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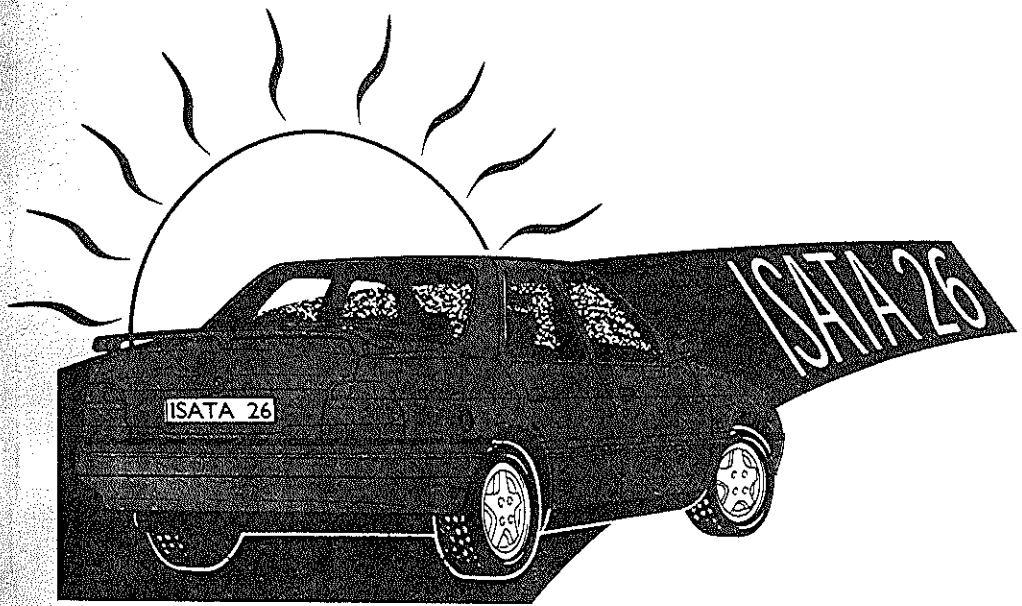

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ENEA ENTE PER LE NUOVE TECNOLOGIE, L'ENERGIA E L'AMBIENTE required secondary pulley pressure and the measured pressure are shown in Figure 7. The dynamic pressure response is fast, indicating that the pressure controller functions properly. After a step input of the rate of change, a steady state error of 0.5 bar remains, representing the dead zone which is included in the adaptation laws to prevent parameter drift [4].

Analysis of the pressure ratio is given in Figure 8. A step response at a magnified time scale is analyzed. The pressures show a small ripple, which is caused by tilting of the rotating pulleys. The rotational frequency of the pulleys is reflected in the frequency of the pressure fluctuations. This plot shows that the clamping pressure on the secondary pulley reaches its new value within 0.03 sec. The pressure ratio shows limited overshoot and a small rise time, indicating that the required bandwidth has been obtained.

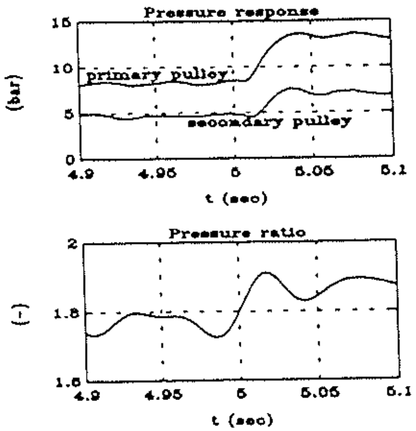

Figure 8. The measured pulley pressures and the corresponding pressure ratio due to a step input of the CVT ratio change at t = 5.00 sec.

5 Conclusions

An electronic CVT controller is presented which yields an accurate rate of CVT ratio change while the clamping pressures are maintained at a safe minimal level. The rate of change of CVT ratio is closed-loop controlled through the oil flow into a hydraulic cylinder. A Kalman filter is employed to provide noise free estimates of the realized oil flow. The clamping pressures are controlled by an adaptive pressure controller which yields a stable and accurate response during all operating conditions. The controller has been verified by various test-rig experiments and is proven to be adequate for use in a flywheel-hybrid vehicle transmission.

