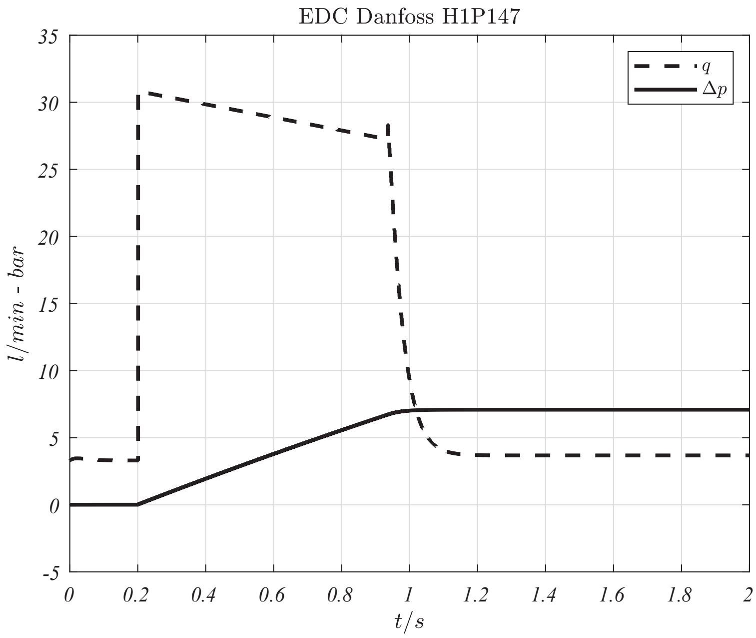

Figure 10: Step response after automatic fitting (refine) - supply flow q (dashed) and servo differential pressure (solid)

1.4 Case study - Prinoth Bosch Rexroth A4VG

The servo circuit of the **Bosch Rexroth A4VG** may be supplied by on of the two **boost pumps**. For our dimensioning of the servo components we will consider the pump with the lowest nominal flow rate, as this is a worst-case scenario.

For this component a step response is not available, hence, we will use the same step response of the Danfoss H1P147.

As before, we assume the following operating condition

- the boost pump has a nominal flow rate of $q_{nom} = 57\,\mathrm{L\,min^{-1}}$ at $2000\,\mathrm{min^{-1}}$ of engine speed.
- the nominal working pressure of the linear actuator is about
$$\Delta p = \frac{p_{nom}}{2} \approx 12\,\mathrm{bar}$$
- the nominal flow rate (at maximum piston speed) is about
$$q = \frac{q_{nom}}{2} \approx 28.5\,\mathrm{L\,min^{-1}}$$
.

and the following physical data

- The rise time (as shown in Figure 6) $t_r = 0.7\,\mathrm{s}$.
- `Piston stroke (both ways)` $= 0.04\,\mathrm{m} = 2L$
- $\tau = \frac{L_{coil}}{R_{coil}} = \frac{1}{2\pi 35\,Hz} = 0.004\,55\,\mathrm{s}$

The `Piston area` and `Spring coefficient` are calculated as follows

$$\begin{cases} \texttt{Piston area} = A = \dfrac{q_{nom} t_r}{2L} = 0.0167\,\mathrm{m^2} \\ \texttt{Spring coefficient} = k = \dfrac{p_{nom} A}{2L} = 999 \times 10^3\,\mathrm{N\,m^{-1}} \end{cases} \tag{1.9}$$

The dimensioning of the valve parameters `Max. orifice opening` and `Max. orifice area` are selected to guarantee the maximum speed of the servo piston which results in the following flow requirement

$$\begin{cases} \texttt{Max. orifice opening} = x_v^{max} = \sqrt{\dfrac{q_{nom}}{2} \dfrac{1}{0.7\sqrt{\frac{p_{nom}}{2}}\sqrt{\frac{2}{\rho}}}} = 0.003\,58\,\mathrm{m} \\ \texttt{Max. orifice area} = (x_v^{max})^2 = 1.28 \times 10^{-5}\,\mathrm{m^2} \end{cases} \tag{1.10}$$

Starting from these initial values, an automatic fitting procedure has been done, giving the following results

