MODELING AND CONTROL OF WET CLUTCHES BY PRESSURE-CONTROL VALVES

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Abstract: This paper proposes an energy-based detailed dynamic model of an hydraulic wet clutch actuator controlled by a pressure-control valve. The proposed system model has been validated by comparison with experimental measurements. Based on the main features of the electro-hydraulic clutch actuator system, the advantages of a proper choice of the pressure-control valve are tested through simulation experiments.

Keywords: Modelling, Hydraulic actuators, Valves, Pressure control, PID control.

1. INTRODUCTION

Multi-layer wet clutches are often used in many automotive applications where high torques, high resistance and high durability are required. Common applications include automatic transmission gearboxes and limited slip differentials.

Modern electronic controls operate the clutch by an hydraulic actuator and an electro-valve. The torque transmitted through the shafts depends mainly on the clutch-actuator internal oil pressure and this pressure is controlled by means of the electro-valve. To achieve good functioning performances an accurate pressure control is necessary, see (Haj-Fraj and Pfeiffer, 2000).

This paper proposes an energy-based detailed dynamic model of an hydraulic clutch actuator with a pressure-control electro-valve. The proposed system model has been validated by comparison with experimental measurements.

Based on the main features of the electro-hydraulic clutch actuator system, a proper choice of the pressure-control valve is proposed and the corresponding tracking performances with a simple PI controller are tested through simulation experiments.

The paper is organized as follows: Sec. 2 introduces the concepts of the modeling technique used to describe dynamic behavior of the systems. The clutch actuator is described in Sec. 3. The system model is proposed in Sec. 4 and it is validated through experimental measurements in Sec. 5. Finally, a proper choice of the pressure-control valve is proposed and tested in Sec. 6.

2. POWER-ORIENTED GRAPHS

The POG modeling technique is based on the same Bond Graph idea of using the power interaction between sub-systems as basic concept for modeling, see (Karnopp and Rosemberg, 1975). Compared with the Bond Graph, the POG technique has a more intuitive modular graphical notation essentially based on the two blocks shown in Fig. 1. The basic characteristic of this modular structure is the direct correspondence between pairs of system variables and real power flows: the product of the two variables involved in each dashed line of the graph has the physical meaning of power flowing through the section.

The two basic blocks shown in Fig. 1 are named "elaboration block" (e.b.) and "connection block" (c.b.). The black dot means a change of sign of the corresponding input variable in the sum node. There is no restriction on variables $\mathbf{x}$ and $\mathbf{y}$ other than the fact that the inner product $\langle \mathbf{x}, \mathbf{y} \rangle = \mathbf{x}^T \mathbf{y}$ must have the physical meaning of a power.

The e.b. and the c.b. are suitable for representing both scalar and vectorial systems. In the vectorial case, $\mathbf{G}(\mathbf{s})$ and $\mathbf{K}$ are matrices: $\mathbf{G}(\mathbf{s})$ is always square, $\mathbf{K}$ can also be rectangular. While the elaboration block can store and dissipate energy (i.e. springs, masses and dumpers), the connection block can only transform the energy, that is, transform the system variables from one type of energy-field to another (i.e. any type of gear reduction). Since the energy is simply transformed from an energetic field to another, it is $\langle \mathbf{x_1}, \mathbf{y_1} \rangle = \langle \mathbf{x_2}, \mathbf{y_2} \rangle$ and consequently the connection block must show the matrices $\mathbf{K}$ and $\mathbf{K}^T$.

Main characteristics of the POG technique are: a direct correspondence between the POG blocks and the real parts of the system; the POG schemes can be easily transformed, both graphically and mathematically; the state space mathematical model of a system can be directly obtained from the corresponding POG representation. For a more detailed description of the POG graphical technique, please refer to (Zanasi, 1991) and (Zanasi, 1993). The POG schemes are particularly suitable to electro-hydraulic mechanical systems where the power flows through different energetic domains, see (Morselli et Al, 2002).

