24th COBEM - 2017

ABCM
Associação Brasileira de Engenharia e Ciências Mecânicas

24th ABCM International Congress of Mechanical Engineering
December 3-8, 2017, Curitiba, PR, Brazil

COBEM-2017-0761
MODELLING THE HYDRAULIC CIRCUIT OF AN AUTOMATIC TRANSMISSION

Tiago Simino Macedo
Victor Juliano De Negri
Universidade Federal de Santa Catarina – Departament of Mechanical Engineering – LASHIP – Florianóplis, SC, Brazil.
tiagosimino@gmail.com
victor.de.negri@ufsc.br

Leonardo Serfert
Ford Motor Company – Camaçari, BA, Brazil.

Abstract. *Is well known that the application of automatic transmissions in passenger cars in the world have been increased since its first appearance on mid's 1940. The hydraulic circuit of a conventional automatic transmission plays the major role due to its main functions: lubrication and actuation. The actuation function can be related to pressure control, gear shift timing control, shifting quality and consequently the drivability, passenger and driver's comfort and fuel save. Due to the relevance of the hydraulic circuit in these systems, most of transmission failures are related to fail of its components. In this article, a Matlab/Simulink modeling of the hydraulic circuit of the conventional automatic transmission is shown. The objective in this paper is to reproduce, with dynamic modeling, the behavior of the hydraulic system and the components movements within it, to observe the characteristics of the system operating in perfect conditions and to compare with the effects of when some failure in the components inserted in the model. Differently from other researches in this area, whose objective is to model the transmission and improve its control system during the shifting process, this present work aims to develop a tool that helps the fault detection, furthermore the detail knowledge about the hydraulic system and its behavior.*
Keywords: *Automatic Transmission, Modelling, Hydraulic circuit, System failure*

1. INTRODUCTION

Aspects like safety, fuel economy, ease of operation and comfort are encouraging the use of automatic transmissions in the world. In United States, less than 5 % of vehicles uses manual transmissions, while in Europe and Japan 80% of sold vehicles have manual transmissions yet. Nevertheless, the sales of cars equipped with automatic transmissions is growing significantly. In the United States, this technology is totally widespread, considering the large number of users and pioneering equipment development in the 1940s. In many other countries, including Brazil, the maintenance sector requires more knowledge, test equipment and tools development to deal with these kinds of transmissions.

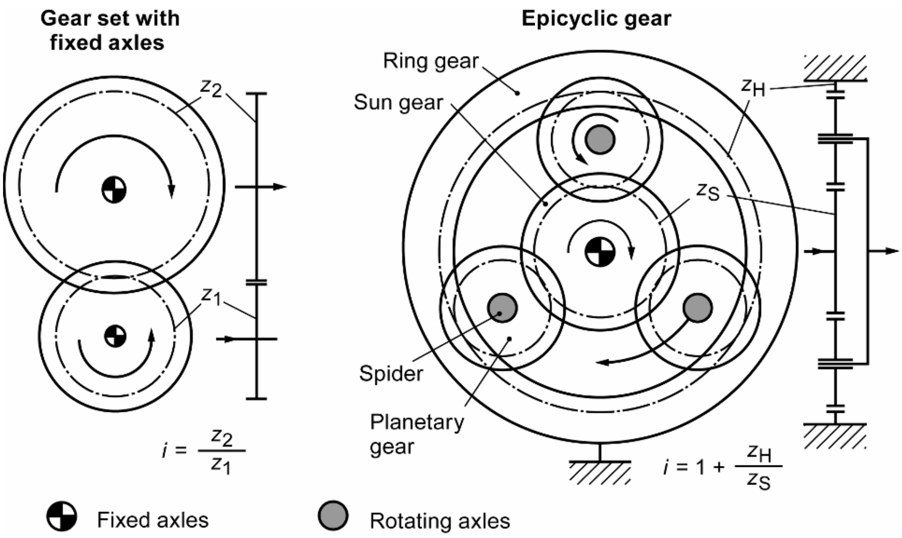
In the same way as any hydraulic circuit, the automatic transmission hydraulic circuit can be divided in three principal parts (von Lisingen, 2013): 1) Primary energy conversion, 2) Energy control and limitation and 3) Secondary energy conversion. In the automatic transmission analyzed in this paper, as primary conversion one has an internal gear hydraulic pump that is responsible to transform the mechanical energy that comes from the engine through a rotative shaft in to hydraulic energy, pressurizing the fluid inside the gearbox hydraulic pipes. On the second stage, the hydraulic circuit of a conventional automatic transmission is comprised by solenoid valves, pilot operated valves, relief and directional valves. Those valves are assembled in the so called valve body and are responsible for controlling the gearshifts according to the combustion engine speed, vehicle speed and throttle position. The solenoid valves are electronically controlled by the vehicle module, transmission control unit (TCU) which is responsible to handle all the information collected by transducers and manage the gear shifting time through electric signals.

