DYNAMICS OF THE ELECTROHYDRAULIC TRANSMISSIONS FOR AUTOMOTIVE APPLICATIONS

Daniela VASILIU[1], Claudia Georgiana VASILIU[2], Constantin Călinoiu[3]

The paper contains a realistic study of the modern structures, and of the dynamic performances of the hydrostatic transmissions controlled by electrohydraulic servovalves. The practical interest for this subject is generated by the wide implementation of these heavy-duty continuous variable transmissions in different technical systems, which can be digitally controlled in real time according to the technological needs. The authors have combined the complete mathematical model of the transmission with the four wheels car model, in order to obtain a global image of the typical transients' occurring in a complete mobile system. The super components from Simcenter Amesim libraries were completed with original models of the different real specific components. A special attention was paid to the cavitation problem, and to the effect of the equivalent bulk modulus of compressibility of the hoses needed for solving cinematic degrees of freedom. The numerical results can be used directly in the synthesis of different new types of automotive systems.

Keywords: modeling, simulation, anti-cavitation protection, automotive continuous transmissions

1. Structure and applications of hydrostatic transmissions electronically controlled

The relatively stiff mechanical characteristics of the automotive engines are adapted to the automotive variable needs by mechanic, electric, hydraulic, pneumatic, or hybrid transmissions. The mechanical characteristic of an engine represents the relation between the torque delivered by the shaft M, and the shaft speed, ω. The relation may be a curve or a surface, depending on the engine control facilities. For example, a Diesel engine steady-state behavior may be described by a family of curves (Fig. 1) obtained by cutting the characteristic surface, $M=f(\omega,\varphi)$ with planes defined by a constant relative fuel mass injected during each active stroke in the cylinders, $\varphi=ct.$ [1]. The dimensionless traction

[1] Prof., Dept. of Hydraulics, Hydraulic Machinery, and Environmental Engineering, University POLITEHNICA of Bucharest, Romania, e-mail: daniela.vasiliu@upb.ro
[2] Lecturer, Dept. of Hydraulics, Hydraulic Machinery, and Environmental Engineering, University POLITEHNICA of Bucharest, Romania
[3] Ass. Prof., Dept. of Hydraulics, Hydraulic Machinery, and Environmental Engineering, University POLITEHNICA of Bucharest, Romania

characteristics (Fig. 2) shows the variation of the relative traction force Z/Z_0 and running resistance F/F_0 as functions of the car relative speed, v/v_0 for different stages selected in a six stages mechanical gear box.

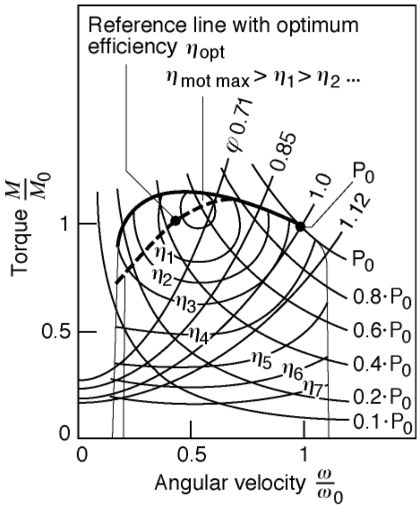

Fig. 1. Universal dimensionless characteristics of a thermal engine. [1]

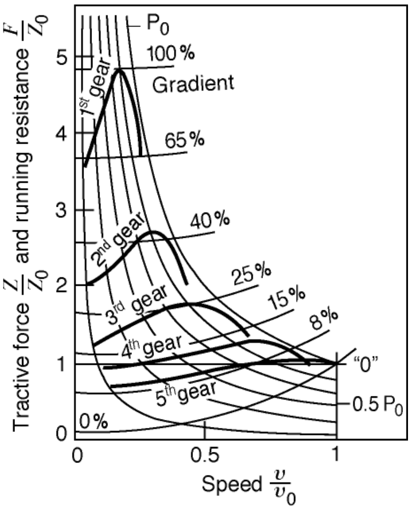

Fig. 2. Traction characteristics of a classical car with six-speed stages gearbox. [1]

A classical heavy-duty automotive speed control system replaces the multistage mechanical gearbox by a hydrostatic transmission, which includes a servopump and a hydraulic motor, connected in a closed loop. Such a widely used combination is shown in Figs. 3 and 4. [2]

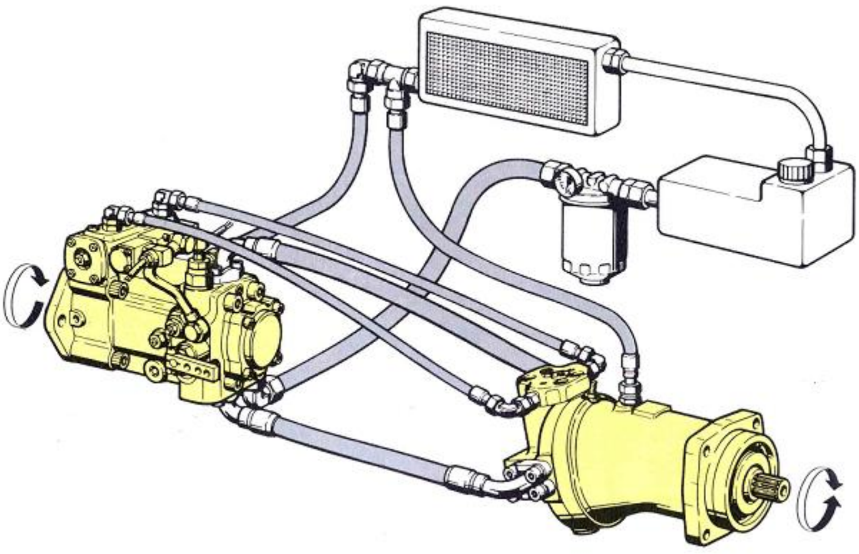

Fig. 3. General-purpose hydrostatic transmission with primary and secondary control [2].

