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Pressure Control of Haldex Limited Slip Coupling

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Abstract

The work we have done is very practical. Half of it, the identification phase, was to understand the coupling and make simple experiments and measurements. The other half, the synthesis phase, was to design a controller, implement and test it. Tests were carried out both in lab and in a car.

The Haldex Limited Slip Coupling is a solution for the traction of an all wheel drive vehicle. The coupling is mounted between front and rear differentials of the driveline. It consists mainly of a mechanical and hydraulic system. To adjust the coupling for different driving situations it is possible to control it with a valve. This is done in a microcontroller with an interface to the CAN-bus. The bus provides information from the ABS and the engine.

We added a pressure sensor to the coupling and designed a controller to see how well the pressure may be controlled. It was easy to reach an accurate pressure level. Thereby, we fulfilled the requirements for the thesis. Also, speed performance was improved by the controller, for the case of torque distribution. The linearization table we have used for the valve was frequently saturated, so the improvement was limited. The table is one of several things that we did not study very carefully.

A more accurate design procedure should give us even more speed improvement. This includes better identification of the coupling, changes in the linearization table or a complete replacement of the valve, more careful construction of the pressure controller, integration of wheel slip control and more testing.

We have some simple suggestions for further development. One is to identify the elasticity of the coupling as carefully as possible. Another is to calculate the control authority which depends on the capacity of the pump, the efficiency in the disc clutch and the elasticity of the hydraulic system. To summarize, the dynamics of the coupling is crucial for further development.

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Torkel & Bengt

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1 Background

In this chapter we make a short introduction to the Haldex Limited Slip Coupling and compare it to other 4WD-systems on the market.

One of the most important features of a 4WD-system is control of the torque transmitted to the wheels. In order to get as much power as possible on to the ground, the torque has to be divided among the four wheels to minimize wheel spin. The Haldex Limited Slip Coupling, HLSC, deals with this problem in a new and, from many points of view, better way.

The first prototype of the coupling that later would become the Haldex Limited Slip Coupling was invented by the Swedish rally legend Sigvard Johansson and his son Peter. Their goal was to make a coupling that adapted itself to the engine speed and distributed the engine torque between the front and rear wheels.

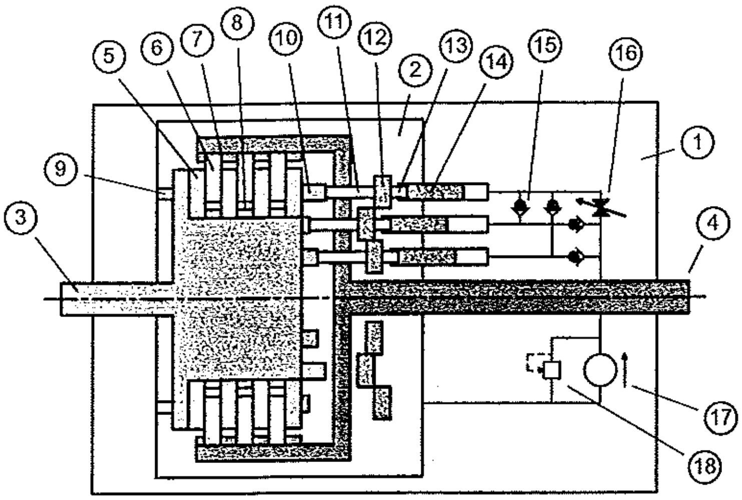
The result was a great success and in 1988 it was patented by the company Haldex under the name Ipumatic. The name Ipumatic was changed to Haldex Limited Slip Coupling (HLSC) in 1996. The HLSC is one of the most advanced 4WD-couplings on the market today, it has good torque control which can be adjusted easily.

Compared to other couplings, it can easier be integrated with other electronic systems provided in a car today. Safety systems like ABS (Brake control), ESP (Stability control) and TCS (Traction control) require rapid activation and deactivation of a 4WD coupling. These are abilities that distinguish the HLSC from other 4WD-systems and make the HLSC superior in many aspects.

1.1 System introduction

As a brief introduction to the HLSC, the coupling can be compared to a hydraulic piston pump which is driven by the slip between input and output shafts (see Figure 1-1). The oil flow from the pump results in an oil pressure that is controlled by a throttle valve. The oil pressure compresses a wet multi-plate clutch that delivers torque to the output shaft. The torque on the output shaft is proportional to the oil pressure. Without the throttle valve the coupling would transfer torque whenever there is differential rotational speed between the two shafts. This is however not always wanted, for instance during parking and tight curves or towing the torque transfer should be kept to a minimum. This is the main function of the coupling, there are however many dynamics and nonlinearities present. A more detailed description of the HLSC is made in chapter 3.

The HLSC can be mounted between the front and rear differentials for torque distribution, but also on a front or rear axles to distribute torque between the two wheels. The second of these possibilities is less common and not discussed in this master thesis.



• Figure 1-1. Haldex Limited Slip Coupling.

1.2 Today's control

The HLSC uses signals from the CAN-bus, which is used in most modern European cars. CAN-signals used are for instance wheel speeds, engine torque, accelerator position together with status flags for brake light, ABS and ESP. This information is used to calculate the torque wanted on the output shaft. The throttle valve is then set.

Different tires and tight curves result in a slip between the input and output shafts of the coupling and this results in non-desired pressure and torque. In these situations, the throttle valve is then set differently to give the desired pressure.

The solution of using CAN-signals and a throttle valve has the advantage that no big hardware changes have to be made to make it fit different cars. Also, if changes in the performance of the car is wanted, most of the changes can be made in the software.

The software in today's version uses three different strategies for controlling the throttle valve. During normal driving condition, when there is limited wheel slip, the throttle valve position is set depending on the motor torque, accelerator position, wheel speed, oil temperature etc. Whenever the slip becomes to great, the valve is opened until the slip decreases. Whenever the brakes are used, the valve is increasingly opened depending on the vehicle speed. Whenever ABS or ESP is active the valve is fully opened.

During normal driving conditions at relatively high speeds and when torque control is active, the differential rotational speeds of the input and output shafts is difficult to calculate with good accuracy. To circumvent this problem, the supposed differential speed is calculated based on the vehicle speed. To achieve the desired torque transfer, this function is trimmed for different driving situations.

Some work is done during manufacturing to calibrate the throttle valve. Calibration is also necessary if the oil type is changed.

1.3 Other 4WD-systems

During the early 1980's, 4-wheel driven cars started to appear on the automobile market. First in line was the Audi Quattro which was very successful in rallying. By the end of the decade almost all serious car manufacturers had a 4WD version of at least one model. During the early 1990's, trends changed and the 4WD cars on the market were quite few compared to before. But now, in 1999, the off-road trend is back and with it, the number of 4WD cars on the market increase.

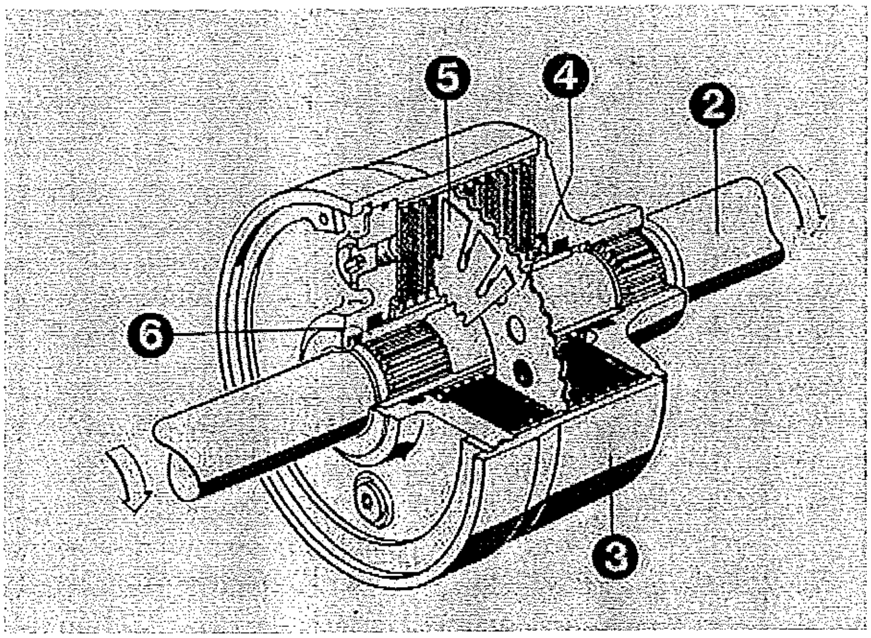
The ways to design a 4WD-system are many. The original way, used in Jeeps and military vehicles, is to lock the front and rear axles together forcing them to the same rotational speed. The lock mechanism is controlled by a lever next to the gear lever. This solution works satisfactory when driving off-road or on a surface which allows the wheels to spin like gravel or snow. But when driving on-road, making tight turns, the axles have a different rotational speed and if they are locked together, problems occur. Another problem is during hard braking when a wheel becomes locked. The other running wheels will tend to lock as well and in worst cases, all four wheels are locked, resulting in loss of car control. To avoid these problems, many drivers choose to only engage the 4WD while driving off-road or on snow. By doing this, they lose the extra safety a good working 4WD system provides.

This type of 4WD system is still used on many cars like pickup trucks and Jeeps. But on most 4WD cars on the market today the solution is another. A viscous-coupling, a planetary gear, a Torsen differential or a combination is often used.

1.3.1 The viscous coupling

The viscous coupling (see Figure 1-2) has many construction similarities to an automatic gearbox. The coupling is constructed with many discs (typically a total of 60) which are surrounded by a viscous silicone fluid. The housing (3) is connected to the input shaft (2) and has its punched discs pointing inwards (4). The center shaft has its slitted discs pointing outwards (5) and is connected to the output shaft (6). Differences in the rotational speeds of the two shafts will cause friction in the oil and torque transfer between the discs.

A viscous coupling can be used to connect the front and rear axle of a vehicle, usually on a front wheel drive. Torque is then transferred to the rear axle whenever there is wheel spin on the front wheels.



• Figure 1-2. *The viscous coupling*

Some disadvantages of the viscous coupling are the following:

- It requires a rather great differential rotational speed of the two shafts in order to transfer any torque. Also, the response time is rather slow.
- In order to make the shafts spin independently, without any torque transferred, a special release mechanism is needed. The release mechanism is often electro-magnetically or hydraulically controlled and is not integrated in the coupling. Without the mechanism, problems occur if, for instance, the car is being towed with one wheel pair still on the ground. The coupling will lock up, resulting in overheating or, in worst cases, a destroyed coupling. If the car is equipped with ABS or ESP this mechanism is also needed to ensure that the wheels spin independently when a safety system is activated.
- Changes in the characteristics of the coupling can not be made without great hardware changes.

1.3.2 The planetary gearing

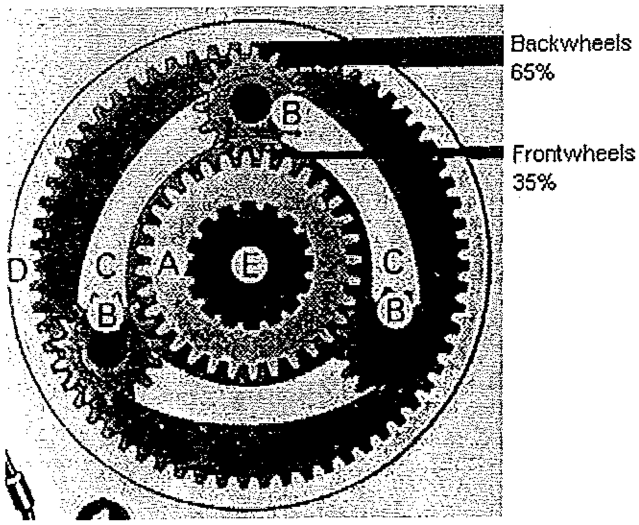
The planetary gearing (see Figure 1-3) consist of a sun-wheel (A), surrounded by three planet-wheels (B) held together by a planet-holder (C) and a gear-ring (D) with its cogs pointing inwards. The gear has one input shaft and two output shafts, all parallel. The input shaft is usually connected to the planet-holder, one output shaft to the sun-wheel and the other to the gear-ring.

Under normal conditions, all three shafts rotate at the same speed and the planet-wheels are not rotating.

The most common mounting of a planetary gearing is with the car's gearbox connected to the planet-holder, the rear axle to the gear-ring and the front axle to the sun-wheel but other solutions are possible depending on what torque distribution is wanted.

- One problem with the planetary gearing is that if one wheel loses traction and starts spinning, all the other wheels will then also lose torque.

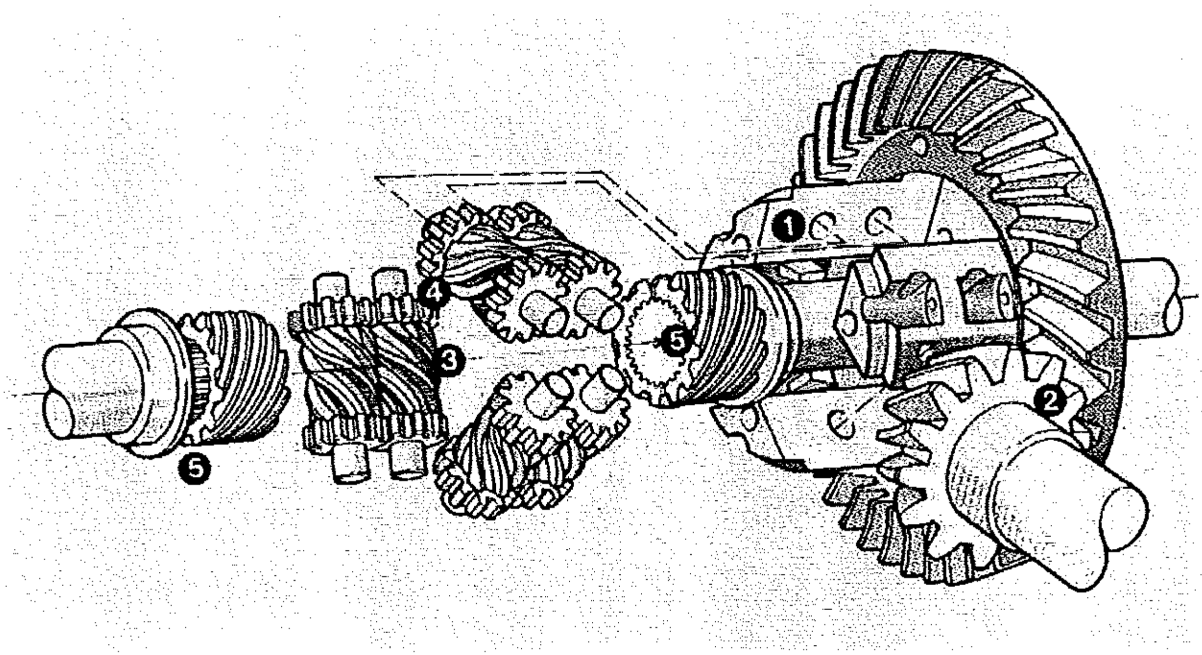
This problem can be solved by combining the planetary gearing with a viscous coupling on the two output shafts. The geometry of the planetary gearing makes this relatively easy since the shafts have the same center point. The problems of the viscous coupling, however, are still present.



