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Traineeship report:

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Preface

This report is written as a traineeship report within the study of mechanical engineering at the Technical University Eindhoven and is part of the master degree Automotive Engineering Science.

First of all I want to mention that due to confidentiality some of the data are modified or masked. This report gives an overview of my traineeship at Toyota Motorsport GmbH, with respect to the signed contract.

During the past 4 months I have worked within Toyota Motorsport GmbH in Cologne. During this time I have worked in the Hydraulic & Integration group, which is part of the Car Design and Development department (CDD). In this group the hydraulic system of a formula-1 car is developed. Further more the group is responsible for many parts (on chassis side) of the car for which there is no specialised group.

Like in all sports the aim of all the F-1 teams (the top of the Motorsport) is to be the best. The aim of this goal is achieved by working with the latest production methods, materials and knowledge. All the car components and production methods are being improved constantly. Due to this, many of the techniques in production cars are initially developed for F-1 cars. Working in a very competitive segment at the top of automobile industry, has made my traineeship at TMG very special and overall educational.

I would like to take this opportunity to thank all people I have been working with and all the people that made my traineeship at TMG possible. I want to thank particularly Klemens Pollmeier, my supervisor within TMG. Klemens is head of the Hydraulics & Integration group, but found time to guide me during my traineeship. Further more I want to thank John Litjens, John Steeghs and Maurice Billekens, with whom I travelled from Venlo to Cologne each day and a special thanks to John Litjens without whom I would not have this traineeship.
Summary

The aim of the project is to develop several software and data evaluation tools for the condition monitoring of (mainly hydraulic) components and sub-systems of the car. Data available in WinTax (the on the car data recording system), needs to be evaluated and compared to known and simulated characteristics. The necessary simulations can be performed with Matlab/Simulink. The goal is be able to see and predict malfunctions as well as degradation parts of the hydraulic system. Some statistical analysis of previously recorded data may also be required in order to find ‘patterns’ of acceptable and borderline performance. Overall mainly the following signals need detailed analysis: speed/velocities, displacements, pressures, valve currents and temperatures.

Auszug

Das Ziel des Projektes ist es, einige Software- und Datenauswertungswerkzeuge für die Analyse (hauptsächlich hydraulischen) Komponenten und Baugruppen des Autos zu entwickeln.

Samenvatting

Het doel van de opdracht is software en data analysis tools te ontwikkelen voor de conditie monitoring van (voornamelijk hydraulische) componenten en sub-systemen van de auto.

Data beschikbaar in WinTax (het data record systeem in de auto), moeten worden geverifieerd en vergeleken met test-bench en simulatie data. De nodige simulaties kunnen met Matlab/Simulink gedaan worden. Het uiteindelijke doel van deze software is het beter inzicht verschaffen en zelfs het voorspellen van storingen en voortekens van fouten in het hydraulisch systeem.
Statistische analyse van grote datasets kan nodig zijn om patronen van acceptable grenswaarden te ondervinden. Vooral de volgende signalen worden gedetailleerd geanalyseerd: Snelheid, verplaatsingen, drukken, klepstromingen en temperaturen.
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1 Introduction

The Toyota Motorsport Company has a long history in motor sport. It all began with Ove Andersson who started his fruitful association with Toyota back in 1972 as a Rally driver. Since that time, he has gone on to develop his partnership with the Toyota Corporation including the founding of Andersson Motorsport in 1975. Results include four World Rally Championships for drivers and three Manufactures World Titles. Along the way, the giant Japanese corporation made a full acquisition of TMG in 1993.

In 1997, Toyota in Japan entrusted the team with the design, build and operation of its new entry in the Le Mans 24 hour race. It was the first time Toyota had given full responsibility for such a two-year program to any non-Japanese based team.

A handpick team of engineers, technicians and designers from the highest level of the sport and industry came together to create the Toyota GT-One. In 1998 the Toyota team led the likes of: Mercedes, Porsche, Nissan and BMW for much of the race. In 1999 Toyota gained a phenomenal second position after setting Pole position and the fastest lap of the circuit.

In 1999 the decision was made to enter the Formula-1. Inauguration of new facility at Cologne started in 2000; the facility is increased from 18,000 sqm to over 30,000 sqm in 2001. The team made its first Grand Prix debut at Albert Park, Melbourne in 2002.

In the Car Design & Development department all parts of the chassis are designed, the hydraulic & integration group is part of this department The group takes care of hydraulic components, e.g. the power steering, parts of the engine, the clutch, fuel flap, etc.

‘Integration’ stands for everything for which there is no specialized group, e.g. the engine and transmission cooling, the fuel cell, tyre pressure, fuel pump, etc.

The work of the engineers in this group is all-round; they guide the project from A to Z.

The aim of the traineeship is to develop several data analysis tools. One goal of these tools is be able to see and predict malfunctions as well as degradation of parts of the hydraulic system. Another goal is to analyse and compare test results from various (mainly hydraulic) car-components.
# Hydraulic in Automotive

In automotive industry a lot of hydraulic components are used. Not only in the car, also with the production of car parts. The plates of the car are made by huge rolls, were the clamp forces are generated through hydraulic cylinders, with speeds over 2000 m/min; tolerances of micrometer are achieved at these processes.

[Figure 1](#): Hydraulic in automotive

Also on board of the car are a lot of hydraulic components. Figure 1 shows a quick overview of the on board hydraulic of a production car.

## 2.1 Servo valves

Initially servo valves were developed for the airplane industry. In the start of the 50er years large progress is made, when the torque motor and not much later the mechanical feedback were introduced.

The within TMG used servo-valves are the one with hydraulic feedback. Figure 2 shows a schematically representation of such a valve.

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1 figure from: Stetighydraulik, Verslag Moderne Idustrie
Figure 2: Servo valve with mechanical feedback

Figure 3 shows the working principle from this valve, when the electrical input is changed: An electrical current in the torque motor coils creates magnetic forces on the ends of the armature. The armature and flapper assembly rotates about the flexure sleeve support. The flapper closes off one nozzle and diverts the flow to the end of the spool. This causes that the spool moves and opens $P_S$ to one control port and opens R to the other control port.

Figure 3: Valve Responding to change in electrical input

When the electrical input stops changing and the valve has to follow a fixed condition: The spool pushes against the ball end of the feedback spring, creating a restoring torque on the armature/flapper. When the feedback torque becomes equal to the torque from the magnetic forces, the armature/flapper moves back to the centred position. The spool stops at a position where the feedback spring torque equals the torque due to the input current. Therefore the spool position is proportional to the input current and with a constant supply pressure; the flow to the load is proportional to the spool position.

Figure 4: Valve condition following change

Figure 5 shows the usual application of flow control servo-valves. The valve is supplied with a constant pressure, $P_s \approx 210$ bar. The valve controls the flow to and from the piston end chambers in response to an electrical signal. The pistons drive the load.

1 figure from: Stetighydraulik, Verslag Moderne Idustrie
2 figure from: Nozzle-Flapper Flow Control Servo valves, MOOG
The position feedback is an error signal; it is the difference between the command signal and the position feedback signal. The error signal is amplified and the amplified signal drives the servo-valve. The error will be reduced to near zero.

2.2 Hydraulic accumulators

Hydraulic accumulators store fluid under pressure and can serve a number of functions within a hydraulic system including:

- leakage compensation
- emergency power source
- pulsation and shock absorption
- noise elimination
- load counter-balance

The required gas pre-charge pressure is dependent on the application of the hydraulic accumulator and a function of the minimum hydraulic system pressure. The accumulators are sized according to their effective or actual gas volume when all oil is discharged. The volume of oil available from a given size accumulator depends on the volume of gas available to push it out, known as the working volume. Working volume varies as the pressure and temperature of the gas varies. Gas temperature and therefore volume, are influenced by the speed at which the gas compresses and decompresses as the accumulator is charged and discharged. For this reason the rate of charge needs to be considered when sizing an accumulator.

In most of the accumulators the gas behaviour is isothermal when the accumulators are charged or discharged slowly, while the gas behaviour is assumed to be adiabatic by fast rates of charge. The accumulators for an F-1 car have the same functions, but are designed specifically for their application.

2.3 The hydraulic system of the Toyota F-1 car

The hydraulic system in the Toyota F-1 car is a closed system. The total amount of hydraulic oil in the system is quite small (about a glass of Kölsch) and the working pressure is above 200 bar. This pressure is supplied by a variable displacement axial piston pump. Components driven by this system are e.g. the power steering, differential as well as clutch, gear shift and throttle actuators.
3 WinTax

During test session and even during the race a formula-1 car contains a lot of sensors. During a test session, several hundred signals are measured. Some of these data are sent to the pit-box constantly, by radio. Others are collected on the ECU and downloaded during a pit-stop. A cable is plugged in and the data can be transferred. An F-1 car does also contain some data collecting boxes from the FIA (Federation Internationale de L’Automobile). In this way they can check conformity with the rules.

Within TMG these data are collected with WinTax. WinTax is data acquisition and analysis software. The program is used on the track as well as of the track. WinTax is integrated in the entire car; the software is even integrated in the hardware, like the ECU. WinTax makes it possible to monitor the collected data and is specialized for racing applications. E.g. it is possible to make a drawing of the circuit and comparing the data from different laps is also easy. Multiple signals can be displayed in one figure and it is possible to show multiple figures on the screen. Moving the cursor in one figure causes that the cursor moves in all the figures.

It is also possible to create ‘virtual channels’, these ‘virtual channels’ are functions of the measured data. The race engineers use the WinTax software on the track to monitor the data before and during the test or race. Specialized engineers create the virtual channels and screen layouts, what makes it for the race engineers possible to monitor these data quickly.

Figure 6: Screen-shot of WinTax, with this layout the behaviour of the power steering system is monitored.

Figure 6 gives a screenshot of WinTax. The figure contains multiple kinds of plots, a x-y plot, a plot of the track, a plot with a single channel and a plot with multiple channels. Also all the relevant information as name of the race track, drivers name, time, etc. is saved in a big data base with these data.
4 Data analysis

The data analysis is necessary to get a better understanding of the hydraulic system. It is also important that easy to use routines are developed with which the data can be compared. The goal is to make some data analysis tools and investigate the (gas) behaviour of the hydraulic accumulators.