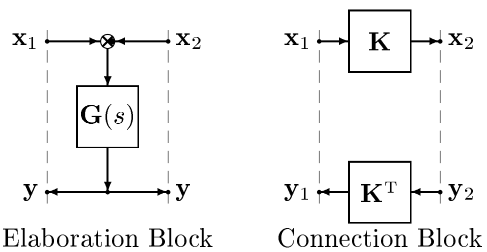

Fig. 1. Power Oriented Graph basic blocks.

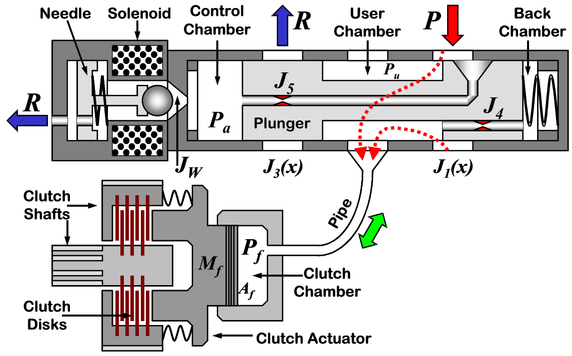

Fig. 2. Clutch control system: control valve (enlarged and partially connected to the hydraulic power supply P) and wet-clutch schematic representation.

3. SYSTEM DESCRIPTION

The pressure-control valve is basically a current-to-hydraulic power amplifier with an internal pressure feedback. As sketched in Fig. 2 the internal plunger moves mainly under the effect of the oil pressures P_a and P_b in the control chamber and in the back chamber respectively. The pressure P_a in the control chamber behaves as a reference for the output user pressure P_u, it can be changed modulating the hydraulic resistance of the orifice J_w by means of the solenoid current. The back chamber pressure P_b is the feedback signal coming from the user pressure P_u. In a steady state condition $P_b = P_u$. The input current sets the resistance J_w and consequently the pressure P_a. If $P_a > P_b$ the plunger moves and connects the output orifice to the hydraulic power supply P, the user pressure P_u rises and consequently P_b increases. As soon as $P_a \simeq P_b \simeq P_u$ the plunger, thanks to the spring, moves back to the central position closing both connections towards the supply pipes (P and R) and the output pressure is set to a value close to the reference P_a. The spring in back chamber pushes the plunger towards the control chamber and keeps the output pipe connected to the oil tank pressure R when the control solenoid is non-energized. To avoid the contemporary openings of the supply orifices (P and R) a positive lap (dead zone) is used.

The valve output pipe is connected to the clutch chamber C_f. The clutch piston is pushed by the pressure P_f (in the chamber C_f) against the clutch springs and disks. The force that compresses the clutch disks determines the amount of torque transmitted through the clutch. The solenoid current and the valve allow to control the pressure P_f and, consequently, the compressing disks force and the torque transmission.

This kind of clutch control system is commonly used in applications as automatic transmission gearboxes and limited slip differentials where high torques have to be quickly and precisely controlled. In a steady state operating mode, a pressure sensor for the implementation of an electronic pressure feedback control is not necessary thanks to the valve internal feedback. When fast and precise pressure transitions and immunity to parameter variations are required, an electronic pressure feedback becomes necessary.

4. POG MODELS

The main device of the clutch control system is the pressure-control valve. The control valve is similar to the diesel injector presented in (Morselli et Al, 2002) and some functioning principles are common to both devices, but the mechanical similitude hides many differences: the injector works at higher pressures and with a very fast on-off mode, while the control valve works in continuous mode with lower pressures. Therefore, due to the different application environment, the control valve model is quite different from the injector model.

The physical system can be divided into four interacting subsystems: the valve plunger, the control chamber, the user chamber and the actuator.