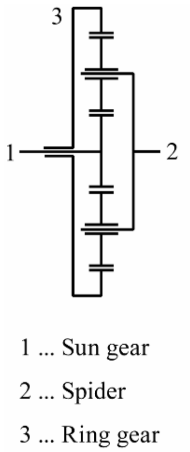
The performance of automatic transmissions depends on the combined behavior of several valves and actuators, resulting on a highly nonlinear dependence between variables. Consequently, several authors have reported how difficult is to understand the hydraulic system behavior of an automatic transmission such as Quan Zheng (1999), Watechagit e Srinivasan (2003), Naunheimer et al. (2011) and Samanuhut (2011). Modelling the hydraulic circuit of an automatic transmission is necessary to analyze and understand the behavior such as valve spool and actuators displacements andmain line pressures. In this paper, a dynamic model of a partial hydraulic circuit implemented in Matlab/Simulink® is presented. The objective is to analyze the behavior of the hydraulic pressures during the actuation of a breaking clutch and, consequently, to be able to understand the effects in order to diagnose possible failures.

The paper text is divided in four main sessions, the transmission presentation, the modeling development and main equations, analyzes and results from simulation, and conclusion.

2. AUTOMATIC TRANSMISSION FUNCTIONALITY

The conventional automatic transmission is equipped with a torque converter and planetary gear sets, depending on the combination of speed input component, output, brake and clutch actives. The transmission is able to perform a specific input/output speed ratio. The Figure 1 shows how to represent, in a diagram, a single planetary gear (epicyclic) in the right side, in comparison with a simple counter shaft couple gears on the left side of the image. The figure includes a table showing all the possible combinations of input and output and their ratios calculation based on the gears size (Z).





State of motion	Type of gearbox	Input	Output	Frame	Planetary step ratio
a)	Gear set with fixed axles	1	3	2	$i_P = \frac{n_1}{n_3} = i_S = -\frac{z_3}{z_1}$
b)		3	1		$i_P = \frac{n_3}{n_1} = \frac{1}{i_S} = -\frac{z_1}{z_3}$
c)	Planetary transmission single drive	1	2	3	$i_P = \frac{n_1}{n_3} = 1 - i_S = 1 + \frac{z_3}{z_1}$
d)		2	1		$i_P = \frac{n_2}{n_1} = \frac{1}{1 - i_S} = \frac{1}{1 + \frac{z_3}{z_1}}$
e)		2	3	1	$i_P = \frac{n_2}{n_3} = \frac{1}{1 - \frac{1}{i_S}} = \frac{1}{1 + \frac{z_1}{z_3}}$
f)		3	2		$i_P = \frac{n_3}{n_2} = 1 - \frac{1}{i_S} = 1 + \frac{z_1}{z_3}$

Figure 1. Epicyclic gear diagram representation and step ratio combinations and calculations (NAUNHEIMER *et al.*, 2011).

It can be observed that one planetary gear is able to reach six different combinations of input/output combination, but not all of its combinations are applicable on a vehicle transmission. Due the necessity of output torque and wheel speed adaptation, the vehicles transmissions has more than five gear ratios nowadays, such that one of the applicable solutions is combine two or more planetary gear sets to achieve all the required design ratios. The conventional transmission used in this study has three planetary gear sets, frontal (1), middle (2) and rear (3), connected to each other as seen in Figure 2 and it is able to perform six different gear ratios.

