BULETINUL INSTITUTULUI POLITEHNIC DIN IAŞI
Publicat de
Universitatea Tehnică „Gheorghe Asachi” din Iaşi
Tomul LVI (LX), Fasc. 4, 2010
Secţia
AUTOMATICĂ şi CALCULATOARE

DYNAMIC SIMULATOR FOR AN ELECTRO-HYDRAULIC WET CLUTCH[1]

BY

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Abstract. The dynamic behaviour of an electro-hydraulic wet plate clutch offers many advantages regarding the reduction of fuel consumption, shifting quality and improvement of driving comfort. In this paper, an analytic model of a wet plate clutch actuated by a pressure reducing valve is developed. Next this model is converted into a Simulink model, with which simulation can be performed in an easy way. To validate the model a test bench, developed by Automotive Continental Romania, was used.

Key words: automatic transmission, clutch, valves, hydraulic actuators, displacement, simulators, nonlinear systems.

2000 Mathematics Subject Classification: 53B25, 53C15.

1. Introduction

The automotive industry changed people ideas of boundaries, limits, and distance. With the arrival of the automobile, people suddenly had the freedom to go farther faster than ever before. Then engineers had to improve efficiency and comfort. This was the time when automatic transmissions come into play.

The automatic transmission system was the subject for many other publications [3], [9], [11], [18], where the main idea was to improve the control algorithms or mechanical design to reduce energy losses. Attempts were made to keep the traditional clutch and gear stick untouched but removing the clutch pedal [8]. This was replaced by an actuator controlling the clutch position and

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[1] This is an extended version of the paper: Feru E. *et al.*, *Dynamic Simulator of a Wet Plate Clutch System for Automatic Transmission*. Proc. ICSTC 2010, 207−212, 2010.

therefore the torque transmission. After some precautions were taken, a two-level cascaded feedback design was used to control the vehicle speed with this clutch system. Another type of approach was made by [16] where the automatic transmission is modeled in detail for each component. There, a dynamic model of a power transmission system for gear ratio changed was proposed. Later, [4] developed a mathematical model of a one-way clutch in belt-pulley systems, where a wrap-spring type of clutch is modeled as a nonlinear spring with discontinuous stiffness. In [17] a model is presented that includes the non-linear nature of the diaphragm spring, kinematics of the pedal motion during clutch release, and the dynamics of the driveline and the overall vehicle.

Researches made until now [1], [15], tell that torque can be varied by modifying the clutch position and although they are mostly based on using dry clutches, they give a good input for the modeling of some automatic transmission elements. Some major advantages of the wet plate clutches can be mentioned [14]: they are built with multiple clutch disks that give a better grip which can be controlled by the pressure inside the clutch. The shifting speed can be considerably reduced by using independent clutch assemblies known as the dual-clutch system. A wet clutch is immersed in a cooling lubricating fluid, which keeps the surfaces clean and gives smoother performances and longer life. Disadvantages are related to the additional energy losses by always actuating a hydraulic pump and the use of high fluid pressure which is unsafe for the environment.

Electro-hydraulic valve models were developed in many other research areas [6], [13], in order to demonstrate that designed control systems [12], [19] assure stability robustness and corresponding performances. In the automotive field [2], [7], solenoid valves as electro-hydraulic actuators which have high pressure and large flow capacity have been developed. Furthermore, mathematical models and control design algorithms were proposed for the studied electro-hydraulic systems.

In this paper, a dynamic model of a wet plate clutch based on mathematical equations obtained with the help of physical laws and experimental measurements is developed. A hydraulic valve actuator is considered in order to analyze the general effect of the automatic transmission system. For validating the model and to procure all the experimental data needed, a test bench developed by Automotive Continental Romania was used. What brings new this model described by the mathematical equations is that a new simulator was developed for a non standardized type of pressure reducing valve and wet plate clutch. This simulator can be then used to make diverse simulations in order to understand the behavior of the system, to design complex strategies of control or to test and improve a controller performances already implemented.

The paper is organized as follows. In Section 1 the modeling of a pressure reducing valve and a wet plate clutch is discussed. In Section 2 both elements are evaluated and results are discussed by means of simulation in Section 3. Finally, conclusions are drawn in Section 4.

2. Wet Clutch System Modeling

In this section the modeling of a wet plate clutch actuated by a pressure reducing valve is presented. Hydraulic control valves are devices that use mechanical motion to control a source of fluid power and are used as actuators in many control applications for automotive systems. Basically there are three types of control valves: directional control valves, pressure control valves and flow control valves. The type of valve and clutch analyzed in this paper are not standardized, they are designed for a certain application where some performances need to be achieved. In what follows it is presented the dynamics of the valve, then the dynamics of the clutch.