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A MECHATRONICS APPROACH TO THE DESIGN AND CONTROL OF AN ELECTRONICALLY CONTROLLED CONTINUOUSLY VARIABLE TRANSMISSION[1]

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93ME057

Abstract

Although the idea to build a continuously variable transmission (CVT) arose hundred years ago, basic research with respect to the possibilities of this type of transmission started only a few decades ago ([7], [6]). The strict regulations of the government with respect to fuel consumption and pollution, as well as the increasing demands by clients are the most important reasons for the intensive research to improve the classical gear box.
From the characteristics of a combustion engine, it follows that the fuel consumption of a car with a well-controlled CVT decreases drastically, compared to a car with a manually shifted classical gear box. It is our goal to develop an electronic control system for the CVT, such that the actual hydraulically controlled CVT (HC-CVT) can be replaced by an electronically controlled CVT (EC-CVT).
To design such an electronic control system, different control levels are considered simultaneously. At first, to study the behaviour of the system at the different levels, models of different complexity are built, using bondgraphs and identification techniques. Afterwards, these models are used in computer-aided control system design (CACSD): first, linear feedback controllers are designed, next, these linear controllers are extended by non-linear feedforward control, to increase the accuracy of the control system.

1 Introduction

As a vendor of CVT's (transmission and control system), VCST tries to fulfill all clients' wishes. Basically, three types of specifications can be recognized:

- Safety and comfort specifications are obviously the most complex specifications, since they are difficult to measure in an objective way.
- Fuel consumption, emission and noise specifications are mainly imposed by the government, to constrain the environmental load.
- Fuel consumption and acceleration time specifications are important trumps for the car vendor when promoting a car, for the car buyer when selecting one.

On the other hand, as a producer of CVT's, VCST wants to adapt a 'basic CVT' towards all client's specifications, since this is probably the cheapest, easiest and fastest way to answer all questions. When using such a strategy, it is silently assumed that the structure of the 'basic CVT' is rich enough to cover a wide range of specifications: to reach sub-optimality, it is sufficient to tune the

[1] This research was sponsored by VCST and the Flemish Institute for Promotion of Scientific and Technological Research in Industry (I.W.T.). The scientific responsability rests with the authors.

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parameters, while the structure may remain unchanged. Furthermore, numerous hardware tasks will be taken over by the control software, since an electronical control unit will be used. If the previous assumptions are correct, it follows from the table that only 'low-cost changes' are needed.

Relative cost-of-change	Software changes	Hardware changes
Structural changes	5	20
Parameter changes	1	5

Although it is possible to use the well-known proto-typing methods to design and tune the 'basic CVT' (as in HC-CVT design), VCST searches for a more fundamental approach to solve the EC-CVT problems.
This paper is organized as follows: in Section 2, an optimal shifting path for combustion engine cars is derived, while Section 3 treats our 'basic CVT'. Section 4 states the main ideas behind the mechatronics approach. Finally, sections 5 and 6 show some results in modelling, design and control of the powertrain and the actuator.

2 An optimal shifting path for combustion engine cars

For combustion engines, two characteristics are of major importance:

- The torque-speed characteristic gives the engine torque T_{eng} as a function of the rotational speed N_{eng} and the throttle position α.
- The consumption characteristic gives the specific fuel consumption S_{eng} as a function of the rotational speed N_{eng} and the throttle position α.

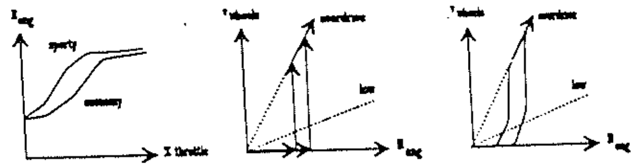

Figure 1: *The curves of 'optimal engine speeds' for economy mode and sporty mode (l). Ideal shifting paths in economy mode, for various throttle positions (m). Due to comfort specifications, the optimal paths are smoothed. These smoothed, sub-optimal paths form the so-called variogram (r).*

From these characteristics, the optimal shifting path for each throttle position can easily be derived (figure 1). For example, to have a car with a sporty behaviour, it is important to accelerate fast: for each throttle position α, a high engine power $P^*_{eng}(\alpha)$ must be delivered. The corresponding rotational speed $N^*_{eng}(\alpha)$, that follows from the torque-speed characteristic, can be seen as a curve of 'optimal engine speeds' for the *sporty mode*.
Similarly, by minimizing the specific fuel consumption for each throttle position, a curve of 'optimal engine speeds' for the *economy mode* is derived. To realize a compromise between a sporty and an economic behaviour, intermediate curves are designed.