4.1 Collecting the data

All the collected data from the races and tests are saved on the network; this makes it possible for the engineers within TMG to use these data. Another software tool enables the data export from WinTax to Matlab. This software converts the selected WinTax data into a ‘structure’ with all the relevant information connected to it.

For analyzing the hydraulic system it is not necessary to have the data at each time step during a race or a test. Averaging the data on a specific interval is often required. For this a routine was written which enables the filtering of a selection of data. Figure 7 shows the output of such a routine. The raw signal is filtered first, after which the data which are in range of the criteria (idling and small oil flow in this case) are selected. Finally the average of the first (large enough) set of data is determined and saved into another structure, which contains only the data of interest.

![Figure 7: Visualization of the routine for collecting the needed data.](image-url)

Because the relevant information is connected to the data (i.e. car, driver, etc. saved with the structure), it is possible to track the source of irregularities and/or problems.
4.2 Analyzing the data

The collected data are scattered points, to find any structures in these data, Matlab routines are developed. 2D or 3D relationship can be visualized with these routines. Figure 8 shows an output plot from the 2D analyse routine. The names of the input channels are directly put on the right axis. The 2d analyze routine has also the possibility to add an n\textsuperscript{th} order trend line (polynomial fit).

![Figure 8: output plot of the 2d analyze routine](image_url)

Figure 9 shows an output plot of the 3d analyze routine. This routine also puts the names of input channels directly on the right axis.

![Figure 9: output plot of the 3d analyze routine](image_url)

There is also made a routine to filter out specific data. E.g. data for one specific chassis number on a specific track can be filtered out.
4.3 Results data analyze

Figure 10 shows the hydraulic low pressure respectively during four stages, before cranking (bc), cranking (cr), idling (id) and after idling (ai), versus the temperature.

![Figure 10: The hydraulic low pressure during the selected four stages versus the temperature.](image)

The oil expansion in the hydraulic system can not cause these changes in pressure. The reason for these expansions is that the hydraulic system contains a low and high pressure accumulator. These accumulators contain a vapour, nitrogen, which expands when the temperature increases. This expansion causes the temperature dependency low pressure site of the hydraulic system.

One of the functions of the low pressure accumulator is keeping the pressure before the pump above a critical pressure, so that cavitation does not occur. Liquid flowing through an orifice will cavitate whenever its velocity causes the pressure in the throat of the orifice to drop below the vapour pressure of the flowing liquid. The effects of cavitation are unstable flow and erosion. To prevent cavitation, sufficiently high throat pressure must be maintained either by: Applying sufficiently high pack pressure or reducing the velocity of the liquid as it flows through the restrictor.

Figure 11 shows the hydraulic high pressure, for the same stages as for the hydraulic low pressure, versus the temperature. As shown in the figure the high pressure is positive dependent of the temperature before cranking and after idling, negative dependent of the temperature during cranking and independent of the temperature during idling.

The positive dependency before cranking and after-idling has the same cause as the low pressure dependency. Theoretical the low and high pressure (before and after the pump) have to be the same at these two stages, but some differences in the measurement range of the sensors cause a small difference.
The negative dependency of the high pressure during cranking can be explained by the decreasing viscosity of the oil. The upper plot of Figure 12 shows this high pressure temperature dependency during cranking again. The lower figure shows the high pressure dependency of the engine revs. The high pressure is strongly dependent on the temperature while it is almost independent of the engine revs. The reason for this independency of the engine revs is that the pump is variable and adjusts itself according to the flow demand.

Figure 12: Dependency of the high pressure from the temperature, upper figure, and the engine revolutions, lower figure.
The pump keeps the pressure in a constant range between 190 and 207 bar. The function of the high pressure accumulator is to spring by when the flow demand of the system is larger as the maximum flow of the pump. When this occurs, oil will flow out of the high pressure accumulator to satisfy the flow demand. This causes a pressure drop, but the pressure drop is smaller then when there is no accumulator.
5 Low pressure accumulator (LPA) gas pressure

The hydraulic system contains a low pressure accumulator and a high pressure accumulator. Both are filled with at an initial gas pressure. The gas pressure of the high pressure accumulator should be: 110 ±2.5 bar @ 293 K and the gas pressure of the low pressure accumulator should be: 2.5 ±0.1 bar @ 293 K.

The hydraulic system is a closed system; this means there is no oil sump. When the oil on the high pressure side reaches the gas pressure, oil flows into the high pressure accumulator, what causes a pressure drop at the low pressure side.

![Figure 13: determine the gas pressure in the system](image)

When these accumulator gas pressures are out of range, this can have some consequences: It can affect the performance of the hydraulic system; the system can even be damaged. Because of this, it is important to check these gas pressures each time the engine is started.

When the engine is started the pressure, at the high pressure side, increases and the measured signal will indicate the gas pressure of the high pressure accumulator (see Figure 13). Once the engine/pump is running the high pressure accumulator is filled. At first the pressure increases quickly until the gas pressure is reached. Afterwards the pressure rises slower due to the oil flow into the accumulator. The gas pressure can be found at the change of the slope. Sometimes the high pressure drops slightly before increasing again, in these cases the pressure after the spike indicates the gas pressure. This is shown in Figure 14:

![Figure 14: in figure a), the pressure drops slightly after which the gradient decreases, at the moment the high accumulator gas pressure is reached, in figure b) there is no pressure drop, only the gradient decreases.](image)
When the oil pressure reaches the gas pressure in the high pressure accumulator, the low pressure starts dropping. Monitoring the low gas pressure is not as easy as monitoring the high gas pressure. Because of this it has happened several times that this low gas pressure was out of range. What can be critical for the performance of the system and even can damage the system, as mentioned before.

In the next session a theoretical model is introduced to calculate this initial gas pressure, with the (already) available measurement data. The goal of this analytical model is to predict the (initial) pressure in the low pressure accumulator.

5.1 Determining the initial low pressure in the LPA

5.1.1 Theoretical model

It is theoretically possible to determine the initial pre-fill pressure in the LPA with the pressure drop at the low pressure side. This occurs when the hydraulic oil at the high pressure site reaches the gas pressure and oil flows into the HPA. In this analysis the low and high pressure accumulator are respectively called LPA and HPA.

![Figure 15: Schematic representation of the hydraulic system](image)

Figure 15 shows the schematic representation of the hydraulic system for these analysis; the irrelevant parts are substituted by a variable restriction. Estimation for the flow through this restriction is available.

Assuming that the nitrogen gas behaves like an ideal gas, it is possible to derive an analytic equation for the initial gas pressure in the low pressure accumulator.

Theoretical the gas behaviour is isothermal during slow changes and adiabatic during fast changes. In this model the general ideal gas law has also been taken into account. This has been done because the volume of the accumulators is very small and they are placed very near to the engine. This means that they are very close to a heat source, what can influence the gas behaviour during the time.
The gas behaviour can be described by using one of the state equations shown hereunder:

- The general way, with the general ideal gas law:
  \[ \frac{P \cdot V}{T} = \text{Const} \]  
  - 1

- In the case the pressure drop is assumed as an isotherm process:
  \[ P \cdot V = \text{Const} \]  
  - 2

- And for the case that the pressure drop is an adiabatic process:
  \[ P \cdot V^k = \text{Const} \]  
  - 3

The oil volume in the LP and HPAs is the total volume minus the gas volume:

\[
VL_{oil} = VL_0 - VL_{gas} 
\]  
- 4

\[
VH_{oil} = VH_0 - VH_{gas} 
\]  
- 5

With:

- \( VL_{oil} \): Oil volume in low pressure accumulator.
- \( VH_{oil} \): Oil volume in high pressure accumulator.
- \( VL_{gas} \): Gas volume in low pressure accumulator.
- \( VH_{gas} \): Gas volume in low pressure accumulator.
- \( VL_0 \): Total volume of low pressure accumulator.
- \( VH_0 \): Total volume of high pressure accumulator.

The assumptions for this analysis are:

- The oil is incompressible and does not expand by rising temperature
- The pipes do not expand by rising temperature or pressure
- The temperature is uniform distributed true whole the system

We first look at a point A, see Figure 16, where there is no oil in the high pressure accumulator, i.e.:

\[ PH_A = PL_A << PH_{gas} \]

At this point the volume of oil in the low pressure accumulator is, according to eq. 4:

\[ VL_{oil,A} = VL_0 - VL_{gas,A} \]  
- 6

Assuming that the gas has had enough time to settle, the gas volume can then be calculated according to equation 1:

\[ VL_{gas,A} = \frac{PL_0 \cdot VL_0}{T_0} \cdot \frac{T_A}{PL_A} \]  
- 7
With:

- $P_{L_0}$: Initial gas pressure in LPA.
- $T_0$: Initial gas temperature in LPA.
- $T_A$: Gas temperature at time A.
- $P_{L_A}$: Gas pressure in the LPA at time A.

Figure 16: Defining the measurement points.

Next we look at time B, see Figure 16, where the high pressure has increased and the low pressure decreased: $PH_B > PH_{gas}$ and $PL_A < PL_B$. The high pressure is larger than the HPA initial gas pressure. This means that oil has flown from the low pressure accumulator to the high pressure accumulator.

At this point the oil volume in the LPA is, according to eq. 4:

$$VL_{oil,B} = VL_0 - VL_{gas,B}$$

And the oil volume in the HPA is, according to eq. 5:

$$VH_{oil,B} = VH_0 - VH_{gas,B}$$

Assuming that the gas temperature in the HPA is equal to the temperature of the hydraulic oil (based on the design of the HPA), the gas volume can be estimated, with eq. 1:
\[
VH_{\text{gas,B}} = \frac{PH_g \cdot VH_0}{T_g} \cdot \frac{T_B}{PH_B}
\]

With:

- \( PH_g \): Pressure in HPA at time B.
- \( T_B \): Hydraulic temperature at time B.
- \( T_g \): Initial gas temperature.
- \( PH_g \): Initial gas pressure in HPA.

From the mass balances, between time A and B, follows the following equation:

\[
VL_{\text{oil,A}} = VL_{\text{oil,B}} + VH_{\text{oil,B}}
\]

Solving the equations for the three different assumptions, starting with the first using the basic ideal gas law.