2.1. Pressure Reducing Valve Model

Schematics of the three way pressure reducing valve used as clutch actuator is presented in Fig. 1. The input for this hydraulic system is represented by the line pressure p_s and current i passing through a solenoid which generates a magnetic force F_{mag}, while the output of the system is represented by the pressure p_r. The physical construction of the valve implies the presence of three pressure variables corresponding to each chamber: p_c, p_r and p_d for the left chamber, the middle chamber and the right chamber respectively.

The hydraulic valve has a self controller by using a mechanical feedback through left and right tubes. Because differences in pressure appear between the middle chamber and the left/right chambers, flows Q_c and Q_d are generated. The feedback force is composed from the force F_c applied on the left sensed pressure chamber, and the force F_d applied on the right sensed pressure chamber. These two forces composed with the magnetic force F_{mag} are used to actuate the spool valve which controls the pressure p_r inside the middle chamber. In the charging phase (Fig. 1 *a*), the magnetic force given by solenoid current, is grater than the feedback pressure force moving the plunger to the left, connecting the source with the clutch. In the discharging phase (Fig. 1 *b*), the magnetic force is less than the feedback force implying that plunger will move to the right, connecting the clutch to the tank.

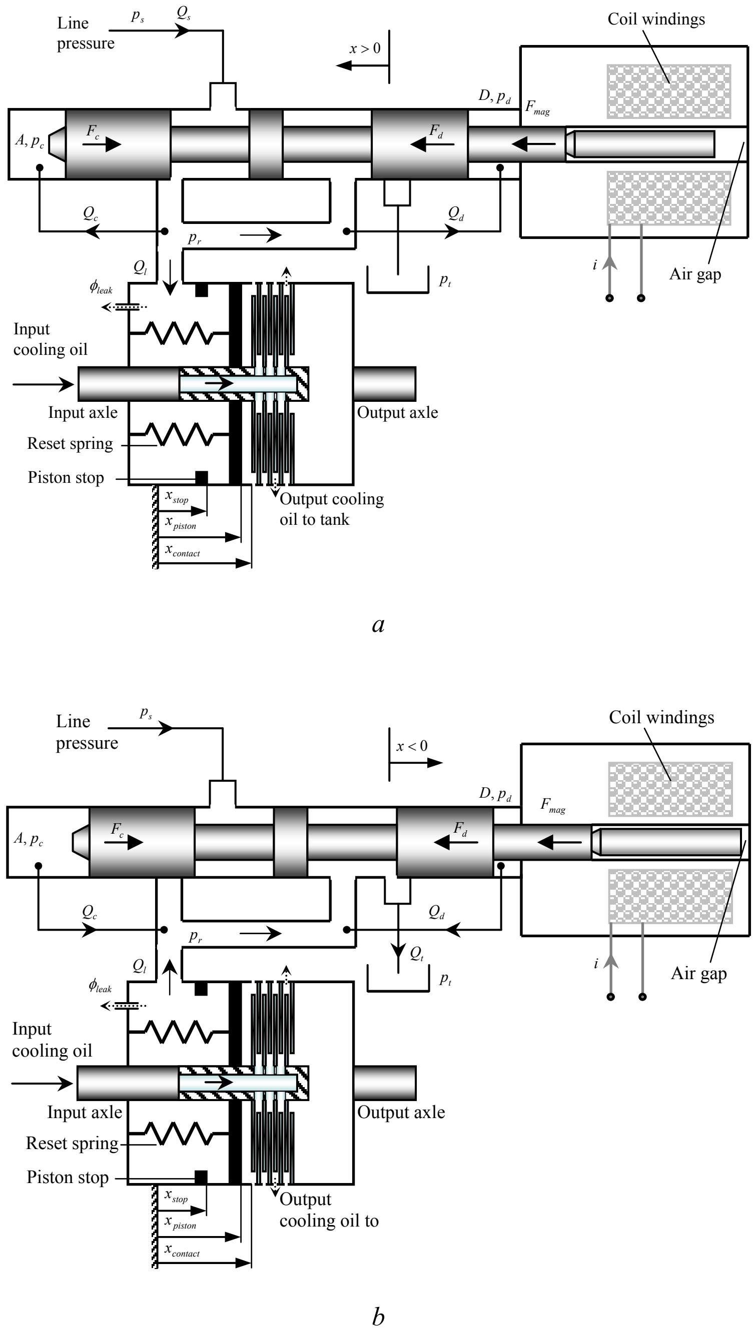

Fig. 1 – Schematic draw of the valve connected to the clutch:
a – Charging phase; *b* – Discharging phase.

The mathematical connection between electric current trough solenoid and magnetic force generated by the magnetic flux is given in [10].