3 A CVT with a metal V-belt and a wet-disc clutch

The wet-plate disc clutch consists of a set of wet friction plates: if the oil pressure p_{clutch} is high, more torque can be transmitted. By varying the clutch pressure p_{clutch}, the torque is varied and the speed difference $\Delta N = N_{eng} - N_{prim}$ can be controlled. To drive away and ride out smoothly, the clutch has to slip ($\Delta N \neq 0$). Most of the time however, the clutch is closed ($\Delta N \simeq 0$) to avoid low transmission efficiency and oil heating.

The variator realizes the shifting actions. Basically, the variator contains two pulleys, connected by a metal V-belt. By changing the position of the belt on the pulleys, the transmission ratio i_G can be varied. Because the belt position can be varied continuously over a limited range, the variator can shift between low and overdrive. By changing the ratio of the oil pressures $\frac{p_{prim}}{p_{sec}}$, the variator ratio is varied. To avoid slip of the belt, the value of both pressures must be sufficiently high (proportional to the torque to be transmitted). However, if the pressures are too high, the pump consumes more power than necessary and the efficiency of the transmission decreases.

The control system is used to realize the optimal shifting strategy, derived in section 2. To do so, the controller uses all measured signals from the driver, the powertrain and the actuator, to calculate electrical pulse-width modulated servosignals for the actuator, that transforms them into hydraulical pressures p_{sec}, p_{prim} and p_{clutch} for the transmission.

However, the hydraulic servo valves in the actuator are sensitive for changes in the oil characteristics (due to temperature variations) and for variations in the pumpflow (due to the changing engine speed N_{eng}). Therefore, the control system is not only responsible for the realization of the shifting strategy, it also surveys the actuator and compensates for its time-variance.

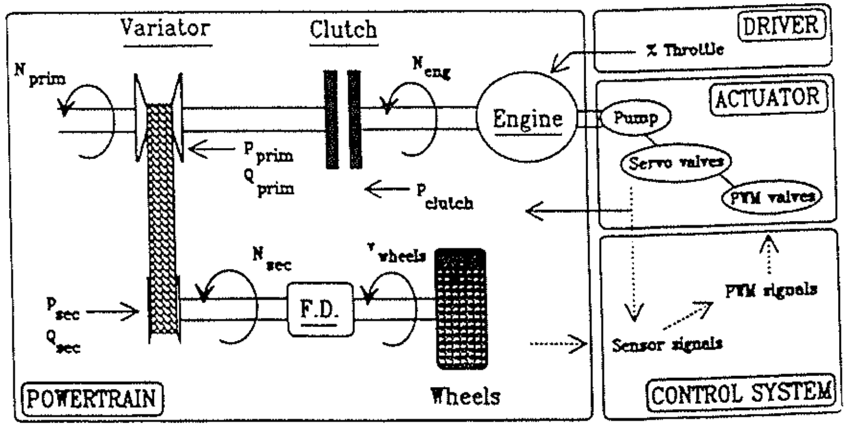

Figure 2: *A principal scheme of the EC-CVT. Although a small part of the engine power is immediately consumed by the external gear pump (a component of the actuator), most of the power goes through a wet-disc clutch, via the variator and the final drive reduction to the wheels.*

4 A mechatronics approach

Although there's still no consensus about the exact meaning of mechatronics, we claim our approach is a mechatronics one, since it satisfies most of the stated definitions. The main ideas of our approach are mentioned below:

A multi-level approach: the specifications on vehicle performance are translated into specifications on systems and signals, at different hierarchical levels. While maintaining as much interconnection as required, it is possible to do modular analysis, design and control. Three levels are recognized:

- The powertrain level: to analyze and control the interactions between the powertrain elements.
- The powertrain element level: to analyze, design and control the different powertrain elements, e.g. the engine, the clutch, the variator.
- The actuator level: to analyze, design and control the actuator valves.