• Figure 1-3. *The planetary gearing*

1.3.3 Torsen differential

Another way to implement 4WD is by using a Torsen differential. Torsen stands for TORque-SENsing. The differential is usually mounted as a central differential on the propeller shaft or as in Figure 1-4, in a rear axle differential. While a conventional differential, during wheel spin, limits the torque distribution to the level of the spinning wheel, the Torsen differential does the exact opposite, up to 80% of the torque can be transferred to the output shaft with the most traction.



• Figure 1-4. *The Torsen differential mounted in a rear axle differential*

The worm-gear (5), connected to the output shafts, are not fixed to the shafts, but connected by splines which let them move sideways if needed. The worm-gears are powered by two packages of worm-wheels, with three worm-wheels (3) each. The packages of worm-wheels are mounted in the differential case (1) and are connected to each other by cogwheels (4) in the ends of the worm-wheels. The differential case is then powered by the input shaft (2). Because of the gear ratio from worm-wheel to worm-gear, the worm-gears can easily turn the worm-wheels but not vice-versa.

During driving situations with no slip, the differential acts as a conventional one. But when a wheel starts slipping, because of the gear rotation and the splines, the worm-gears start to rotate and the worm-gears are pressed together. This transfers more torque to the wheel with greater traction.

Because of its mechanical solution, the Torsen differential has more rapid activation and deactivation compared to the viscous coupling. Since the torque transfer is only activated when there is positive torque on the input shaft (during acceleration), car towing and ABS is possible on a car equipped with a Torsen differential since no torque is transferred. The negative aspects are few but, like the viscous coupling, the mechanical construction makes it difficult to change the characteristics.

1.3.4 4-Matic

Mercedes-Benz has developed a rather complex 4WD system which uses a planetary gearing combined with two disc-clutches called 4-Matic (see Figure 1-5). The disc-clutches are controlled hydraulically by a microprocessor.

The gear-ring (2) of the planetary gearing is connected to the output shaft of the car's gearbox (1). The first disc-clutch (9) locks the planetary gearing's gear-ring (2) to the planet holder (4), which is connected to the rear axle (6).
The second disc-clutch (10) locks the front axle (14) to the sun-wheel (8).

Added to this is a third disc-clutch on the rear axle differential (not illustrated) which locks the differential, thus the rear wheels to each other. The third disc-clutch can only be activated at speeds lower than 38 km/h.

This makes for eight different combinations of locked disc-clutch of which four are used.

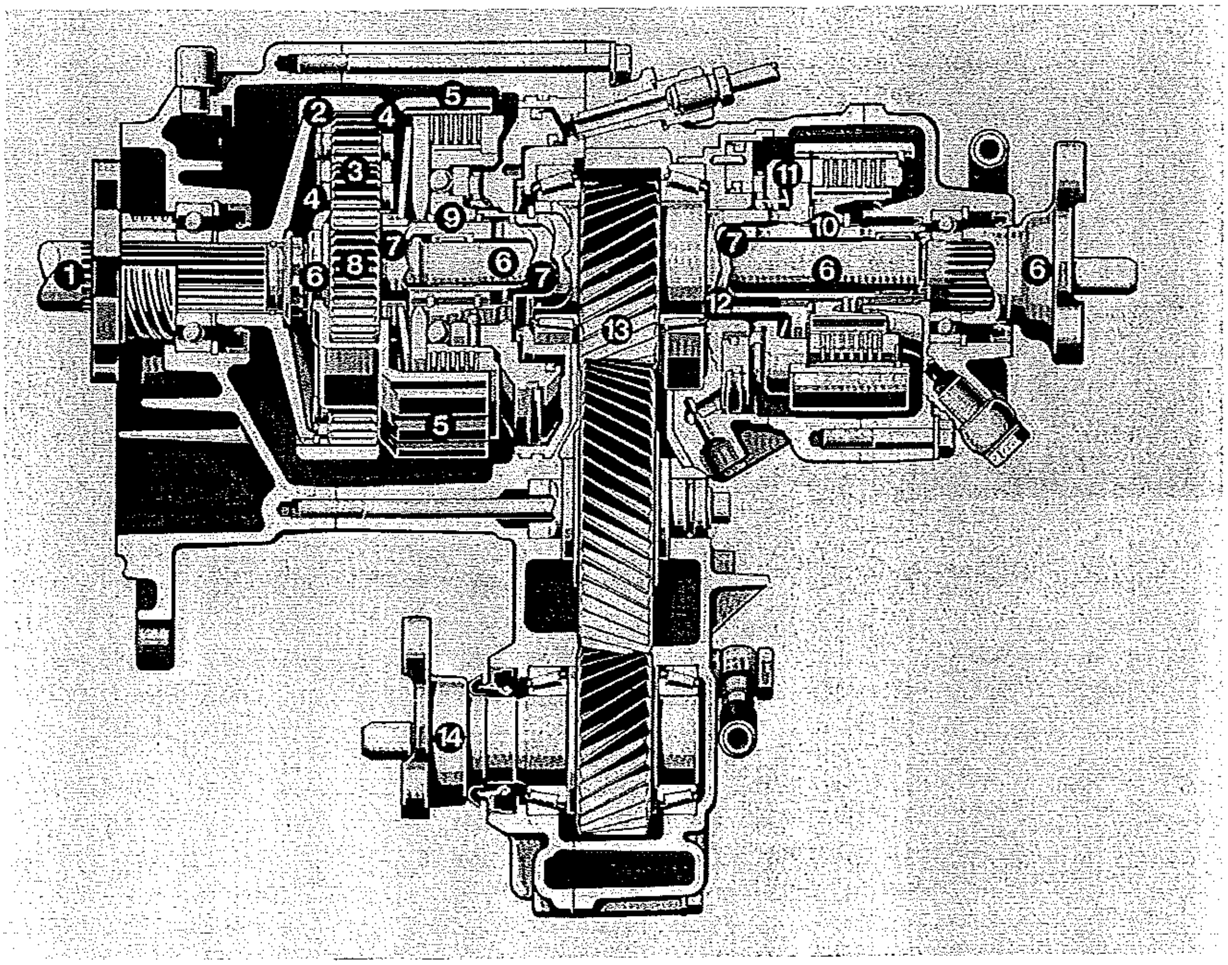
1. The first disc-clutch (9) locked and the second (10) open. All the torque is delivered to the rear wheels.
2. The first disc-clutch (9) open and the second (10) locked. 4WD with 65% of the torque on the rear axle and 35% on the front axle.
3. Both disc-clutches locked. The front and rear axles are locked together.
4. Same as 3 but the third disc-clutch is activated. With this combination, at least three wheels are always supplied with torque (one front wheel could still slip).

Which alternative is active is controlled by the microprocessor and depends on the driving situation.

The disadvantages of this system, besides the high weight, are few but the complexity makes it rather expensive.

One can also wonder the reason of the second disc-clutch (10). In which situations is it wanted to disengage the front wheel drive? Should not the front-wheel drive always be engaged? Answers to these questions have not been found in the material.

On later car models, the 4-matic system has been replaced by another type of computer controlled system. The new system uses a planetary gearing between the front and rear axles where the torque is distributed between the wheels by using the ABS system of the car. Whenever wheelspin is detected, the brake of this wheel is activated. This method makes it possible to control wheel slip on each wheel individually and combining it with ESP is possible. The disadvantage is that much power is lost using the brakes, power that can be used in better ways. Compared to the old 4-matic the new one is less mechanically complex but with the electronic control maintained. This makes it a good competitor to the HLSC.



• Figure 1-5. The 4-matic.

2 Project goals

The final goal of this project is to design a pressure controller that will be evaluated in comparison to the existing system. Improvements will be explored later in chapter 7.

In order to do this, one has to fully understand the function and complexity of the coupling and, to some extent, the application software used today. To do this, a mathematical model of the coupling will be made and tests to verify it will be made in a testbench. After this the controller will be designed and trimmed in the testbench. The controller will then be modified so that it can be tested in a car. At this point, the first comparisons the controller used today can be made.

3 Hardware

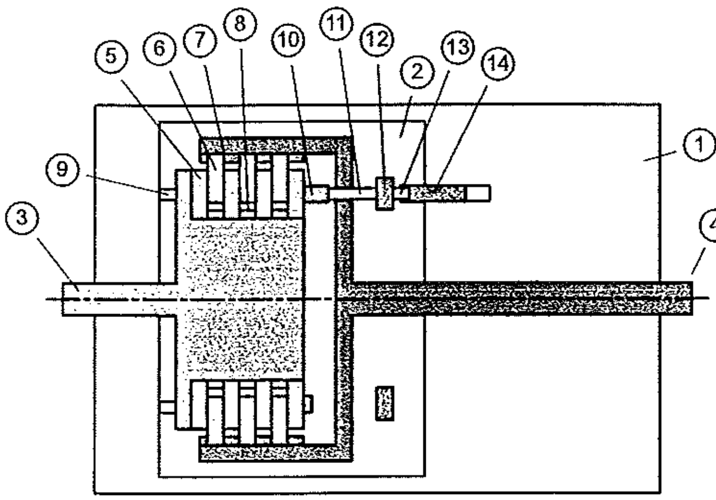
The purpose of the coupling is to transfer torque between front and rear differentials of the driveline. Torque transfer depends mainly on difference in rotational speeds of the shafts, but it is also influenced by the control of a built-in microcontroller.

Here below, we will first explain the parts of the coupling. Then a simple mathematical model is derived. The purpose is to model the most important behaviour of the coupling. The model is then validated to some extent and differencies and equilities are discussed. All this is done with the object of pressure control in mind.

3.1 Mechanical parts

In Figure 3-1 some parts of the coupling are shown. The incoming shaft (3) is connected to the front differential and the gearbox and engine. The outgoing shaft (4) is connected to the rear differential. In this configuration, a front wheel drive vehicle becomes an all wheel drive.

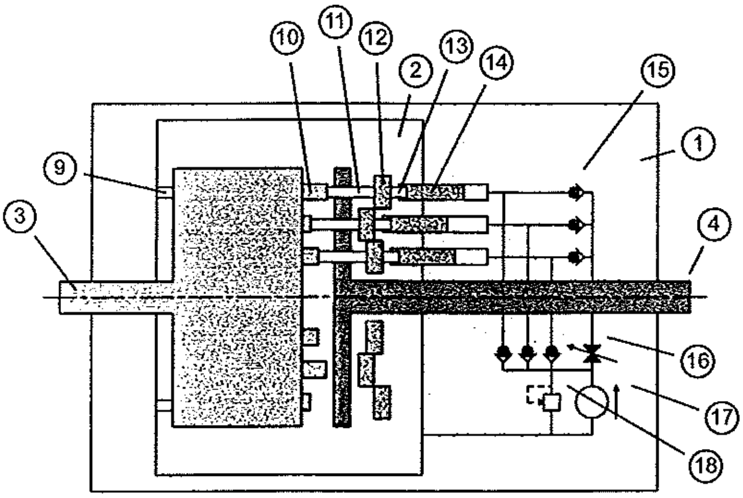
Torque transfer is accomplished with a disc clutch. Some discs are mounted on the incoming shaft (5) and some on the outgoing shaft (6). They are squeezed together by a hydraulic piston (14). The piston is mounted in the housing (1) and connected with rollers (11 and 13) and a ring (12) to the discs. The housing is filled with oil (2).



• Figure 3-1. *Principle of clutch.*

To create pressure for the disc clutch, a hydraulic pump is needed (see the parts in Figure 3-2). The pump is driven by different rotational speeds of the incoming and outgoing shafts. Three pistons (14) are mounted in the housing. Each piston is connected with a roller (13) to a ring (12). Another roller (11) is forced to move around with the outgoing shaft and is squeezed between a cam curve (10) on the incoming shaft and the ring. The three cam curves are designed so the pistons will move back and forth interchangeably.

When a piston moves back, oil flow in through a check valve (15). When the piston moves forth, the oil is forced through another check valve. A constant feed pressure is maintained by an electric pump (17) and an overflow valve (18). The pressure is increased further by the differential pump and a throttle valve (16). This pressure is used for the operation of the clutch piston mentioned before.



• Figure 3-2. *Principle of the differential pump.*

The two different parts described above, the clutch (Figure 3-1) and the differential pump (Figure 3-2), are combined to form the Haldex limited slip coupling (see Figure 3-3). The clutch is mounted to one of the cam curves, and the check valves are arranged so the function described above is maintained. The purpose of this is to save one piston in the design. The combination of the differential pump and the clutch gives fast activation response whenever the differential speed rises. This design in conjunction with the control of the throttle valve form the HLSC. The purpose of the valve is mainly to adjust the coupling for different driving situations.

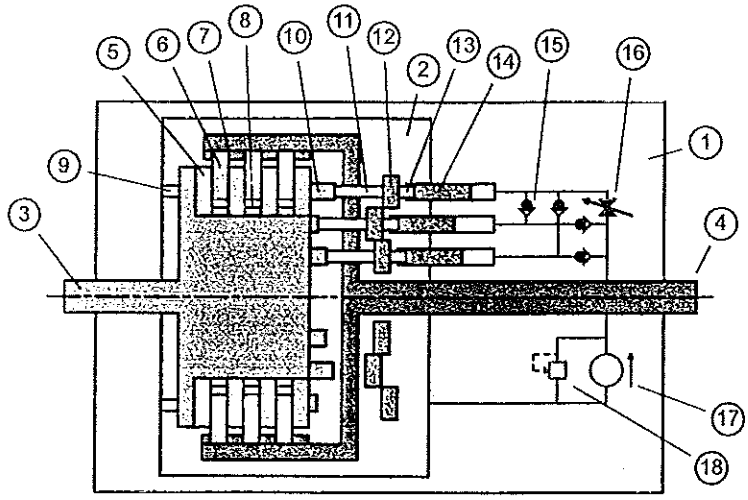


Figure 3-3. The Haldex Limited Slip Coupling.

The type used, a needle valve, is difficult to characterize. Flow depends on pressure fall over the valve, needle position and viscosity of the oil. Flow rate is almost linear with respect to pressure difference but highly nonlinear to needle position. A linearization table compensates for the nonlinear behaviour and also changes in viscosity. Oil temperature is measured and used for the table. The output from the table is actuated by a stepper motor which moves the needle to the correct position. The quantization of the motor is small. The motor takes one step per millisecond. If the motor windings are demagnetized, a coil spring returns the needle to fully opened.

We have used the existing linearization table for the valve, so we will not go into the subject of the valve any more. The notion of the valve characteristics is what is important. Some comments are given later on the qualities of the linearization table.

The assignment for us were to investigate pressure control of the coupling so we have added a pressure sensor in the hydraulic system. It is mounted next to the clutch piston. The signal from the sensor is connected to one of the AD-converters of the microcontroller.

The microcontroller is connected to the CAN-bus of the car. The bus provides information from various units. This is very important for the operation of the coupling. Wheel speeds and brake activation signals are read by the microcontroller. Also, engine torque and information from the clutch is read. The bus has some time delay.

3.2 Mathematical model

The model we develop is as simple as possible, describing the most important mechanisms of the coupling. The resulting equation is nonlinear and of first order.

The differential speed pump delivers oil flow proportional to differential speed.

$$\varphi_1 = k \cdot \omega$$

The factor k can be calculated if we know the areas and strokes of the pistons. The linearization of the throttle valve makes it easy to give an equation for the oil flow. Stiffness is a definition relating pressure fall to flow rate in the valve. The dynamics of the stepper motor is ignored.

$$\delta \equiv \frac{p - p_f}{-\varphi_2} \Leftrightarrow$$

$$\varphi_2 = -\frac{p - p_f}{\delta}$$

The pressure p is imposed on the clutch piston and p_f is the feed pressure from the electric pump. The oil, the housing and the pistons are elastic.