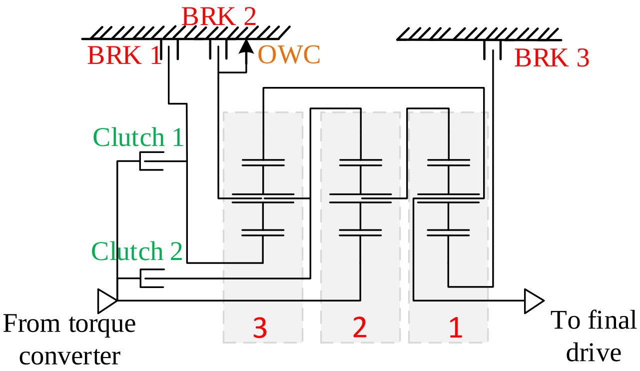


Figure 2. Diagram of the six speed conventional automatic transmission.

To hold a gear stationary or transfer the input torque, the automatic transmission uses hydraulic actuators to apply or release the brakes or clutches discs. Due to that, the hydraulic circuit is considered the main system of an automatic gearbox. There are other systems that makes interface with the hydraulic, for instance electronic system which communicates through TCU and solenoid valves and the mechanical system with the gears, carriers, discs in mechanical contact with the hydraulic actuators. Altogether there are five hydraulic actuators, of which three brake actuators and two clutches and their location is represented in Figure 3.

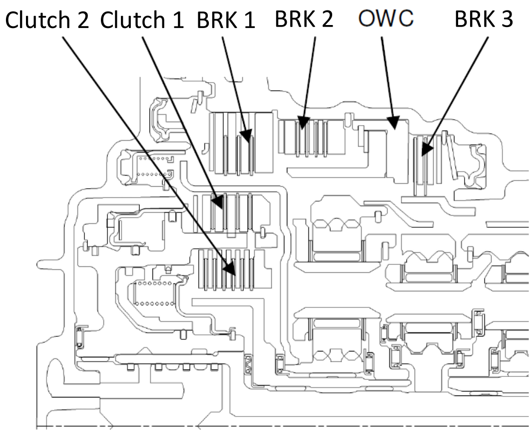


Figure 3. Clutch and brakes location.

Those components shown in Figure 3 can combined, as shown in Table 1, to reach the gear ratios shown on Table 2. The hydraulic system is considered to play the major role on the automatic transmission, controlling the shift timing, shifting quality and, consequently, vehicle consumption and emissions.

Table 1. Actuators scenarios to reach each gear ratio.

Gear		Actuators				
		BRK 3	**BRK 1**	**Clutch 1**	**BRK 2**	**Clutch 2**
R				x	x	
N					x	
D	**1st**	x				
	2nd	x	x			
	3rd	x		x		
	4th	x				x
	5th			x		x
	6th		x			x

Table 2. The transmission gear ratios.

Gear	Ratio		
1st	4,58	:	1
2nd	2,96	:	1
3rd	1,91	:	1
4th	1,44	:	1
5th	1	:	1
6th	0,75	:	1
Reverse	2,94	:	1

The relevance of the hydraulic circuit function inside a gearbox motivates the modelling presented in this paper. In the next section, the main equations of the hydraulic system are presented as well as the Matlab/Simulink block diagram.

3. HYDRAULIC CIRCUIT MODELING

The whole automatic transmission hydraulic circuit modeled have sixteen valves and five actuators, although the five cylinders are equally asymmetrical and spring returned, the sixteen valves are not the same. One strategy to create the model was to divide the hydraulic circuit in groups, more specifically five groups, one for each actuator. During the division was realized that there was a common group of valves for each actuator, and this group repeat four times on the circuit, as a standard circuit for actuator pressure control as shown in Figure 4.

As a simplification, the hydraulic pump with relief valve and the outlet of the pressure control valve were assumed as a constant pressuresupply sources, designed as p_s and p_L in Figure 4. The p_s port supply fluid tothe solenoid valves and the other source to the remaining circuit, until the actuator.