A model-based approach: instead of doing proto-typing on the real systems, we build a library of mathematical models, that can be used for analysis, design and control. Multiple design and control techniques can be applied to these models of various complexity and accuracy, to generate and evaluate a number of alternatives, at a lower cost and at a higher speed as in classical proto-typing methods. Once the 'optimal' solution is found, it can be implemented and validated. Two different model types are recognized:

- White box models describe the complete physical behaviour of the system. The technique of causal bondgraphs is used to derive these physical models. Generically, the models are non-linear and have high order. The main advantages of these models are the high accuracy and the fact that the model is parametrized by well-known physical and geometrical parameters.
- Black box models have only the same input-output relation as the original system. By using linear identification techniques, a linearized, low order model of the dynamics of the system is found.

An integrated approach: although it is very tempting to solve the total problem by solving isolated problems on isolated levels, this is definitely not the right strategy. The following questions define an iterative process, to maintain as much interconnection as required, not only between problems on different hierarchical levels, but also between different problems at the same hierarchical level.

- Are the controller's wishes inside the system's limits of performance? If the controller asks more then the physical structure is able to give, either the control law or the physical structure is changed.
- Can the wishes of a high-level controller be fulfilled by the low-level controllers? If the high-level controller asks more then the low-level controllers are able to give, either the high-level control law or some of the low-level control laws are changed.

A multi-domain approach: experience from many fields (mechanics, hydraulics, electronics, computer sciences) is required, to fully understand the complete system.

A multi-skill approach: to treat the problem in a fundamental way, various skills must be mastered: modelling, identification, signal conditioning, signal processing, control design and control implementation.

5 Modelling and control of the powertrain and its elements

The global model of the powertrain is derived in [3] and is similar to the bondgraph model obtained in [9]. The following table states the modelled and the unmodelled effects:

	Modelled effects	Unmodelled effects
Engine	Static torque-speed characteristic	Engine dynamics (delayed injection, sequential firing)
Road load	C_x, slope γ, rol	
Clutch	Wet friction, stick-slip ([5], [3])	
Variator	Belt stress, centripetal forces micro-slip, clamping forces ([8], [3])	Variator dynamics
Dynamics	Inertias (shafts, pulleys, car)	Shaft torsion

As mentioned earlier, the powertrain will perform optimal under variogram control. In practice, the car speed v_{wheels} and the throttle position α are measured and used to determine the optimal engine speed N^*_{eng} from the variogram.
Below the low-line (clutch mode), variogram control is realized by controlling the clutch, while the variator remains in low. Belt slip must be avoided by variator control. However, between low and overdrive (variator mode), variogram control is mainly realized by the variator, while the clutch is closed most of the time. To have a smooth transition between these modes, both controllers cooperate around low (transition mode).
The following table summarizes the most important control features:

	Clutch	Variator
Setpoint	$N^*_{eng} - N_{prim}$	N^*_{eng}
Control law	PI	PI, with logics
Constraints	$p_{clutch} \leq 15$ (bar)	$i_{OD} \leq i_G \leq i_{LOW}$ $4 \leq p_{prim} \leq 25$ (bar), $4 \leq p_{sec} \leq 40$ (bar) Minimize pressures, avoid belt slip
Tuning	Trial-and-error	Trial-and-error

In figure 3, the main results for the clutch are shown, while figure 4 presents the main results for the variator. It is shown that the models can be used for analysis and control of the powertrain elements and their interactions. Furthermore, it seems possible to predict the required actuator signals. How a suitable actuator is designed and controlled, is explained in the next section.