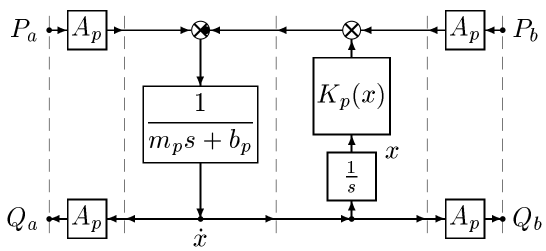

Fig. 3. Valve plunger subsystem POG model

The valve plunger subsystem POG model is shown in Fig. 3. Let x and $\dot{x}$ denote, respectively, the plunger position and velocity. The plunger mass m_p moves subject to the forces coming from the viscous friction coefficient b_p, the return spring $K_p(x)$ and the pressures P_a and P_b of the control chamber and of the back chamber, respectively. The nonlinear force $K_p(x)$ models both the force of the return spring and the contact force between the plunger and the plunger case at the two extreme plunger positions. The plunger motion causes the oil flows Q_a and Q_b through the control chamber and the back chamber, respectively.

$$\begin{aligned} m_p\ddot{x} &= (P_a - P_b)A_p - b_p\dot{x} - K_p(x) \\ Q_a &= Q_b = A_p\dot{x} \end{aligned} \tag{1}$$

The pressure P_a in the control chamber is determined by the integration of three oil flows in the hydraulic capacity C_a: the oil flow Q_5 coming from the hydraulic power supply P, the flow Q_a due to the plunger motion and the flow Q_w through the variable discharge orifice. The very small hydraulic capacity C_a stores potential energy in terms of oil pressure and it takes into account the high oil stiffness and the small elastic deformation of the valve case:

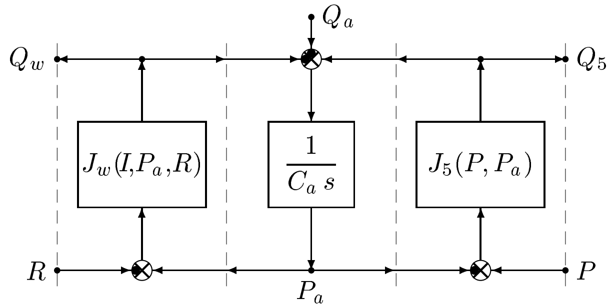

Fig. 4. Control chamber POG model

$$C_a\dot{P}_a = Q_5 - Q_a - Q_w \tag{2}$$

The flows Q_5 and Q_w are nonlinear functions of the pressures P, P_a and R:

$$\begin{aligned} Q_5 &= C_{d5}\sqrt{|P - P_a|}\,\mathrm{sgn}(P - P_a) = J_5(P, P_a) \\ Q_w &= C_{dw}(I)\sqrt{|P_a - R|}\,\mathrm{sgn}(P_a - R) = J_w(I, P_a, R) \end{aligned} \tag{3}$$

The effective discharge coefficient C_{dw} can be varied by the control current I. In the real system C_{dw} is a function of the sphere position which is determined by the pressure P_a and by the needle force due to the current I. Experimental measurements showed that the discharge coefficient C_{dw} can be approximated as a nonlinear function of the control current $C_{dw} = C_{dw}(I)$ with negligible errors. Equations (2) and (3) lead to the POG model shown in Fig. 4.

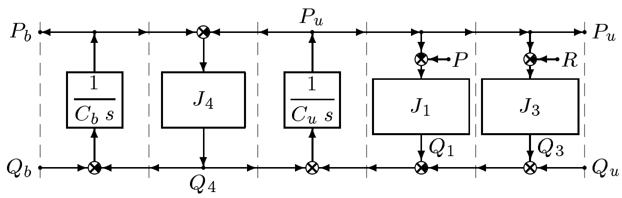

Fig. 5. Back chamber and output user chamber POG models

Depending on the plunger position x, the output user chamber is connected either to the power supply P through the variable orifice J_1 or to the oil tank by the orifice J_3, see eq. (4). The user chamber is connected to the back chamber through the orifice J_4. This orifice plays two fundamental roles: it implements the feedback action since P_b becomes a measure of the user pressure P_u, and it has a damping effect that avoids plunger oscillations. The orifice J_4 must be carefully designed finding the best trade-off between a fast measure of P_u (wide orifice) and a good oscillations rejection (small orifice).