Insert equations 6, 7, 8, 9 and 10 into equation 11:

\[
VL_0 - \frac{PL_0 \cdot VL_0}{T_0} \cdot \frac{T_A}{PL_A} = VL_0 - VL_{\text{gas,B}} - VH_0 - \frac{PH_g \cdot VH_0}{T_g} \cdot \frac{T_B}{PH_B}
\]

1. When we assume that the process can be described by the basic ideal gas law, the gas volume in the low pressure accumulator, at time B, is according to equation 1:

\[
VL_{\text{gas,B}} = \frac{PL_0 \cdot VL_0}{T_0} \cdot \frac{T_B}{PL_B}
\]

Insert equation 13 in equation 12:

\[
VL_0 - \frac{PL_0 \cdot VL_0}{T_0} \cdot \frac{T_A}{PL_A} = VL_0 - \frac{PL_0 \cdot VL_0}{T_0} \cdot \frac{T_B}{PL_B} + VH_0 - \frac{PH_g \cdot VH_0}{T_g} \cdot \frac{T_B}{PH_B}
\]

This leads to:

\[
\frac{PL_0 \cdot VL_0}{T_0} \left( \frac{T_B}{PL_B} - \frac{T_A}{PL_A} \right) = VH_0 - \frac{PH_g \cdot VH_0}{T_g} \cdot \frac{T_B}{PH_B}
\]

and

\[
PL_0 = \frac{T_0}{VL_0} \left( \frac{PL_A \cdot PL_B}{PL_A \cdot T_B - PL_B T_A} \right) \left( VH_0 - \frac{PH_g \cdot VH_0}{T_g} \cdot \frac{T_B}{PH_B} \right)
\]

2. In the case of a pure isotherm process, the temperatures at time A and B are equal so:

\[ T_A = T_B = T \]
Therefore eq. 16 leads to:

$$PL_0 = \frac{T_0}{VL_0} \left( \frac{PL_A \cdot PL_B}{T \cdot (PL_A - PL_B)} \right) \left( VH_0 - \frac{PH_g \cdot VH_0}{T_g} \cdot \frac{T}{PH_B} \right)$$

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3. When it is assumed that the pressure drop in the low pressure is an adiabatic process, the relation between time A and B can be described in the following way, according to equation 3:

$$PL_A \cdot VL_{gas,A}^k = PL_B \cdot VL_{gas,B}^k$$

- 18

Where $k$ is the specific heat ratio, for a diatomic gas (like nitrogen) this ratio is 7/5.

With this relation, the gas volume at time B can be expressed as a function of the gas volume at time A and the pressure ratio:

$$VL_{gas,B}^k = \left( \frac{PL_A}{PL_B} \right)^{\frac{1}{k}} VL_{gas,A}$$

- 19

Inserting this relation in equation 12 gives

$$VL_0 - \frac{PL_0 \cdot VL_0}{T_0} \cdot \frac{T_A}{PL_A} = VL_0 - \frac{PL_0 \cdot VL_0}{T_0} \cdot \frac{T_A}{PL_A} \left( \frac{PL_A}{PL_B} \right)^{\frac{1}{k}} + VH_0 - \frac{PH_g \cdot VH_0}{T_g} \cdot \frac{T_B}{PH_B}$$

- 20

Hence $P_0$ can be determined as:

$$PL_0 = \frac{T_0}{VL_0} \cdot \frac{PL_A}{T_A} \left( \frac{PL_A}{PL_B} \right)^{\frac{1}{k}} - 1 \left( VH_0 - \frac{PH_g \cdot VH_0}{T_g} \cdot \frac{T_B}{PH_B} \right)$$

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5.1.2 Optimise the model

Hereunder the tuning process will be discussed. This will be done for the analytic equation based on the general ideal gas law (equation 16). We will initially think that the behaviour of the gas in the accumulator was isothermal or adiabatic, dependent of the charge rate of the accumulators, does the general gas law describe the behaviour the best.

Figure 17 shows the initial pre-charge pressure calculated by equation 16, as shown the values for the LPA initial pressure are dependent on the temperature. In Appendix A all the measurement data are summarized.

Knowing that the calculations are dependent on the temperature, the model can be tuned. The following equation for the temperature at point B is introduced:

\[
T_B = \alpha \cdot T_M + \beta (T_M - T_E)
\]

With

- \(T_M\): The measured temperature at a time after the pressure drop, in °C.
- \(T_E\): is here 140°C
- \(\alpha, \beta\): Tuning parameters.

By tuning the model it is possible to compensate the temperature dependency which leads to an initial pressure which is independent of the temperature.

Figure 18 shows a set of data which is tuned. There are still some large differences, but the data in Figure 18 are from different accumulators. Figures 22 and 23 shows the separated sets of data, these will be discussed later.
When all the data are tuned, it is possible to look at the separated data. These are for one car and one engine during a whole test or race session. This means that (normally) the accumulators should stay the same and that the calculated initial pressures should also be the same. Details from the data can be found in Appendix A.
There is a large temperature drop when the ‘outstanding’ point @ [65 °C, 2.2 bar] occurs. The temperature drop is described in the section ‘REMARK’. DATA: WE Abs94

As shown in Figure 20 and Figure 21 the calculated value of the initial pressure is relatively constant. Appendix B contains more figures of the separated and tuned sets of data, for these data the outstanding points are explained. This explanation is used for the implementation into WinTax.
The initial goal is to implement the formulas into WinTax. The HPA gas pressure and temperature have to be select manual. In order to avoid this, the influence of fixing the HPA gas pressure is investigated. The high pressure accumulator pressure has to be around 110 bars at 293 Kelvin. Figure 22 and Figure 23 show the same sets of data as Figure 20 and Figure 21, only with a fixed gas pressure.

![Figure 22: Initial pressures, according to equation 16 and 22.](image)

![Figure 23: Initial pressures, according to equation 16 and 22.](image)

As shown in Figure 22 and Figure 23 the values of the calculated initial pressures with fixed gas pressure and temperature are still in range. But they are more spread than the calculations with variable gas pressure. The data from Figure 23 seems to be slightly temperature dependent again, but this can be adjusted by setting the gas pressure to the correct value.

Figure 24 shows the complete set of data again for the fixed and variable gas pressure, with the same tune for equation 34. Figure 24 shows also that some ‘outstanding’ points are springing back into the right direction. So the reason for the different values of these points is that the gas pressure is inserted wrongly. This shows the risk of inserting the gas pressure manual. Other ‘outstanding’ points are staying at their position; the reason for this should be investigated.
It can also be interesting to look at the ‘time’ that the engine is running. This can be done by looking at the temperature before the engines start.

First we can look at the temperature difference between $T_A$ and $T_B$. This temperature difference is related to the time, because when the engine starts, the hydraulic oil starts heating. This heating is not only dependent on the time, but it is a simple approximation.

When we define the temperature $T_B$ as:

$$T_B = T_M - \alpha \cdot (T_M - T_A)$$  \hspace{1cm} (23)

As shown in Figure 25, the temperature shift, by equation 23, is quit independent of the ‘time’, because either a positive or a negative shift causes more widely scattered points.
Assuming that the temperature at point B, $T_B$, is only a function of the measurement temperature at point M. Different formulas for the temperature at point B are investigated; the most interesting is discussed below:

$$T_B = \alpha \cdot T_4 - \beta \frac{1}{T_4^{\gamma}} + \varepsilon \cdot T_4^{\lambda}$$  - 24

With the tuning parameters: $\alpha$, $\beta$ and $\gamma$. By manipulating the temperature, using equation 36, the calculated pressure for the low temperatures is also adjusted. Figure 26 shows the temperature shift, compared with the measured temperature, at point M.

![Figure 26: Temperature switch using formula XIX, with the parameters: $\alpha = 0.8$, $\beta = 2e4$, $\varepsilon = 0.002$, $\gamma = 2$ and $\lambda = 2$.](image)

When the temperature shift with equation 34 is tuned, the temperature shift out of Figure 26 is very similar to the one shown in Figure 18. Only with this temperature shift it is possible to shift the lower temperature more detailed.
5.1.3 Different Approach

When the process is analysed more detailed, the different stages can be analysed Figure 27 shows a Pressure-Volume diagram (P-V diagram); in the figure all the stages are separated:

0-1 The accumulator is adiabatically filled with oil
1-2 The gas temperature increases, this causes a constant-pressure process (isobar).
2-3 The gas temperature increases, this causes a constant-volume process (isochoric).
3-4 The pressure drops, by one of the three equations of state, described above.
4-5 The temperature increases after the pressure drop, this causes a constant-pressure process (isobar).
5-6 The temperature increases after the pressure drop, this causes a constant-volume process (isochoric).

![Figure 27: P-V diagram of the low pressure accumulator process.](image)

The equations that describe these 2 variable processes are:

Volume change due to a temperature change:

\[
\frac{T_B}{T_A} = \left( \frac{V_B}{V_A} \right)^{1-k}
\]

Pressure change due to a temperature change:

\[
\frac{T_B}{T_A} = \left( \frac{P_B}{P_A} \right)^{\frac{k-1}{k}}
\]

Adiabatic process, according to equation 3:

\[
\frac{P_B}{P_A} = \left( \frac{V_B}{V_A} \right)^{-k}
\]
The time between point 0 and 3 in Figure 27 is relatively large, the gas has time for heat exchange. Because of this it is possible to describe the relation between these points by the basic ideal gas law. Figure 28 shows also a P-V diagram, but here the relation between point 1 and 2 is a straight line, because of the long time between the processes.

![P-V diagram of the simplified low pressure accumulator process.](image)

**Figure 28:** P-V diagram of the simplified low pressure accumulator process.

The stages after the pressure drop can be simplified as well. A reason for a relative small volume change in the LPA is shown in Figure 29.

![Temperature change after the pressure drop](image)

**Figure 29:** Temperature change after the pressure drop

If the volume stays constant a temperature change causes a pressure change. Because the pump controls the pressure at the high pressure site, the gas volume at the HPA site has to increases. This leads to an oil flow from the HPA flows into the LPA and what causes a compression of the gas. The gas temperature in the LPA increase also, this causes an expansion of the gas.
Assumed is that the result of these effects leads to no change in volume. So the pressure in the LPA increases, while the volume of gas doesn’t change (isochoric process).