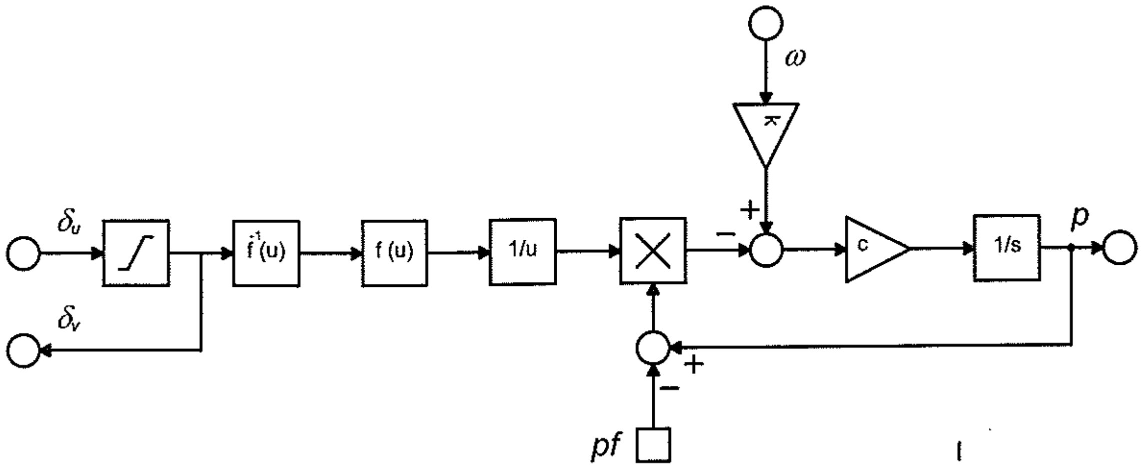
$$\varphi_3 = -\frac{1}{c} \cdot \frac{dp}{dt}$$

The sum of the three flows must be zero.

$$\varphi_1 + \varphi_2 + \varphi_3 = 0 \Leftrightarrow$$

$$\frac{dp}{dt} = c \cdot \left[k \cdot \omega - \frac{p - p_f}{\delta} \right]$$

This is the state equation for the valve. The pressure p is state variable of the process and is measured directly with the pressure sensor. The differential speed is considered to be a load disturbance. Measurement noise is not modeled.



• Figure 3-4. Nonlinear model of the coupling.

3.2.1 Equilibrium conditions

It may be interesting to linearize the state equation and see how the behaviour changes for different operating conditions. A linearization is valid only around an equilibrium point. For the coupling, this is when the pump flow is equal to the flow through the valve and the elasticity of the oil is balanced by the pressure over the valve. If we set the derivative in the system equation to zero, we get the equilibrium condition.

$$p^* - p_f = k \cdot \omega^* \cdot \delta^* \Leftrightarrow$$

$$\omega^* = \frac{p^* - p_f}{k \cdot \delta^*}$$

3.2.2 Linearization

Assume that we write the system on the form

$$\frac{dp}{dt} = f(p,\delta,\omega)$$

with the function

$$f(p,\delta,\omega) = c\left[k \cdot \omega - \frac{p - p_f}{\delta}\right]$$

Then the partial derivatives are

$$f_p' = -\frac{c}{\delta^*}$$

$$f_\delta' = c \cdot \frac{p^* - p_f}{(\delta^*)^2}$$

$$f_\omega' = ck$$

and the linearization around the equilibria becomes

$$\frac{dp_\Delta}{dt} = -\frac{c}{\delta^*} \cdot p_\Delta + ck \cdot \omega_\Delta + c \cdot \frac{p^* - p_f}{(\delta^*)^2} \cdot \delta_\Delta$$

This can be written with the Laplace operator on the form

$$\begin{cases} P_\Delta(s) = G_p(s) \cdot \delta_\Delta(s) + G_l(s) \cdot \omega_\Delta(s) \\ G_p(s) = \dfrac{\beta}{s+\alpha} \\ G_l(s) = \dfrac{\gamma}{s+\alpha} \end{cases}$$

where

$$\begin{cases} \alpha = \dfrac{c}{\delta^*} = \dfrac{ck \cdot \omega^*}{p^* - p_f} \\ \beta = c \cdot \dfrac{p^* - p_f}{(\delta^*)^2} = \dfrac{ck \cdot \omega^*}{\delta^*} \\ \gamma = ck \end{cases}$$

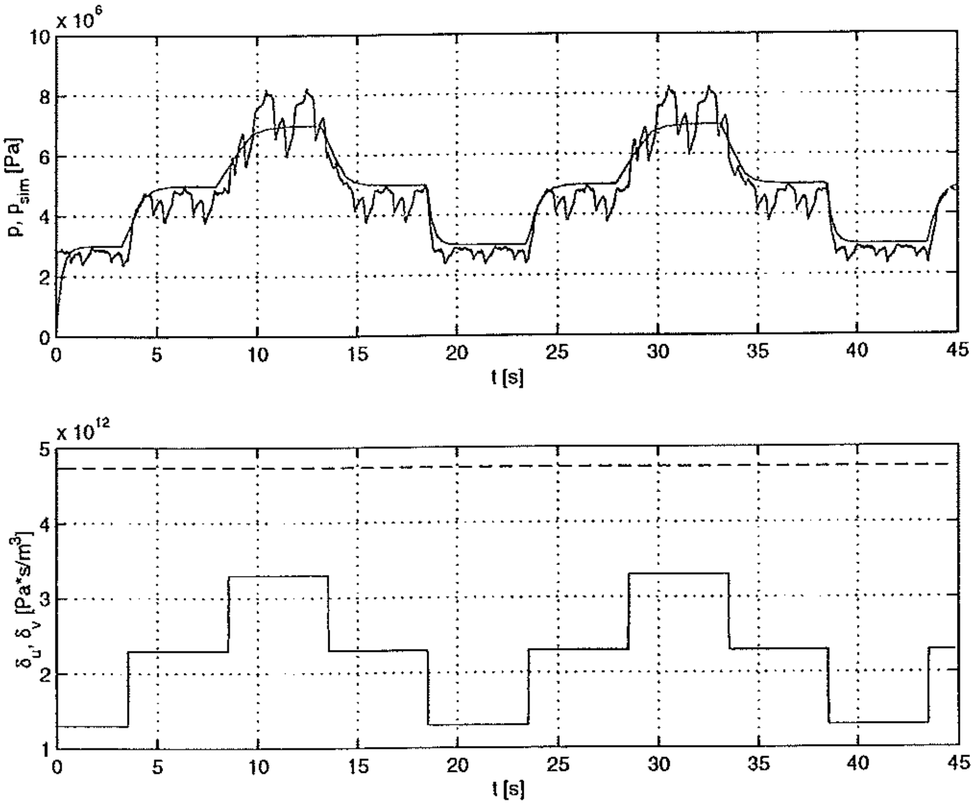
From this we may conclude that the system time constant changes with different stiffness values. Also, the gain from stiffness to pressure changes.

3.3 Measurements

We made some first simple measurements in open loop to see how the coupling behaved. The model above is a result from this. The stepper motor responds as it should to the input signal, of course. The valve needle reaches its correct position in some milliseconds.

The pressure on the other hand, responds sometimes slowly. The time constant lies between some tenths of a second and a second, so it is varying a lot. This corresponds to the time constant α . The system is well damped. A first order model should be good enough as an approximation. The disturbances seen in the measurement depends on the differential speed pump. They arise when the pistons change direction.

The k parameter in the model has been calculated from the piston areas and strokes. The c parameter is difficult to calculate, as it depends on elasticity in both oil, housing and clutch discs. We have simulated the model with the same input as the real coupling and compared the outputs graphically. The c parameter was chosen so the outputs should be as equal as possible. Attempts were also made to identify the c parameter with help of least squares identification with both the linearization of the process and with the nonlinear process and nonlinear regressors.

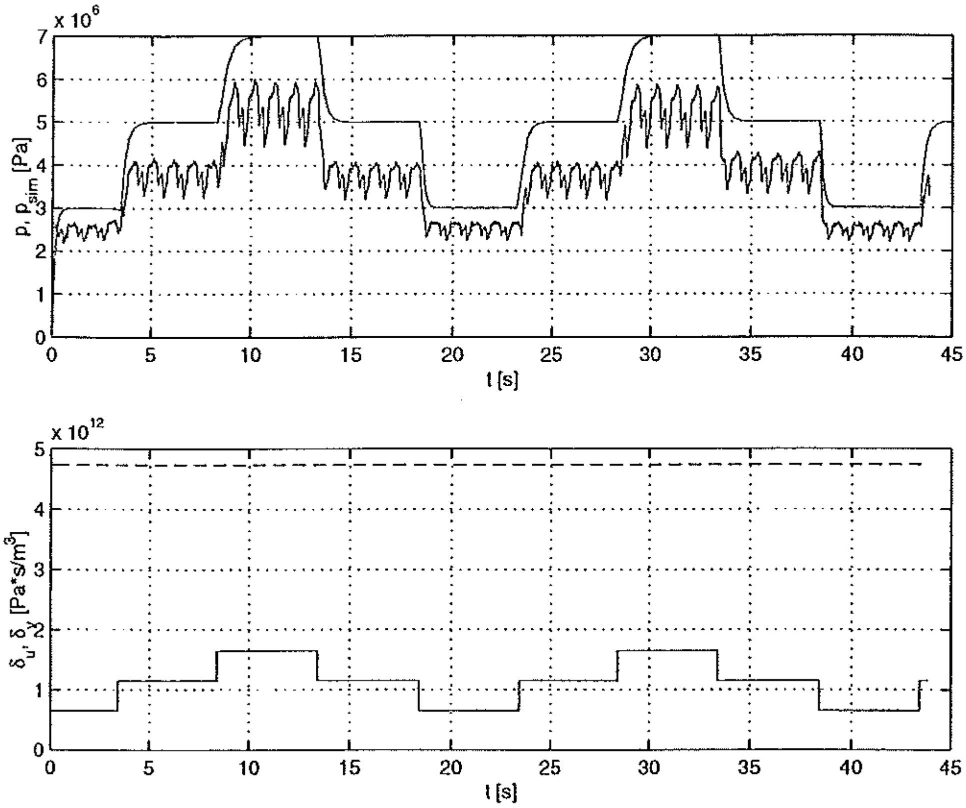


• Figure 3-5. *Measurement and corresponding simulation of the coupling at 10 rpm.*

3.4 Validation

The validation we have done is not extensive. However, it shows some qualities of the model.

The variations in the time constant seem to agree, but the outputs of the model and the real coupling settle at different values. This is probably caused by imperfections of the linearization table and the throttle valve. The table and valve are known to be sensitive. So this is probably not a problem of the model, but rather the throttle valve. These variations can be dealt with, simply by making the controller more robust. Disturbances are not part of the model. In the real coupling they are an important part of the behaviour.



• Figure 3-6. *Measurment and corresponding simulation of the coupling at 20 rpm.*

3.5 Conclusion

The model is a rough approximation of the coupling. It is important to know the other problems related to control the pressure. When the coupling is mounted in the car, the flow variations of the differential pump are lower but still present. The uncertainties in the throttle valve need more care. The controller must be robust enough to handle this. The pressure signal and the differential speed signals are noisy. The pressure signal is oversampled and can easily be filtered, but the differential speed signal is harder to cope with. It is also delayed during the transfer on the CAN-bus.

The model we have still give us indications of how fast the control may be. As seen before, the time constant varies in different operation modes, so the controller should adapt to the process.

4 Vehicle dynamics

The purpose of this section is to describe the demands on an all wheel drive. All wheel drive cars have the obvious advantage of better power distribution from engine torque into motion. Also, better handling of the vehicle is possible. The desire for fast reaction and accurate torque transfer for the coupling is the key issue in our work.

Resulting from this section will be three different cases. The first one is fast deactivation of the coupling when the brakes activate. This is important since the anti locking system needs total control of the wheels. The second case considered appears when at least one tire looses its grip with the ground. The coupling must redistribute the force within tenths of seconds. Last but not least, accurate torque distribution is demanded for nice handling of the vehicle.

- **Configuration** This configuration we used is suitable since the front wheels almost always rotate faster than the rear. With positive engine torque, traction on all wheels is always possible even if turning. If a rear wheel drive is modified to an all wheel drive, traction is mostly not possible in sharp turns.
- **Braking** The anti locking system of newer cars need full control of the wheels. As soon as the brakes activate the coupling must be deactivated. This demands fast control and is already solved in the existing coupling.
- **Slippery roads** High engine torque and slippery roads will make the traction difficult. Normally, the tires of the vehicle should have grip to the ground. If a tire looses its grip, the friction decreases and the wheel will spin. The most effective way to solve this problem is to distribute the engine torque to all wheels of the vehicle, thereby utilizing the friction to the ground at a maximum. In some cases, though, some of the wheels will spin. Then we need to detect this and take some action. The coupling should act fast on this situation and distribute the torque to the other wheels of the vehicle. One additional problem of this is the varying differential speeds when turning. If the coupling is to stiff when making a turn, the car will be very difficult to handle.
- **Torque distribution** Normally, when the wheels do not spin and the brakes are inactivated, it is possible to influence the handling of the car. Just like the handling is influenced by the weight distribution on the wheels, so is it influenced by the torque distribution. There is a demand for accurate torque control as the difference for 10 % change in the power distribution ratio between front and rear wheels may be large. More torque to the rear wheels will make the car power oversteered. Here are some possibilities for different designs.
- **Special problems** A coupling can transfer torque between two shafts only when they rotate with different speeds. Torque can only be transferred from the faster rotating shaft to the slower one. If the rear tires of the vehicle are more worn or for some other reason rotates faster than the front ones, it may be impossible to transfer torque from the engine to the rear wheels. This should be considered as abnormal. An additional constraint on the coupling is the varying differential speeds depending on turning the vehicle. A spare wheel may also be a problem.

5 Control strategy

5.1 Choice of control signal and regulator

The process is a rather complex one with fast process variations. During the early stages of this master thesis, the ambitions were to make an identification of the throttle valve and totally replace the stiffness control and instead, use the needle position as control signal. The temperature dependencies were to be handled by process estimation and the nonlinearities of the throttle valve were to be handled by a nonlinear regulator. But as the function became more and more clear, one realized that the rapid changes in the process and pulsation of the pressure (see Figure 3-5 and Figure 3-6) would make it difficult to estimate the process. Furthermore, the complexity of the valve would have to be analyzed for our needs. Including this in the master thesis would result in a lot of work, too much work for the time offered.
Therefore, stiffness is still to be used as control signal. By assuming that the linearization-tables (see chapter 3.3) are quite accurate, the problem of the temperature dependencies and valve nonlinearities is thereby solved. This also makes simulation in MATLAB possible. The process dependencies of ω and δ (see chapter 3.2) are however still present.