- `Servo-actuator parameters`
  - `Piston area` $= 0.0356\,\mathrm{m}$
  - `Piston stroke (both ways)` $= 0.0334\,\mathrm{m}$
  - `Spring coefficient` $= 645 \times 10^3\,\mathrm{N\,m^{-1}}$
- `Valve parameters`

  - `Max. orifice area = (Max. orifice opening)`^{2} $= 2.5 \times 10^{-5}\,\mathrm{m}^2$
  - `Max. orifice opening` $= 0.005\,\mathrm{m}$
- `Spool control`
  - `Proportional gain` $= 1.18$
  - `Integral gain` $= 0.122$

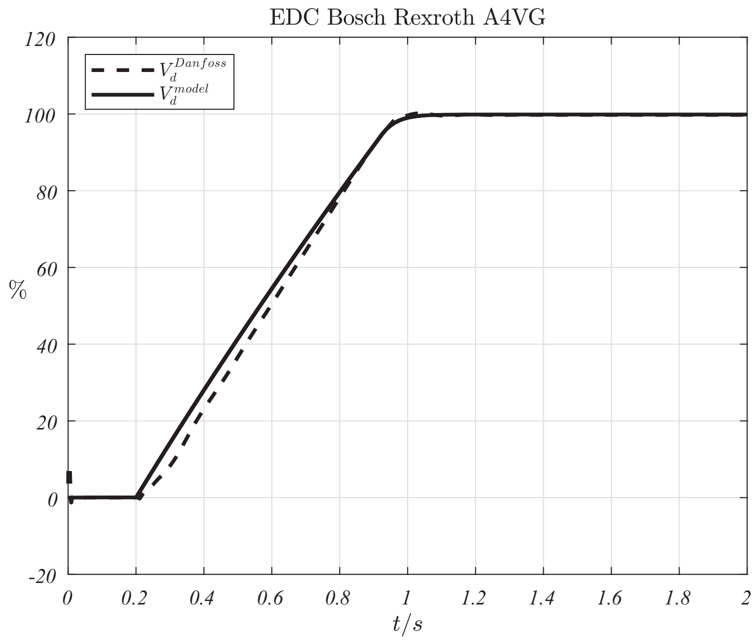

Figure 11: Step response before automatic fitting (refine).

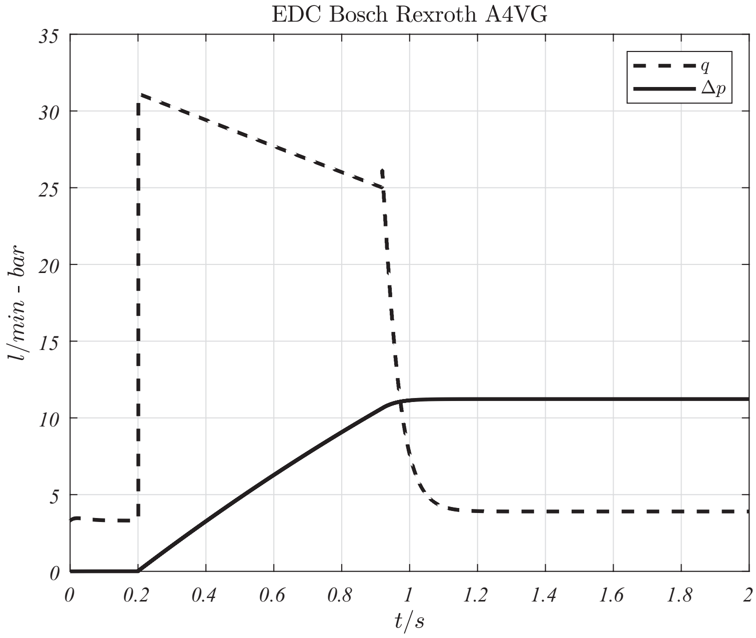

Figure 12: Step response before automatic fitting (refine) - supply flow q (dashed) and servo differential pressure (solid).