If the pressure-drop (X→Y) is adiabatic, the drop can than be described, using equation 27:

$$\frac{P_X}{P_Y} = \left(\frac{V_X}{V_Y}\right)^{-k}$$  \hspace{1cm} - 28

Because the volume does not change, the process after the pressure drop can be described with the general ideal gas law:

$$\frac{P_Y}{T_Y} = \frac{P_Z}{T_Z}$$  \hspace{1cm} - 29

Combining equation 28 and 29 leads to:

$$P_X = \frac{T_Y}{T_Z} \cdot \left(\frac{V_X}{V_Y}\right)^{-k} \cdot P_Z$$  \hspace{1cm} - 30

The temperature and pressure at point X and Z are measured; this means that the temperature at point Y and the volumes are still unknown.

In the same way as done for equation 10 the volume of oil in the HPA can be calculated. With this volume the volume ratio between point X and Z can be described as:

$$\frac{V_X}{V_Y} = \frac{V_X}{V_Y} \cdot \frac{Z}{V_{oil}}$$  \hspace{1cm} - 31

This means that only the volume at point X and the temperature after the pressure drop, at point Y, are unknown.

The pre-charge pressure from the LPA should be around 1.5 bar relative. When we assume that the average pressure equals 1.5 bar, the volume at point X can be calculated. This can be done according to the general gas law:

$$V_X = \frac{P_o \cdot V_{L_o}}{T_o} \cdot \left(\frac{T_X}{P_X}\right)$$  \hspace{1cm} - 32
With equation 30, 31, and 32 we can calculate the gas temperature in the LPA, just after the pressure drop:

$$T_Y = \frac{P_X \cdot T_Z}{P_Z} \left( \frac{P_O \cdot V_{L_O}}{P_O \cdot V_{L_O} - \frac{Z V_{H_{PA}}}{T_D} \cdot T_X} \right)^k$$

With the measurement data out of Appendix A, it is now possible to calculate the temperature just after the pressure drop.

![Figure 30: Temperature after the adiabatic pressure drop versus measured temperature](image)

From these measurement data, the temperature shift can be expressed, as function of the measured temperature:

$$T_Y = 0.0048 \cdot T_Z^2 - 2.9427 \cdot T_Z + 742.1930$$

Were the temperatures are in Kelvin.
With this relation between the temperature at point Y and the measurement temperature, the initial pressure from the LPA can be calculated:

\[
P_o = \left( \frac{VH_O - \frac{PH_O \cdot VH_Q \cdot T_Z}{T_X \cdot PH_Z}}{1 - \left( \frac{P_Z \cdot T_Y}{P_X \cdot T_Z} \right)^k} \right) \cdot \frac{P_X \cdot T_Q}{T_X \cdot VL_O}
\]

Were the initial gas pressure in the HPA is assumed to be 110 bar @ 293 K.

Figure 31 shows these calculations of the initial pressure versus the measurement temperature. As shown in the figure the spread of points is relative low.

This model predicts the pre-charge pressure also pretty well; however with this model it is more difficult to see leakages. Because of this, the other model is implemented into WinTax; this will be discussed in the next session.
5.1.4 Implementation into WinTax

For implementation into WinTax it is much easier to fix the gas pressure (the calculated values for the initial gas pressure are still in the expected range +/- 0.5 bar), this method is used for the implementation.

For the implementation into WinTax, 5 parameters are needed. These are: \( T_A, P_A, T_B, P_B \) and \( PH_B \). Which are respectively: the temperature and the hydraulic low pressure at point A (before the pressure drop), the temperature, hydraulic low and hydraulic high pressure at point B (after the pressure drop, Figure 16).

For the calculation at a particular time in WinTax, the 5 signals are needed at each time step. WinTax can not recall data from the past. Because of this, the temperature and pressure at point 1 have to be made available at each time step:

Introducing a signal \( n(t) \) which has the following characteristics:

\[
  n(t) = \begin{cases} 
    n(t) = 1 & \text{if } N_{-144} = 0 \\
    n(t) = 0 & \text{if } N_{-144} > 0 
  \end{cases}
\]  

This is possible because the engine is started each engine test.

Multiplying the available low pressure and temperature signal by \( n(t) \), gives the signals \( P_n \) and \( T_n \) with the following characteristics:

\[
P_n = \begin{cases} 
  P_n = P_{HYD-O} & \text{if } n(t) = 1 \\
  P_n = 0 & \text{if } n(t) = 0 
  \end{cases}
\]  

and

\[
T_n = \begin{cases} 
  T_n = T_{HYD} & \text{if } n(t) = 1 \\
  T_n = 0 & \text{if } n(t) = 0 
  \end{cases}
\]

Now we can integrate \( P_n \) and \( T_n \) over the time and divide those by the signal \( n(t) \) integrated over the time. This gives us the signals \( P_A \) and \( T_A \), which are the averaged values of the pressure and temperature before the pressure-drop. These are exactly the values that are needed and they are available at each time step during the time:

\[
P_A = \frac{\int P_n dt}{\int n dt} \quad \text{and} \quad T_A = \frac{\int T_n dt}{\int n dt}
\]

Now all signals are available and it is possible to calculate the initial pressure for the values at each time interval, when the pressure has dropped.

When the formula is implemented into WinTax it becomes clear that there are some boundary conditions for this calculation. These conditions are also explained in Appendix B.

The conditions for this calculation are that there has to be no large flow demand, the temperature at point 2 has to be larger than 40 °C and the gradient of the temperature needs to be small. Due to the condition that the temperature has to be larger than 40 °C, the temperature shift of equation 22 is used.
The criterion that the temperature has to be larger as 40 degrees can directly be set in the Conditions of WinTax. The condition that the flow demand has to be small is more complex.

With the available signals $Q_{TOT\_MOOG}^4$ and $Q_{TOT\_ACT}^4$ the problem can not be solved, because the default value is different for each car. If the flow demand increases, this can be caused by three main causes: there is a steering input, the clutch is released or the engine revs-up, what causes a larger throttle demand. This throttle demand is assumed to be directly related to the flow demand through these channels. The signal $CLU\_DEM^5$ is directly related to the flow needed to actuate the clutch. For the steering input, the pressure difference between $P\_STR\_L^6$ and $P\_STR\_R^6$ is directly related to the flow demand of this channel. Both signals are smaller as 1, when the flow demand is negligible and are much larger for a large flow demand. With those signals the condition for a small flow demand is monitored. Sum-up these signals gives an index for the flow demand at the moment, because these signals are more constant it is a usable value. This index can also be implemented in the WinTax Conditions.

The gradient of the temperature has to be small. The gas temperature is not equal to the hydraulic temperature, but the hydraulic temperature is an estimation of the gas temperature. When the hydraulic temperature changes, it takes a while before the gas temperature equals this temperature. When the gradient of the temperature is relatively large, there will be a large difference between the gas temperature and the hydraulic temperature. This condition can be accounted by determining the derivative of the hydraulic temperature. Because a measurement channel, measured on a certain frequency, has to be derivate, it is important to avoid large steps in the derivation, because this is impossible. This problem has been solved by determining the derivative at a lower frequency than the channel is measured. The derivative is also filtered, so that the signal becomes smoother. After this, it can easy be implanted in the Conditions. Appendix C gives the programmed WinTax formulas. Finally the alarm can be set if the calculated value for $P_0$ is out of range. The OK range is set to $1.5 \pm 0.5$ bar, with the conditions defined above.

When these conditions are implemented into the WinTax Conditions, the initial pressure can be calculated according to the model. Another option is to integrate the conditions into the virtual channels, which means that the calculations are only made when possible, according to the conditions. For a visible representation, this is better because only the relevant points are displayed.

Figure 32: snap shot WinTax screen with calculations for initial pressure.

---

4 channels out of: Estimation of the hydraulic circuit flow

* WinTax measurement channels
As shown in Figure 32, the initial pressure before the cursor is pretty constant, after the cursor the calculation drops, but we also see that the flow-index value increases. The large flow demand has effect on the calculation.

When these conditions are set, the initial pressure can be calculated. Table 1 shows the result of equation 16 with the temperature shift equal to equation 22 programmed in WinTax. Table 2 shows the results of equation 16 with the temperature shift of equation 24.

<table>
<thead>
<tr>
<th>TRACK</th>
<th>MARKER</th>
<th>CAR</th>
<th>MAX P&lt;sub&gt;L&lt;/sub&gt;&lt;sub&gt;_0&lt;/sub&gt;</th>
<th>MIN P&lt;sub&gt;L&lt;/sub&gt;&lt;sub&gt;_0&lt;/sub&gt;</th>
<th>ΔP&lt;sub&gt;max&lt;/sub&gt;</th>
</tr>
</thead>
<tbody>
<tr>
<td>Shanghai</td>
<td>RACE</td>
<td>410</td>
<td>1.8</td>
<td>1.3</td>
<td>0.5</td>
</tr>
<tr>
<td>Shanghai</td>
<td>RACE</td>
<td>409</td>
<td>1.6</td>
<td>1.0</td>
<td>0.6</td>
</tr>
<tr>
<td>Silverstone</td>
<td>T29W38</td>
<td>402</td>
<td>1.9</td>
<td>1.3</td>
<td>0.6</td>
</tr>
<tr>
<td>Silverstone</td>
<td>T29W38</td>
<td>403</td>
<td>1.9</td>
<td>1.1</td>
<td>0.8</td>
</tr>
<tr>
<td>Monza</td>
<td>RACE</td>
<td>406</td>
<td>1.7</td>
<td>1.2</td>
<td>0.5</td>
</tr>
<tr>
<td>Monza</td>
<td>RACE</td>
<td>409</td>
<td>1.6</td>
<td>1.3</td>
<td>0.3</td>
</tr>
<tr>
<td>Monza</td>
<td>T28W36</td>
<td>411</td>
<td>1.6</td>
<td>1.1</td>
<td>0.5</td>
</tr>
<tr>
<td>Monza</td>
<td>T28W36</td>
<td>403</td>
<td>1.5</td>
<td>1.2</td>
<td>0.3</td>
</tr>
</tbody>
</table>

Table 1: WinTax results of equation 16, using the temperature shift of equation 34, with: α =1 and β = 0.23.