5.2 PI-controller

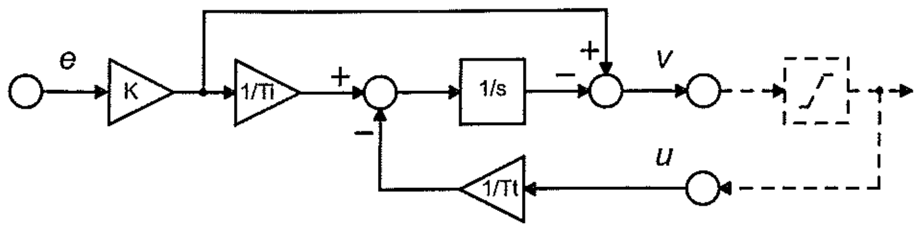
The problem is now of the complexity so that it can be controlled by a linear controller. Due to the absence of liquidity index and the problem of debugging the real time programming in the microprocessor, the choice is made to use a discretizised PI-controller. A discretizised PI-controller has the advantage that it can easily be developed step by step, adding and verifying one feature at a time. Features referred to are antiwindup, bumpless transfer and gainscheduling. Gainscheduling is used to handle the remaining nonlinearities.

The choice of using a PI- and not a PID-controller is because of the great pulsation of the pressure (see Figure 3-5 and Figure 3-6). Derivative action in the controller would make greater compensation for the pulsation and is also sensitive to noise. The choice not to try to minimize the pulsations was made at an early stage in the project. This due to the limited lifetime of the valve. Therefore, the derivative action is excluded.

The controller can be described by the equation.

$$v(t) = K\left[e(t) + \frac{1}{T_i}\int_0^t e(s)ds\right] + \frac{1}{T_t}\int_0^t u(s) - v(s)ds$$

or Figure 5-1.



• Figure 5-1. *The time continous version of the PI-controller.*

where

- e is the pressure error
- v the desired stiffness by the controller

u the true stiffness

This is a standard PI-controller with a tracking signal added to eliminate integral windup (see chapter 5.2.1).

Because of the fast sampling (h=10ms) where $h/T_i \leq 0.16$ for all resulting T_i (see Figure 5-4), no great consideration has to be made for the discretization. Euler discretization can therefore be used resulting in the following equation:

$$v(k) = K\left[e(k) + \frac{h}{T_i}\sum_{i=0}^{k} e(i)\right] + \frac{h}{T_t}\sum_{i=0}^{k-1} u(i) - v(i)$$

where h is the sampling interval.

5.2.1 Antiwindup and bumpless transfer

To implement antiwindup and bumpless transfer, it is desired that the stiffness u is to include both theoretical limitations done in the software and the physical limitations of the stepper motor. This is done by including all limitations in the saturation and limitation block from Figure 5-1.

The physical limitations of the stepper motor can be included since the true quantitized stiffness-value can be read from the stepper motor control.
However, by doing this, the integration won't act as intended since u and v can differ even if there is no saturation or limitation made other than the always present quantization.

The quantization makes it difficult to distinguish a quantization from a saturation and the integrator behavior will depend on how much the control signal is quantitized and choice of tracking time constant, T_t.

If instead the desired continuous stiffness demanded from the control-software to the stepper motor control is chosen for feedback, this problem is solved (see Figure 5-1). By doing this, the dynamics of the stepper motor is not taken into consideration. However, the dynamics of the stepper motor are so rapid, compared to the process of the coupling, that neglecting it is a fair assumption.

Therefore the desired, continuous stiffness demanded from the control-software is chosen for feedback of u.

T_t is set fixed (T_t =30ms), so that it is faster than the integration time constant, T_i.

Since gainscheduling is implemented, bumpless transfer for changes in the regulator parameter K is also needed. This is not handled by the tracking signal but by updating the integrator as

$$I_{PartNew} = (K_{old} - K_{new})e + I_{PartOld}$$

whenever K is changed.

5.2.2 Design and gainscheduling

A first attempt to trim the controller is made in rig that resulted in satisfactory performance for a certain pressure and differential rotational speed.

As the pressure setpoint and differential rotational speed are then changed and another trim is made, the resulting K and T_i are changed from the first trim. This indicates that the variations in process dynamics has to be taken into consideration.

Given the linearized process transfer function $G_p(s)$

$$G_p(s) = \frac{\beta}{s+\alpha}$$

where

$$\alpha = \frac{c}{\delta^*} = \frac{ck\omega^*}{p^* - p_f}$$

and

$$\beta = c\frac{p^* - p_f}{(\delta^*)^2} = c\frac{k\omega^*}{\delta^*}$$

The PI regulator has the simplified transfer function $G_c(s)$

$$G_c(s) = K\frac{1+sT_i}{sT_i}$$

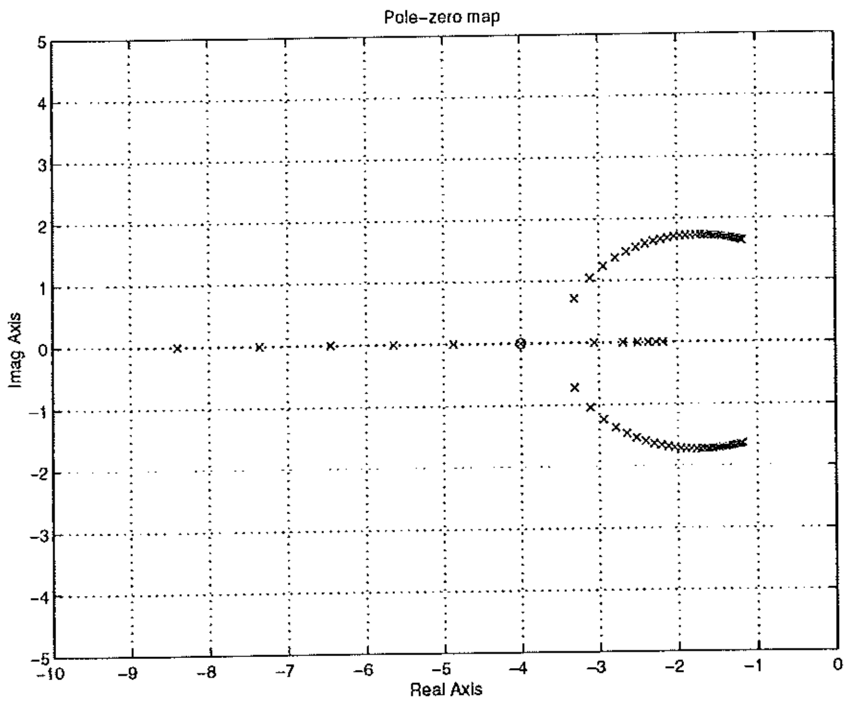
where the tracking function is excluded since the tracking signal is equal to zero during control without saturation.

The closed loop transfer function is then given by $G_{cl}(s)$,

$$G_{cl}(s) = \frac{G_c(s)G_p(s)}{1+G_c(s)G_p(s)} = \frac{\frac{\beta}{s+\alpha}K\frac{1+sT_i}{sT_i}}{1+\frac{\beta}{s+\alpha}K\frac{1+sT_i}{sT_i}} = \frac{sKT_i\beta + K\beta}{s^2T_i + sT_i\alpha + sKT_i\beta + K\beta} =$$

$$\frac{sKT_i\beta + K\beta}{s^2T_i + sT_i(\alpha + K\beta) + K\beta} = \frac{sK\beta + K\beta/T_i}{s^2 + s(\alpha + K\beta) + K\beta/T_i}$$

Using this transfer function the zero-pole plot for different δ using the controller parameters trimmed by hand is illustrated in Figure 5-2.



• Figure 5-2. *Pole-zero plot for different δ with constant K and T_i .*

Introducing the desired characteristic polynomial for the closed loop transfer function as
$s^2 + 2\zeta\omega_{cl}s + \omega_{cl}^2$

where

ω_{cl} is the natural frequency of the closed loop system and

ζ is the damping factor,

and requireing unit static gain, identification results in design parameters as

$$\begin{cases} \alpha + K\beta = 2\zeta\omega_{cl} \\ K\beta / T_i = \omega_{cl}^2 \end{cases} \Rightarrow \begin{cases} K = \dfrac{2\zeta\omega_{cl} - \alpha}{\beta} \\ T_i = \dfrac{2\zeta\omega_{cl} - \alpha}{\omega_{cl}^2} \end{cases}$$

Since the process dynamics have great variations it is difficult to set a constant ω_{cl} .The design parameter *d* is therefor introduced as how many times faster the closed loop controlled process should be compared to the open process. This results in:

$$\omega_{cl} = d\alpha$$

which, applied on the equations from chapter 3.2.2, result in

$$K = \frac{(2\zeta d - 1)\alpha}{\beta} = \frac{(2\zeta d - 1)}{k}\frac{1}{\omega^*}$$

and

$$T_i = \frac{2\zeta d - 1}{d^2 \alpha} = \frac{2\zeta d - 1}{d^2 c}\delta^* .$$

In the tests, four different designs have been used where

- ζ =0.7, which gives the system nice damping and
- d =[1.50 1.75 2.00 2.50].

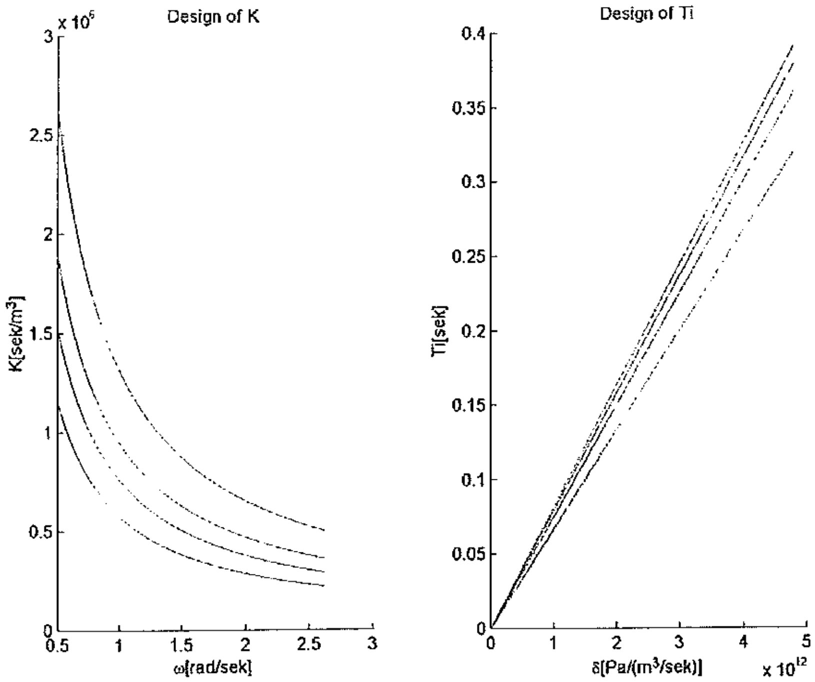
The resulting K and T_i are illustrated in

- For the gainscheduling
- K assumes 5 different values for different intervals in ω . In each interval, the ω with the highest process gain is used for the design.

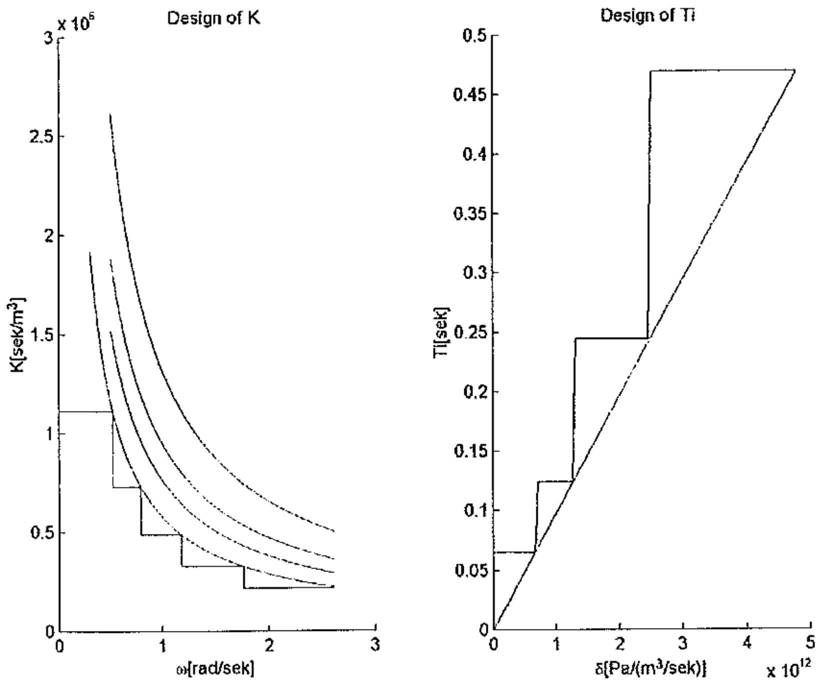
T_i assumes 4 different values for different intervals in δ .In each interval, the δ with the highest time constant is used for the design.

The range of the schedules are chosen so that the different boundary layers will not interact with each other. On the other hand, the schedules should adapt the controller to the different conditions of the process, so the schedule boundaries should not be to far apart. Gainscheduling for ζ =0.7 and d=1.50 is illustrated in Figure 5-4.

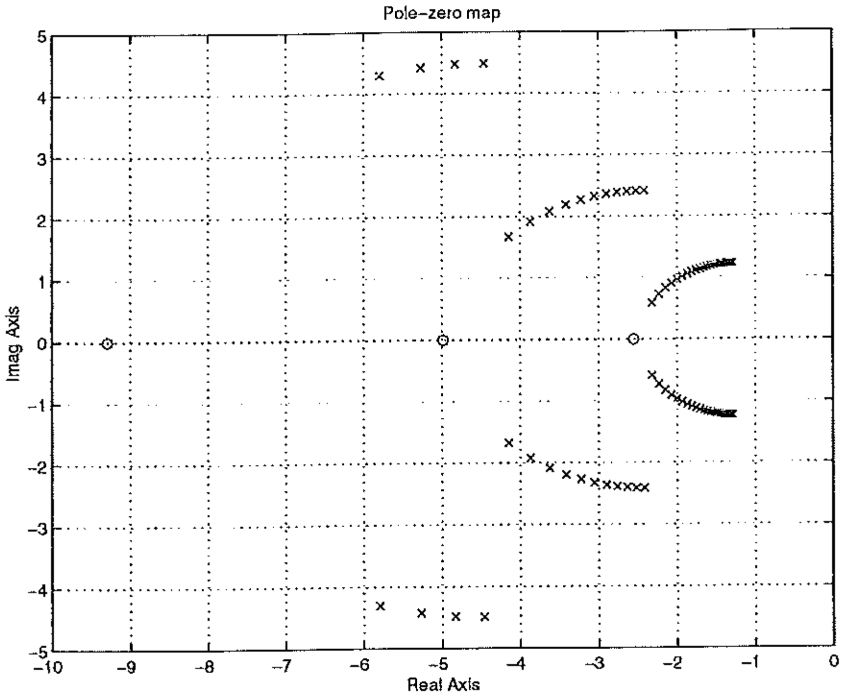
The zero-pole plot for different δ using gain scheduling for ζ =0.7 and d=1.50 is illustrated in Figure 5-5. Also, the sensitivity function for different stiffness schedules is shown in Figure 5-6, it is valid for all differential speed schedules. The sensitivity should be as low as possible, preferably less than two. The plot shows that the sensitivity never goes above one so it is satisfactory. We should not have to worry about stability. Probably, the actual behaviour is the opposite. As the schedules are designed for worst case, we will have well damped behaviour in most conditions.