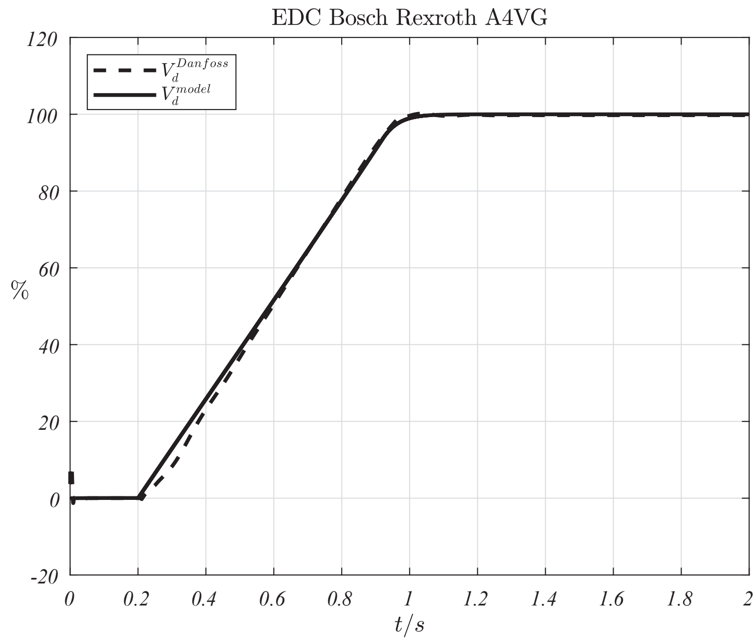

Figure 13: Step response after automatic fitting (refine).

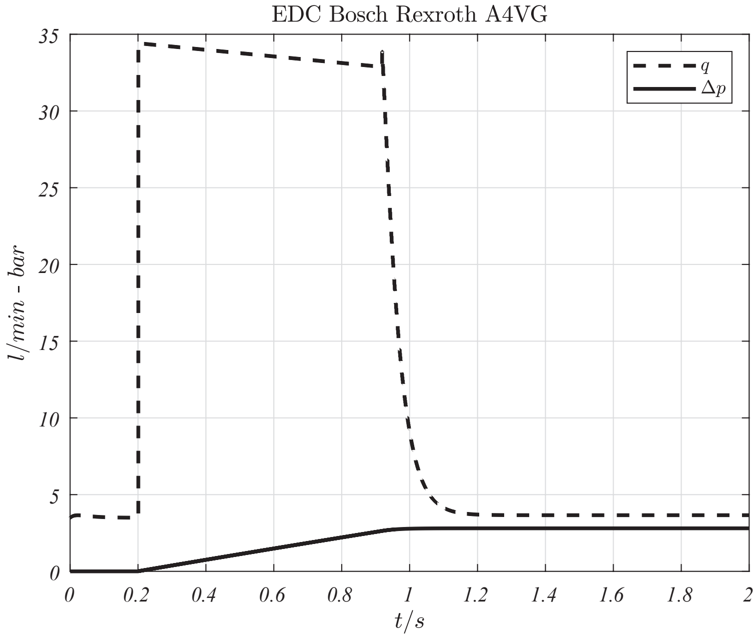

Figure 14: Step response after automatic fitting (refine) - supply flow q (dashed) and servo differential pressure (solid)

1.5 Some additional comments to the model parameters

The model contains some additional parameters reported here

- `Servo-actuator parameters`
  - `Piston mass` $= 1\,\mathrm{kg}$
  - `Damping coefficient` $= 250\,\mathrm{N\,m^{-1}\,s}$
  - `Viscous coefficient` $= 250\,\mathrm{N\,m^{-1}\,s}$
- `Valve parameters`
  - `Orifice opening offset` $= 1.9 \times 10^{-4}\,\mathrm{m}$
  - `Flow discharge coefficient` $= 0.7$
  - `Leakage area` $= 1 \times 10^{-12}\,\mathrm{m^2}$
- `Spool control`
  - `Antiwindup factor gain` $= 1$
  - `Scaling factor` $= 5 \times 10^{-4}$