Feedback pressure force and magnetic force are the most important forces that give the forces balance applied on the valve plunger. Based on Pascal equations which transform the hydraulic pressure into mechanic force we can simulate the feedback pressure by next equation:

(1) $$F_{feedback} = Ap_c - Dp_d \,,$$

where: p_c , p_d are the pressures of the left and right chambers, A − the left plunger area of pressure contact, D − the right plunger area of pressure contact.

Using the magnetic force and feedback force created by the pressure inside the left and right chambers the movement of the plunger, which governs the output pressure, can be approximated with a mass spring damper system given by:

(2) $$F_{mag} - F_{feedback} = M_v\ddot{x} + c\dot{x} + K_e x \,,$$

where: $F_{feedback}$ is the feedback force, M_v − the spool mass, x − the plunger displacement, $\dot{x}$ − the plunger speed, $\ddot{x}$ − the plunger acceleration, K_e − the flow force spring rate and c is the damper coefficient.

To restrict the motion of the plunger between the left and the right bounds a double-side mechanical translational hard stop was implemented:

(3) $$F_{restrict} = \begin{cases} K_p x + D_p\dot{x}, x \geq x_{\max} \\ 0, x_{\min} < x < x_{\max} \\ K_n x + D_n\dot{x}, x \leq x_{\min} \end{cases} \,,$$

where: $F_{restrict}$ is the dynamic balance force between the plunger and the mechanical limits, $x_{\min}$, $x_{\max}$ − the gap on the right and left side, K_p, K_n − the contact stiffness at left and right restriction, D_p, D_n − damping coefficient at left and right restriction.

The linearity of pressure drop and flow, a characteristic of all laminar flows, is desirable in many circuits. Capillary tubes are often used to stabilize pressure control valves, as upstream restrictors in hydrostatic bearing. For this reason flow through left and right tubes were considered, for simplicity, to be laminar.

Flows Q_c and Q_d through a tube for both charging and discharging phase are given by [5]:

(4) $$Q_c = \frac{\pi r_c^4}{8\varepsilon L_c}(p_r - p_c)\,,$$

(5) $$Q_d = \frac{\pi r_d^4}{8\varepsilon L_d}(p_r - p_d)\,,$$

where: r_c, r_d are the left tube radius and right tube radius, L_c, L_d − the passage length of the tubes, ε − the kinematic viscosity of fluid.

To describe the dynamics given by the left and right chamber the flow continuity equation for each chamber is given by:

(6) $$Q_c = \frac{V_c - Ax}{\beta_e}\frac{dp_c}{dt} - A\frac{dx}{dt}\,,$$

(7) $$Q_d = \frac{V_d + Dx}{\beta_e}\frac{dp_d}{dt} + D\frac{dx}{dt}\,,$$

where: V_c, V_d are the sensing chamber volumes for the initial spool displacement, β_e − the effective bulk modulus.

In the (6) and (7) equations the sign for the left and right chamber volume were chosen based on the fact that total volume variation of these chambers depend accordingly to spool displacement. Therefore, a movement of the plunger to the left will decrease the left chamber volume and increase the right chamber volume.

From the (6) and (7) equation the derivative of the pressure sensed in the left and right chamber respectively can be written as follows:

(8) $$\frac{dp_c}{dt} = \frac{\beta_e}{V_c - Ax}\left(Q_c + A\frac{dx}{dt}\right),$$

(9) $$\frac{dp_d}{dt} = \frac{\beta_e}{V_d + Dx}\left(Q_d - D\frac{dx}{dt}\right),$$

An important remark can be made regarding the (6) and (7) equations, that direction of flow Q_C and Q_D are given accordingly to the plunger movement.

The flow behavior for the orifices that connects the clutch to the source and the clutch to the tank was modeled considering variable orifices created by a cylindrical sharp-edged spool. The flow rate through the orifice is proportional to the orifice opening and to the pressure differential across the orifice. The model accounts for the turbulent flow regimes because fluid passes through these orifices with a high speed leading to high Reynolds numbers.

For the charging phase the flow rate from the source to the clutch is determined according to the next equation:

$$Q_s = C_d A_s(x)\sqrt{\frac{2}{\rho}\left|p_s - p_r\right|}\,, \tag{10}$$

$$A_s(x) = bx + A_{leak}\,, \tag{11}$$

where: C_d is the flow discharge coefficient, $A_s(x)$ − the instantaneous orifice area, b − the orifice slot width, A_{leak} − the orifice leakage area, ρ − the fluid density. For simplicity, A_{leak} was considered to be 0.