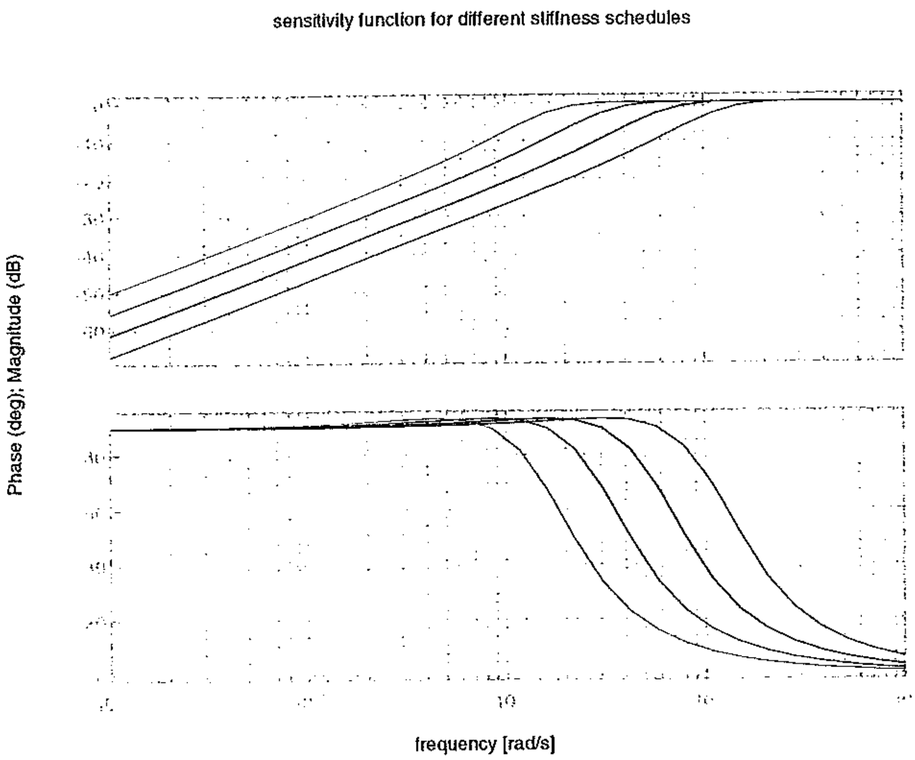
• Figure 5-3. *The resulting controller parameters.*



• Figure 5-4. *The controller parameter for d=1.50 after gainscheduling.*



• Figure 5-5. *Pole-zero plot for different* δ *with gainscheduling.*



• Figure 5-6. *Sensitivity function for different stiffness schedules.*

6 Testing

During the various stages of the project we've used different equipment for testing, trimming and evaluation.

The first step is to get the correct pressure value into the application software. The control algorithm is then developed step by step and tested in a testbench. Last but not least, the control is tested in the real application, e.g. in a car, and comparisons to today's control can be made for different conditions.

6.1 Reading the pressure

The pressure sensor we've chosen to use is a sensor normally used in air-condition systems for cars. It is capable of handling the oil pressure in the HLSC (0-10 MPa) and has a 0-5 V output.

- In order to read the pressure signal in the software, two approaches come to mind:
  Using the CAN bus for sending the pressure value to the processor.

Using an A/D converter already implemented in the processor.

Since communication over the CAN bus result in great time delays (up to 4 samples) this solution would make the control problem even more difficult. Added to this, the pressure value has to be sent on to the CAN bus. This requires some kind of computer communication, an area which both of the master thesis participants have limited knowledge of.

The second solution, using the A/D converter, involves no time delays and seems like a much more appealing solution. Therefore, the solution with A/D converter is chosen.

6.1.1 A/D converter

The processor is equipped with two 10 bit A/D-converters with a sample interval of 2ms. The A/D-converters are, in today's version, used for one brake light signal and one handbrake-signal. The brake light signal is a backup signal for the CAN signal and since handbrake signal on the CAN-bus is not implemented on all cars it is available as an analog value.

If the A/D converter is to be used for evaluating pressure feedback, the handbrake signal is the one that is least essential and the one to replace.
The A/D-converter in itself needs 0-5 V input but since it is used for a handbrake signal (12 V), it is equipped with a 5:1 divider on the input (see appendix C). This means that we need a 0-25 V signal in order to use all the 10 bits of the converter. To do this we design a non-inverting amplifier that amplifies the signal 5 times (see figure in appendix C)

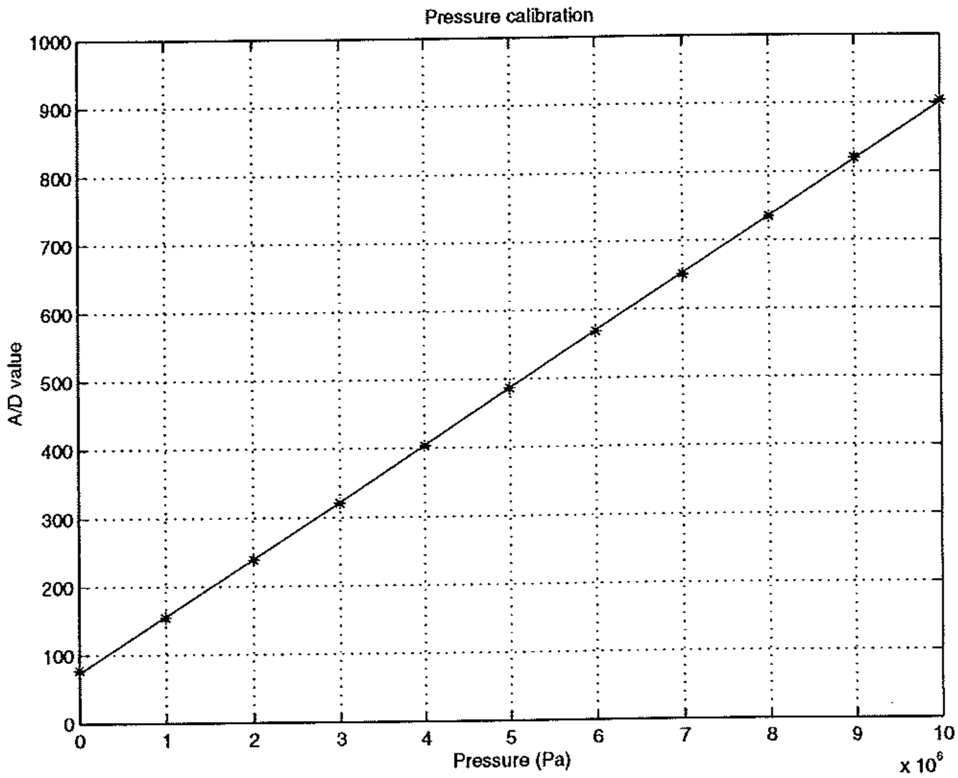
6.1.2 Calibration

In order to transform the A/D value to a pressure value it is calibrated and linearized. This is done in a separate procedure in the program that returns the true pressure value. The signal is filtered by a digital low pass filter with the corresponding time continuos transfer function H(s).

The filter has little influence on the process since its time constant is much faster than the one of the coupling. It can therefore be neglected.

During calibration we use a pressure calibration pump used for this special purpose. A pressure sensor can be calibrated by mounting it in the pump and pumping up the desired pressure manually. The correct pressure is then read on a display and the voltage value can be measured.

In our case, we connect the circuit in the way it is to be used during control (amplifier included) and mount the pressure sensor in the calibration pump. The value from the A/D converter is then read in the application software and transmitted over the CAN bus for presentation (for result see Figure 6-1). The result is an almost linear function that is linearized and then inverted in the software.



• Figure 6-1. *The pressure calibration*

6.2 CANalyzer

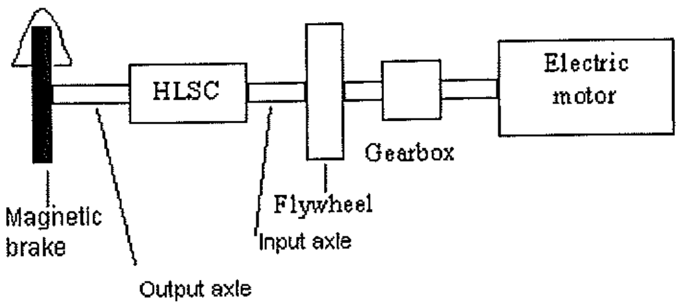
A very useful tool when you're dealing with CAN bus signals is the software CANalyzer, developed by the company Vector Informatik GmbH. On a PC, equipped with a special CAN card, CANalyzer can be used to read and transmit CAN messages. The signals are presented in a data window and a graphic window, logging to file for later analysis is also possible.

In CANalyzer, writing to the CAN bus is also possible. In order to do this a program has to be written, specifying what values and which addresses are to be sent. This is written in a special programming language called CAPL(CAnalyzer Programming Language). At

Haldex Traction, such a program has been made especially for the HLSC. By using this program, the CAN signals, needed by the HLSC, can be transmitted without having the HLSC mounted in a car. This feature is very useful when using the HLSC in a testbench. We have used CANalyzer during almost all the testing stages of this project. The possibility to send values to the HLSC came in very handy while trimming the controller.

6.3 Testbench

At Haldex Traction there are a few different types of testbenches of different sizes and performances. For our purposes we've chosen a rather small and easy to handle testbench. The testbench is called "Mini-Hägglundaren".
The testbench(see Figure 6-2) is equipped with an electric high power motor connected to the input shaft of the HLSC and a flywheel. A magnetic brake is connected to the output shaft.



• Figure 6-2 The "Mini-Hägglundaren"

By using a PC connected to the testbench, the HLSC can be tested in different situations, useful for both development and production tests. The magnetic brake (on/off) and the rotational speed of the input shaft can be controlled by the PC. The PC is also equipped with CAN communication for controlling the coupling during testing.
The testbench is very easy to use once you have got the coupling in place and all the cables connected. For our purpose we've used the testbench only to control the disc brake and the rotational speed of the input shaft, the CAN communication is not used. Instead, the HLSC is connected to a laptop with CANalyzer for trimming and evaluation.

6.3.1 Problems

- For testing, a testbench has the advantage that an exact situation can be repeated a multiple of times. This makes it ideal for comparing different control designs for the same situation. However, there are still some situations that are hard to realize in the "Mini-Hägglundaren".
- The disc brake is not infinitely variable, either it is locked or it is running free without any torque transferred. This makes the disc brake difficult to use for testing step responses in differential speed, ω. It can only be used for testing step responses from $\omega=0$. Since there is zero process gain at $\omega=0$, the control signal will be saturated.
  The other way to test step responses in ω is using the electric motor. But since the motor and flywheel have some inertia and the rotational speed is set by a slider on the screen, steps are almost impossible to realize.

- Since ω is essential for the control design, it is important that its value, calculated in the software and used for gain scheduling, is the correct one. The value of ω in the software is based on CAN signals which are not available on a testbench. This means that ω has to be set by CANalyzer and it is sometimes difficult to set the correct value in CANalyzer and the testbench at the same time.

6.3.2 Procedure

- Because of the problem with getting different ω, mentioned above, and since there is no good way to log the true value of ω, the tests have been done for fixed ω. Two types of tests have been done:
- Tests on the open loop system, analyzing step responses for stiffness directly on the coupling.

Tests on the closed loop system where step responses for the pressure setpoint have been analyzed.

The stiffness values in the open loop measurement has been chosen so they should give the same pressure levels as pressure controller.

6.4 Car

One of the goals during this project was to be able to test the result in a car. These tests are difficult to use scientifically but are very useful in another aspect. Since today's control uses a lot of signals from the CAN bus, which are almost impossible to simulate in a testbench, this is the only way to make fair comparisons to today's coupling.

In order to get good performance of the car, we transferred 45% of the torque to the rear axle. Therefore, the pressure set point is calculated so that 45% of the driving torque is transferred to the rear axle (T_r =0.45*T_e) where the driving torque is obtained from the CAN bus.

The pressure control is only to be active during normal road conditions. Should the brakes be activated or the slip of the front wheels become to great, the control algorithm of today's system is to take over.

Since many different control systems are present, bumpless transfer is desired. Bumpless transfer to pressure control is handled by the tracking signal in the PI-controller. But bumpless transfer to the other algorithms needs great changes in these algorithms, changes that could alter the function of the algorithm radically. Since the control program of today's system is rather complex we chose not to implement bumpless transfer in these cases.

The car used in the tests is a Volvo V70 AWD (All-Wheel-Drive). A 4WD car equipped with a viscous coupling in the original, but the people at Haldex have replaced the viscous coupling with the HLSC for evaluation purposes. The car is therefore not a production car and the final software trimmings have not been done yet. This means that the performance might not be fully representative, but for comparison to our control, also a prototype version, it will do.

6.4.1 Problems

- The HLSC mounted in the car is not of the same specification as the one tested in testbench. The differential pump has half the capacity of the one used in the lab. This is because of different differential speeds of different cars. This means that the identification done in testbench might differ from the coupling in the car. Since the coupling in the car is the only one of its kind, identification is difficult. Based on the known differences, the control design is changed and the unknown factors are not taken into consideration.

While driving, reproducing the exact same driving conditions is difficult, both on account of the traffic and the sometimes long time period between the tests.

- During the hectic time for testing in car, other more practical problems arose:
- While testing in testbench, a 220 V power converter can be used to supply the amplifier for the A/D-converter with 30 V supply voltage. The car is equipped with a 220 V outlet. However, this outlet doesn't deliver enough current for the power converter used in the testbench. Some other power supply has to be used. In a pile of used electronic equipment a 12/24 V power converter was found. This could, just barely, be "tuned up" to deliver the 30 V needed.
- Even though the coupling is mounted to the rear axle, it proved not to be thoroughly grounded to the chassis. This was solved by simply placing a banana-cable between the coupling an the chassis.

6.4.2 Procedure

In order to make a fair comparison we have used a few basic test procedures which should not differ too much from test to test. The tests prodecures are:

1) Full throttle acceleration from 30km/h on the second gear on a dirty road.
2) Full throttle acceleration from 30km/h on the second gear on a paved road.
3) Full throttle acceleration from 50km/h on the third gear on a paved road.
4) Driving in a tight circle at constant speed.

The first two tests are for comparing the same acceleration for different road conditions. Since today's version does not take any consideration to this, the result should differ pending on the road condition, while the result of the pressure control should not differ as much.