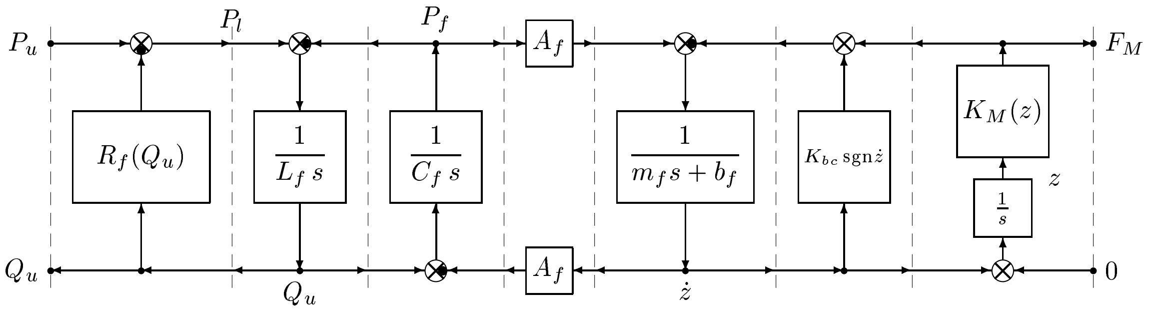

Fig. 6. Clutch actuator POG model

$$
\begin{aligned}
Q_1 &= C_{d1}(x)\sqrt{|P-P_u|}\,\mathrm{sgn}(P-P_u) = J_1(x,P,P_u)\\
Q_3 &= C_{d3}(x)\sqrt{|P_u-R|}\,\mathrm{sgn}(P_u-R) = J_3(x,P_u,R)\\
Q_4 &= C_{d4}\sqrt{|P_b-P_u|}\,\mathrm{sgn}(P_b-P_u) = J_4(P_b,P_u)
\end{aligned}
\tag{4}
$$

The back chamber and the user chamber are modeled as two small hydraulic capacities as for the control chamber:

$$
\begin{aligned}
C_b\dot P_b &= Q_b - Q_4\\
C_u\dot P_u &= Q_1 + Q_4 - Q_3 - Q_u
\end{aligned}
\tag{5}
$$

Equations (4) and (5) are graphically represented by the POG dynamic model of Fig. 5

A pipe connects the valve user chamber to the clutch chamber. The dynamic effects of this pipe cannot be neglected and they are described by four elements: the user chamber capacity C_u, the hydraulic resistance R_f, the pipe hydraulic inductance L_f and the clutch chamber capacity C_f:

$$
\begin{aligned}
L_f\dot Q_u &= P_l - P_f = P_u - P_{Q_u} - P_f\\
P_u - P_l &= \frac{Q_u\,|Q_u|}{C_{df}} = R_f(Q_u)\\
C_f\dot P_f &= Q_u - A_f\,\dot z
\end{aligned}
\tag{6}
$$

Equations (6) defines the left part of the POG model of Fig. 6. The right part represents the motion of the clutch actuator under the effects of the pressure P_f, of the elastic force $K_M(z)$ and of the viscous friction b_f:

$$
\begin{aligned}
m_f\ddot z &= P_f A_f - b_f\dot z - K_M(z) - K_{bc}\,\mathrm{sgn}(\dot z)\\
K_M(z) &= K_F(z) + K_D(z)
\end{aligned}
\tag{7}
$$

The elastic force $K_M(z)$ is the sum of two contribution: $K_F(z)$ represents the force of the return springs and the contact with the gearbox at the two extreme actuator positions; $K_D(z)$ is the force generated by the compression of the clutch disks that determines the maximum transmissible torque through the clutch. Since $K_F(z) + K_{bc} + b_f\dot z \ll K_D(z)$ and $\ddot z \simeq 0$, when the actuator presses the disks, the clutch chamber pressure P_f is the main contribution to determine the torque transmission.