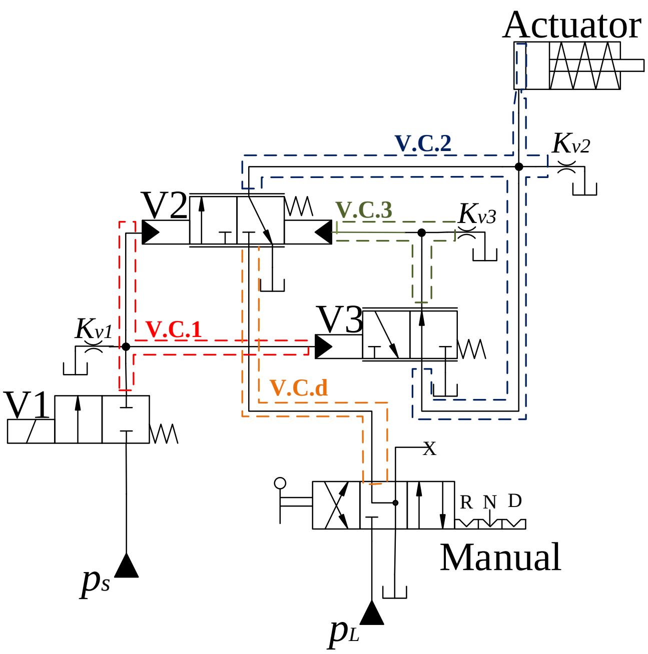


Figure 4. Standard valve group, control pressure hydraulic circuit.

The manual valve is common to whole circuit, the V1 valve is a solenoid valve spring returned 2/2 normally closed, the V2 and V3 valves are hydraulic piloted valves 3/2 spring returned normally closed and opened. respectively. The Figure 5 shows the internal cut view from the valve V1, and the spools from valves V2 and V3.

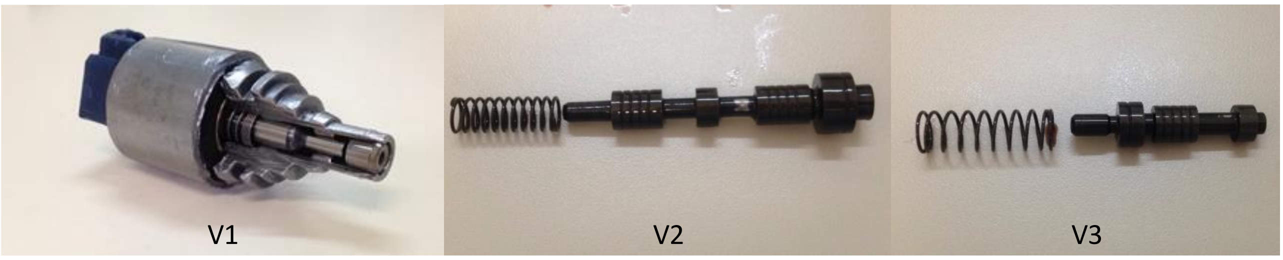


Figure 5. Cut internal view from valve V1 and valves V2 and V3 spools.

The procedure of modeling is common to all valves regardless its function. Here will be presented the modelling procedure to V2 valve as an example. The equations and analysis can be replicated to other hydraulic valves belonging to the automatic transmission hydraulic circuit.

The free body diagram of the valve spool and valve body is shown in Figure 6, from where it is possible to write the movement equation regarding to the valve spool:

$$p_1.\mathrm{A}_1 - p_3.\mathrm{A}_3 = m_r.\frac{d^2x_r}{dt^2} + B_r.\frac{dx_r}{dt} + \mathrm{F_{kr}} + F_{rend}, \tag{1}$$

where p_1 and p_3 are pressures in control volumes V.C.1 and V.C.3, respectively, x_r the spool displacement, m_r the spool mass, B_r the viscous damping coefficient, F_{kr} the spring force and F_{rend} a wall force as a stop force described by Bacca *et al.* (2010) as:

$$F_{rend} = \begin{cases} k_{rend}.x_r + B_{rend}.\dfrac{dx_r}{dt}, & x_r \le 0 \\ k_{rend}.(x_r - x_{rmax}) + B_{rend}.\dfrac{dx_r}{dt}, & x_r \ge x_{rmax} \end{cases}, \tag{2}$$

where k_{rend} is the elasticity coefficient of the wall, representing a wall force to stop the spool at the end of his stroke.

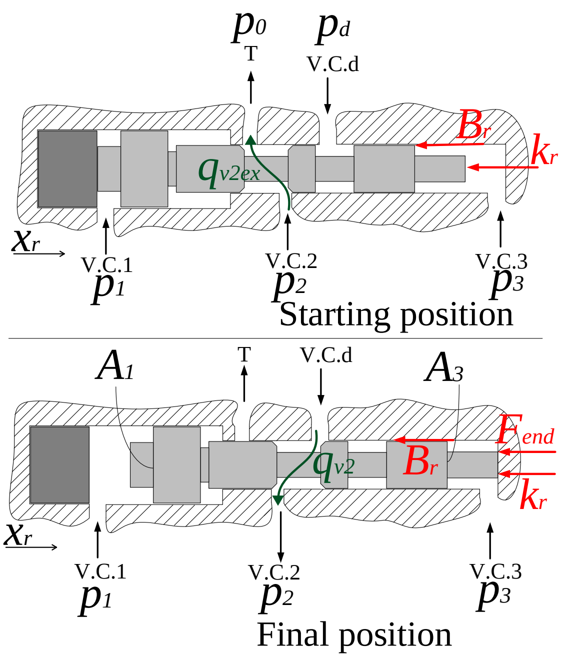


Figure 6. V2 free body diagram.