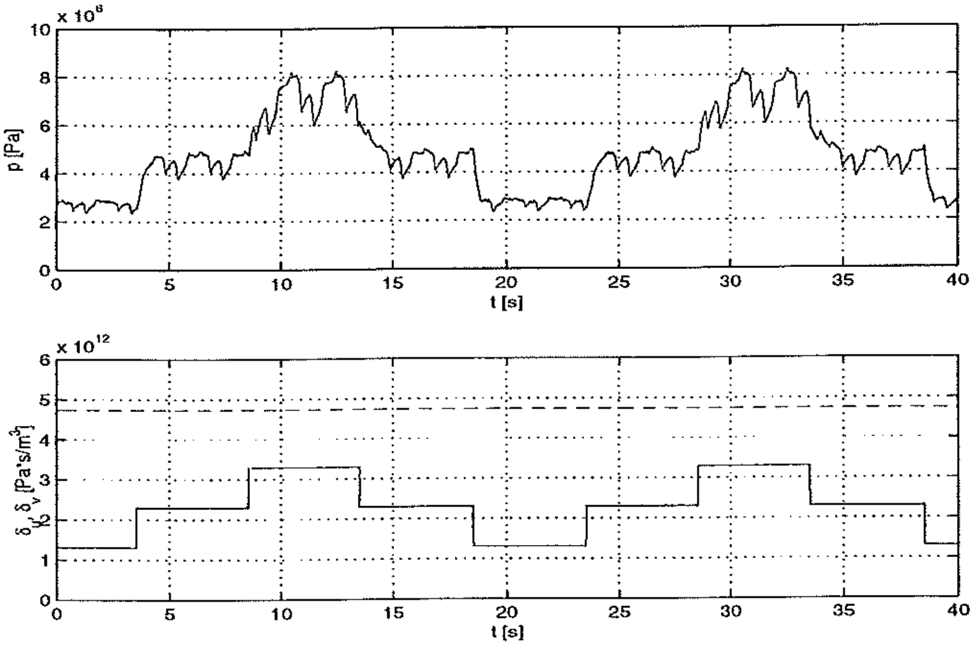
The third test is just to compare highway driving at higher speed.

The fourth test is for testing the control at constant differential rotational speed, ω, and constant driving torque, T_e.

The first two test procedures proved to be the ones most important for the evaluation, therefore they are the only ones presented in this report.

7 Performance

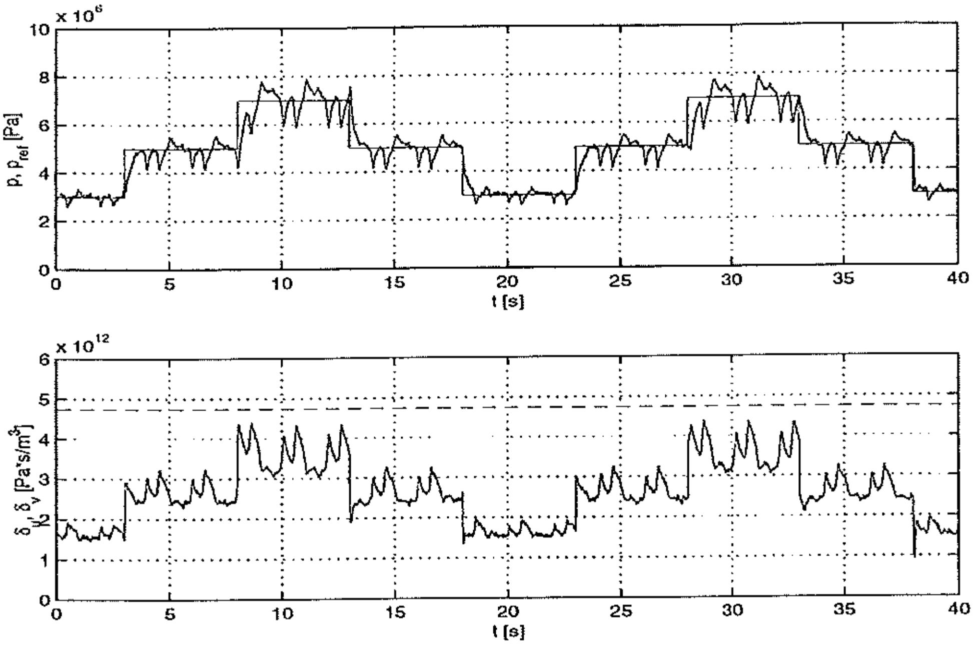
7.1 Testbench



• Figure 7-1. Open loop measurement at 10 rpm. The pressure levels should be 30, 50 and 70 bar ($1bar = 10^5\,Pa$).

The performance of the original system has been discussed before. The major improvement of the pressure control is the elimination of stationary error. The errors originating from the linearization table are regulated by the controller. The uncertainties of the table are handled by using a robust controller. It is also possible to speed up the system with the pressure controller. In the car, we experienced that the linearization table was saturated so the speed improvement of the controller was limited.

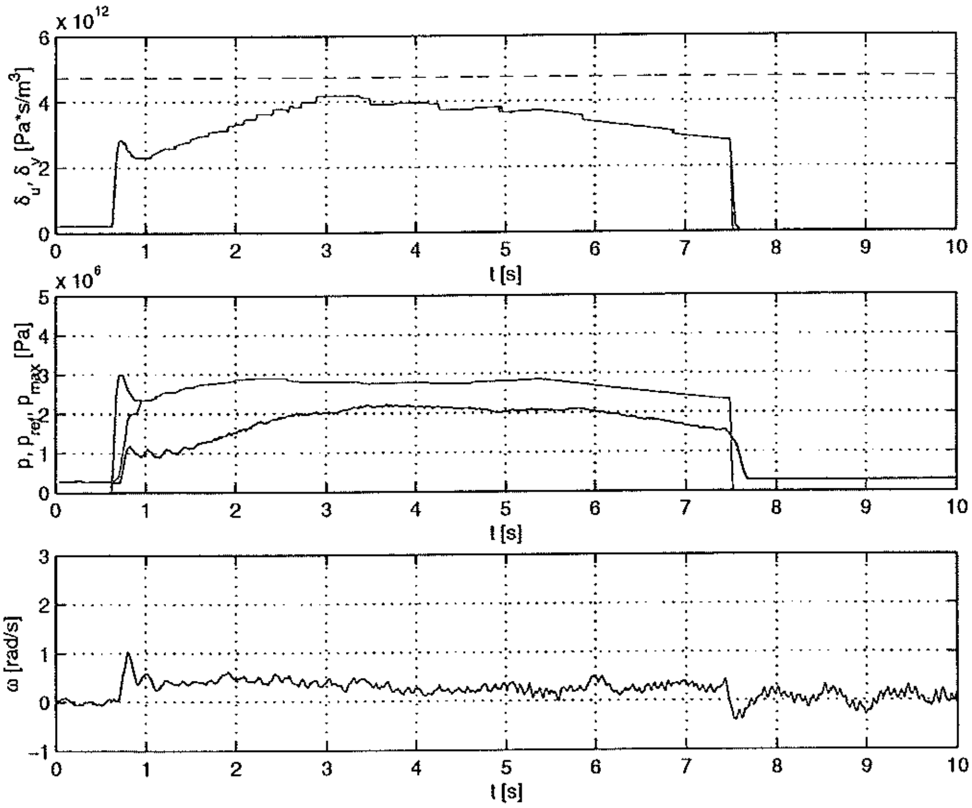
The problem of variations in the flow from the differential speed pump is hard to cope with. The variations were a bit lower at low differential speeds, but at higher speeds they are a problem. They actually set a limit for how fast the system may be controlled. If the controller is made faster, the stiffness control signal will vary too much, almost without improvement in speed. The speed is also limited by saturation of the control signal. We get more movements on the needle with control, so the stepper motor wears out faster. Some nonlinear effects are present in the results from the testbench. Short pressure peaks may be seen when a negative step is applied. Also, the pressure seems to respond slower on positive steps.



• Figure 7-2. *Pressure controller at 10 rpm.*

In car, the existing control has stationary error, especially on tarmac. It takes a couple of seconds for the pressure to rise, which is far too much. Variations in pressure, which were clearly visible in the test bench, are very small in car. It is because of the elasticity in the driveline. See Appendix B for more measurements.

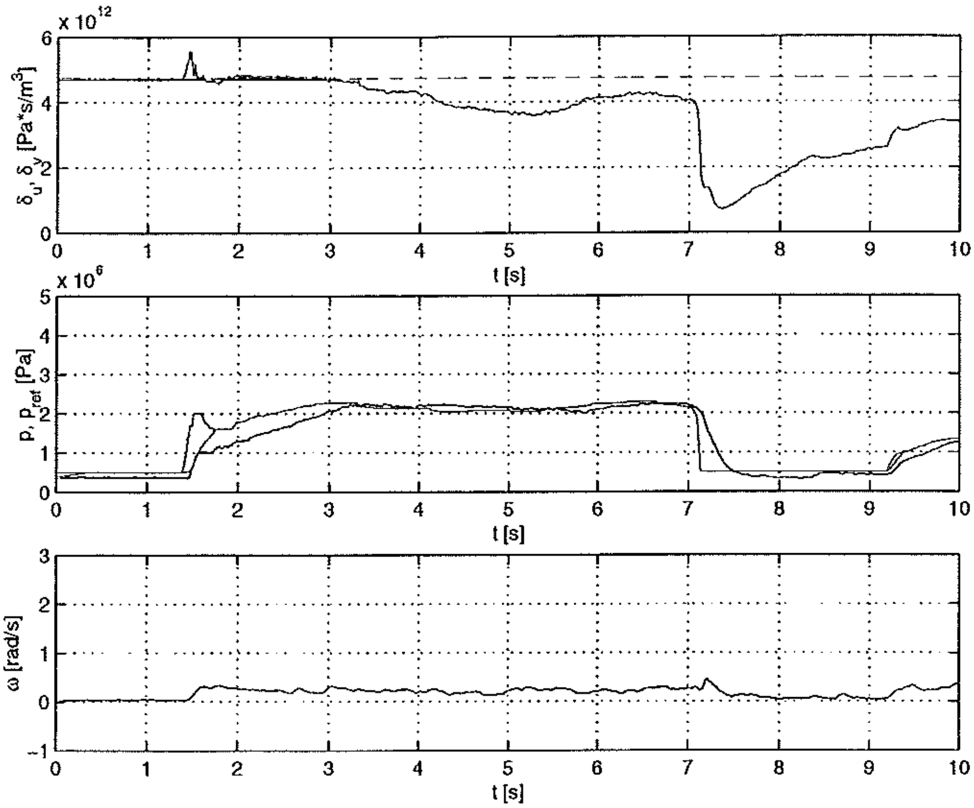
7.2 Car



• Figure 7-3. *Existing control on tarmac, second gear.*

With pressure control in the car, stationary errors are eliminated. The control makes the coupling a bit faster. The minimum pressure setpoint is a bit higher than ground pressure. This makes our controller saturate at maximum stiffness when the car is at rest. This should make the response faster when starting on slippery surfaces. The minimum pressure setpoint resulted in some problems when driving in circles. A very low pressure should have to be set, so there should actually be no limit. It seems like the pressure control needs less adjustments to work nicely as it regulates errors in the pressure by itself.

The fourth test with driving in a circle was not a problem either for the existing program or the pressure control. Also see Appendix C for more plots.



• Figure 7-4. *Pressure control on tarmac, second gear.*

8 Conclusions

We have gained insight into most parts of the product. Mechanics, hydraulics, electronics, the control valve, the pressure sensor and the control program are parts of the pressure control loop. This makes the performance of them essential for us. We have studied these parts the most but also the driveline and the bus information in the car.

Simple tests in open loop were made at first to see the dominating behaviour of the coupling. Then, after making a physically based model, more extensive experiments was planned and performed. To fit the model to the data, the unknown elasticity parameter was given a value. Then, the model was compared to data sets from various measurements to see if it was reliable.

With the model at hand, some different controllers were tried. We used some simulation but primarily actual measurements in testbench. At first, we tried a three-point controller, then a simple P-controller, a PI-controller and the final PI-controller with gainscheduling. Also, different gainschedules were tested. The last consideration was on how fast the closed loop system could be made. When the controller was good enough, we adjusted the microcontroller program for use in a car and tested it. The existing control program has been tried parallel to ours.

As we have worked with several different problems, the individual parts of the identification and synthesis process are not completely investigated. We will give some hints of how to develop the control even further. But first, the results obtained so far should be commented.

Our controller eliminates the stationary error of the pressure. The difference between existing control and pressure control is greatest when driving on tarmac. By using a pressure controller, the program can also be adjusted to the car in less time, because it is less sensitive to external parameters. Calibration of the valve is not as important any more. A disadvantage is the increased cost due to purchase and mounting of the pressure sensor.

The control developed by us gives faster response than before, provided no wheel is slipping. Theoretically, the speed increase of the process should be up to 75 %, but as the linearization table is saturated frequently, the actual increase is less. By extending the linearization table to the physical limits of the valve, this problem should be less frequent. This would be the first thing to improve.

The control speed can not exceed the limits imposed by the capacity of the differential pump, the elasticity of the hydraulic system and the efficiency of the clutch. Therefore, these limits - the control authority - should be calculated. The valve and the controller should be as near these limits as possible.

The most important parameter of the coupling, the elasticity, should be identified more carefully. The parameter is used for the synthesis of the pressure controller so an error would influence the resulting system. The object of our thesis is to make a pressure controller. In a car, there are more signals than the pressure to use in the control. We added the existing wheel slip control parallel to the pressure controller. The slip control could be better integrated. Maybe the needle valve should be replaced by another valve type. If the needle valve is kept, the linearization tables should be redesigned, perhaps by fitting measurement data to a model with least squares or a similar method.

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A. Variables

9.1.1 Physically related

p	[Pa]	pressure
p_f	[Pa]	feed pressure
ω	[rad/s]	differential speed
δ	[Pa·s/m^3]	stiffness (definition)
k	[m^3/rad]	capacity of the differential pump
c	[Pa/m^3]	elasticity of the hydraulic system
φ_1	[m^3/s]	flow created by differential pump
φ_2	[m^3/s]	flow through throttle valve
φ_3	[m^3/s]	flow due to elasticity of hydraulic system

9.1.2 Process

G_p	[-]	transfer function, open loop
α	[1/s]	natural frequency, open loop
β	[m^3/s^2]	gain, open loop
γ	[Pa/rad]	load disturbance gain, open loop

9.1.3 Design

G_c	[-]	transfer function, controller
G_{cl}	[-]	transfer function, closed loop
ω_{cl}	[rad/s]	frequency constant of closed loop system
ζ	[-]	damping of closed loop system
d	[-]	ratio between closed loop and open loop frequency
K	[s/ m^3]	controller gain
T_i	[s]	controller integration time constant
T_t	[s]	controller traction time constant
v	[Pa·s/m^3]	desired control signal
u	[Pa·s/m^3]	actual control signal
e	[Pa]	regulation error
h	[s]	sampling time

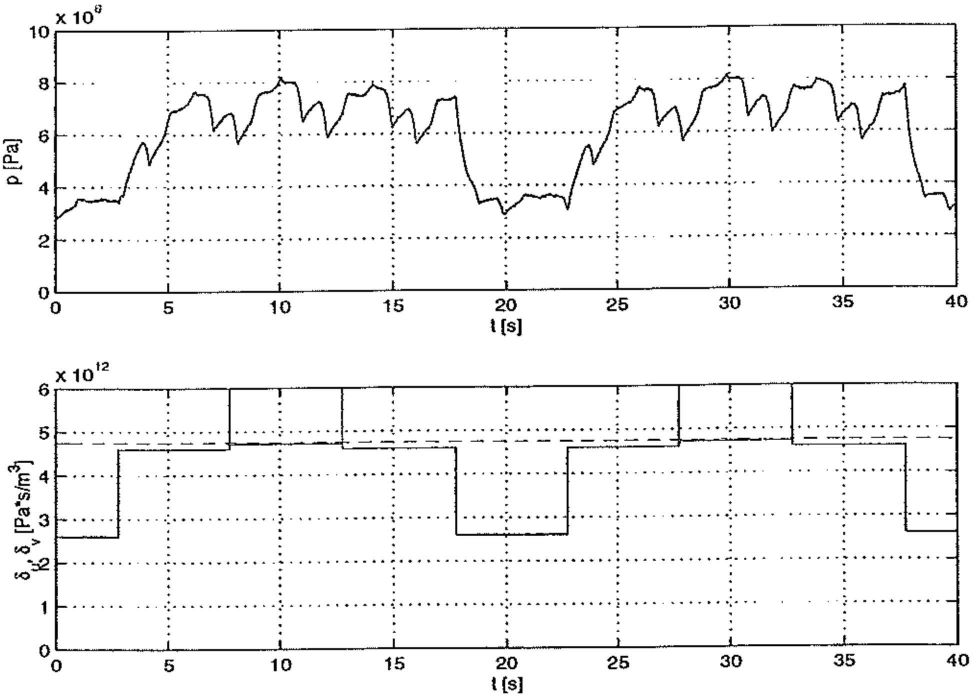
9.1.4 Other

Equilibrium condition is denoted by placing a superscript on the variable, like this, x^*.