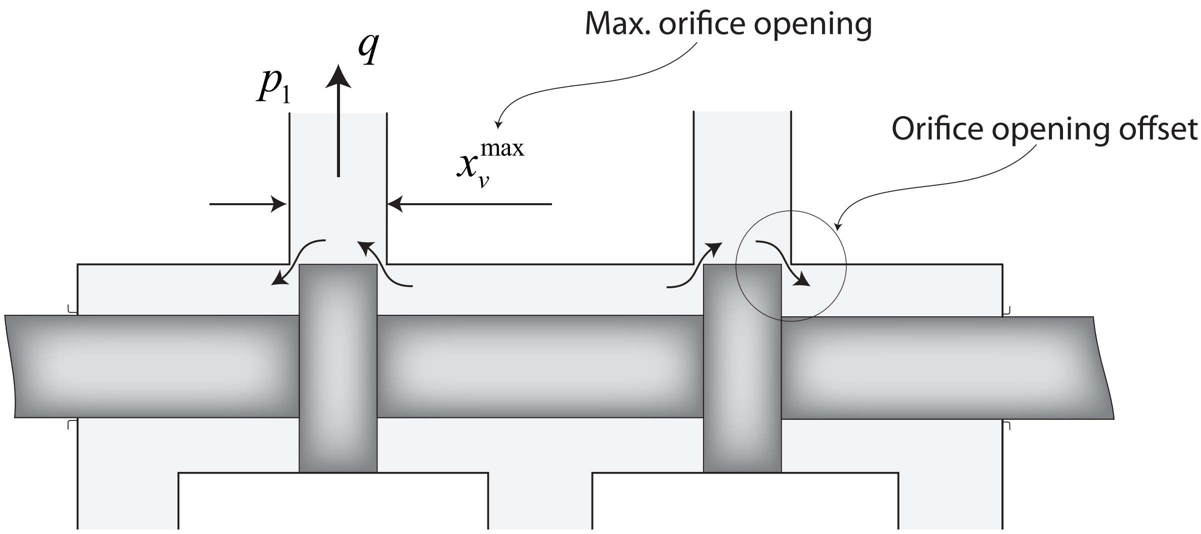

Figure 15: Representation of the `Orifice opening offset` of the valve. The offsets generate an always available flow across the valve.

All these parameters, except for `Orifice opening offset`, have very low impact on the dynamics of the system and hence we will keep them constant. It is supposed that these values will not change until the order of the "power" of the components changes up to two orders of magnitude.

The parameter `Orifice opening offset` is used to introduce a certain amount of orifice always available in a valve, as shown in Figure 15.

The value of the parameter `Orifice opening offset` has been selected in order to have a continuous flow below of $5\,\mathrm{L\,min^{-1}}$ where boost pressure is about $29\,\mathrm{bar}$ as also shown in Figure 14.

2 Motor Swash Plate Control

2.1 Introduction

This section describes a simplified version of the *motor swash plate control* presented in the *digital twin* model of Liebherr vehicle. The aim of this analysis is to identify the main physical quantities which affect the dynamic and steady state behavior of the servo.

2.2 Description of the system

Figure 16 shows the layout of the motor swash plate servo: the **Double Check Valve** provides to the servo cylinder B-chamber (**Hydromotor Servo**) always the maximum pressure available among the drive line. That permits to keep the motor volumetric displacement, in case of none control action, as close as possible to the maximum volumetric displacement. The maximum volumetric displacement permits, for a given value of delta line pressure, the maximum load available. That represent a safe working condition. The swash plate is controlled by an external PI control by the valve **3-2 Way Control Valve** which regulates the pressure to the A-chamber of **Hydromotor Servo**. The difference among the pistons area of A-chamber and B-chamber generates the moving force which is contrasted by the servo spring. The **3-2 Way High Pressure Cut-off Valve** is enabled when the maximum drive pressure become greater than a certain threshold. When it happen, the **3-2 Way High Pressure Cut-off Valve** drops pressure from the **Hydromotor Servo** A-chamber to permit the piston to move toward to the maximum volumetric displacement.