<table>
<thead>
<tr>
<th>TRACK</th>
<th>MARKER</th>
<th>CAR</th>
<th>MAX P&lt;sub&gt;L&lt;/sub&gt;&lt;sub&gt;_0&lt;/sub&gt;</th>
<th>MIN P&lt;sub&gt;L&lt;/sub&gt;&lt;sub&gt;_0&lt;/sub&gt;</th>
<th>ΔP&lt;sub&gt;max&lt;/sub&gt;</th>
</tr>
</thead>
<tbody>
<tr>
<td>Shanghai</td>
<td>RACE</td>
<td>410</td>
<td>1.9</td>
<td>1.5</td>
<td>0.4</td>
</tr>
<tr>
<td>Shanghai</td>
<td>RACE</td>
<td>409</td>
<td>1.7</td>
<td>1.1</td>
<td>0.6</td>
</tr>
<tr>
<td>Silverstone</td>
<td>T29W38</td>
<td>402</td>
<td>1.9</td>
<td>1.3</td>
<td>0.6</td>
</tr>
<tr>
<td>Silverstone</td>
<td>T29W38</td>
<td>403</td>
<td>2.0</td>
<td>1.2</td>
<td>0.8</td>
</tr>
<tr>
<td>Monza</td>
<td>RACE</td>
<td>406</td>
<td>1.7</td>
<td>1.3</td>
<td>0.4</td>
</tr>
<tr>
<td>Monza</td>
<td>RACE</td>
<td>409</td>
<td>1.7</td>
<td>1.3</td>
<td>0.4</td>
</tr>
<tr>
<td>Monza</td>
<td>T28W36</td>
<td>411</td>
<td>1.7</td>
<td>1.2</td>
<td>0.5</td>
</tr>
<tr>
<td>Monza</td>
<td>T28W36</td>
<td>403</td>
<td>1.6</td>
<td>1.3</td>
<td>0.3</td>
</tr>
</tbody>
</table>

Table 2: WinTax results of equation 16, using the temperature shift of equation 36, with: α =0.8, β =2e4, ε =0.002, γ =0.5 and λ =2.

As shown in the tables both equation predict the pre-charge pressure between the tolerance of ± 0.5 bar. Because the difference between the temperature shift from equation 22 and 24 is negligible, there is chosen to use equation 22. In the conditions is set that the low oil temperature is not taken into account and this was the main advantage of equation 24.
5.1.5 Remark

A remark that has to be made is that the calculation is incorrect when there is a large temperature drop in the measured temperature, see Figure 33.

Figure 33: Screenshot WinTax, of a large temperature drop. Upper line: Measured temperature. Middle line: Temperature according to eq. 34. Lower line: index for the oil flow.

The temperature drop can be explained by a van, which blows into the cooler. When the engine starts the cold oil passes the temperature sensor. When this sort of heating occurs, the measured temperature is not related to the gas temperature in the accumulators.
5.2 Results

With the final optimised model, the pre-charge pressure of the LPA can be estimated within the required tolerances. This means that with the model a leakage in the LPA can be detected, what can save a lot of time.

The result of the made model is demonstrated in the example below.

Example

During the Shanghai race practice there was a gas leakage in the hydraulic low pressure accumulator. This leakage can be detected with the made model and is demonstrated in the following example:

<table>
<thead>
<tr>
<th>MARKER</th>
<th>ABS-LAP</th>
<th>MAX $PL_0$</th>
<th>MIN $PL_0$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fire-up (Thursday)</td>
<td>4</td>
<td>1.8 [bar]</td>
<td>1.0 [bar]</td>
</tr>
<tr>
<td></td>
<td>7</td>
<td>1.1 [bar]</td>
<td>1.0 [bar]</td>
</tr>
<tr>
<td></td>
<td>9</td>
<td>0.6 [bar]</td>
<td>0.4 [bar]</td>
</tr>
<tr>
<td>P1 (Friday)</td>
<td>1</td>
<td>0.6 [bar]</td>
<td>0.5 [bar]</td>
</tr>
<tr>
<td></td>
<td>20</td>
<td>1.5 [bar]</td>
<td>1.5 [bar]</td>
</tr>
</tbody>
</table>

Table 3: Low pressure accumulator pressure during the first two days of the Shanghai race of the 411, calculated with equation 16 and 34.

As shown in Table 3 the leak of the low pressure accumulator pressure should have been detected during the fire-up on Thursday. During the fire-up the pre-charge pressure decreased and becomes smaller than the lower limit (1 bar). This should have meant that there was at least $\frac{1}{2}$ an hour more for testing during the first free practise, what can be critical in F-1!

The model is implemented in WinTax and the layout is made in such way that the race engineers can monitor the accumulator pre-charge pressure in an eye wimp. The final layout is shown in Figure 34. This layout also contains the hydraulic temperature, this because of the problem discussed in the remark.
Figure 34: Layout to check to hydraulic pressures. The figure shows the limits for the hydraulic high pressure and the calculation points of the hydraulic low pressure between their limits.
5.3 Recommendations

The model should be tested on the test-bench:

- When there is a structural error in the pre-charge pressure, this error is also in the model. The model has been tuned on race and test data.

- When the oil temperature is relative low, the viscosity can influence the results. The resistance of the oil flowing out of the low pressure accumulator and into the high pressure accumulator is larger. This effect should also be investigated.

- The temperature shift should also be investigated on the test bench. The gas temperatures in the accumulators have to be measured. During this test the behaviour of the gas (isotropic, adiabatic, general gas law) during the time has to be investigated.
6 Steering System

The power steering system of an F-1 car is a complex system. To understand this system, first some literature over power steering systems is studied. The final goal of this project is to make software to compare different power steering systems. This software must have the possibility to compare the steering system on different corners of the track. During the test not only different power steering system were tested, but also some different steering columns. Because of this the routine must also have the possibility to compare these steering columns.

6.1 Conventional power steering systems

Since people use vehicles with more then one axle, they had to think about finding ways to steer. The easiest steering systems are those on motorbikes. This system directly guides the actuation of the steering towards the wheel.

Nowadays, these systems are barely found in vehicles with more than two wheels. This is due to the not so comfortable steer feeling and the large amount of assembly space required by these systems.

When a steering system is designed, important parameters are: the forces that act on the vehicle, the drivability and the required steer axle.

In the 40ies of the last century, the first power steering systems were introduced on the American market. The European market followed in the 50ies. Nowadays almost all cars are equipped with power steering systems.

Next to the conventional hydraulic power steering systems, there are other power steering systems introduced during the years, i.e. the electro-hydraulic systems and the fully electric systems. Nowadays much research is done on ‘Steer by wire’ systems. These systems are based on a mechanical decoupling of the steering-wheel and the wheels. All these systems are discussed below.

6.1.1 Hydraulic systems

One of the oldest power steering systems, the hydraulic system, has proved itself on rally sport. A compact system, small weight and a high reliability are the most important features which make a vehicle constructor to choose a hydraulic system.

The rack and pinion steering system can be supported hydraulically. The working principle of this power steering system is shown in Figure 35:

![Figure 35](image-url): hydraulic rack and pinion power steering system
As shown in Figure 35: when a steering torque is applied, this will act on the torsion bar (3) in the housing (1). Because of the relative displacement between the input shaft and the valve sleeve, in the housing the hydraulic supply flow is connected to either one of the two chambers (10 or 11). And the hydraulic will support the mechanical steering input.

Another hydraulic power steering system is the **hydraulic re-circulating-ball steering system**. This system is shown in Figure 36. The workings principal of this system is much different to the system shown above. Also in this system, the hydraulic valve is actuated by the relative displacement between the input shaft and the valve sleeve. And the hydraulic oil flows to one of the two chambers.

![Figure 36](image)

**Figure 36**: hydraulic re-circulating ball steering system

A steering wheel input is directly guided to the screw (3), what leads to an axial movement of the piston (5). Due the oil pressure in one of the two chambers the hydraulic actuation supports the steering.

These hydraulic systems were developed more sophisticated along the years. The rack ratio made variable versus the travel. Also the torsion bar can be stifled or pre stressed, with these methods the ratio from steering wheels to steering rod can be varied.
6.1.2 Electro-hydraulic systems

Figure 37 shows the setup of an electro-hydraulic rack and pinion system. The working principle of this system is almost the same as the fully hydraulic rack and pinion system. With the difference that the hydraulic support is velocity dependent in these systems.

Figure 37: electro-hydraulic system.

Figure 38 shows this velocity dependency. The extra parts needed for an electro-hydraulic system are: an electric velocity control device, an electro-hydraulic converter and a steering valve with feedback.

As shown in Figure 38, this velocity feedback results in a velocity dependent steering support, what improves the drivability of a vehicle.

Figure 38: Velocity dependent pressure built-up.

The electro-hydraulic rack and pinion and the re-circulating ball systems are shown in Figure 39 and Figure 40.
As shown in Figure 39 and Figure 40 the servo-valve is different. The goal of this valve, in combination with the electronic, is to create the velocity dependent pressure build-up as shown in Figure 38.

Figure 39: electro-hydraulic rack and pinion system.

Figure 40: electro-hydraulic re-circulating-ball steering system.
6.1.3 Electric systems

In the rack and pinion steering system it is possible to replace the hydraulic part by an electric actuator. The electric systems contain an electromotor, a transmission and sensor-electronic. The working principal is: The input torque is measured and transformed in an electric signal; the electro engine transforms the signal into a servo moment at the steering rack. Some different designs for electric systems are shown in Figure 41.

![Different setups for electric power steering systems](image)

Figure 41 3): Different Setups for Electric power steering systems.

Advantages of the electric system (compared to the hydraulic system) are that there are overall fewer components needed and there is no oil required. The durability is higher and the energy savings can be over 85 percent. The system works also when the engine doesn’t run, which can be useful when the engine is broken.
6.1.4 **Steer by Wire**

In a *steer by wire system* the mechanical connection between the steering wheel and the steering rack is decoupled. Figure 42 shows a schematic representation of a steer by wire system.