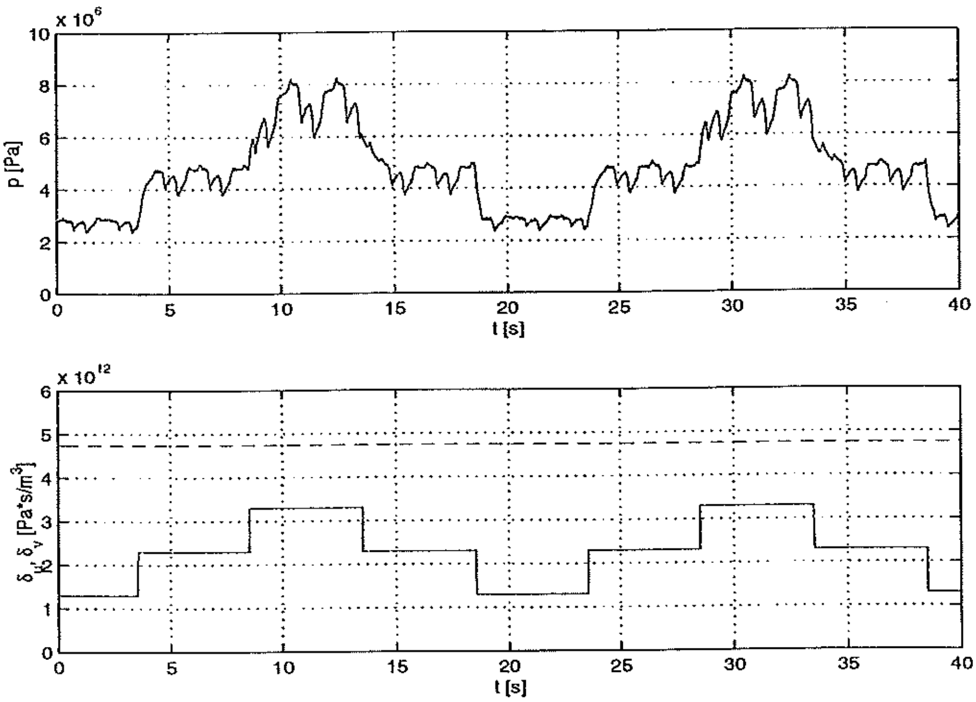
B. Testbench measurements

These measurements were made in testbench, the so called 'mini-hägglund'. Constant differential speed was applied and different stiffness values were applied. Four measurements are presented here. The first subplot of each measurement show the pressure response, the lower subplot show the wanted (blue) and actual (green) stiffness values. The dashed line (red) show the saturation limit for the stiffness. The stiffness function is chosen so that the pressure should reach the values 30, 50 and 70 bar. This is not the case for the first plot, as the stiffness is saturated. The pressure seems to be to low at high differential speeds and to high at low differential speeds.

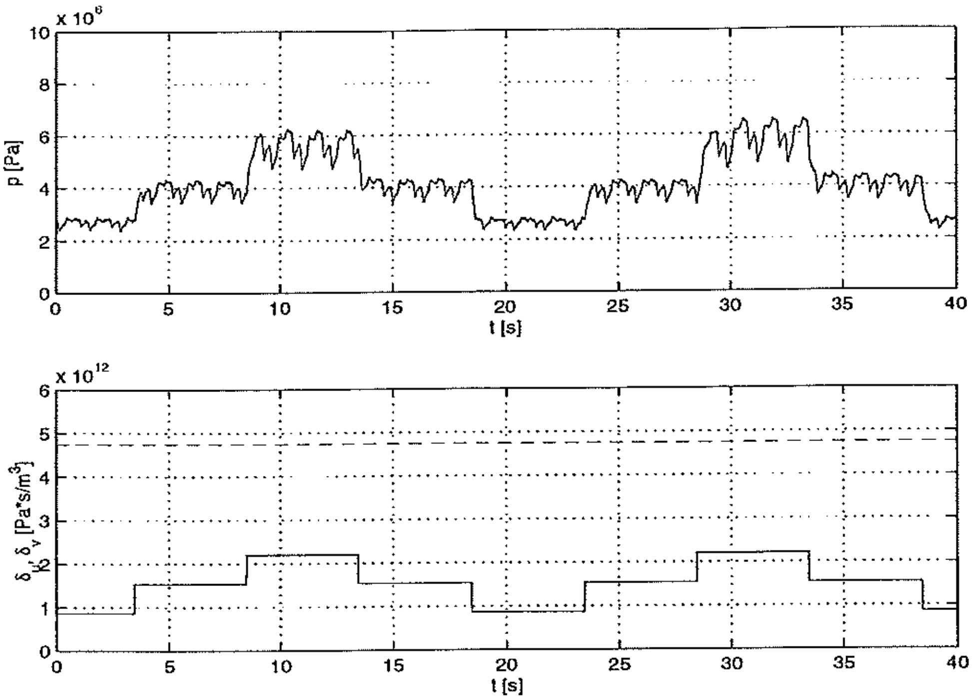
Four measurements were made for a pressure controller with d=1.75. They were done at the same differential speeds as the open loop measurements. The first subplot of each measurement shows wanted and actual pressure and the second subplot shows wanted and actual stiffness. The stiffness is saturated in the first measurement.



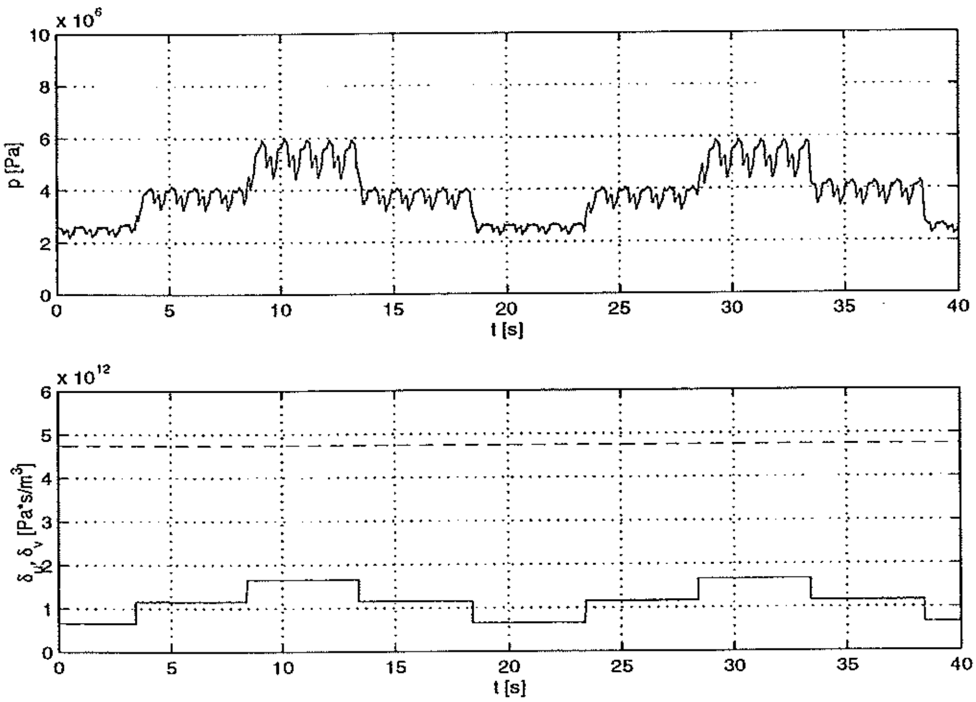
• Figure B-1. *Open loop measurement at 5 rpm.*



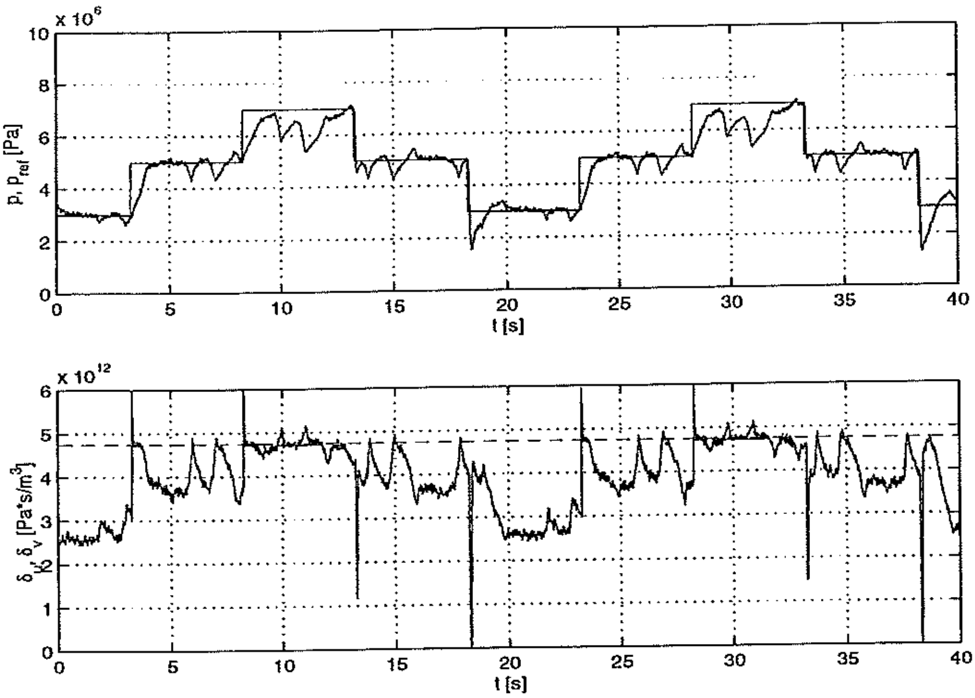
• Figure B-2. *Open loop measurement at 10 rpm.*



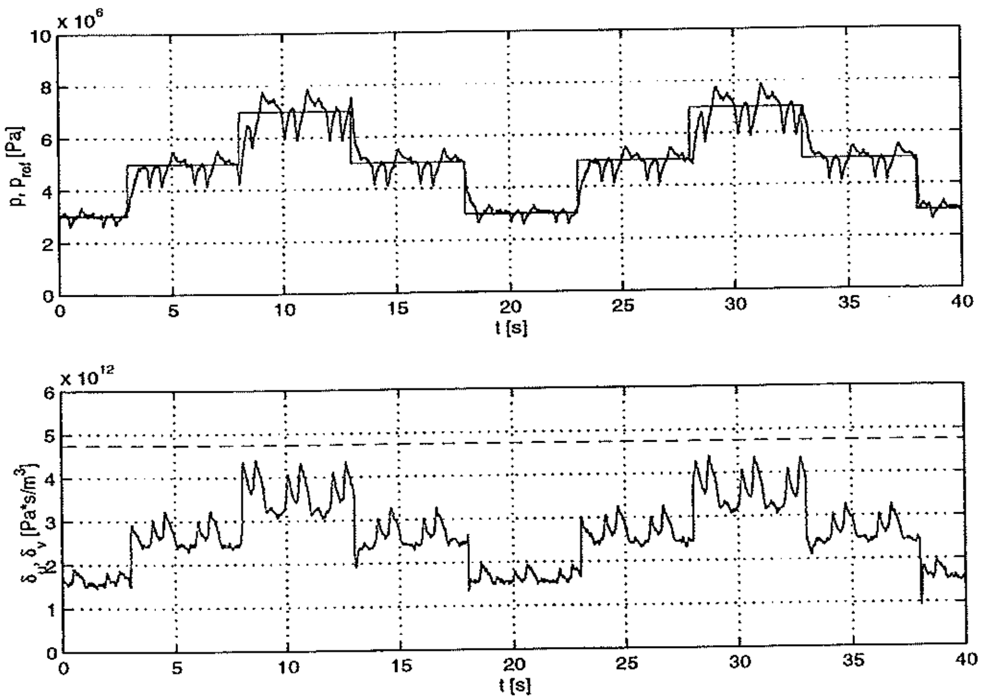
• Figure B-3. *Open loop measurement at 15 rpm.*



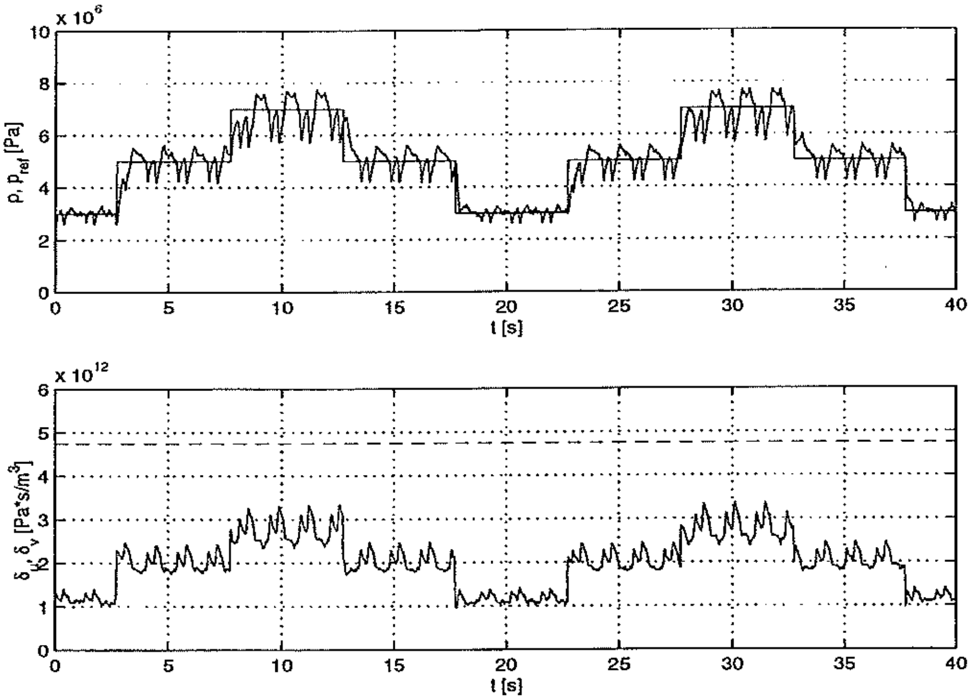
• Figure B-4. *Open loop measurement at 20 rpm.*



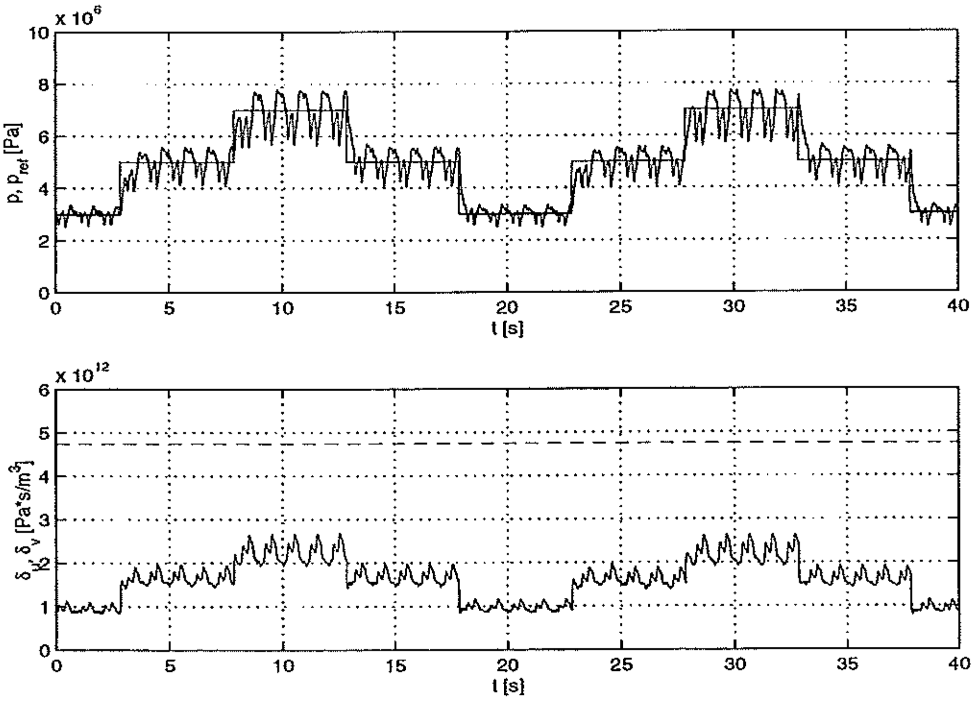
• Figure B-5. Pressure controller at 5 rpm.