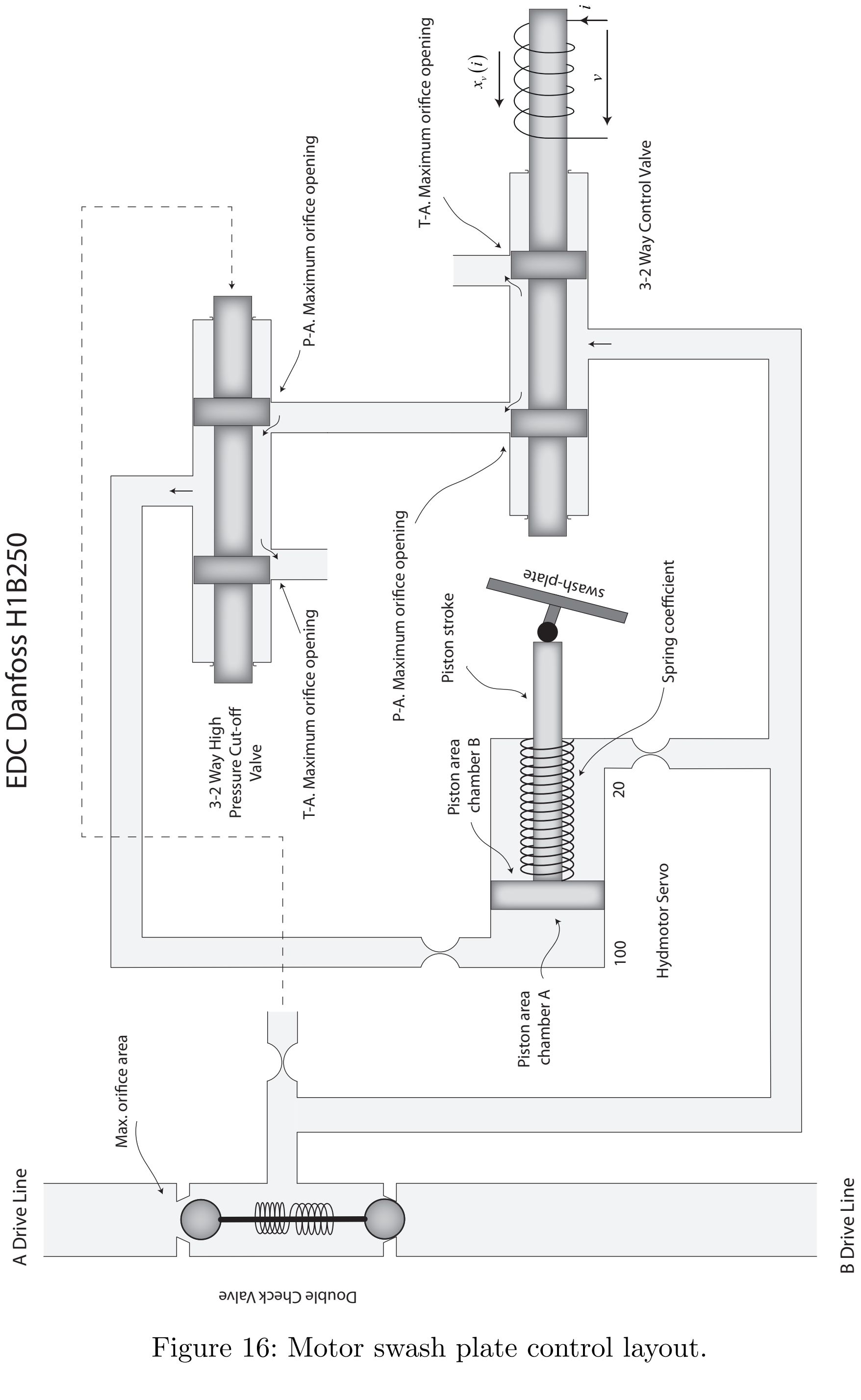

Figure 16: Motor swash plate control layout.

Coupled to the motor, there is also the **Loop Flushing Valve System** which extracts from the low pressure side of the driveline fluid used to cool down the whole components. The cooling chamber is called **Case**. The **Loop Flushing Relief Valve** operates in a way to keep constant the extracted flow from boost line toward the case.