![Schematic representation of a 'Steer-by-Wire' system](image)

**Figure 42**: Schematic representation of a 'Steer-by-Wire' system

A *steer by wire system* contains two parts, the steering part and the steering module. The steering part contains a conventional steering wheel. The input signal, characterised by the steering angle is measured. The remained steering axle is connected to steering wheel motor and the hydraulic pump. The steer module contains of a steering rack, which is actuated by a steer motor. For safety back up, the system contains a hydraulic part. When the electric system fails, the steering is ensured by the hydraulic system. The *steer by wire system* is not yet on the market and illegal in formula-1 cars.

6.1.5 **The Toyota F-1 power steering system**

For the power steering system in a formula 1 race car different properties are required. I.e. the required steering support for production cars is high at low speeds and low at high speeds. For a formula 1 car this is just the opposite, the required support is high during high speed corners and low during by low speeds.

Another criterion is that there is no flow through the system when the engine is cranked, this because there has to be a hydraulic pressure built up for starting the engine. Also the steering angle of a formula 1 car is limited and the weight and reliability of the system is quite important.

*Due to confidentially of this system no details can be disclosed.*
6.2 Comparison of different setups

During test 33 in Jerez different steering setups were tested. The steering setup contains a steering column and a power steering system, from this power steering system the steering rack was changed. The column and rack were changed systematically; the tested setups are shown in Table 4:

<table>
<thead>
<tr>
<th>Jerez T33W42TU 402 Zonta</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>steering column</strong></td>
</tr>
<tr>
<td>V1</td>
</tr>
<tr>
<td>V1</td>
</tr>
<tr>
<td>V1</td>
</tr>
<tr>
<td>V1</td>
</tr>
<tr>
<td>V3</td>
</tr>
<tr>
<td>V3</td>
</tr>
</tbody>
</table>

Table 4: Tested steering system setups

In order to compare the steering systems, the measurement data are exported to Matlab. A program is written to compare the different steering systems at different corners. Figure 43 shows the corners and the numbers, these numbers are commonly used within TMG.

If the steer is moved slightly to the end stops, the v-curve looks like shown in Figure 44. Figure 44 shows on the horizontal axis the steering torque and on the vertical axis the pressure difference. During a race of test the steering wheel is not moved slightly, this causes that the measurement points in the v-curve are more scattered.
Figure 44: the v-curve when the steering wheel is moved slightly.

Figure 45 shows the measurements from the tested steering systems, were the steering column and power steering system are varied. Figure 45 also shows the calculated mean line out of these scattered measurement data. With these mean lines the various system can be compared. The support of road cars often is velocity dependent, while the support of a race car has to be as (proportional to steering torque) linear as possible. This is because of the drivability of these cars, so that the driver has a linear feedback. This is also visible in the measurement data.

As shown in the figure the v-curve has an overlap, which can be different for the various systems. This overlap is caused by a servo-valve (hysteresis), for the hydraulic part, and a pre-load spring causes this overlap for the mechanical part.

Figure 46 shows the comparison of the mean lines, calculated out of the measurement data. The gradient of the various systems are different, as well as the overlap. These differences are caused by slight changes of the power steering system. Figure 47 is a detail plot of Figure 46, which shows the various overlap even better.

In Figure 48 two different systems are compared. Figure 48a and Figure 48b show the measurement data and the calculated mean values again. The other figures only contain the data from the selected corner. In this way the various systems can be compared for different corners (high, low speed and left, right corners).

The figures are different visualisations of the three main parameters for comparing power steering systems. These are the steering torque, the steering angle and the steering support (pressure difference).

In Figure 48 two different power steering systems are compared. As shown in the figure the overlap of both systems is the same, but the occurring pressure difference and gradient of the slope is different. These are major differences in the power steering system.

Figure 48b,c and e show the main values plotted against each other. The arrow heads in these plots are directed into positive direction. As shown in these figures it seems that the C rack power steering is heavier as the A rack.

Figure 48f shows the steering angle input versus telemetric distance. With this plot it can be compared if the moment of steering of the compared systems is at the same position of the track. This moment of steering is also velocity dependent, because of this the minimum, maximum and mean velocity are shown in the legend.
Figure 45: Measured data and the calculated mean values.
Figure 46: The calculated averaged values of the measured data.
Figure 47: Detail plot of Figure 46.
Figure 48: Comparison between two different power steering systems.
6.3 Results
Software is developed to compare the different steering setups. As shown above, the software has the possibility to compare the measurement data on different corners of the track. During several tests more steering systems are tested on different circuits, the software is made usable for all these tests. Again some of these results are implemented into WinTax.

*Due to confidentiality the comparisons and results can not be disclosed.*
7 Tyre pressure

The tyre pressure is dependent of many things, e.g. the temperature, sort of tyre and the velocity. The velocity dependency is caused by down force and centrifugal forces. These forces work in opposite direction. Figure 49 shows that the pressure is dependent of the temperature and the velocity. The low frequency trend is caused by the temperature change, while the high frequency fluctuations are caused by the velocity changes.

Figure 49: Dependency of velocity and temperature for the tyre pressure.

The goal form this analysis is to get a better understanding of the tyre behaviour. Especially when the tyre heaters are removed and the car takes-off, during the start or after a pit stop. Due to the next season regulation (only one set of tyres per race), it is even more important to monitor the tyre behaviour constantly, e.g. for leakage, tyre pressure and tyre temperature.

7.1 Velocity Dependency

It is possible to investigate the velocity dependency of the pressure by looking at a few points. This has been done in Table 5. In the table two time points after each other are compared, so that the temperature difference is minimal. The first point is around 100 km/h and the second around 300 km/h.

Although it seems that there is a relation, this does not become clear out of the measurement data. The Michelin Data from Appendix D show that a velocity-increase causes a volume increase. This leads, according to the ideal gas law, to a decrease of pressure by increasing velocity. This is in line with the measurement data shown in Table 5.
Silverstone T29W38TH  ABS 85-89 403 FL-tyre

<table>
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<tr>
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<th>V [km/h]</th>
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<th>P [bar]</th>
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<td>102</td>
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</table>

Table 5: velocity dependency of the tyre pressure. (* : Modified)

The velocity dependency does not become clear out of these measurements, the temperature dependency does.

7.2 Temperature Dependency

For the temperature dependency also the Silverstone T29W38TH 403 data are used. Figure 50 shows the measurement temperature at the rim. Figure 50 also shows the filtered temperature, this has been done in WinTax, using a low pass filter with a cut-off frequency of 0.01 Hz. Now only the pressure changes caused by the temperature change are visible.

![Figure 50: Measured temperature and filtered temperature versus the time.](image-url)
Figure 51 shows the measured tyre pressure and the filtered tyre pressure versus the time. Again a low pass filter with a cut-off frequency of 0.01 Hz was used.

Figure 51: Measured pressure and filtered pressure versus the time.

Figure 52 shows the filtered temperature and filtered pressure plot against each other. This is a characteristic figure for the tyre behaviour.

Figure 52: Tyre-Pressure versus rim temperature of the FL tyre.
Figure 52 shows the following:

A: The tyre-heaters are on the tyre. The tyres are at pre-heat temperature.

B: The tyre heaters are off, and the car is driving on the track for the first (1 or 2) laps. The temperature and the pressure drop.

C: The car has driven a few laps, the temperature and pressure stop decreasing and start increasing. There is an offset in the temperature sensor, what causes that the pressure is higher, at the same temperature.

D: The pressure is stabilised while the temperature keeps increasing, this may be caused by escaping vapour or changing volume.

When the car has driven several laps, and leaves again after a short stop, the tyres heaters are not put on. Part A and B are than not visible in the figure.

We can now compare some data from the season 2003 and the season 2004. Table 6 shows the used data.

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<th>TRACK</th>
<th>SEASON</th>
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<th>CAR</th>
<th>ABS LAP</th>
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<td>T29W38TH</td>
<td>403</td>
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</tr>
<tr>
<td>Jer_03</td>
<td>Jerez</td>
<td>2003</td>
<td>T26W38FR</td>
<td>303</td>
<td>80-84</td>
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<tr>
<td>Jer_03**</td>
<td>Jerez</td>
<td>2003</td>
<td>T26W38FR</td>
<td>303</td>
<td>80-91</td>
</tr>
</tbody>
</table>

Table 6: Used data for the comparison.

Table 6 shows two sets of data for the test in Jerez. The second set of data (Jer_03**) contains more laps than the first (Jer_03). These sets are separated so that they can be compared more easily.

Figure 53 to Figure 55 show the measurement data from the 2003 season and data from the season 2004. It seems that the temperature of the tyre heaters is different; this causes as much shorter ‘stabilisation time’, the initial tyre pressure is also much higher.

In the figures one can see that the temperature drop of the right-hand tyres is much larger then the temperature drop from the left-hand tyres during the Silverstone test. The tyre pressure of the Sil_04 test seems also higher, but the Jer_03** shows that they slightly approach this pressure.

In Appendix D the measured and filtered data are shown.
Figure 53: Pressure versus Temperature. The data from the 2003 season are compared with data from the season 2004.
Figure 54: Pressure versus the time. The pressure increases during the time.
Figure 55 Temperature versus the time. The temperature increases during the time.
7.3 Pressure relief rim

When the research on the ‘pressure relief rim’ is a success, it is possible to control the tyre pressure even more precise. By letting air escape out of the tyre, the pressure is kept constant. A disadvantage of this system is that it can’t keep the pressure constant when the temperature decreases. A decrease of the tyre temperature can be caused by a safety car situation, when this happens, the pressure decreases too. Due to the escape of air out of the tyre, the pressure drop will be larger. Because refilling the tyre electrically during the race is illegal, there has to be found another solution for this disadvantage. A possible solution will be discussed in the next section.

7.3.1 Theoretical model

The criteria for this system are: simple, reliable and it can not contain any moving mass. This because of the acceleration forces that work on the tyre (over 800 G). A possible solution conforming to these criteria is discussed below:

The rim contains two volumes, chamber A and chamber B. Chamber B contains pressurised air, while chamber A is the largest chamber and is sealed by the tyre. The relation between the pressures and volumes is: $V_A > V_B$ and $P_A < P_B$.