• Figure B-6. Pressure controller at 10 rpm.



• Figure B-7. *Pressure controller at 15 rpm.*

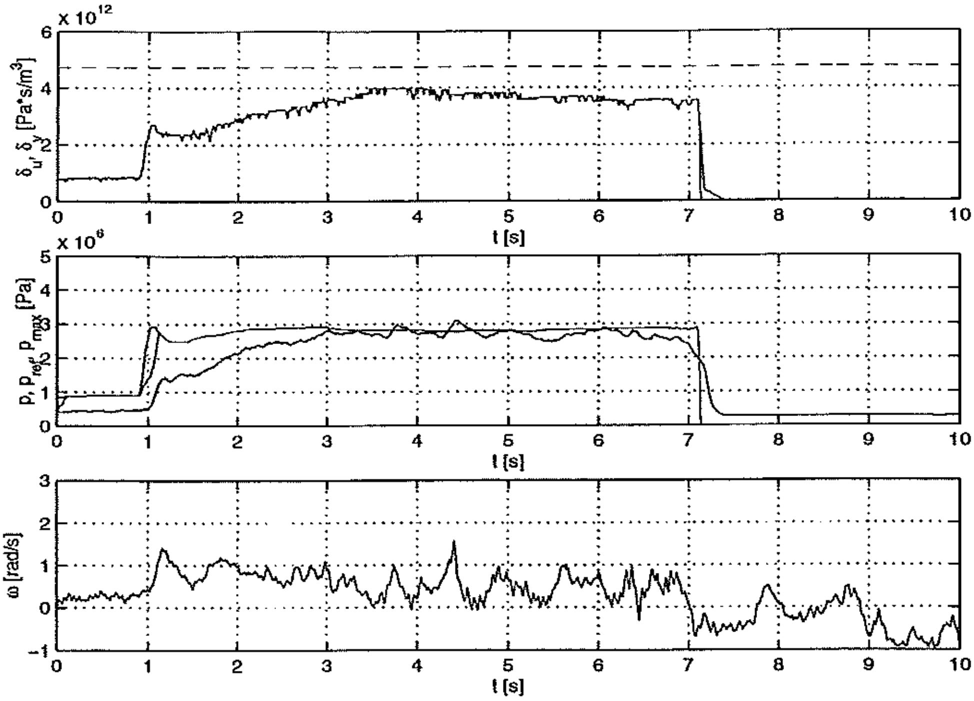


• Figure B-8. *Pressure controller at 20 rpm.*

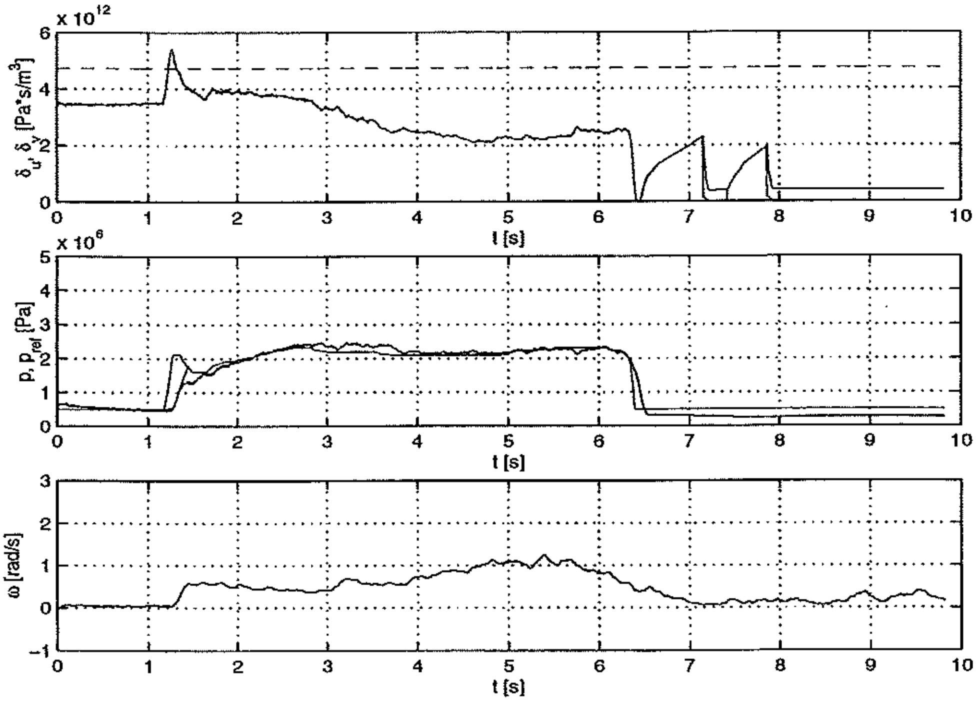
C. Car measurements

Some measurements in car were made. We have tried to compare the existing program with the pressure control, so the same test is performed for both systems, the conditions are not completely equal though. The first subplot shows the stiffness, wanted (green) and actual (blue). The second subplot shows actual (blue), wanted (green) and maximum (red) pressure. The maximum pressure level is there to illustrate the control authority of the coupling. It shows how fast the pressure would grow if the valve was completely closed. It is calculated from the integral of the differential speed.

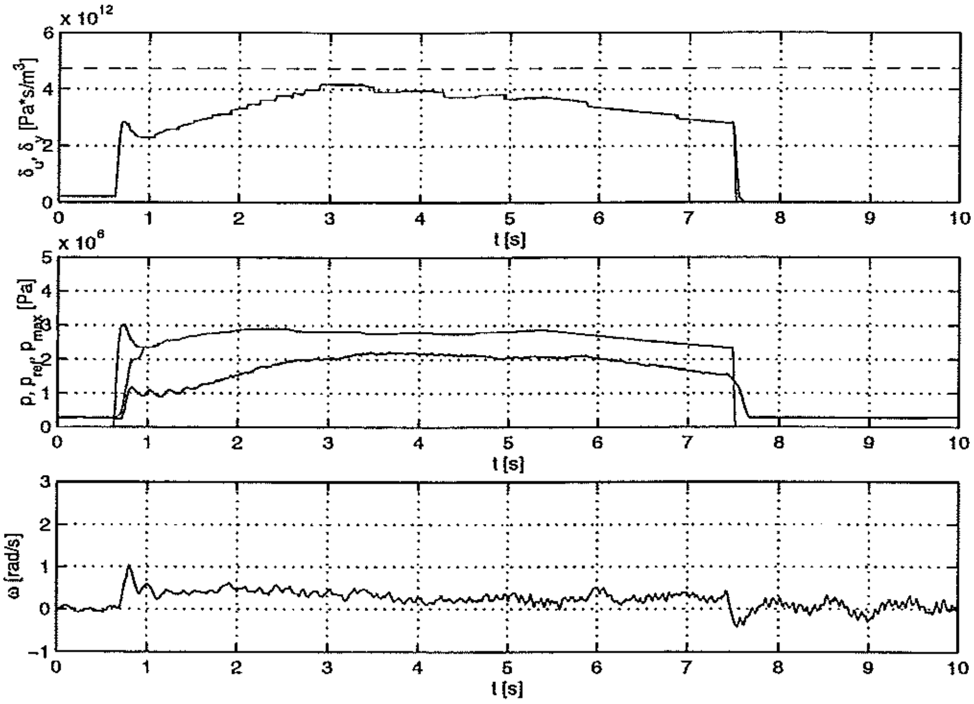
The first set of measurements show a full-throttle acceleration on gravel in second gear from 30 km/h. Second set of measurements show a full-throttle acceleration on tarmac in second gear from 30 km/h.



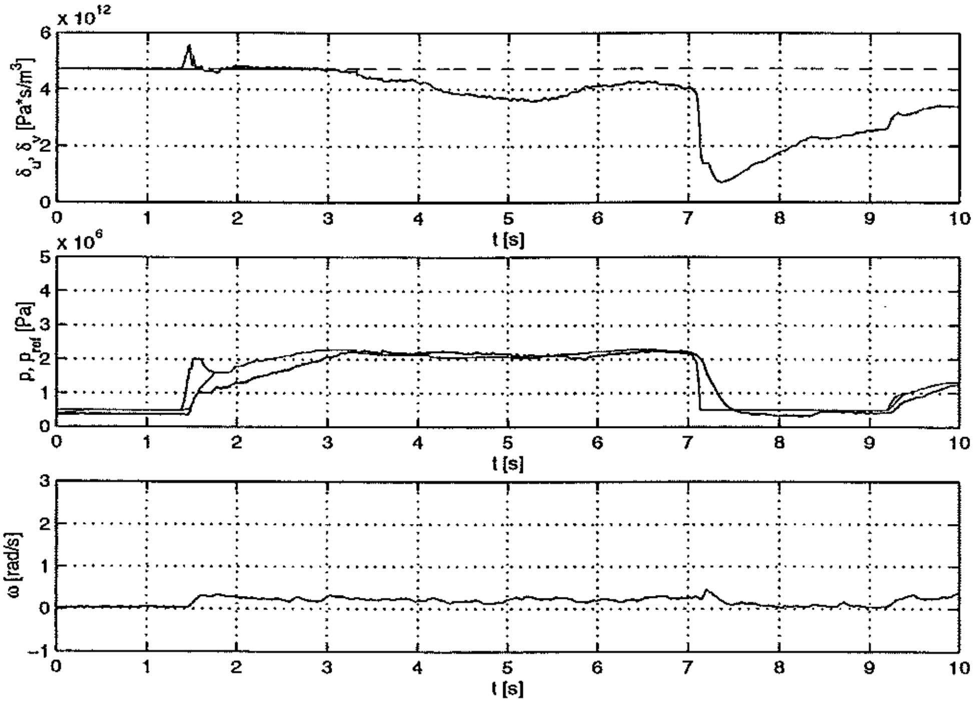
• Figure C-1. *Existing control on gravel, second gear.*



• Figure C-2. *Pressure control on gravel, second gear.*

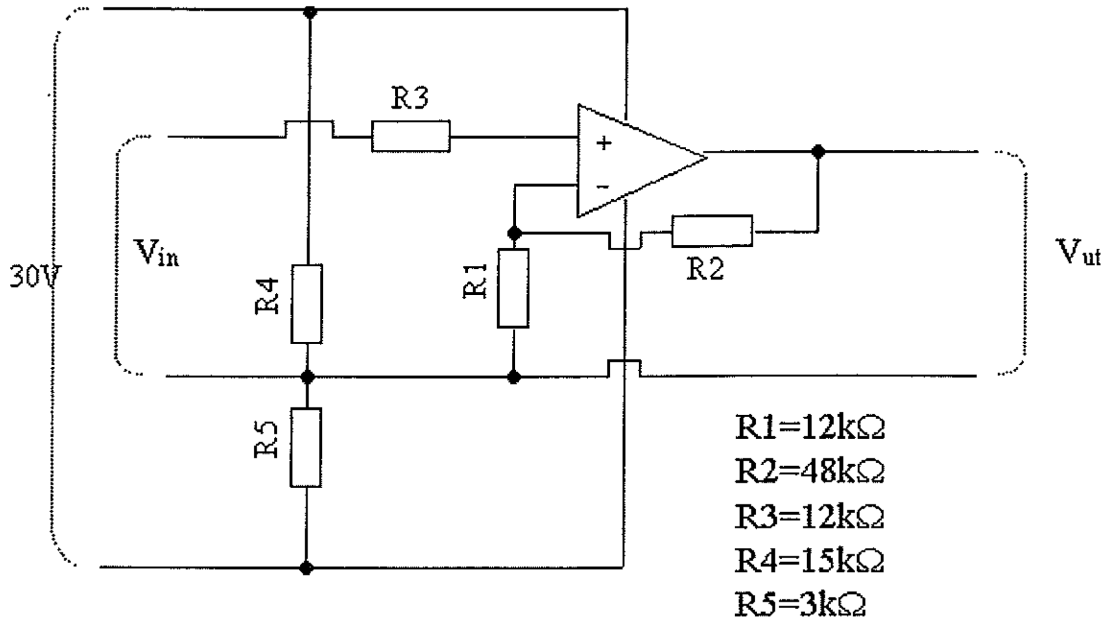


• Figure C-3. *Existing control on tarmac, second gear.*

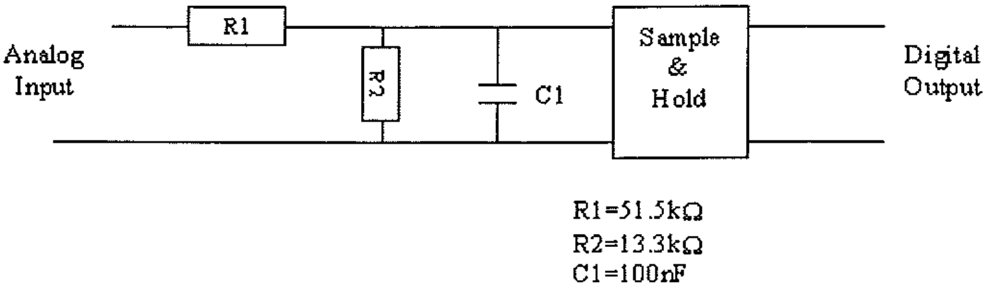


• Figure C-4. Pressure control on tarmac, second gear.

D. Electronics



• Figure D-1. *Amplifier circuit for the pressure signal.*



• Figure D-2. *The ADC of the microprocessor of the coupling.*