6 Modelling, design and control of the actuator

As mentioned before, two techniques (bondgraphs and linear identification) are used to model the actuator valves. In this section, only some results obtained with bondgraph models are discussed. In [3] and [4], a bondgraph model for the secondary valve was derived in detail. This valve transforms a servo pressure p_{servo} into an hydraulic pressure p_{sec}. The measured pressure and the simulated pressure (for the same input sequence under the same operating conditions) are compared in figure 5. It turns out that this model is able to do good predictions, for a wide range of oil temperatures, pump flows and model parameters. This makes the model usable for valve design.

However, the bondgraph model is also usable for control design ([3]). An excellent static feedforward is easily generated by inverting the static model. Since the oil temperature and the engine speed are measured, this feedforward is able to compensate for variations in the oil temperature and in the pump flow. The increase in dynamic accuracy is also advantageous. On the other hand, dynamic feedback (mainly PID-like controllers) is designed using heuristic tuning rules or multi-objective optimization techniques ([1], [2]). In [3], a gain-scheduled PI controller is designed, using the transient response method of Ziegler and Nichols.
When combining both controllers, an actuator with excellent closed-loop properties is obtained.

Time domain	Step response	Polynomial response	Frequency domain	
Steady state error (bar)	0	0	Phase margin (°)	≥ 60
Overshoot (%)	0	0	Gain margin (dB)	≥ 2
Settling time (s)	≤ 0.050	≤ 0.060	Bandwidth (Hz)	≤ 30
Error-norm (%)	$\leq 10\%$	$\leq 10\%$		
Actuation signal (bar)	$\geq 0, \leq 4$	$\geq 0, \leq 4$		

This section showed that the bondgraph models can be used in actuator design and control. Once an 'optimal' design/controller is obtained, it will be implemented and tested.

7 Conclusion

To conclude, let us point out that the mechatronics approach allows us to formulate and to solve the problem as a multi-level problem. By constructing a library of mathematical models, computer aided analysis, design and control of powertrains and actuators can be done easily. Reduced development times, lower design costs, increased reliability, improved quality and more flexibility towards clients are the main advantages of this innovating approach.

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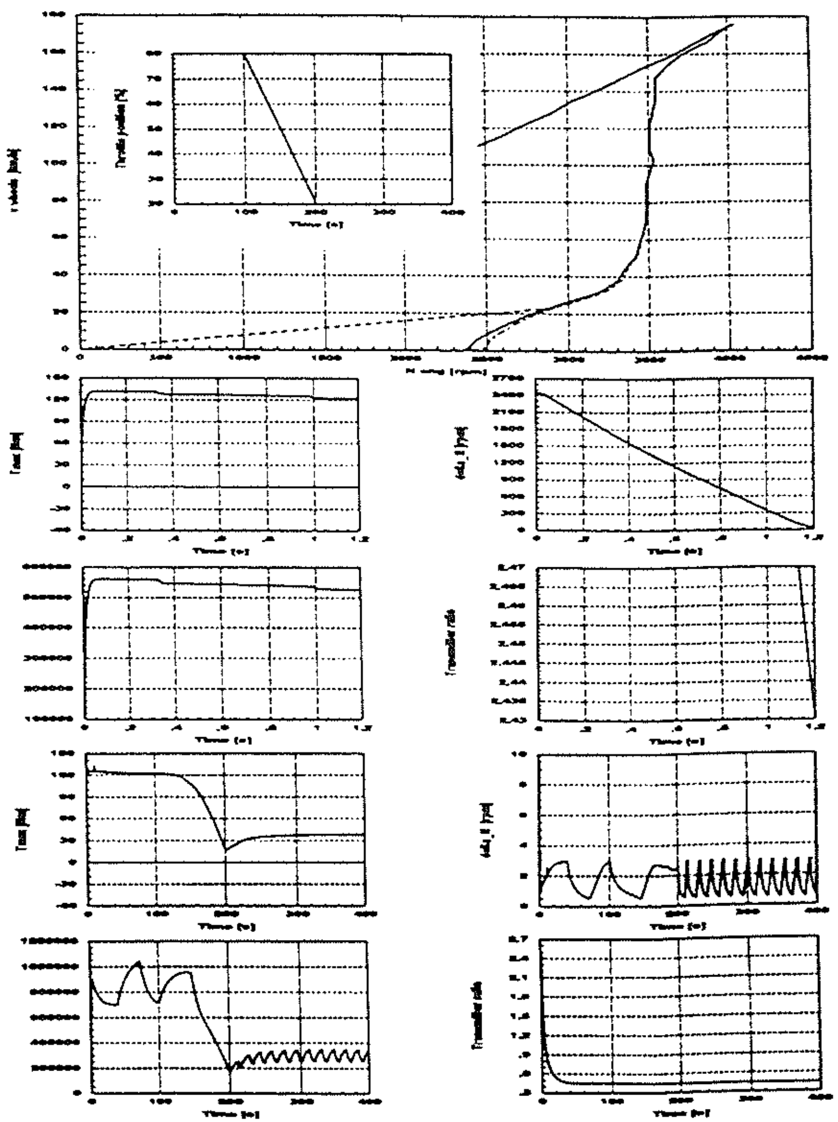