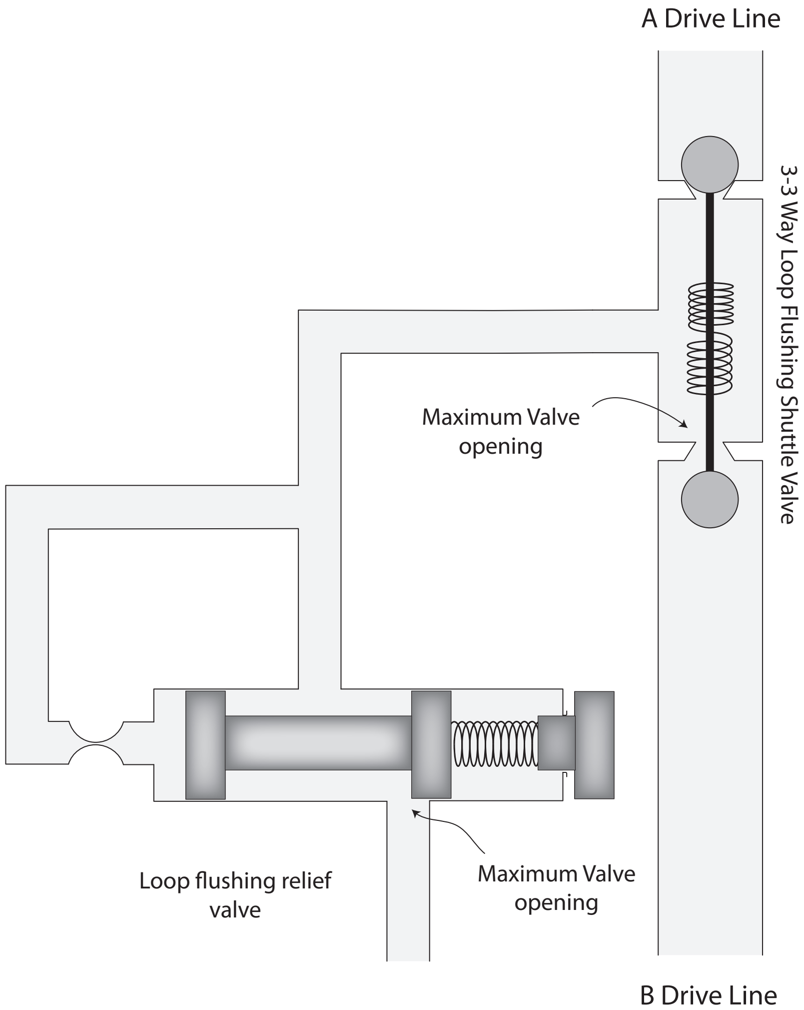

Figure 17: Loop flushing valves system.

3 Driveline Checks Valves

3.1 Introduction

This section describes the checks and relief valves which are implemented among the driveline of the *digital twin* models of Prinoth and Liebherr vehicles. The aim of this analysis is to identify their main physical quantities which affect the dynamic and the steady state behavior.

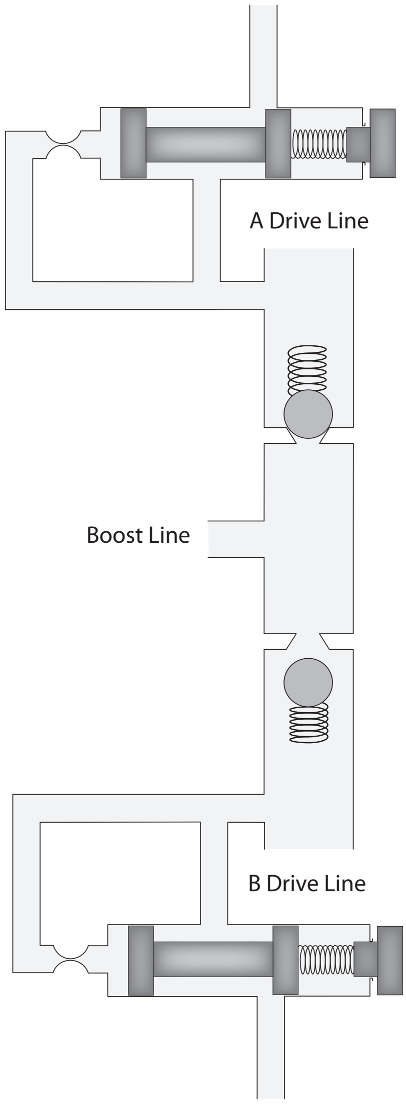

Figure 18: Driveline check and relief valves.

As shown in Figure 18 among the drive lines a pair of check and relief valves are present. The aim of the check valves is to feed the drive line in order to charge them at the boost pressure. During the normal operating condition one check valve is always closed (the one connected to the high pressure side). On the other hand the pressure relief valve drop out flow in case its pressure line exceed a certain pressure threshold.

it to keep the low pressure side of the drive line close to the boost pressure and to

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