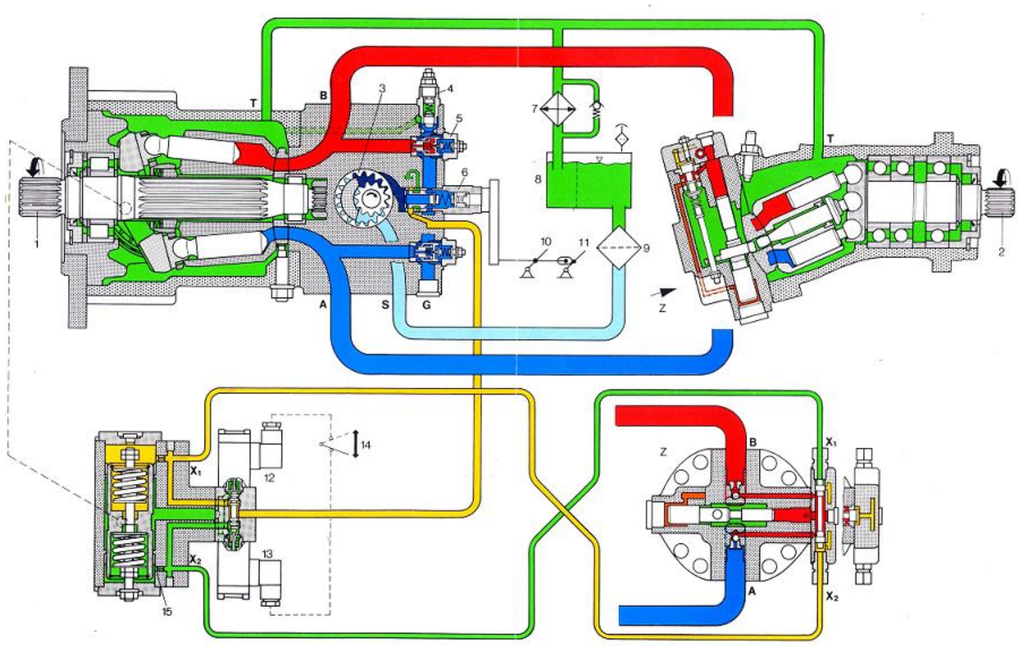

Fig. 4. Hydraulic diagram of a typical hydrostatic transmission composed by a swash plate axial pistons servopump and a variable displacement bent axis hydraulic motor [2].

Different hybrid devices can control continuously the torque and the speed of the hydraulic motors according to the needs of the working machine. The flexibility is the main advantage of the hydrostatic transmission regarding the mechanical ones, although their overall efficiency is affected by a double energy conversion.

The most important scientific problem regarding the hydrostatic transmissions is the optimal implementation in complex control systems, using mathematical modeling, numerical simulation, identification, and real time simulation. The first practical objective of the researches is the significant reduction of both fuel consumption and the pollutant emissions level by the common digital control of the thermal engine and the hydrostatic transmissions, with distributed microcontrollers, interconnected by CAN or other digital bus. This target can be reached by detailed dynamic analysis of all the components only, using modern theoretical and experimental procedures.

Thirty years ago, a "paper and pencil" analysis was always done before numerical simulation needed for final refinements. *The sound engineering judgments was the key of the success*. In the computer’s era, the facilities of the simulation languages allow more emphasis on the physics and mathematical formulation of problems, and less on the technical solution. The "dual" approach is still applied in a complementary manner, but the mouse replaced the pencil. This paper presents a systematic research of the hydrostatic transmissions aiming to develop some embedded mathematical models, which can be integrated in the hydraulic library of Simcenter Amesim, developed by Siemens PLM Software Corporation. The designers of complex fluid power systems will have the possibility to promote easy the hydrostatic transmissions without considering the mechanical design details.

2. Design criteria for the hydraulic scheme

The design of a hydrostatic transmission can be gradually performed: from the simple choice of the whole assembly, offered by the dedicated manufacturers based on some preliminary data obtained from similar vehicles, to a complete hydraulic, thermal, mechanical, electrical, control computation, passing through the following stages: a) Design of the main components of the transmission; b) Sizing the main components needed for specified performances; c) Choice of the common components from the catalogs of the available suppliers; d) Optimization of the operation parameters of all the components by systematic numerical simulations.