The flow continuity equation at the chamber of the pressure being controlled for the charging phase is represented by:

$$Q_s - Q_c - Q_d - Q_l = \frac{V_s}{\beta_e}\frac{dp_r}{dt}\,. \tag{12}$$

For the discharging phase the flow rate from the clutch to the tank is obtained in the same manner and given by:

$$Q_t = C_d A_s(x)\sqrt{\frac{2}{\rho}\left|p_r - p_t\right|}\,, \tag{13}$$

where: p_t is the tank pressure.

For this functioning phase of the valve the flow Q_l, in this case called the discharging flow, will be in opposite direction and included in the following equation:

$$Q_l + Q_c + Q_d - Q_t = \frac{V_s}{\beta_e}\frac{dp_r}{dt}\,. \tag{14}$$

From (12) and (14) equations it can be noticed that flows Q_c and Q_d will be considered as a perturbation for the flow Q_l which depends on the load connected to the output of the valve. In our case the load is a wet plate clutch and flow Q_l will be obtained accordingly to the dynamics given by this hydraulic element.

These equations define the valve and can be used to construct the block diagram in Fig. 2.

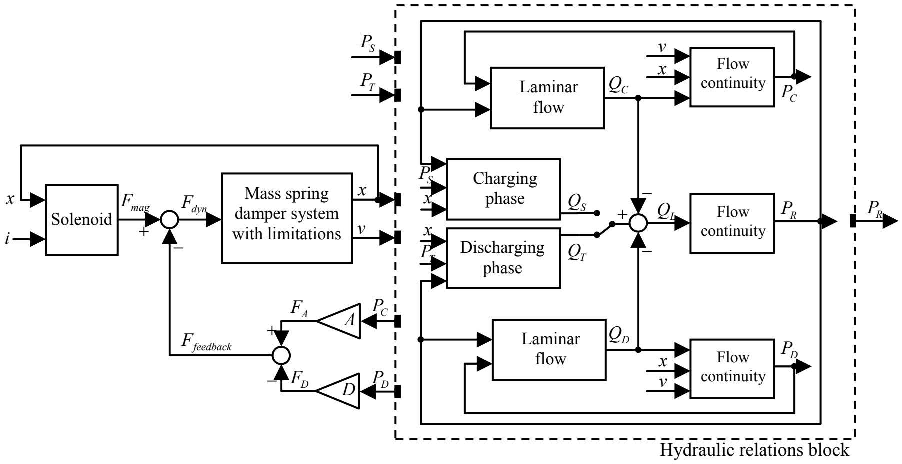

Fig. 2 – Nonlinear valve system representation.

The solenoid block is considered to be a look-up table, obtained with the help of the experimental measurements, having as input the current and displacement of the plunger and as output the magnetic force. The mass spring damper system is implemented accordingly to the eq. (2) introducing the limitations (3) given by the left bound and right bound of the pressure reducing valve. In the charging phase and discharging phase blocks eqs. (10) and (13) were materialized to obtain the flow which comes from source and the flow going to the tank, respectively. The flows Q_c and Q_d were computed using relations (4) and (5) in the blocks called laminar flow. Finally, to obtain the pressures p_c, p_d and the output pressure p_r eqs. (8), (9) and (12), (14) were implemented in the flow continuity blocks. The pressures p_c, p_d and p_r were then used as inputs for the laminar flow blocks and only p_r as input for the charging/discharging phase blocks.

2.2. Wet Plate Clutch Model

A wet plate clutch is schematically given in Fig. 1. The clutch itself is a chamber with a piston. Behind the piston the clutch plates are fixed on the input axle, respectively on the output axle. The friction surfaces of these plates are covered with an organic material, lubricated with cooling oil, which is brought in through the input axle.

When the clutch closes, considering a step input, four stages can be distinguished for the clutch pressure (Fig. 3):

1) filling of the clutch, without significant pressure increase;

2) pressure increase until the pressure is large enough to exceed the force applied on the piston by the reset spring;

3) movement of the piston towards the clutch plates;

4) pressure rise to the set point value p_{set} after the clutch plates resume contact.

Initially the clutch is considered to be open having no fluid inside the chamber. The piston is pulled back to the piston stops (Fig. 1) by the reset springs with total stiffness k_{reset}. Their elongation in this case is denoted by x_{stop}. The piston itself is assumed to be mass less with a cross sectional area A_{piston} on which the clutch pressure is effective.

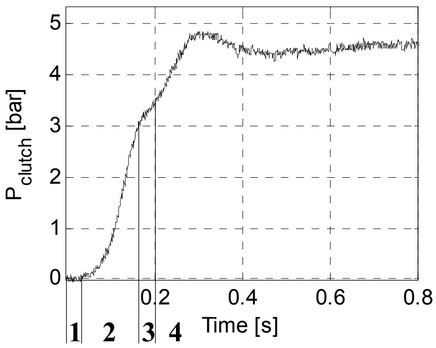

Fig. 3 – Step response of the clutch pressure.