The flow rates, q_{v2} and q_{v2ex}, which represents the flow rates through the V2 valve depending on its spool position can be modeled as the following orifice flow equations:

$$q_{v2} = cd_r.A_r.\sqrt{\frac{2}{\rho}.|(p_d - p_2)|}.sgn(p_d - p_2), \tag{3}$$

$$q_{v2ex} = cd_{rex}.A_{rex}.\sqrt{\frac{2}{\rho}.|(p_2 - p_0)|}.sgn(p_2 - p_0), \tag{4}$$

where *cd* are the discharge coefficients, ρ the fluid specific mass and *A* the proportional opening areas to the fluid passage, those areas can be calculated in function of the spool displacement, and its geometrically obtained.

Finally, to find the pressures behavior is necessary to make the mass balance through the control volumes. Considering the mass balance equation, or continuity equation, for instance to the V.C.2 control volume (Figure 7), the following equation can be derived:

$$q_{v2} - q_{2ex} - q_{v3} - q_{vL2} = \frac{dV_2}{dt} + \frac{V_2}{\beta}.\frac{dp_2}{dt}, \tag{5}$$

where, q_{vL2} is modeled as a leakage flow rate from the line with flow coefficient *Kv2* as showed in Figure 4. Rearranging Equation (5) and considering the volume variation associated to actuator displacement, results on:

$$p_2 = \int_0^t \frac{\beta}{V_2}.(q_{v2} - q_{2ex} - q_{v3} - q_{vL2} - A_c.\frac{dx_c}{dt})\,dt. \tag{6}$$

where, *Ac* is the actuator area and xc its displacement.

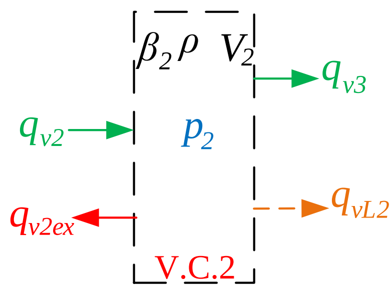


Figure 7. Control Volume V.C.2.

All those equations were implemented at Matlab/Simulink. Figure 8 shows an example of block diagram modeling regarding to the circuit showed in Figure 4.

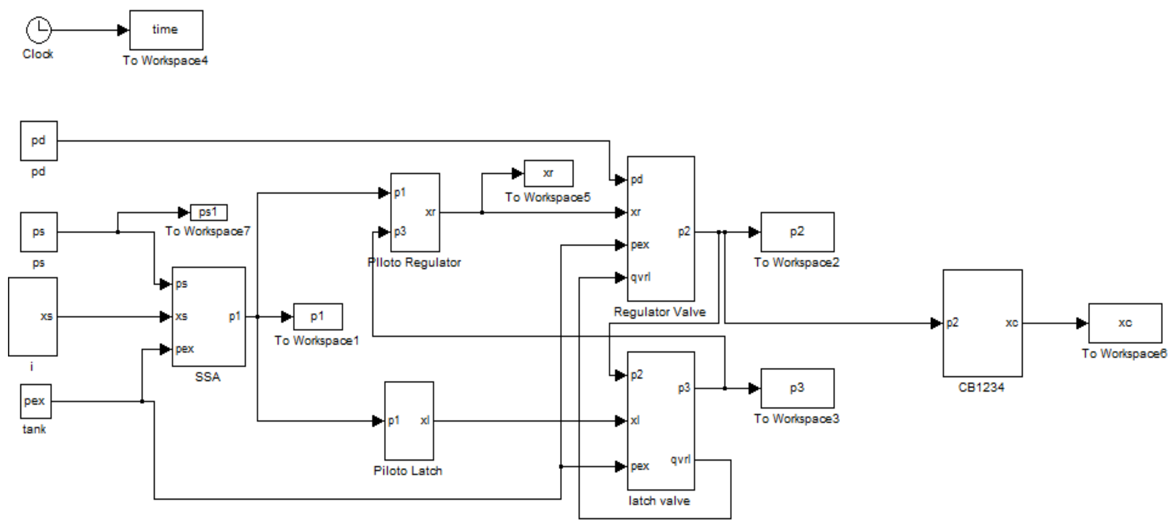


Figure 8. Matlab/Simulink block diagram.