The present paper treats mainly the first and the last stage, the target being to point out the capabilities of the numerical simulation for optimization the whole system's behavior without taking into account the concrete details of the components. The structure of a hydrostatic transmission essentially depends by the following specifications of the automotive equipment [3], [4] and [5]:

a) application field; usually, this is the main transmission of a mobile equipment: forklifts, front loaders, mobile cranes, concrete mixer trucks, agricultural machines, articulated steering forestry tractors, civil works machines like land levelers etc.;

b) Range of steady state velocity control with good enough overall efficiency;

c) Type of positive displacement hydraulic machines available on the market (usually – with axial or radial pistons);

d) Type of the displacement control system: mechanical, hydro mechanical, electrohydraulic or hybrid one;

e) type of the servopump driving engine; usually, Diesel engines are used, with mechanical or electrohydraulic fuel injection systems;

f) Maximum moving vehicle mass, depending on the destination and the size;

g) Maximum slope of the accessible roads;

h) Maximum velocity on the maximum slope road;

i) System nominal pressure;

j) Maximum velocity on horizontal road;

k) Dynamic radius of the motor wheels;

l) Mechanical transmission ratio of the gears placed between the hydraulic motors and the wheels;

m) Universal characteristics of the servopump driving engine (Diesel).

3. Optimisation of a hydrostatic transmissions by numerical simulation

In order to illustrate the capabilities of the numerical simulation to help the designer to obtain the specified performance, the authors studied a complete dynamic model of a hydrostatic transmission composed mainly by an electrohydraulic swash plate axial pistons servopump and one electrohydraulic bent axis servomotor, which drives a vehicle through a planetary gear and a classical differential. The four wheels' vehicle mathematical model was chosen from the AMESim library $AME/demo/Libraries/TR/ ManualGearbox.ame.

In a first stage, using the complete model from Fig. 5 it was studied the variation of the car body longitudinal velocity during the start generated by a linear increasing of the servopump displacement (0 to 100%) in 4 seconds, followed by the decreasing of the servomotor displacement from 100% to 50% (Fig. 6). The following parameters were inspected during such a transient:

- The relative variation of the pump displacement (from 0 to 100%), servomotor displacement (from 100% to 50%) and the variable sine wave source relative output corresponding to a sine road slope variation between 0 and 15% (Fig. 7);

- Variations of the car body longitudinal acceleration and longitudinal velocity (Fig. 8) during the motion cycle from Fig. 6;

- Variation of the pressure in the servopump main ports (Fig. 9) during the motion cycle from Fig. 6;

- Variation of servopump shaft torque and the servomotor shaft torque (Fig. 10) during the motion cycle from Fig. 7;

- Influence of the equivalent bulk modulus of the main connection hoses of the transmission (1200 bar, 2000 bar, and 4000 bar) on the pressure from the servopump output port during the first stage of the vehicle start, when the servopump displacement is continuously increased (Fig. 11).

Fig. 12 shows the variation of the pressure in the servopump delivery port during the complete acceleration of the car in the first 10 s of the transient, for different equivalent bulk modulus of the main connection hoses of the transmission (1200 bar, 2000 bar, and 4000 bar). The influence of the stiffness and the length of the main connections is very important, especially in the case of a large amount of micron air bubbles in the hydraulic fluid.