An important fact in clutch functioning is that piston movement takes place only in stage three and it moves from x_{stop} to $x_{contact}$.

The piston position x_{stop} can be derived from the equilibrium equation:

$$p_{clutch} A_{piston} + F_{stops} = k_{reset} x_{piston} + F_{friction}, \tag{15}$$

where: F_{stops} is the reaction force of the stops on the piston.

Furthermore, the piston is equipped with a sealing ring which seals the connection between the compartment to the left of the piston and the volume to the right of the piston. Therefore the leakage flow, present between piston and clutch housing, can be neglected. The friction force $F_{friction}$ represents the friction between this sealing ring and the clutch housing.

As soon as the force on the piston exerted by the clutch pressure becomes larger than the sum of the friction force and the force applied on the piston by the reset springs, the piston starts moving towards the clutch plates. This pressure, denoted by $p_{prestress}$, is given by:

$$p_{prestress} = \frac{k_{reset} x_{stop} + F_{friction}}{A_{piston}}. \quad (16)$$

The plates are pressed together with a compression force $F_{compress}$, expressed by:

$$F_{compress} = (p_{clutch} - p_{contact}) A_{piston}. \quad (17)$$

The flow continuity equation written for our application is:

$$Q(t) - Q_{leak}(t) = \frac{dV(t)}{dt} + \frac{V(t)}{k_s}\frac{dp_{clutch}(t)}{dt}, \quad (18)$$

where: $V(t) = V_0 + A_{piston} x_{piston}(t)$ is the volume variation.

After making some calculations in eq. (18) we obtain a more convenient form, for the wet plate clutch model:

$$\dot{p}_{clutch}(t) = \frac{k_s}{V_0 + A_{piston} x_{piston}(t)}\left[-A_{piston}\dot{x}_{piston}(t) + Q(t) - Q_{leak}(t)\right] \quad (19)$$

Furthermore, eq. (19) can be specialized for each of the four stages mentioned before.

Stage 1, filling of the clutch:

$$\begin{gathered} V_{oil,0} + \int_{t=t_0}^{t} Q(\tau)d\tau < V_0 \\ p_{clutch}(t) = \dot{p}_{clutch}(t) = 0 \end{gathered} \quad (20)$$

Stage 2, pressure increase without piston displacement:

$$\begin{gathered} 0 \le p_{clutch}(t) < p_{prestress} \\ \dot{p}_{clutch}(t) = \frac{k_s}{V_0 + A_{piston} x_{stop}}\left[Q(t) - Q_{leak}(t)\right]. \end{gathered} \quad (21)$$

Stage 3, movement of the piston:

$$p_{prestress} \le p_{clutch}(t) < p_{contact}\,. \tag{22}$$

With the relations for the piston displacement $x_{piston}(t)$ and speed $\dot{x}_{piston}(t)$:

$$\begin{aligned} x_{piston}(t) &= \frac{A_{piston}\,p_{clutch}(t) - F_{friction}}{k_{reset}} \approx \frac{A_{piston}\,p_{clutch}(t)}{k_{reset}}\,, \\ \dot{x}_{piston}(t) &= \frac{A_{piston}}{k_{reset}}\,\dot{p}_{clutch}(t)\,, \end{aligned} \tag{23}$$

eq. (19) becomes:

$$\dot{p}_{clutch}(t) = \left(\frac{V_0 + A_{piston}x_{piston}(t)}{k_s} + \frac{A_{piston}^2}{k_{reset}}\right)^{-1}\left[Q(t) - Q_{leak}(t)\right]. \tag{24}$$

Stage 4, compression of the clutch plates:

$$p_{clutch}(t) \ge p_{contact}\,, \tag{25}$$

$$\dot{p}_{clutch}(t) = \frac{k_s}{V_0 + A_{piston}x_{contact}}\left[Q(t) - Q_{leak}(t)\right]. \tag{26}$$

Because the clutch is represented by a closed chamber the flow $Q(t)$ through the valve will decrease as the clutch pressure increases.

Eqs. (20),…,(22), (24),…,(26) represent the wet plate clutch model.

3. Model Validation

The model of the valve actuator and the wet plate clutch were validated and tested using a test bench from Automotive Continental Romania (Fig. 4). Simulation results were recorded and compared with the experimental data provided by this test bench.

The primary function of the test bench is to evaluate the behavior of various hydraulic elements, designed models and control mechanisms. Furthermore it is equipped with several measurement and control devices.