Chamber A contains a ‘pressure relief valve’ and there is a nozzle between the chambers. This causes a constant flow of air between the chambers. If the pressure in chamber A reaches the set pressure, it is kept constant by a pressure relief valve (or ‘pressure relief rim’). During the pit stops only the volume in chamber B has to be pressurised. The benefit of the system is that the pressure in the tyre can be kept even more constant.

The heating of the air in the tyres is a relative slow process. Because of this the behaviour of the air in the tyre can be seen as an ideal gas. The gas behaviour can be described with the ideal gas law:

$$P \cdot V = m \cdot R \cdot T$$
Where
\[ P \quad = \quad \text{Pressure} \ [ P_A ] \]
\[ V \quad = \quad \text{Volume} \ [ m^3 ] \]
\[ m \quad = \quad \text{Gas mass} \ [ kg ] \]
\[ R \quad = \quad \text{Specific gas constant} \ ( R_{air} = 286.9 ) \ [ J/kg \cdot K ] \]
\[ T \quad = \quad \text{Temperature} \ [ K ] \]

The rim is separated into two volumes, A and B. Both of these volumes can be prescribed with equation 42:

\[ P_A \cdot V_A = m_A \cdot R \cdot T \quad - 42 \quad \text{and} \quad P_B \cdot V_B = m_B \cdot R \cdot T \quad - 43 \]

The mass flow between the areas is dependent of the pressures and the temperature.
If the pressure ratio is larger as the critical pressure ratio, the flow is called ‘sonic’. The critical pressure ratio is given in eq. 45.

\[ \left( \frac{P_A}{P_B} \right)_{critical} = r_{critical} = \left( \frac{2}{\kappa + 1} \right)^{\frac{\kappa}{\kappa - 1}} \quad - 44 \]

(\( \kappa = 1.4 \) for an ideal gas)

A sonic mass flow is prescribed as shown in equation 46.

\[ \dot{m}_{BA} = \alpha \cdot A_z \cdot \psi \cdot P_B \cdot \sqrt{\frac{2}{R \cdot T_z}} \quad - 45 \]

With:
\[ \alpha \quad = \quad \text{Correction factor} \]
\[ A_z \quad = \quad \text{The nozzle area, with diameter} \ D_z \]
\[ T_z \quad = \quad \text{Temperature ‘in’ the nozzle} \]

The correction factor \( \alpha \) has to be determined experimental. In this model \( \alpha \) is kept constant, assumption of an ideal orifice.

Were:

\[ \psi = \sqrt{\frac{\kappa}{\kappa - 1} \left[ \left( \frac{P_A}{P_B} \right)^{\frac{2}{\kappa}} - \left( \frac{P_A}{P_B} \right)^{\frac{\kappa + 1}{\kappa - 1}} \right]} \quad - 46 \]

is the flow function.

Figure 57 shows the characteristics of this flow function. The flow is called subsonic. If the pressure ratio is smaller than this critical value \( r_{critical} \), \( \psi \) is constant and equal to \( \psi_{max} \).

\[ \psi_{max} = 0.484 \quad - 47 \]
This means that if the pressure ration is smaller than the critical value, the parameter $\psi$ is constant. The mass flow between the chambers is only dependent on the pressure in chamber B and the temperature.

Assumed is that the nozzle between the chambers is a simple nozzle. This means that the maximum velocity of the nozzle can’t become larger than critical speed (velocity of sound), because of ‘choking’. This critical speed is given by:

$$v_{\text{critical}} = \sqrt{\kappa \cdot R \cdot T_z}$$  \hspace{1cm} - 48

When the pressure ratio is larger as the critical pressure ratio, the temperature ‘in’ the nozzle $T_z$ is equal to the gas temperature. If the pressure ration is larger as the critical pressure ratio (subsonic region) the temperature in the nozzle can be prescribed as:

$$T_z = T \cdot \left( \frac{P_1}{P_n} \right)^{\frac{\kappa-1}{\kappa}}$$  \hspace{1cm} - 49

$P_n$ is the pressure inside the nozzle.

When the critical pressure ratio is in the nozzle and assumed is that the isentropic exponent is constant, the temperature in the nozzle can be prescribed as:

$$\frac{T_z}{T} = \left( \frac{2}{\kappa+1} \right)^{\frac{\kappa}{\kappa-1}} = \frac{2}{\kappa+1} = 1.2$$  \hspace{1cm} - 50
Summarized this means:

\[
\dot{m}_{BA} = \begin{cases} 
\frac{P_A}{P_B} \leq r_{critical} \quad \text{and} \quad v \leq v_{critical} & : \quad \dot{m}_{BA} = \alpha \cdot A_z \cdot \psi_{max} \cdot \sqrt{\frac{2}{R \cdot T_z}} \\
\frac{P_A}{P_B} > r_{critical} \quad \text{and} \quad v \leq v_{critical} & : \quad \dot{m}_{BA} = \alpha z \cdot \psi \cdot P_B \cdot \sqrt{\frac{2}{R \cdot T_z}} \\
v > v_{critical} & : \quad \dot{m}_{BA} = \alpha z \cdot \rho \cdot \sqrt{\kappa R T_z} 
\end{cases}
\]

There is a sort of ‘pressure relief valve’ in the rim, this means that there is a flow out of the tyre, called \( \dot{m}_{prv} \), when the pressure reaches the set pressure (\( P_s \)), the mass in both chambers is described as:

\[
m_B = m_B^0 - \int \dot{m}_{BA} \cdot dt \quad - 51 \quad \text{and} \quad m_A = m_A^0 + \int \dot{m}_{BA} \cdot dt - \int \dot{m}_{prv} \cdot dt \quad - 52
\]

With:

- \( m_B^0 \) = initial mass in chamber B
- \( m_A^0 \) = initial mass in chamber A

Where:

\[
\dot{m}_{prv} = \frac{(P_A - P_s) \cdot V_A}{R \cdot T} \quad \text{(This is the required flow out of the tyre)} \quad - 53
\]

When is assumed that the flow capacity of the pressure relief rim is larger then the maximum required flow out of the tyre, the mass in chamber A can be prescribed as:

\[
m_A = \begin{cases} 
P_A \leq P_s & : \quad m_A = m_A^0 + \int \dot{m}_{BA} \cdot dt \\
P_A > P_s & : \quad m_A = \frac{P_s \cdot V_A}{R \cdot T} 
\end{cases}
\]

With these equations, it is possible to simulate this system. This is discussed in the section below.
7.3.2 Simulation

The following parameters are used:

\[
\begin{align*}
V_A &= 7.0 \cdot 10^{-2} \ [\text{m}^3] & P_S &= 2.35 \cdot 10^5 \ [\text{Pa}] \\
V_B &= 0.5 \cdot 10^{-2} \ [\text{m}^3] & \alpha, \beta &= 1 \ [-] \\
m_A &= 0.15 \ [\text{kg}] & D_c &= 200 \ [\text{μm}] \\
m_B &= 0.035 \ [\text{kg}] & \end{align*}
\]

The temperature profile for this simulation is shown in Figure 58:

![Temperature profile](image)

**Figure 58: Temperature profile**

During the simulation time the mass in the chambers will change. Figure 59 shows the mass in both of the chambers and the mass flow, \( \dot{m}_{BA} \) between them. The mass flow, decreases during the time, this is caused by the decreasing pressure of chamber B. Figure 60 shows the value of \( \psi \) during the pressure ratio. As shown in this figure with these parameters the flow is 'subsonic' for almost all the 35 minutes. This means that the flow parameter \( \psi \) is constant and the mass flow is totally dependent of the pressure, in chamber B, and the temperature.

As shown in Figure 59, after 35 minutes there is still some pressurised air left in chamber B. The initial amount of air in the chambers is a 'tuning parameter'. If is chosen for a lower initial amount of air in chamber B, the flow will become subsonic earlier. This tuning process will be discussed in the next paragraph.

Figure 59 shows that the flow is sonic for almost all the time, this is because of the relative high supply pressure.
Figure 59: Mass in both chambers.

Figure 60: Flow parameter versus pressure ratio.
Figure 61 shows the pressure in the tyre and in the pressurised volume versus the time. The red line in the upper figure is the tyre pressure when there is just one volume.

![Tyre pressure graph]

Figure 61: Pressure vs. time

As shown in the figure the pressure reaches the set pressure, in this simulation, a few minutes earlier than when there is just one volume. When the pressure drops, the one-volume tyre-pressure drops relative fast, while the two chamber tyre-pressure drops slower and increases again when the temperature stays constant. Differences from almost 0.1 bar are reached.

The limitation on this method is that the ‘pressure relief valve’ in the rim has to have a larger flow capacity. Figure 62 shows the mass flow out of the tyre:
As shown in Figure 62 the maximum mass flow out of the tyre increases. Table 7 shows the maximum flow through the valve according to the simulation data. The conversion to m$^3$/hr is made according to equation 53.

$$Q[m^3/\text{hr}]=\frac{m_{\text{val}}[\text{kg} / \text{s}]}{\rho[\text{kg} / m^3]} \cdot 60 \cdot 60$$

$$\rho_{\text{air}} = \text{const} = 1.00 \left[ \frac{\text{kg}}{m^3} \right]$$

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<th>Max. Flow [kg/s]</th>
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<td>With pressurised volume</td>
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Table 7: Valve flow

As shown in Table 7 the required flows are very low, but the required flow for the rim with pressurised area is much larger as the required for the one-chamber-rim.
7.3.3 Tuning the system

The tuning parameters of this model are: the size of the pressurised chamber, the initial pressure in the pressurized chamber and the nozzle diameter. These tuning parameters will be discussed according to ceteris paribus conditions.

Varying the nozzle diameter

Initially the nozzle diameter is set to 200 µm. Slightly increasing and decreasing this diameter visualizes the effect of changing this parameter.

Figure 63: Influence of the nozzle Diameter.

Figure 63 shows the influence of the parameter D. As shown in the figure, increasing the nozzle diameter leads to a faster response at the start. Increasing the diameter results in a larger air flow, this causes a faster pressure drop in chamber B. As shown in the figure, the support during the temperature drop is less for the larger diameters. Decreasing the diameter leads to a slower response at the start, also the response after the temperature drop is slower.

When it is known at which time the pressure drop occurs (or what the maximum time on the track is) it is possible to tune the nozzle diameter.
Varying the initial pressure in chamber B

The initial pressure of chamber B is around 7 bar. Varying the initial pressure is equal to varying the initial mass in chamber B.