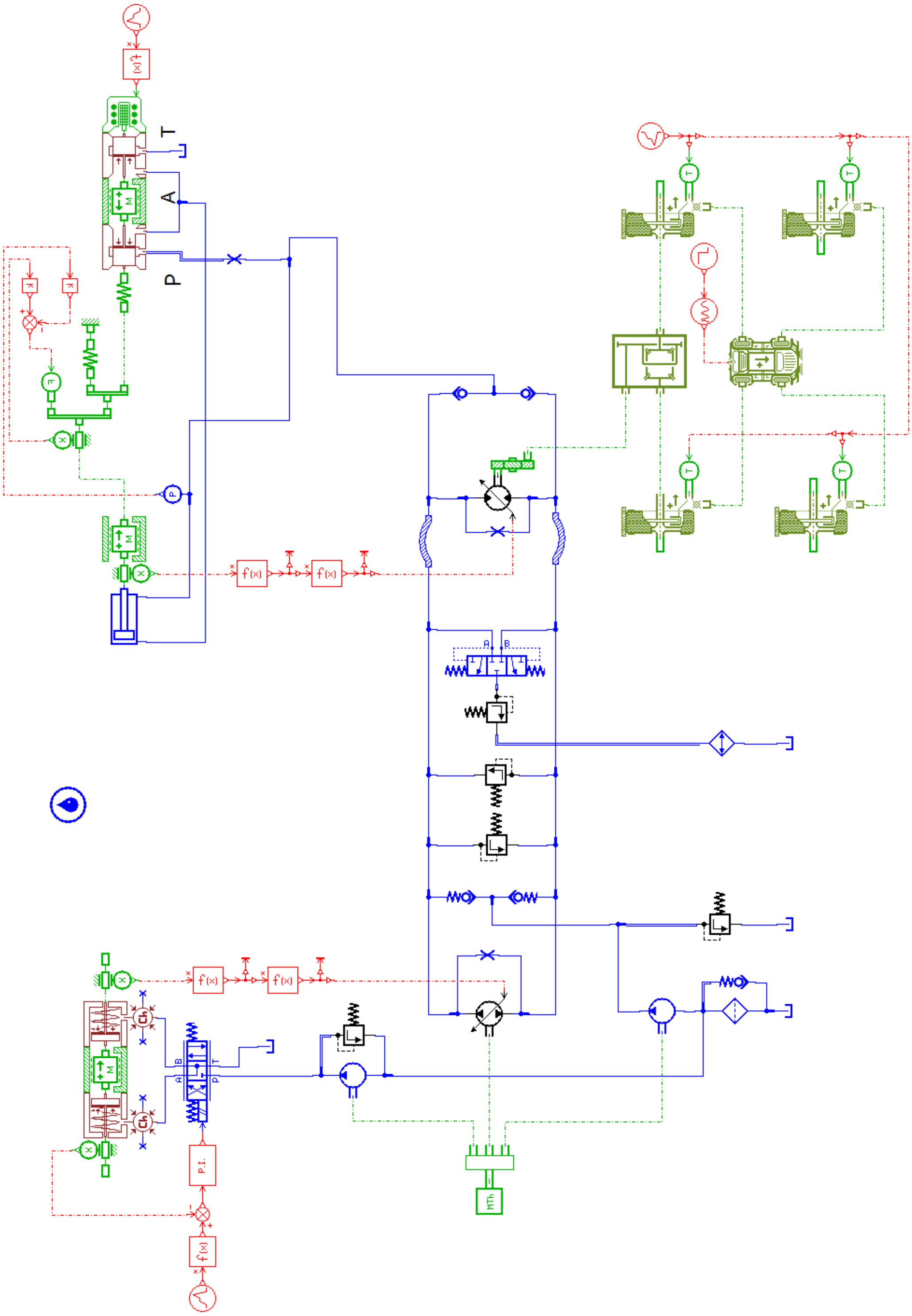

Fig. 5. The complete model of the transmission

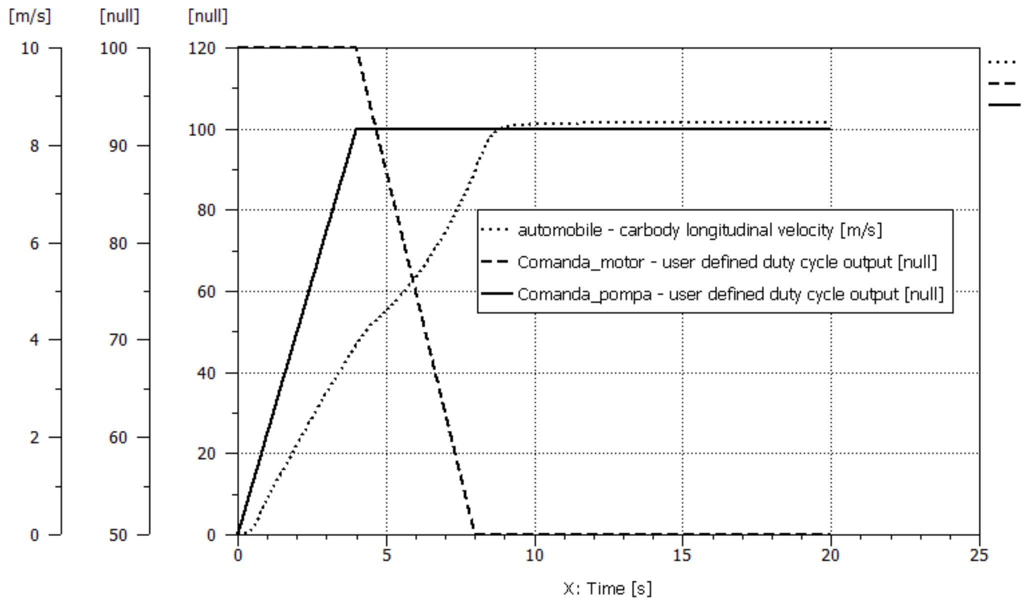

Fig. 6. Variation of the car body longitudinal velocity during the start generated by a linear increasing of the servopump displacement from 0 to 100%, followed by the decreasing of the servomotor displacement from 100% to 50%.

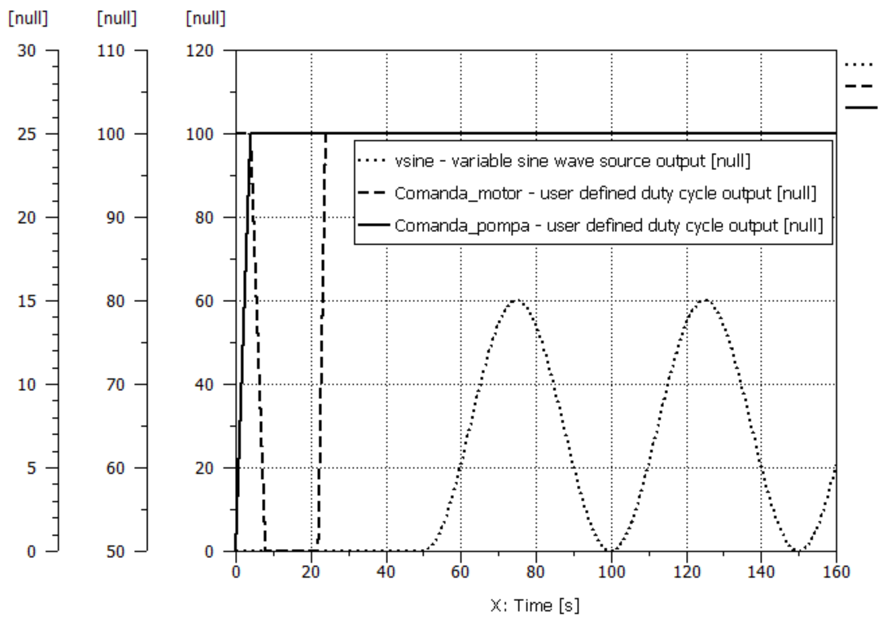

Fig. 7. Relative variations of the pump displacement (0…100%), servomotor displacement (from 100% to 50%) and the variable sine wave source relative output corresponding to a sine road slope variation between 0 and 15%.