Figure 3: *The realized (dashdotted line) and the desired variogram (full line) are almost the same. The dashed line represents the low-line (t). The car starts from stall ($N_{eng} = 2400$ rpm, $v_{wheels} = 0$), with a fixed throttle position (80%). The speed difference ΔN decreases linearly, till the clutch locks 1.2 s later. After a fast increase, the clutch pressure remains almost constant. The variator remains in low, till the low line is approached (m). At that moment, the variator shifts from low to overdrive. To keep the clutch closed ($0 \leq \Delta N \leq 3$), the clutch pressure p_{clutch} oscillates around a mean pressure, which is proportional to the transmitted torque T_{max} (b).*

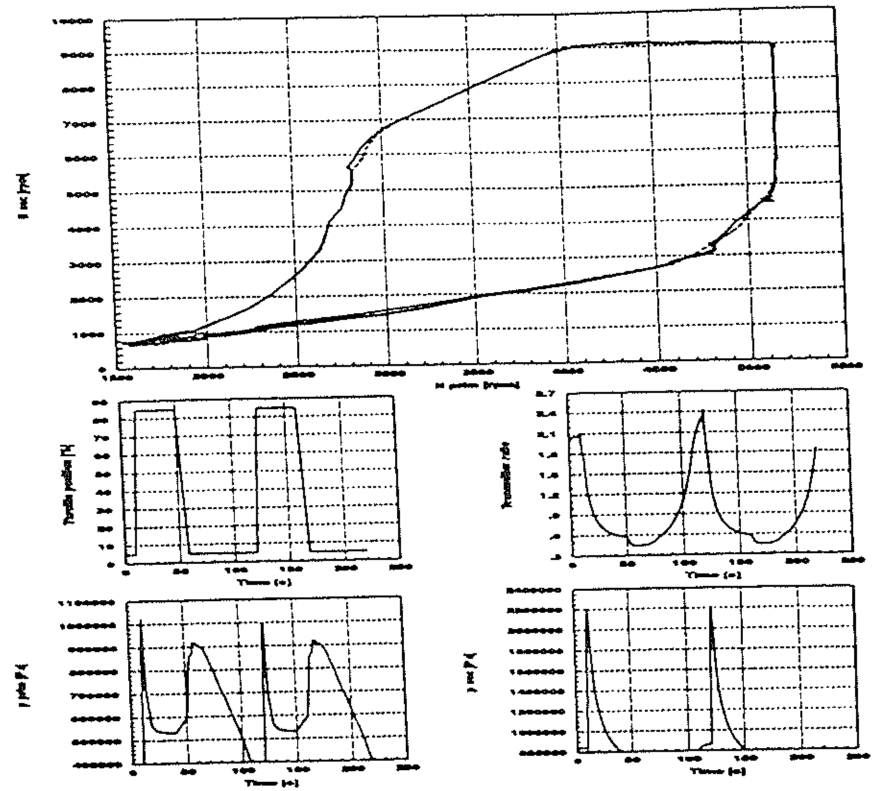

Figure 4: *The realized (full line) and the desired variogram (dashed line) are almost the same (t). The variator shifts between low and overdrive, to realize the desired variogram (m). In most cases, p_{sec} or p_{prim} is critical to avoid belt slip, while the other pressure is adapted to realize the variator's ratio (b).*