5. EXPERIMENTAL SET-UP AND MODEL VALIDATION

The experimental set-up consists in a complete clutch actuator with voltage and current sensors on the valve solenoid, a pressure sensor in the user chamber and a position sensor for the actuator.

Acting on the solenoid supply voltage, two main measurements have been made: a long time (almost steady state) measure that stresses the static parameters of the model and a fast measure for the dynamic parameters. Due to privacy reasons the figure axes are normalized.

The result of a steady state measure compared with the corresponding simulation are shown in Fig. 7. Both the clutch chamber pressure and the clutch actuator position are well replicated by the simulation. The detail shows the pressure peak due to the bump of the actuator against the disks.

Using the same model parameters of the steady state measure, the simulation results for the fast measure fit with the experimental data as shown in Fig. 8.

The proposed model gives reliable simulation results, therefore it is used to test a pressure-control valve together with a simple controller in the next section.

6. CONTROLLER AND PRESSURE-CONTROL VALVE

The first benefit of a pressure sensor mounted close to the clutch chamber is the possibility to measure the steady state relation $P_f = P_{fwd}(I)$ between the control current I and the corresponding pressure P_f. Inverting the relation P_{fwd}, the function $I = I_{fwd}(P_f)$ for the controller feedforward action is found.

Let P_{fr} denote the pressure reference for the clutch chamber. The proposed control law (8) implement a simple standard PI regulator with saturated integral action to reduce windup effects:

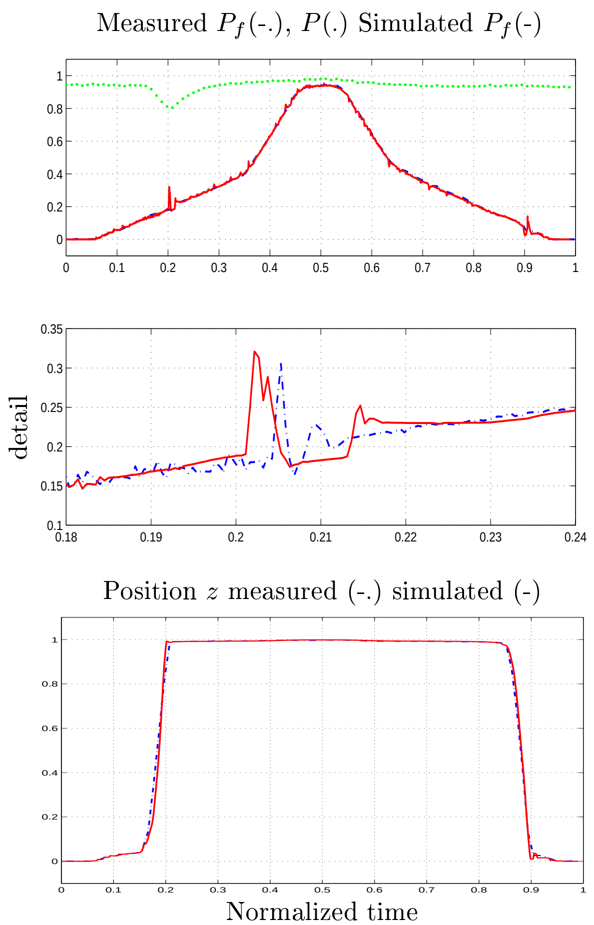

Fig. 7. Steady-state measurements. Supply pressure P (dotted), clutch chamber pressure P_f measured (dash-dotted) and simulated (solid). Clutch actuator position z: measured (dash-dotted) and simulated (solid).