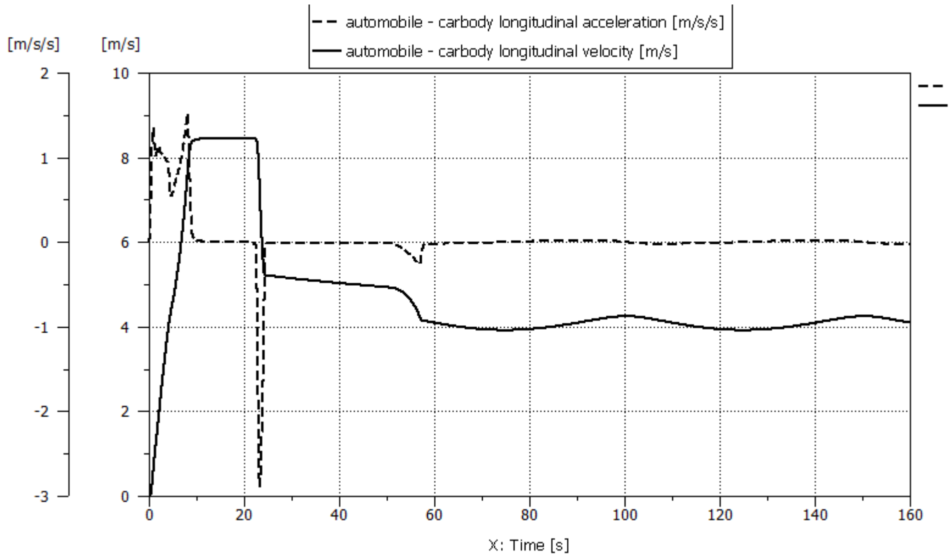

Fig. 8 Variations of the car body longitudinal acceleration and longitudinal velocity during the motion cycle from Fig. 7.

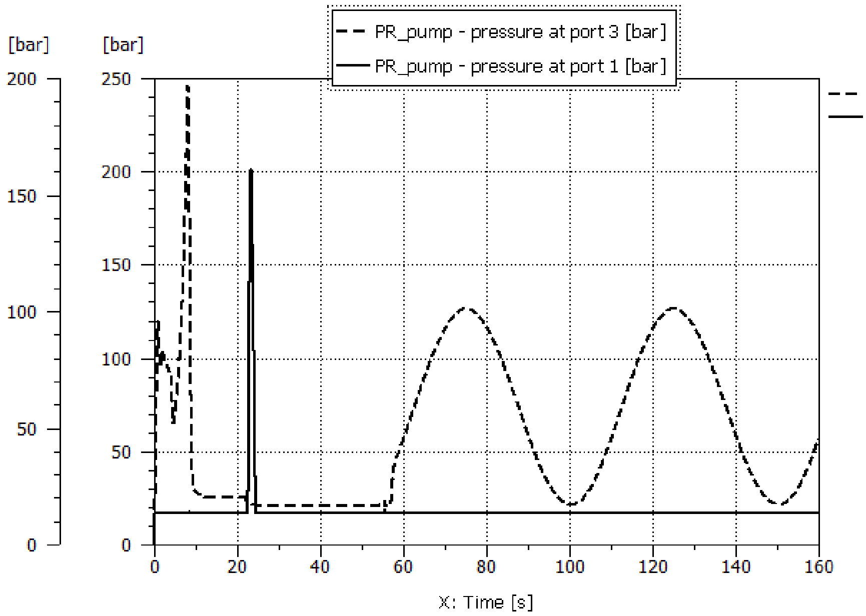

Fig. 9 Variation of the pressure in the servopump main ports during the motion cycle from Fig. 7.

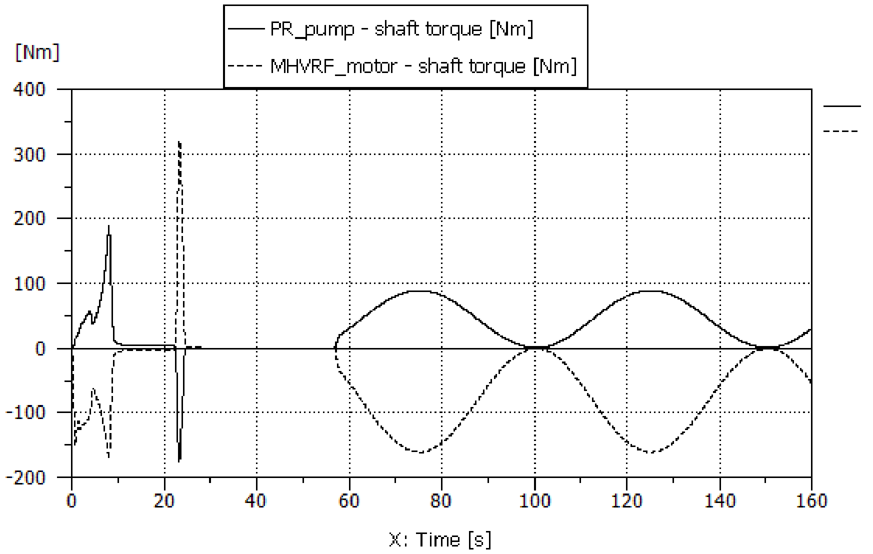

Fig. 10. Variation of servopump shaft torque and the servomotor shaft torque during the motion cycle from Fig. 7.