4. RESULTS AND DISCUSSIONS

The circuit modelling allows to carry out simulations under different operational conditions, resulting on the static and dynamic responses, specially related to pressures component displacements. As a result, it will be shown in this section a comparison between to different signals sent to the solenoid valve. The solenoid communicates with the TCU through pulse width modulation signal (PWM) and the signal width is controlled by the TCU according to several parameters, such as vehicle speed, actual gear, engine speed, and throttle position. Due to hydraulic circuit architecture, there is proportionality between the pulse width and the actuator pressure as seen o Figure 9.

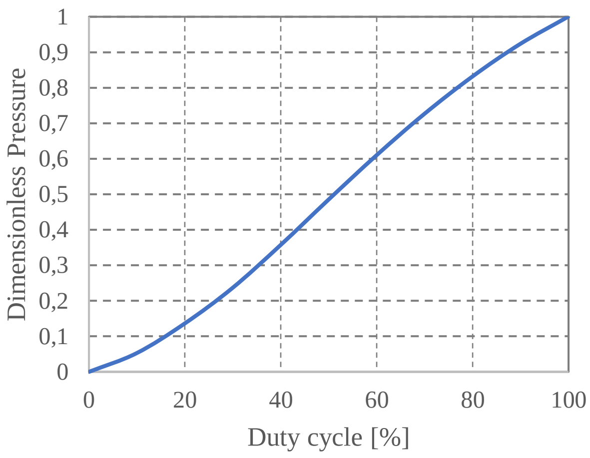


Figure 9. Actuator pressure according to solenoid valve duty cicle applied.

The actuator displacement responses for a PWM controlled signal and a full step signal (duty cycle corresponding to 100 %) are discussed below. Figure 10 shows both input signals, the actuator pressures for both cases and the actuator displacements.

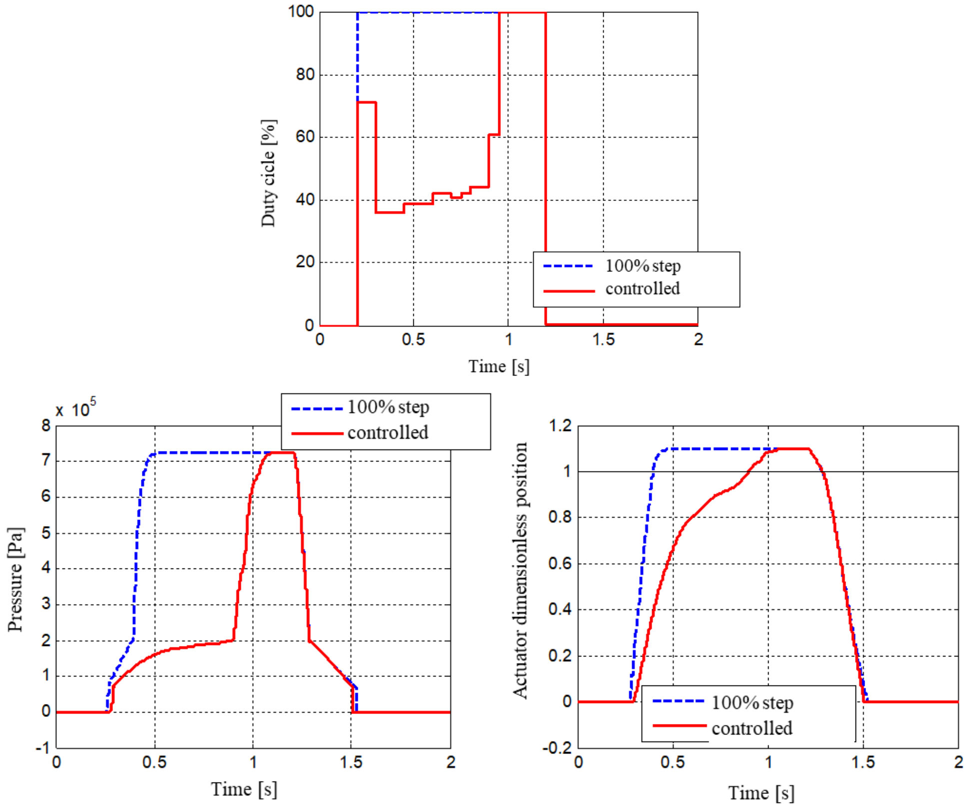


Figure 10. Solenoid valve signal comparison.