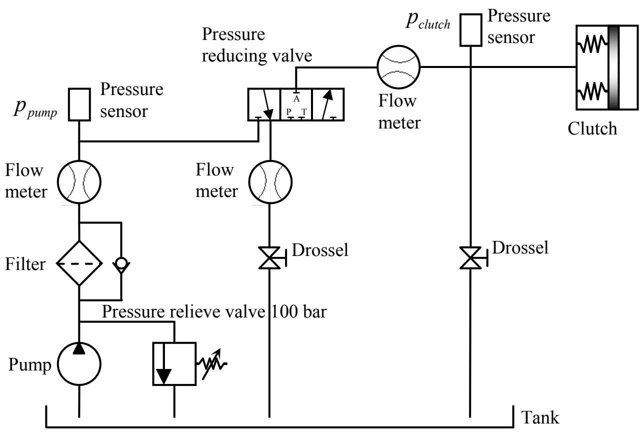

Fig. 4 – Simplified test bench representation.

The created environment for model validation is with the first drossel completely open and the second drossel completely closed. This means the discharging phase of the wet plate clutch system is taking place through the valve, flow meter and first drossel to the tank.

3.1. Simulator of the Valve-Clutch System

The simulator of the valve-clutch system, based on the analytic model described in Section 2, was constructed in Matlab/Simulink and is implemented regarding the connections between the hydraulic valve and wet plate clutch to simulate the system dynamics.

The simulator (Fig. 5) is composed by the valve and clutch model. In the left of this figure are the inputs of the system, while in the right the output variables are recorded in Matlab memory during simulation. The input of the simulator is represented by the line pressure and the current passing through the coil of the valve which generates a magnetic force F_{mag} applied to one end of the plunger. The magnetic force depends on geometry of the valve and this was hard to consider using mathematical support. For this reason, the magnetic force was measured using a force sensor and the results were used in form of a two dimensional look-up table. A low pass filter was used in order to reduce the noise of the current signal.

The resultant of forces represents the input for the mass spring damper block to obtain the plunger displacement, velocity and acceleration. The output of this block is used in eqs. (10), (13) to dictate the flow through the valve and in eqs. (8), (9) for computing the pressure values at the spool end areas.

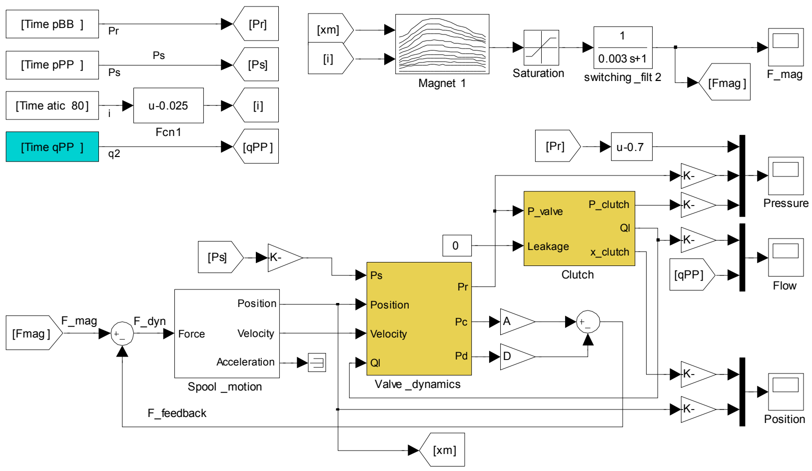

Fig. 5 – Simulator representation of the valve-clutch system.

The model of the valve contains a switch (Fig. 6) whose function is to commute between the charging phase and discharging phase having as a threshold factor the 0 mm of the plunger displacement which means that all the orifices are closed. Next, the flow Q_c and Q_d were implemented based on the (4), (5) eqs. and used as a perturbation for the flow Q_l. In the right side of the Fig. 6 the continuity eqs. (8), (9), and (12), (14) were implemented. The pressure sensed in the left chamber and right chamber creates a feedback force which contributes to the equilibrium of forces.

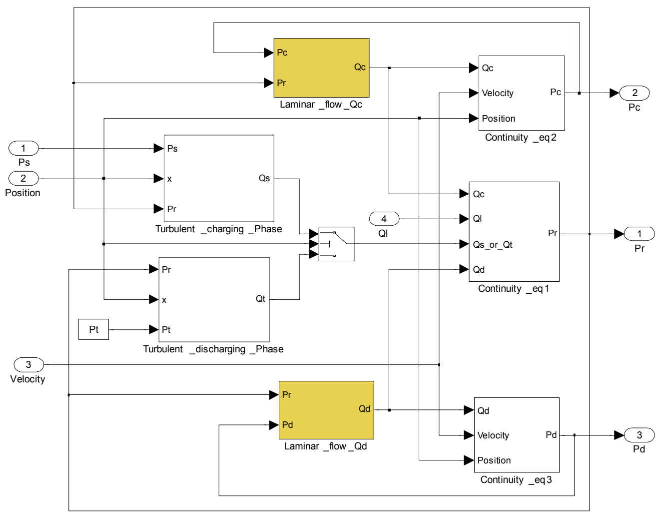

Fig. 6 – Simulink block diagram of the valve.