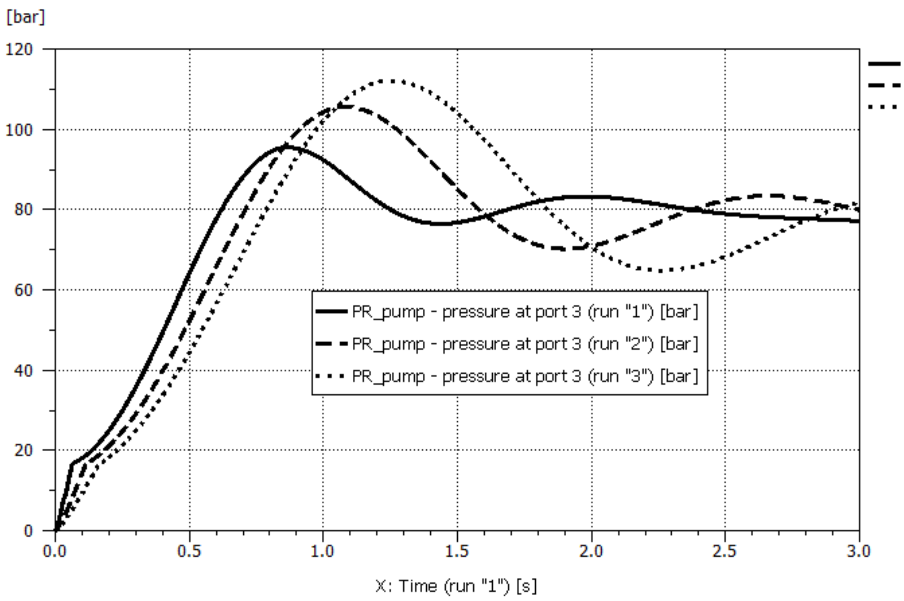

Fig. 11. The influence of the equivalent bulk modulus of the main connection hoses of the transmission (1200 bar, 2000 bar, and 4000 bar) on the servopump output port pressure during the vehicle start.

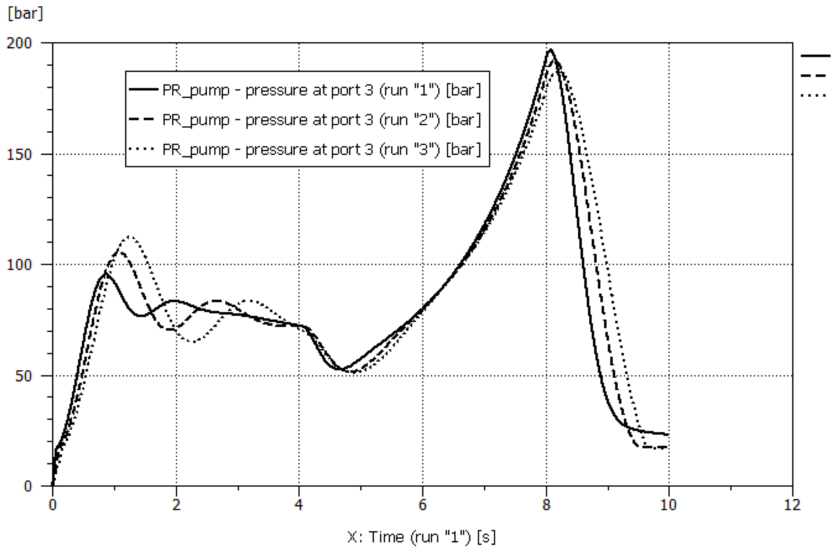

Fig. 12. Variation of the pressure in the servopump delivery port during the complete acceleration of the car in the first 10 s of the transient, for different equivalent bulk modulus of the main connection hoses of the transmission (1200 bar, 2000 bar, and 4000 bar).

The most dangerous phenomenon, which can occur in the operation of a hydrostatic transmission, is the cavitation. The setting of the cracking pressure of the replenishing pump at a low value generates a sudden pressure drop in the servopump suction line in the first second of the increase of the flow rate in the main loop of the transmission (Fig. 13). This is the reason for setting the lowest pressure in the main loop at minimum 16 bar. The heavy-duty transmissions, working at 700 bar, are pressurized at 45 bar!

Another important information obtained by simulation regards the minimum flow rate of the replenishing pump. The use of the swash plate axial pistons reversible machines offers an overall good mechanical efficiency, and a good start torque applied to the load, but the size of the replenishing pumps is greater than for other type of machines. The slippers generate the main leakages when the oil lubricant properties are too poor, at high operating temperature. All the suppliers of compact hydrostatic transmissions are turning any step input signal for the servopump displacement into a ramp (Fig. 14) in order to reduce the cavitation danger.