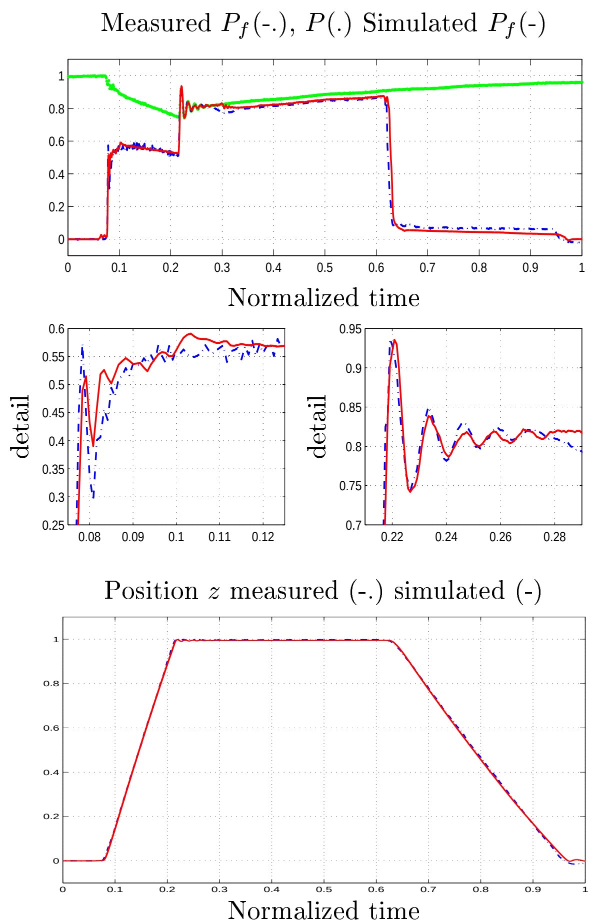

Fig. 8. Step input measurements. Supply pressure P (dotted), clutch chamber pressureP_f measured (dash-dotted) and simulated (solid). Clutch actuator position z: measured (dash-dotted) and simulated (solid).

$$\begin{cases} I = K_P(P_{fr} - P_f) + K_I\ u + I_{fwd}(P_{fr}) \\ \dot{u} = \begin{cases} 0 \ \text{ if } \ (P_{fr} - P_f) > 0 \text{ and } u \geq u_{max} \\ 0 \ \text{ if } \ (P_{fr} - P_f) < 0 \text{ and } u \leq -u_{max} \\ (P_{fr} - P_f) \quad \text{else} \end{cases} \end{cases} \tag{8}$$

The discrete time version of the control law (8) including the computational time has been tested by simulation experiments with two slightly different pressure-control valves.

The torque control is operated modulating the clutch pressure when the actuator presses the disks, the actuator movements are negligible due to the high disks stiffness and little changes in the amount of oil stored in the clutch chamber produce wide pressure modifications.

Usually pressure-control valves have a positive lap: the shape of the internal plunger avoids the contemporary connection between the output pipe and the two supply inputs (P and R), see Fig. 2. This mechanical solution is chosen by the valve manufacturer for safety reasons, in fact this is the safest choice when the application purpose is not known. The positive lap produces a dead-zone from a control theory point of view.

The main strong problem related to the electronic pressure feedback is the highly nonlinear behavior of the system in the torque control condition: to obtain a modulation of the clutch pressure, the control valve plunger must move across the dead zone caused by the positive lap. Moreover, little openings of the sealed clutch chamber produce large pressure changes, therefore the system shows an high intrinsic gain. From a theoretical point of view, it is well known that any computational delay together with an high gain and a dead zone cause oscillations and instability.

This forecasted behaviour has been observed also in simulation. Fig. 9 shows the control performances when the control law (8) is applied to control the modeled system, whose valve has a positive lap. The controlled pressure shows overshoots and oscillations. To improve the controller performances it would be necessary to reduce the gains but slower time responses would be obtained. Moreover an inaccurate feed forward action would cause large steady state tracking errors.

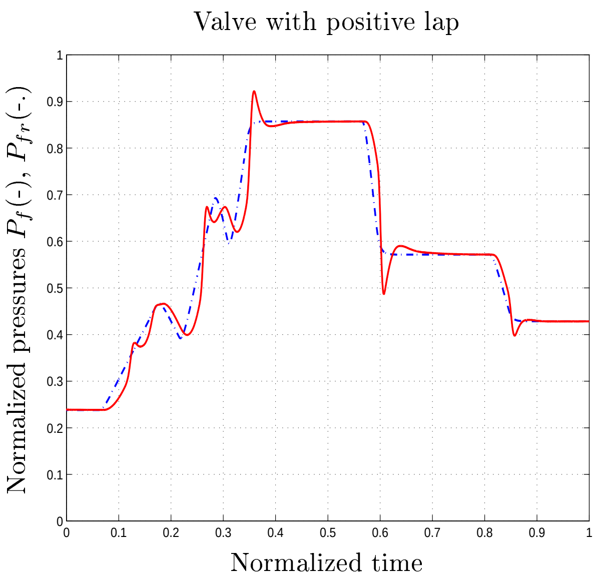

Fig. 9. Simulation results. PI controller and valve with positive lap (dead zone): reference pressure P_{fr} (dash-dotted) and clutch pressure P_f (solid).

To solve this problem a simple mechanical modification is proposed: a small negative lap eliminates the dead zone and it reduces the intrinsic gain since the clutch chamber is not sealed when the plunger is around its central position. This solution does not affect the system behaviour when the clutch is fully engaged or disengaged and it greatly improves the pressure control performances.

Control law (8) with a negative lap valve are used to control the clutch pressure, Fig. 10 shows the corresponding performances. A very accurate reference tracking is obtained.

The negative lap causes an hydraulic power loss during the torque control phase. Since a small negative lap is sufficient to obtain the performances shown in Fig. 10, the power losses are negligible respect to the average hydraulic power consumption.

7. CONCLUSIONS

A model of a wet-clutch hydraulic actuator has been proposed. Comparisons between measured data and simulation results validated the model for both slow and fast dynamic behaviours.

The proposed model allowed to verify that the implementation of a simple mechanical modification on the pressure-control valve could improve the closed-loop tracking performances for the clutch pressure and therefore it could ensure a better modulation of the transmitted torque.

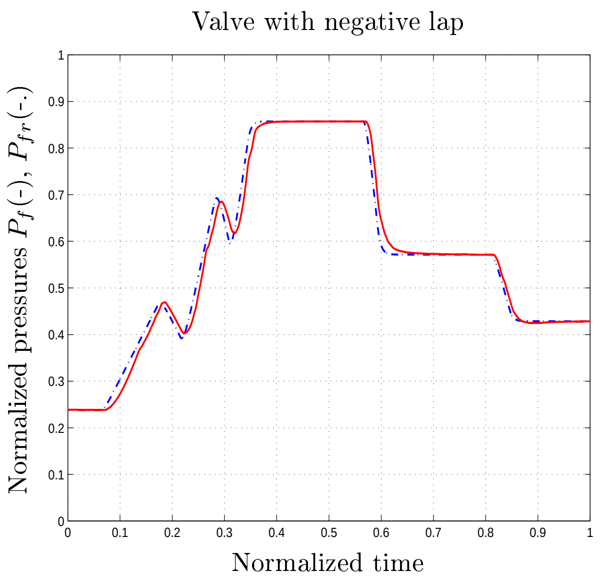

Fig. 10. Simulation results. PI controller and valve with negative lap: reference pressure P_{fr} (dash-dotted) and clutch pressure P_f (solid).

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