The input of the clutch is the pressure obtained with the valve dynamics while the output of the system is represented by the pressure inside the clutch, the displacement of the clutch piston and the flow entering in the clutch chamber. The four stages in the pressure response when closing the clutch, as described in the model, are displayed in Fig. 7 as subsystems. Pressure values, like $p_{contact}$ and $p_{prestress}$ are used as switching parameters between the stages. Next the time derivative of the pressure is fed into an integrator to gain the pressure. In this component also the initial pressure can be enclosed. A limitation block is developed to obtain the clutch piston displacement, the clutch piston speed and to represent the stops. The mathematical model of this block and implementation is represented by eq. (3). The input signal for the limitation block is represented by a force which in this case is the force generated by the pressure inside the clutch chamber applied upon the piston area A_{piston}. Furthermore, the relation between the pressure and overall compressibility is converted into a lookup table.

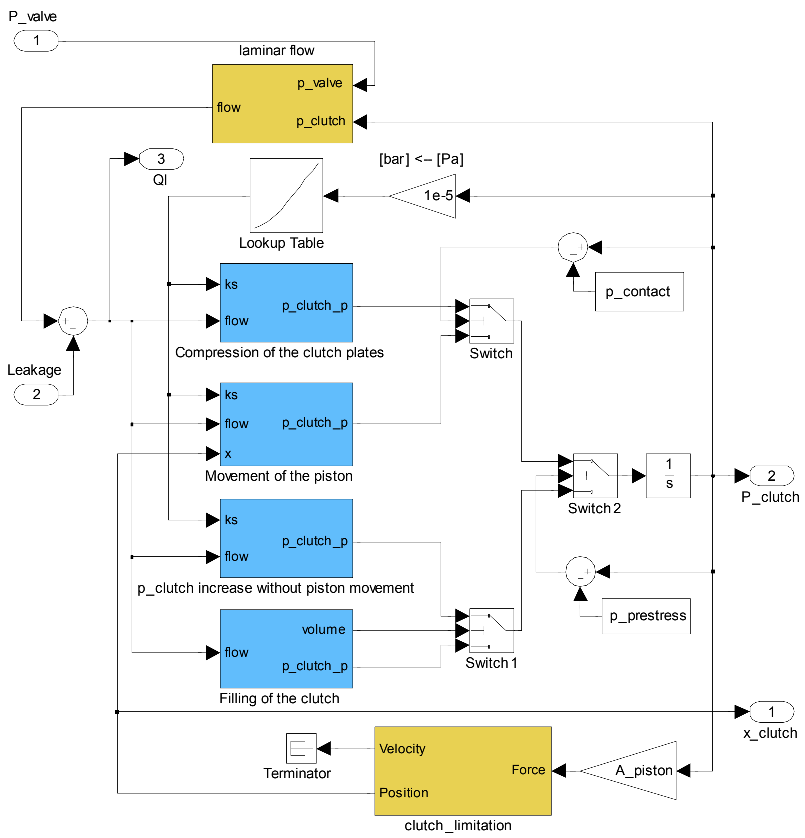

Fig. 7 – Simulink block diagram of the wet plate clutch.

It can be noticed that piston movement takes place only in stage three from x_{stop} to $x_{contact}$. Thereby, a pressure controller can be developed based on piston displacement.

3.2. Simulation Results

In order to validate the model for the solenoid valve actuator and wet plate clutch, a current was applied in coil windings as an input for the simulator described in Section 3. In Fig. 8 the input signal is illustrated, represented by a pulse current from 0 to 630 mA used to obtain the magnetic force through the look-up table. The line pressure is set to 10 bars and the obtained output pressure is about 5 bars.

The simulation results are presented in Fig. 9, where the reduced pressure and flow through the valve were compared with real data obtained from experiments made on the test bench.

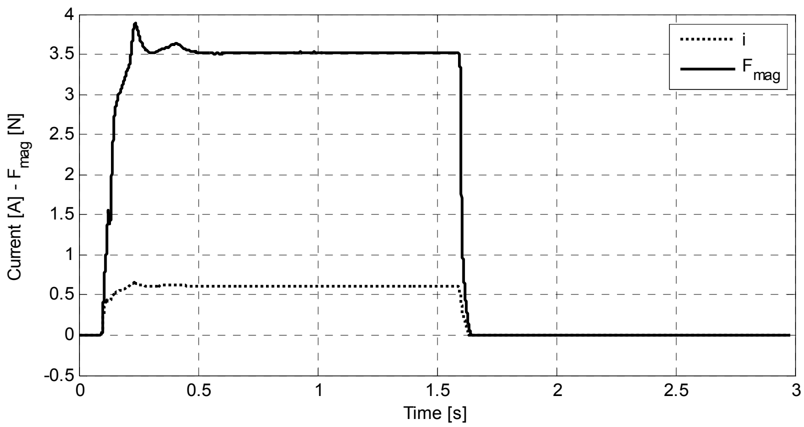

Fig. 8 – Current and magnetic force signals Simulink.