The pressure and, consequently, the actuator displacement is faster for 100% step signal compared with the controlled signal input. When the step signal is applied, the actuator spent less than 0.25 s to reach its final position compressing the clutch discs. In order to represent the clutch discs compression, the 100% position (dimensionless position equal 1) corresponds to the first touched disc caused by the cylinder advance. After that, the disc compression occurs such that the actuator relative position reaches a value greater than1. Possibly, the component life cycle, durability and drivability are increased with a good controlled shifting, preventing wear, shock and abrupt gear shifts.

In addition to this dynamic analysis, the full circuit model allows to observe the steady state behavior of the transmission actuators too. An example of it can be seen in Figure 11, where the results of a transmission running perfectly is shown in the left side. Assuming that the solenoid valve V1 fails (the valve remains closed), 5th and 6th gears ratio will not be engaged, as can be seen in the right side.

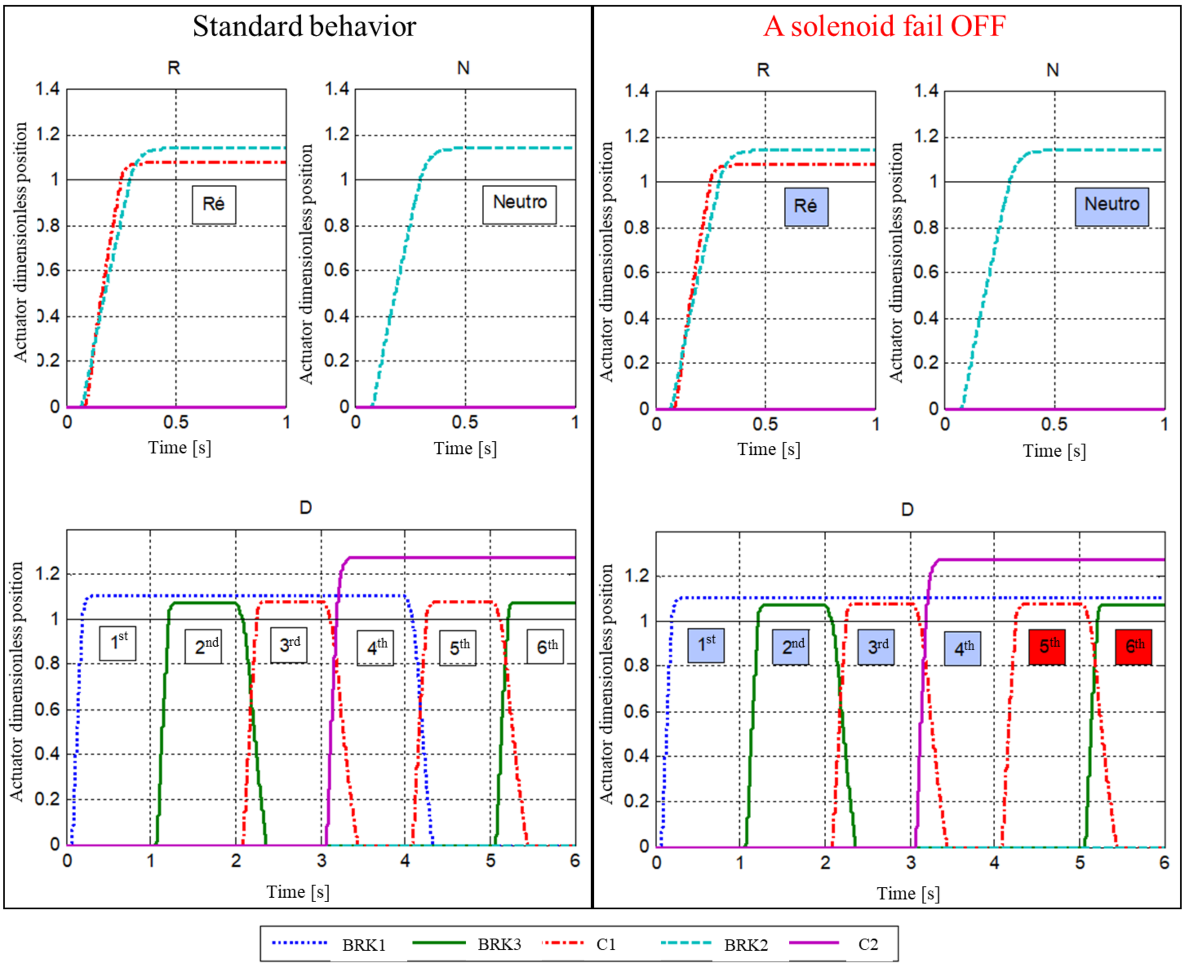


Figure 11. An example of steady state model analysis.

This is an example of a fault simulation on the model that could represent a transmission failure, which the effect is no engaging all the gear ratios. Other faults could be simulated such as valve spool blocked, high and low fluid viscosity, electrical faults and so on.

5. CONCLUSIONS

During the model development, acquiring the feeling of the system behavior and the knowledge related to the system functionality was very important to the project. The model showed to be very versatile and as a result, made possible dynamic and steady state analysis regarding to the hydraulic system behavior, for instance, during gear shifts, solenoid signal analysis, gear engagement and also can be used for failure analysis as exemplified.

The knowledge about the automatic transmission hydraulic circuit is very important, since it plays the major role at the gearbox. Thereby this model has a big value due to the multiple possibilities to analyze the hydraulic circuit behavior, for instance, main lines pressure, spool valves and actuators displacement.