The simulations revealed many other important criteria for accepting the chosen parameters for different components. For example, the maximum value of the pressure in the main loop occurs in the moment of the sudden increasing of the servomotor displacement (at 22s from the beginning of the motion), but it is not overcoming the cracking pressure of the two-stages pressure relief valves which protect the main loop against the burst (350 bar).

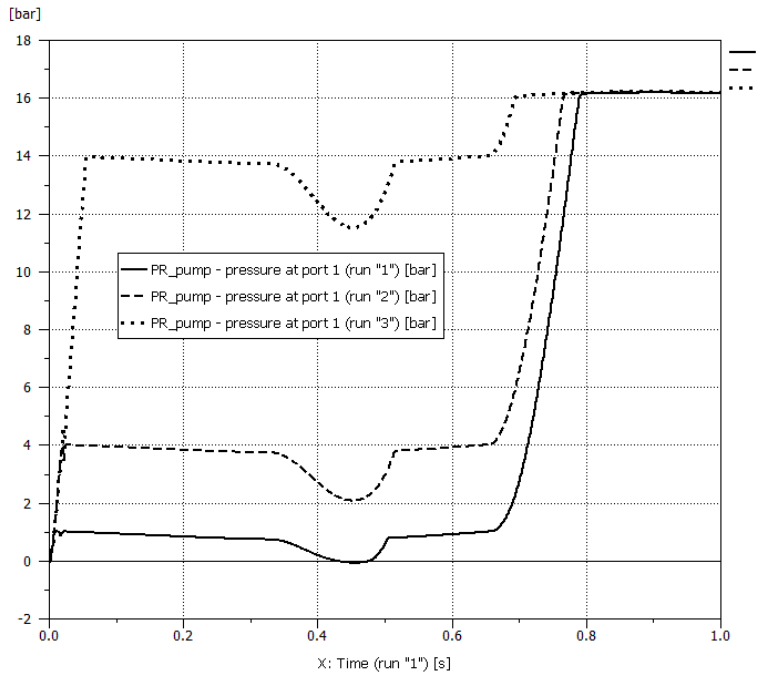

Fig. 13. Pressure variation in the suction line of the servopump during the first second of increasing the displacement for different setting pressures of the replenishing pump (2, 5 and 16 bar).

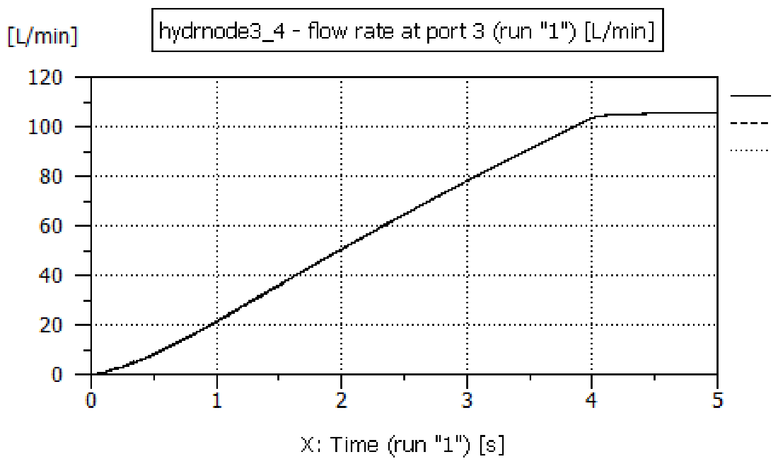

Fig. 14. Servopump flow rate variation during a normal start of the vehicle on a horizontal road.

The authors checked the validity of all the above results by the aid a complex test bench (Fig. 15) devoted to the dynamics of the split power hydrostatic transmission [6]. The good quality of all the components, from a strong DC motor with electronic speed control, to a complex NI PXI controller with LabVIEW RT, offered a good validation environment of the numerical simulations.

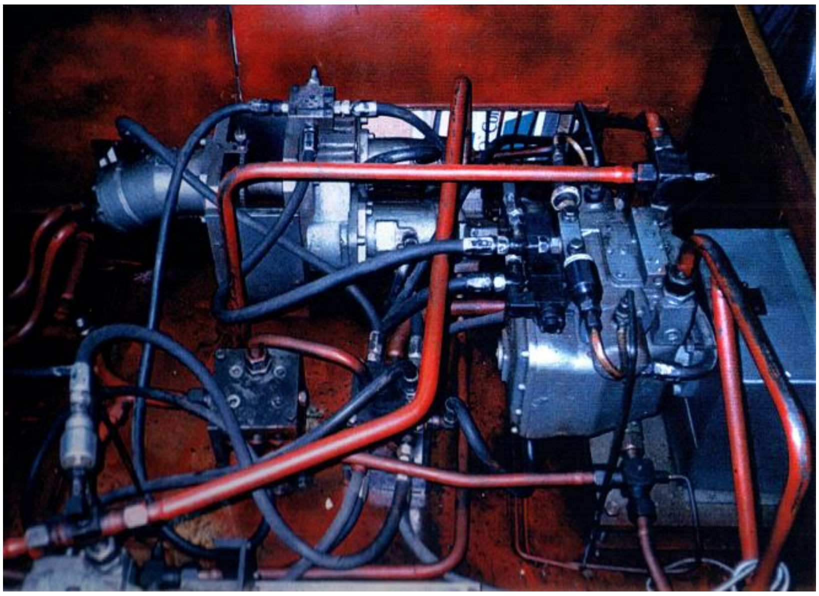

Fig. 15. Split power hydrostatic transmission in the Fluid Power Laboratory of the University POLITEHNICA of Bucharest (partial view). [6]

The researches developed by the manufacturers of high-pressure axial pistons machines on the split power transmissions of the so-called “high-mileage” vehicles heavy-duty vehicles generate high efficiency compact transmissions (Figs. 16 and 17).

Fig. 16. Hydrostatic compact unit A41CTU for power-split transmissions [7].

Fig.17. Planetary gear HYDROTRAC for heavy vehicles with power-split transmissions [8].

All these new hybrid units are electronically driven by general-purpose software as BODAS-drive DRC “which is a software solution embedded in Rexroth controller RC12-10/30 to control hydrostatic drive trains of wheeled vehicles. BODAS-drive covers a wide range of drivetrain types. The drivetrain is always based on an engine with CAN interface and a hydrostatic drive consisting of a pump and at least one motor. The gearbox type can vary between fixed gears, a gearbox shiftable during standstill, a shift-on-fly gearbox, a summation gearbox or radial piston motors mounted at the wheels”. [9]

4. Conclusions

The numerical simulation using realistic models, based on adequate super components like hydrostatic slippers is a very useful tool to choose properly the structure and the parameters of the hydrostatic transmissions [10]. The continuous world progress in the use of the fluid power systems needs more and more the development of new tools for the correct description of the border phenomena occurring in the complex technical systems controlled by digital devices. The compatibility of the electronic, electro mechanic, and fluid power systems remains a real challenge for all the innovative companies, sustained by the modern universities [11], [12], [13], [14], [15]. The study by FEM and other modern digital analysis methods of the new materials behavior in extreme mechanical conditions [16] will increase the performances and the lifetime of the hydrostatic transmissions. The development of the new generation of control algorithms as fuzzy ones, together with the adequate real-time simulation environments [17], can reduce the implementation time of the hybrid control systems.

R E F E R E N C E S

[1]. *** *BOSCH* – Bosch Automotive Handbook, 9th Edition, John Wiley & Sons Ltd, 2014.

[2]. *** *Rexroth*, Axialkolben machinenen, RD 90100, 1981…2019.

[3]. *Vasiliu D.*, Researches on the Hydrostatic Servopump and Servomotors of the Hydrostatic Transmissions. Ph.D. Thesis, University POLITEHNICA of Bucharest, 1997.

[4]. *Negoiță G. Cl.*, Researches on the Dynamics of the Hydrostatic Transmissions. Ph.D. Thesis, University POLITEHNICA of Bucharest, 2011.

[5]. *Ion Guță D.D.*, Researches on the Real Time Simulation of the Fluid Power Systems. Ph.D. Thesis, University POLITEHNICA of Bucharest, 2008.

[6]. *Vasiliu N., Vasiliu D.*, Fluid Power Systems, Vol. I, Technical Publishing House, Bucharest, 2005.

[7]. *** *Rexroth*, HYDROTRAC GFT for mobile equipment Series 8000: Transmission units for high-mileage vehicles, RE98707/10.2014.

[8]. *** *Rexroth*, New size of hydrostatic compact unit A41CTU for power split transmissions in low HP Range, PI 089/13, 2013-09-06.

[9]. *** *Rexroth*, Application software BODAS-drive DRC, RE 95323, 08.2019.

[10]. *Vasiliu N., Vasiliu D., Călinoiu C., Puhalschi R.,* Simulation of the Fluid Power Systems with Simcenter Amesim, CRC Press, Taylor & Francis Group, Boca Raton, Fl, USA, 2018.

[11]. *Ivantysyn, J., Ivantysynova M.*, Hydrostatic Pumps and Motors. ABI, New Delhi, 2001.

[12]. *Grabbel, J., Ivantysynova, M.*, An investigation of swash plate control concepts for displacement-controlled actuators. International Journal of Fluid Power, Vol. 6, No. 2, pp.19- 36, 2005.

[13]. *Ivantysynova M. and Baker J.*, Power Loss in the Lubricating Gap between Cylinder Block and Valve Plate of Swash Plate Type Axial Piston Machines. International Journal of Fluid Power, Vol. 10, No. 2, pp. 29 – 43, 2009.

[14]. *Pelosi M., Ivantysynova M.*, A New Fluid-Structure Interaction Model for the Slipper-Swashplate Interface. Proceedings of the fifth FPNI PhD Symposium, Cracow, Poland, pp 219 – 236, 2008.

[15]. *Milella A., Di Paola D., Cicirelli G.*, Mechatronics Systems, Simulation, Modelling and Control. In-Tech, Croatia, 2010.

[16]. *Vasiliu N., Roşu C.A., Sorohan Şt., Marin S.*, Theoretical and Experimental Researches on the Axial Piston Pumps Yoke. ANSYS Conference, Pittsburgh, 2002.

[17]. *Puhalschi R.*, "Modeling and Real Time Simulation of the Energy Control Systems", Ph.D. Thesis, University POLITEHNICA of Bucharest, 2014.