As it can be seen in the first set of signals (Fig. 9 *a*) the clutch pressure and measured pressure are considerably close, ensuring a small modeling error. The simulation signal for the flow passing through the valve (Fig. 9 *b*) shows that some factors like leakage, pipe dynamics were not taken into account. This fact led to a bigger value for the simulation signal toward the flow signal from experiments. Regarding the flow signals it can be noticed some negative aspects about the experimental response: a persistent noise of reading is present and flow sensor can not read reversed fluid flows. The last set of signals (Fig. 9 *c*) shows that piston position of the clutch synchronizes smoothly with the clutch pressure in stage three.

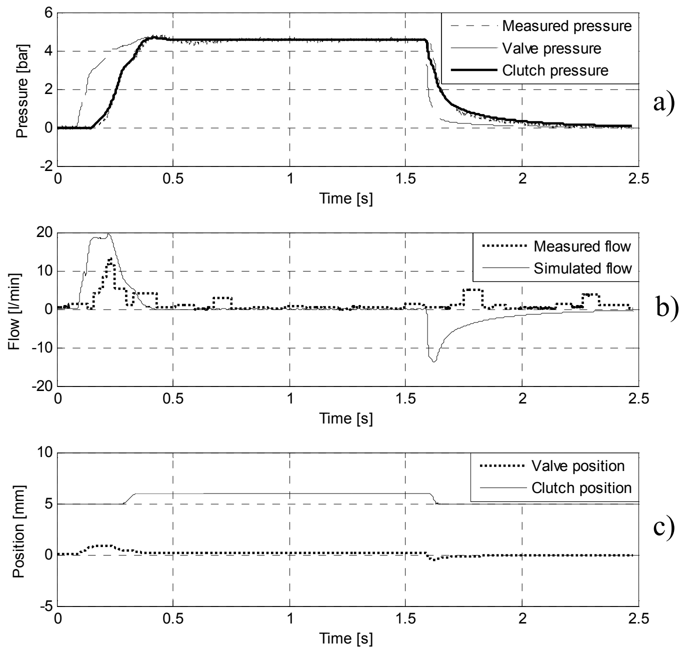

Fig. 9 – Simulation responses compared with the experimental measurements.

Simulations are performed with the valve-clutch system simulator for verifying that model response lie within an acceptable range around the experimental data.

The model shows good performance, being stable for input signals variations and the results of the experiments illustrate a good behavior of the simulator.

4. Conclusions

In this paper a simulator of the valve-clutch system based on mathematical equations, static and transient experiments was developed. The simulator characterizes the dynamics given by the valve, clutch as well as the whole system.

The models are satisfying to the comparison of the simulated and experimental response. It predicts valve pressure, clutch pressure, flow through the system and the position of the hydraulic elements.

A c k n o w l e d g e m e n t s. This work was partially supported by National Centre for Programs management from Romania under the research grant 12100/2008 SICONA.

Received: November 12, 2010

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SIMULATOR DINAMIC AL UNUI SISTEM HIDRAULIC VALVĂ-AMBREIAJ

(Rezumat)

Lucrarea prezintă implementarea unui model matematic pentru un ansamblul valvă-ambreiaj din cadrul transmisiei automate. Acest sistem valvă-ambreiaj este nestandardizat, în sensul că este realizat şi folosit pentru un anumit tip de aplicaţie în care se doreşte atingerea unor performanţe cerute. Cele două elemente hidraulice au fost modelate separat urmând ca în final sa se pună în evidenţă dinamica şi modul în care acestea interacţionează între ele. Pe baza ecuaţiilor matematice a fost construit un simulator utilizând mediul de dezvoltare Matlab/Simulink.

Pentru determinarea parametrilor modelului şi validarea acestuia au fost utilizate date experimentale achiziţionate folosind un stand hidraulic dezvoltat de către firma Continental Iaşi. Rezultatele obţinute în urma simulărilor au fost comparate cu datele experimentale pentru a concluziona validitatea modelului şi a analiza performanţele sistemului valvă-ambreiaj. Simulatorul dezvoltat în această lucrare poate fi utilizat pentru realizarea diverselor simulări cu scopul implementării, testării sau îmbunătăţirii unor strategii complexe de reglare.