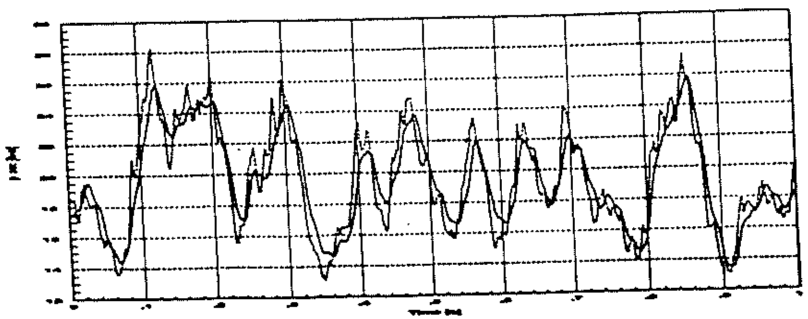

Figure 5: *The measured (full line) and the simulated secondary pressure (dotted line) are almost equal. The parameters of the model are adjusted to the prototype's parameters. The high frequency components in the simulated pressure are not present in the measured signal, since the measurement is done in a volume at a certain distance from the valve output (which means that an hydraulic low-pass filter should be added to the valve model).*

MECHATRONICALLY-CONTROLLED & ELECTRICALLY-POWERED FOUR-WHEEL-STEERING CONVERSION SPHERES

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ABSTRACT

The development of Very Advanced Conversion (VAC) Technology have significant progress during the 1990s. A driving factor behind this have been the increasing requirements for bettering of ride comfort and active safety of automotive vehicles (AV). The steering converts the steering wheel rotary motion into the turn of the wheels of AVs. A major contribution to this progress is the application of mechatronically-controlled and electrically-powered Rack-and-Pinion (R&P) steering gears. As the conventional power boosted Four-Wheel-Steering (4WS) conversion sphere can only provide a constant assistance characteristics, whereas the AV ideally requires a variable boosting, the application of mechatronically-controlled and electrically-powered rack & pinion steering gear will be introduced, mainly to adapt and predict the level of power boosting to the AV speed of travel. This paper to be intended to describe first the current state-of-the-art VAC Technology and then to show steps towards VAC spheres with the higher level of mechatronically control and greater complexity of the 4WS conversion spheres.

INTRODUCTION

The improvement Four-Wheel-Steered (4WS) conversion automotive high-technology (high-tech) has made significant progress during the 1990s. An evolutionary factor behind this has been the increasing requirements for an active safety and a ride comfort of Automotive Vehicles (AV). A major contribution to this progress is the introduction and fast growing application rate of mechatronically controlled and electrically powered Rack-and-Pinion (R&P) steering gears. As the standard incontrolled and fluidically powered R&P steering gear can only provide a constant assistance characteristic, whereas the AV ideally requires a variable support, the introduction of mechatronically controlled and electrically powered R&P steering gear is initiated, mainly to adapt the level of power assistance to the vehicle speed. To achieve these objectives a further increase in the flexibility and controllability of the R&P steering gear is required. Only the expanded use of Electronic Control Unit (ECU) combined with more complex smart Electro-Mechanical (E-M) conversion actuators can make these objectives achievable.

Currently, a great deal of attention has been focused on the Research-and-Development (R&D) of dual-mode hybrid 4WS conversion spheres for AVs in order to enhance their handling properties. At high travel speeds AV response can be enhanced by steering the rear Steered-and-Motorized Wheels (S&MW) in the same sense of steering direction as those in the front, while at low travel speeds AV manoeuvrability can be enhanced by turning front and rear S&MWs in opposite senses of steering direction. It is well known that the addition of dual-mode hybrid 4WS conversion spheres improved the responsiveness of AVs by reducing transient response time, and also reduced undesirable AV motions such as fish-tailing, making an AV easier to conversion control during a potential accident situation. However, it is also well known that the addition of a dual-mode hybrid 4WS conversion sphere did not appreciably extend the overall stability of an AV.