![Figure 64: Pressure in chamber A for different initial gas mass in chamber B.](image)

As shown in Figure 64, decreasing the initial pressure leads to a lower benefit of the system, while increasing the initial pressure does not lead to a better behaviour. This is caused by the limited gas flow through the nozzle.

Figure 65 shows the pressure versus the time in chamber B. As shown in the figure, increasing the initial mass, leads to an increase of initial pressure. The pressure drops parallel; this is also caused by the limited mass flow.

Increasing the initial mass leads (in this case) not to a better behaviour of the system.

The mass that has to be in the pressurised area can be tuned, so that there is no unnecessary mass carried in the wheels.
Figure 65: Pressure in chamber B versus the time for different initial masses.

Varying the volume of Chamber B

The volume of chamber B can also be varied. This can not be done according to the cetirus paribus conditions, because varying the volume leads to either a change in initial pressure or initial mass. Chosen is for a constant mass, so the pressure in chamber B varies.

Figure 66 shows the influence of varying the volume of chamber B on the pressure in chamber A. As shown in the figure the initial pressure does also change by changing the volume. The gradients of the lines after the start are the same, what means that it has no influence on the start, changing the volume of chamber B.

Figure 67 shows a detail plot of Figure 66, the part shown is during the temperature drop. The mass in chamber B is not changed, but the volume is increased. This causes that the pressure in the chamber is faster equal to the pressure in chamber A and the mass flow is equal to zero. There is no difference between the 5L, 4L and 3L volume. The differences in the figure are caused by the different initial pressures.

Figure 68 shows the pressure in chamber B, also in this figure the difference in initial pressure is visible.
Because of the limited flow trough the nozzle, the volume of chamber B can be made very small, but decreasing the volume results in an increasing initial pressure.
Figure 66: Varying the volume of chamber B.

Figure 67: detail plot of Figure 66
Figure 68: Influence of varying the volume of chamber B on the pressure in chamber B.
7.4 Results

The tyre pressure is dependent on the tyre gas temperature and the velocity of the car. The velocity dependency of the tyres is related to the down-force and the centrifugal forces in the tyre, these forces work in opposite direction.

With the measurement data a relation between the temperature and pressure can be found, while it is more difficult to find a (analytic) relation between the car velocity and tyre pressure. Measurement data from the car are compared with the measurements done by Michelin and they are in line.

The heating of the tyres is also dependent on the track. The front and rear tyre heat up different and the left and right tyres also. When the tyre-heaters are released, the temperature in the tyre starts decreasing. Due to this, the pressure also decreases. After a few laps the temperature and pressure start increasing. This process is different for all the tyres. These results of the heating process become more important due to next seasons regulations.

At the moment there is done some research on a pressure relief valve in the tyre. A disadvantage of a pressure relief valve is that there is less mass in the tyre at high temperatures. This causes a larger pressure drop when the temperature decreases. A (theoretical) solution for this disadvantage is found, this solution satisfies the boundary conditions: It is simple and reliable.

Due to confidentiality the comparisons and results can not be disclosed.
8 Conclusion

The goal of this traineeship was to develop several software and data analysis tools for the condition monitoring of components and sub-systems of the car. This has been done for various car-components:

With the knowledge gained by the data analysis, a simplified model of the hydraulic circuit is made to analyse the accumulator behaviour. The behaviour of the pre-charge gas pressure in the accumulators has been described by different equations of state. With these equations, analytic equations are derived to predict the initial pre-charge pressure in the LPA. These analytic equations are tested on the measurement data, after which one of these analytic equations is chosen. This chosen equation has been optimised until it satisfied the required tolerances.

Finally the model is implemented into WinTax, and the layout is made in such a way that the HPA and the LPA pre-charge pressures can be monitored quickly and even automatically.

Software is developed for easy and structural comparison of different steering setups. Various steering columns and steering racks are tested. These results are compared with test bench data. With the software, it is possible to compare the test result separately at each corner. Some of these results are also implemented in WinTax.

The pressure dependency of the tyre on the velocity and temperature is investigated. Results from the velocity dependency are compared with the Michelin data. The temperature dependency is investigated even further, which led some characteristics for the tyre pressure during a start or pit stop.

There is found a solution for a disadvantage of the pressure relief rim. This solution satisfied most of the requirements and is worked out theoretically.
9 References

Literature:

- 1) Stetighydraulik, Verslag Moderne Industrie, Moog
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- 3) Lee Catalogue
- 4) Estimation of the hydraulic circuit flow rate using WinTax data, L.Delvallez (*)
- 5) Servolenksysteme für Pkw und Nutzfahrzeuge, Verslag Moderne Industrie, ZF Lenksysteme
- 6) Grundlagen der Fluidtechnik, Teil 2: Pneumatik, H.Murrenhoff
- Fahrwerktechnik: Lenkanlagen und Hilfskraftlenkungen, Helmut Stoll
- Grundlagen der Fluidtechnik, Teil 1: Hydraulik, H.Murrenhoff
- Hydraulic check, K.Pollmeier, D.Morton, L.Delvallez (*)

* = internal reports

Used software for data analysis:

- Matlab
- WinTax
- Microsoft Excel
# Appendix A

<table>
<thead>
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<th>Track Test Name</th>
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<th>ABS Lap</th>
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Appendix B

Calculations of the initial low pressure accumulator using eq. 16 and 34, with: $T_e=413$ K, $\alpha=1.07$ and $\beta=0.18$.

Monza 410
Po_1

There is a large flow when the ‘outstanding’ point @ [42 °C, 2.2 bar] occurs.
DATA: P1 Abs3

Monza 409
Po_1

There is a large temperature drop when the ‘outstanding’ point @ [50 °C, 1.9 bar] occurs.
The temperature drop is described in the section ‘REMARK’.
DATA: Race Abs3
The temperature is under the critical temperature of 40 °C when the ‘outstanding’ point @ [35 °C, 2.1 bar] occurs.
There is a large temperature drop when the ‘outstanding’ points @ [45 °C, 2.2 bar], [48 °C, 0.8 bar], [48 °C, 1.1 bar] occurs.
The temperature drop is described in the section ‘REMARK’.
There is an input error for the HPA initial pressure @ [40 °C, 1.2 bar]
DATA (respectively): Race Abs5, Fire-up th Abs5 / Abs6 and Race Abs1
There is a leakage, when the ‘outstanding’ point @ [50 °C, 1.9 bar] occurs. See the section EXAMPLE.
Appendix C

\[ CP1_1 = \text{IFLE} (N_{144}, 0) \]
Determine if the engine is running; output is 0 when running and 1 when not running.

\[ CP1_2 = CP1_1 \times P_{HYD\_O} \]
Taking the part of \( P_{HYD\_O} \) were the engine is not running.

\[ CP1_3 = \text{INTEG} (CP1_1) \]
Integrate the samples, digital: total samples during this time.

\[ CP1_4 = \text{INTEG} (CP1_2) \]
Integrate the pressure during this part, sum-up during the time the engine is not running.

\[ CP2_1 = \text{IFEQ} (N_{144\_FILT}, 4000, 200) \times P_{HYD\_O} \]
Determine the low pressure at the moment the engine is idling, with a range of 200 rpm of idling speed.

\[ CP2_2 = \text{MEANS} (CP2_1, 225) \]
Average this part over a certain point-length, so that not every time the engine crosses this speed range is a calculation point.

\[ CP2_3 = \text{IFLE} (DT, 0.75) \times \text{IFGE} (CP2_2, CP2_1) \times \text{IFLE} (FDEM, 2) \times \text{IFGE} (T_{HYD\_FILT}, 40) \times \text{IFEQ} (N_{144\_FILT}, 4000, 200) \times P_{HYD\_O\_FILT} \]
Implementing the conditions.

\[ CPH2_1 = \text{IFEQ} (P_{HYD\_H}, 200, 10) \]
Condition for the high pressure, has to be in range.

\[ CT1_1 = CP1_1 \times T_{HYD} \]
Temperature when engines is out, see CP_2.

\[ CT1_2 = \text{INTEG} (CT1_1) \]
See CP1_4.

\[ P1 = (CP1_4/CP1_3) \times 1E5 \]
Determine the pressure at point 1.

\[ P2 = (CPH2_1 \times CP2_3) \times 1E5 \]
Determine the pressure at point 2.

\[ PH2 = P_{HYD\_H\_FILT} \times 1E5 - 1E5 \]
Set hydraulic high pressure to the right value.

\[ T1 = CT1_2/CP1_3 + 273 \]
Determine T1, by dividing sum up true samples.
\[ T_2 = 1.07 \times T_{HYD\_FILT} + 273 + 0.18 \times (T_{HYD\_FILT} - 140) \]

Formula for \( T_2 \), according to shift

\[ P_0 = \left[ \frac{(298/0.22 \times 10^{-3}) \times ((P_1 \times P_2) / (P_1 \times T_2 - P_2 \times T_1)) \times (0.075 \times 10^{-3}) - ((110 \times 10^5 \times 0.075 \times 10^{-3}) \times T_2) / (298 \times P_{H2})}{10} \right] \times 10^{-5} \]

Formula for \( P_0 \), pre-charge pressure

\[ FDEM = 2 \times CLU_{DEM} + 2 \times \text{ABS}(P_{STR\_L} - P_{STR\_R}) \]

Flow-index

\[ DT = \text{ABS} \left( \text{DERIV}(T_{HYD}) \right) \]

Determine gradient of \( T_{HYD} \)

\[ P_{HYD\_O\_FILT} = \text{MEANT}(P_{HYD\_O}, 1000) \]

Filtering the low-pressure over a wide range
Appendix D

Figure: Temperature increase of all the four wheels, Jer_03**. [Jerez T26W38FR 303 Ab 80-91]
Figure: Pressure increase of all the four wheels, Jer_03**. [Jerez T26W38FR 303 Ab 80-91]
Figure: Temperature increase of all the four wheels, Sil_04. [Silverstone T29W38TH 403 Ab 85-89]
Figure: Pressure increase of all the four wheels, Sil_04. [Silverstone T29W38TH 403 Ab 85-89]