The parameters adjustments were the biggest challenge on modelling this kind of system, the lack of information and specific knowledge requirements made this work very interesting and tough. Besides that, the model validation with a test bench creation would be important to reach dynamic and steady state analysis more confident.

6. ACKNOWLEDGEMENTS

The work was supported by CNPq, Ford Motor Company and LASHIP.

7. REFERENCES

BACCA, G. A.; DE NEGRI, V. J.; ASSAF, Y. Modelo Matemático No-Lineal Para Un Servosistema Neumático De Posicionamiento Preciso. *Tecnológicas*, v. 24, p. 27, 24/06/2010 2010.

BALAU, A.-E.; CARUNTU, C.-F.; LAZAR, C. Simulation And Control Of Anelectro-Hydraulic Actuated Clutch. *Mechanical Systems And Signal Processing*, v. 25, p. 12, 2011.

BURKHART, R. O. et al. Latch Valve For Actuating A Transmission Control Element: Google Patents 2015 < https://www.google.com/patents/US20130056657 >.

CARMO, P. F.. Proposta De Modelo Para Descrição Da Vazão Em Válvulas Direcionais Proporcionais, Com Efeito De Vazamento. 2003. 134 p. *Master Thesis*. Engenharia Mecânica, Universidade Federal de Santa Catarina

MENG, F. et al. Optimal Design Of An Electro-Hydraulic Valve For Heavy-Duty Vehicle Clutch Actuator With Certain Constraints. *Mechanical Systems And Signal Processing*, v. 68-69, p. 13, 2016.

NAUNHEIMER, H. et al. Automotive Transmissions: Fundamentals, Selection, Design and Application. Second Edition. 2011. 740 p. ISBN 978-3-642-16213-8.

QUAN ZHENG, B. S. Modeling And Control Of Powertrains With Stepped Automatic Transmissions. 1999. 446 p. (*Doctor of Philosophy*). Department of Mechanical Engineering, The Ohio State University

SAMANUHUT, P. Modeling And Control Of Automatic Transmission With Planetary Gears For Shift Quality. 2011. 238 p. (*Doctor Of Philosophy*). The University Of Texas At Arlington

VON LISINGEN, I. Fundamentos de Sistemas Hidráulicos. 4ª Edição. Florianópolis, SC, Brasil: Editora UFSC, 2013. 399 p. ISBN 85-328-00646-8.

WATECHAGIT, S.; SRINIVASAN, K. Modeling and Simulation of a Shift Hydraulic System for a Stepped Automatic Transmission. In: *SAE World Congress*, Detroit, Michigan, 2003. p.16.

8. RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper.