Design of a rear axle transmission test rig

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Faculdade de Engenharia da Universidade da Porto

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I would like to thank my parents: Regina and Jorge for giving me the opportunity to study and for supporting me in all my pursuits throughout my life. I appreciate all of their continuous support and advices throughout my academic and personal journey.

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My time at FEUP was made enjoyable in large part due to the many friends that became a part of my life. I am grateful for the time spent on extracurricular activities like sports and parties as well as leisure time.
Vehicle fuel economy relies on different variables like mechanical transmissions. The rear axle is one of the main sources of power loss in the transmission, and hence represents an area where improvements can have a significant impact on the overall vehicle fuel economy. Improvements at the component level include applying higher efficiency lubricants.

In this purpose, CETRIB (Unidade de tribologia, vibrações e manutenção industrial) has interest in manufacture a test bench to evaluate these losses with different types of lubricants with different torques and temperature conditions.

This study describes the mechanical and automation designs of a rear axle transmission test rig.

Objectively, this test rig measures the torque applied at the differential propeller shaft and the torque at the drive shafts. With this values, it is possible to calculate the torque losses during the transmission.

A BMW 318d Saloon was used as a reference to this research because it represents the rear-axle transmissions niche market, diesel engines are an increasingly popular choice amongst european consumers and its engine power is within the average of 2015 passenger cars registrations. Its powertrain was characterized and the power and torque requirements of the test bench were consequently mapped.

Two types of drive systems were discussed: hydraulic drive system and electrical drive system. Considering this application, the electrical drive system was considered the most appropriated due to its efficiency compared to the hydraulic systems for the same power ranges.

Some components were selected, particularly: synchronous motors, drives, line module, torque transducers and couplings while others were designed during this work.

A mechanical structure was also designed.

Finally, apart from the selection of a power architecture and the correspondent choice of the necessary components for its implementation, an automation system architecture was designed based on LabView using a Profinet network to integrate all the components like a PLC and analog data acquisition modules.
Resumo

A economia de combustível de um veículo depende de diversas variáveis tais como transmissões mecânicas.

O diferencial é uma das maiores fontes de perda de potência na cadeia de transmissão existente num veículo e, portanto, representa uma área onde possíveis desenvolvimentos podem causar impactos positivos para a economia de combustível e consequente menor emissão de CO₂ para a atmosfera.

Com este propósito, o CETRIB (Unidade de tribologia, vibrações e manutenção industrial) tem interesse em construir uma banca de ensaios onde consiga medir estas perdas de potência nos diferenciais utilizando diferentes tipos de óleos lubrificantes, sujeitando o diferencial a diferentes cargas e temperaturas de funcionamento.

Este estudo descreve a conceção mecânica e de automação de uma banca de ensaios para diferenciais traseiros.

Objectivamente, esta banca mede o binário aplicado no veio de accionamento do diferencial e o correspondente binário transmitido aos semi-eixos. Com estes valores torna-se possível calcular as perdas de potência que ocorreram na transmissão diferencial.

A berlina BMW 318d foi utilizada como referência deste estudo dado que a BMW é uma marca que apenas fabrica carros com tração traseira, o gasóleo é o combustível em voga na EU e potência do automóvel encontra-se dentro da média da potência dos automóveis na EU em 2015. Todo o seu powertrain foi caracterizado e dessa forma foi possível mapear os requisitos de binário e potência de accionamento do diferencial para executar os testes de laboratório.

Foram discutidos dois tipos de sistemas de accionamento: hidráulico e eléctrico, sendo que o eléctrico foi considerado o mais apropriado para esta aplicação devido à sua melhor eficiência face ao accionamento hidráulico para a mesma gama de potência.

Foram seleccionados componentes para possibilitar a medição da perda de potência. A destacar: motores síncronos, amplificadores de potência para os respectivos motores, fonte de alimentação, transdutores de binário e acoplamentos. Foram também modelados outros componentes, como patas para fixação dos motores, acoplamentos, entre outros.

Fora efectuado o projecto da estrutura mecânica que visa acoplar todos os componentes seleccionados.
Finalmente, para além da concepção da arquitectura de potência e escolha dos componentes necessários à sua implementação, foi também concebida a arquitetura do sistema de automação, baseada em LabView, utilizando uma rede Profinet para integrar os diferentes componentes como um PLC e módulos de aquisição de dados analógicos.
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<tr>
<td>$F_x$</td>
<td>Tractive effort</td>
</tr>
<tr>
<td>$R_a$</td>
<td>Aerodynamic resistance</td>
</tr>
<tr>
<td>$R_g$</td>
<td>Gradient resistance</td>
</tr>
<tr>
<td>$R_r$</td>
<td>Rolling resistance</td>
</tr>
<tr>
<td>$M$</td>
<td>Vehicle’s mass</td>
</tr>
<tr>
<td>$C_d$</td>
<td>Drag coefficient</td>
</tr>
<tr>
<td>$A_f$</td>
<td>Projected area of vehicle in the direction of the travel</td>
</tr>
<tr>
<td>$\rho$</td>
<td>Volumetric mass density of air</td>
</tr>
<tr>
<td>$v$</td>
<td>Vehicle speed</td>
</tr>
<tr>
<td>$C_r$</td>
<td>Rolling resistance coefficient</td>
</tr>
<tr>
<td>$p_t$</td>
<td>Tyre pressure</td>
</tr>
<tr>
<td>$r_{min}$</td>
<td>Minimum turning radius</td>
</tr>
<tr>
<td>$t$</td>
<td>Distance between two rear wheels</td>
</tr>
<tr>
<td>$i$</td>
<td>Final drive ratio</td>
</tr>
<tr>
<td>$v_{out}$</td>
<td>Outer wheel speed</td>
</tr>
<tr>
<td>$v_{in}$</td>
<td>Inner wheel speed</td>
</tr>
<tr>
<td>$D_{out}$</td>
<td>Outer diameter of the tyre</td>
</tr>
<tr>
<td>$T_1$</td>
<td>Propeller shaft torque</td>
</tr>
<tr>
<td>$T_2, T_3$</td>
<td>Torque at each drive shaft</td>
</tr>
<tr>
<td>$n_{2,3}$</td>
<td>Angular speed of each drive shaft</td>
</tr>
<tr>
<td>Symbol</td>
<td>Description</td>
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<tr>
<td>$n_1$</td>
<td>Angular speed of propeller shaft</td>
</tr>
<tr>
<td>$P_{M1}$</td>
<td>Propeller shaft power</td>
</tr>
<tr>
<td>$P_{M2}, P_{M3}$</td>
<td>Drive shaft power</td>
</tr>
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<td>$P_{loss}$</td>
<td>Power loss in the differential transmission</td>
</tr>
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<td>$P_m$</td>
<td>Measured power applied into the differential</td>
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<tr>
<td>$P_{out}$</td>
<td>Output measured power of the differential</td>
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<td>$M_1$</td>
<td>Machine that plays the role of of gearbox</td>
</tr>
<tr>
<td>$M_2, M_3$</td>
<td>Machines that play the role of wheels</td>
</tr>
<tr>
<td>$q_v$</td>
<td>Hydraulic secondary unit flow</td>
</tr>
<tr>
<td>$P_{q_{max}}$</td>
<td>Hydraulic drive power with maximum flow</td>
</tr>
<tr>
<td>$p$</td>
<td>Operating pressure of hydraulic secondary unit</td>
</tr>
<tr>
<td>$\eta_t$</td>
<td>Total efficiency of an A4VSO variable pump</td>
</tr>
<tr>
<td>$P_{supp}$</td>
<td>Power supplied</td>
</tr>
<tr>
<td>$\eta_2$</td>
<td>Efficiency of converting electrical energy into mechanical energy</td>
</tr>
<tr>
<td>$\eta_3$</td>
<td>Line module efficiency</td>
</tr>
<tr>
<td>$\eta_4$</td>
<td>Drive+ electric motor efficiency</td>
</tr>
<tr>
<td>$J_{1_{eq}}, J_{2_{eq}}, J_{3_{eq}}$</td>
<td>Mass moment of inertia felt by machines M1, M2 and M3 respectively</td>
</tr>
<tr>
<td>$M_t$</td>
<td>Twisting moment</td>
</tr>
<tr>
<td>$G$</td>
<td>Shear modulus</td>
</tr>
<tr>
<td>$\theta$</td>
<td>Angle of twist</td>
</tr>
<tr>
<td>$K$</td>
<td>Fraction of the polar moment of inertia</td>
</tr>
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</table>
1.1 Facts about automobile industry

Europe is one of the largest producers and sellers of passenger cars. More than 70% of journeys are made by car resulting in automobile being the largest source of mobility in Europe [1].

Registrations during April 2016 grew by 9.1% compared to April 2015, see figure 1.1. This is the highest result in volume terms since April 2008, just before the economic crisis hit the automotive industry. From January to April 2016, new passenger car registrations increased by 8.5%, totaling more than 5 million units.

As shown in figure 1.2, most cars on EU roads have an internal combustion engine. So, power losses in rear-wheel-drive configurations keeps to be a focus in current researches because this configuration gives worse efficiency due to mechanical losses in the transmission [2].

In spite of most of passengers cars in Western Europe having a petrol engine, figure 1.3 discloses that Diesel engines are an increasingly popular choice amongst
1 Introduction

Figure 1.2: Passenger car fleet by fuel type

consumers with over half now opting for this type of powertrain.

Figure 1.3: Share of Diesel in New Passenger Cars in 2015

Figure 1.4 shows that European cars have become progressively more powerful over the last twenty years [3].

The most relevant passenger cars brands of European market that produce rear axle transmission cars are BMW and Mercedes-Benz [4, 5].

However, the class A, B and CLA marketed by Mercedes-Benz have front-wheel-drive configurations while BMW only have rear-wheel-drive configurations in all commercialized models.

Considering that the main goal of this study is the project and automation design of a rear differential test rig, the BMW 318d Saloon is used as a reference to this
1.2 $CO_2$ emissions from new passenger cars in the EU

The growing need for improved fuel economy is also a global issue due to the relationship between reduced fuel consumption and reduced $CO_2$ emissions [7]. Presently, vehicles with internal combustion engines are by far the most popular choice of powertrain. However, concern for the environment has led to policy targets on carbon emissions both globally and at EU level, and the EU is committed to a 20% reduction in $CO_2$ emission levels by 2020 compared to the level in 1990 [8].

In transport section, this is being achieved by encouraging greater emphasis on increasing the efficiency of existing means, like the differential. The higher the processes inefficiencies, the higher will be the $CO_2$ emissions.

Climate change is a global challenge which demands collective action and international cooperation. The automotive sector is playing a leading role, embracing its responsibility to reduce $CO_2$ emissions from vehicles and production, this sector faces continuously tightening legislation to improve fuel efficiency and reduce emissions [8] [7].

Laboratory tests ensure that cars comply with EU limits for both $CO_2$ and pollutant emissions, before they are put on the market. This lab tests are called driving cycles. The main objective to create this kind of tests was to estimate emissions of air pollutants from road vehicles and reduce them. All conditions for vehicle set-up,
testing and the handling of test results for cars are defined by EU law. By this way, a standard regulation has to be respected by all car manufactures and other players. Additionally, it allows for a repeatable procedure which enables customers to compare emissions between different car models.

As there is no ‘one’ real-world fuel consumption value, only figures measured in lab tests using standardized cycles allow customers to make comparisons between vehicles in terms of their \( \text{CO}_2 \) emissions and fuel consumption [9].

1.3 Driving cycles

A driving cycle is a fixed "vehicle speed profile" which allows an emission test to be conducted under specific conditions. Driving cycles are usually defined in terms of vehicle speed and time [10].

As in European Union, in America and Japan, emission tests are required by law. Light-duty vehicles are tested using a power-absorbing chassis dynamometer. In these tests, a trained driver has to follow the driving cycle on the chassis dynamometer, and a ‘driver’s aid’ is provided to ensure that the driven cycle is as close as possible to the defined cycle.

Depending on the speed profile and load, cycles can be divided in two groups:

- 'steady-state' cycles:
- 'transient' cycles.

The "steady-state" cycles have a sequence of a constant speed and torque. In transient cycles, the vehicle speed and engine load are continuously changing [10].

1.3.1 New European Driving Cycle

In the European Union, the NEDC (New European Driving Cycle) is the most popular and is used today to measure car emissions driving cycle. It is a "transient" cycle which includes two sub-cycles: an urban, ECE, repeated four times; and one extra-urban sub-cycle, EUDC, presented in figure 1.5.

The main characteristics of this cycle are:

- Total distance: 11 016.13 m;
- Driving time: 1180 s;
- Maximum speed: 120.09 km/h;
- Average speed: 42.34 km/h.
1.3 Driving cycles

Figure 1.5: New European Driving Cycle

This driving cycle has periods of constant speed and acceleration which bears little relation to real driving profiles on the road.

With this kind of variations in speed, the engine has a better performance and in consequence, emits less polluting gases than a real-world cycles, that are generally much more transient than stylized cycles such as the NEDC.

The NEDC was developed to represent the typical usage of car in Europe. However, nowadays the industry recognizes and has stated consistently that the current NEDC test cycle is outdated. As it was designed in the 1980’s and deployed from the 1990’s, NEDC does not account for many new technologies in today’s cars that have a strong effect on fuel consumption [9].

1.3.2 Worldwide Harmonized Light Vehicles Test Procedure

Because of the shortcomings of the current car emissions test, the automotive industry is actively contributing to the development of the new global test cycle, called the Worldwide Harmonized Light Vehicle Test Procedure (WLTP) [9].

This new test cycle will include higher speeds, more dynamic and representative accelerations and decelerations, and stricter vehicle set-up and measurement conditions than the NEDC like tyre pressure, gear shift strategy for manual transmission vehicles, warm-up, etc. As all these elements will make the WLTP test more accurate than the current lab test, it will provide a better reflection of today’s drive situation [11].

The Worldwide Harmonized Light-duty Test Cycle is presented on figure 1.6

Table 1.1 quantifies the main descriptive parameters of NEDC and WLTC driving cycles.
1 Introduction

Table 1.1: Descriptive parameters of the driving cycles NEDC and WLTC

<table>
<thead>
<tr>
<th>Units</th>
<th>NEDC</th>
<th>WLTC</th>
</tr>
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<tbody>
<tr>
<td>Duration s</td>
<td>1180</td>
<td>1800</td>
</tr>
<tr>
<td>Distance km</td>
<td>11.03</td>
<td>23.27</td>
</tr>
<tr>
<td>Average velocity km/h</td>
<td>33.6</td>
<td>46.5</td>
</tr>
<tr>
<td>Max. velocity km/h</td>
<td>120</td>
<td>131.3</td>
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**Durations**

<table>
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<th></th>
<th>NEDC</th>
<th>WLTC</th>
</tr>
</thead>
<tbody>
<tr>
<td>Constant driving s</td>
<td>476</td>
<td>66</td>
</tr>
<tr>
<td>Acceleration s</td>
<td>247</td>
<td>789</td>
</tr>
<tr>
<td>Deceleration s</td>
<td>178</td>
<td>719</td>
</tr>
</tbody>
</table>

**Shares**

<table>
<thead>
<tr>
<th></th>
<th>NEDC</th>
<th>WLTC</th>
</tr>
</thead>
<tbody>
<tr>
<td>Constant driving</td>
<td>40.3%</td>
<td>3.7%</td>
</tr>
<tr>
<td>Acceleration</td>
<td>20.9%</td>
<td>43.8%</td>
</tr>
<tr>
<td>Deceleration</td>
<td>15.1%</td>
<td>39.9%</td>
</tr>
</tbody>
</table>

1.3.3 Common Artemis Driving Cycles

The Common Artemis Driving Cycles (CADC) are chassis dynamometer procedures developed within the European Artemis (Assessment and Reliability of Transport Emission Models and Inventory Systems) project, based on statistical analysis of a large database of European real world driving patterns [12].

These cycles include not only urban and rural driving conditions but also motorway. The motorway cycle has two variants with maximum speeds of 130 and 150 km/h.

This cycle is defined by several parameters: speed average, stop frequency, driving time duration among others as presented in figure 1.7.

The main descriptive parameters of Artemis driving cycle are:

- Total distance: 50 886.36 m;
- Driving time: 3143 s;
1.4 Tractive effort

The tractive effort, $F_x$, is the force required to move forward a vehicle.

The tractive resistance is defined by the opposing force that the tractive effort must overcome to create movement. The tractive resistance is composed by two categories: the steady-state resistance and the dynamic resistance. The steady-state resistance occurs when the vehicle is moving through at constant velocity. It can be divided in three resistances:

- Aerodynamic resistance, $R_a$: Depends on the size and shape of the vehicle and increases with the square of air velocity;

- Gradient resistance, $R_g$: Depends on the slope of the road and the weight of the vehicle;

This Artemis cycle, as WLTC, has got a more realistic drive profile and because of that is another start-point for this study.

Figure 1.7: Artemis URM130

- Number of stops: 20;
- Maximum speed: 130 km/h-150km/h;
- Average speed: 58.3 km/h.
1 Introduction

- Rolling resistance, $R_r$: Depends on type of road pavement, vehicle’s weight, tyres and vehicle speed.

The dynamic resistance occurs in addition to the steady-state resistance when the vehicle is accelerating or braking and is caused by the vehicle’s inertia [13].

Figure 1.8: Forces acting on a vehicle when in movement

Figure 1.8 presents a vehicle’s free body diagram, where the forces described above are presented in their point of application.

Assuming the vehicle’s moves along the $xx$ direction, the Newton’s second law establishes that,

$$\sum F_x = Ma_x \quad (1.1)$$

$$F_x = R_a + R_g + R_r + Ma_x \quad (1.2)$$

The equation (1.2) defines the tractive effort, $F_x$, where $M$ is the vehicle’s mass.

1.4.1 Aerodynamic resistance

Under moderate to higher speeds, the aerodynamic resistance has a preponderant effect on the vehicle’s power requirements as this force increases with the square of the speed [14].

Usually it is expressed as,

$$R_a = \frac{1}{2} C_d A_f \rho v^2 \quad (1.3)$$
where,

- $C_d$ is the drag coefficient;
- $A_f$ is the projected area of the vehicle in the direction of travel;
- $\rho$ is the volumetric mass density of the air;
- $v$ is the vehicle speed relative to the wind.

1.4.2 Gradient resistance

The gradient resistance is the force that relates to the pavement road slope and the weight of the car. This force is applied in the center of gravity of the vehicle [15].

It is defined as,

$$R_g = Mg \sin \theta$$  \hspace{1cm} (1.4)

1.4.3 Rolling resistance

The rolling resistance of tires is mainly consequence of the tyre deformation when rolling. It also arises from friction between the tyre and road, the resistance due to air circulating inside and outside the tyre [14].

This force is defined as [16],

$$R_r = C_r Mg \cos \theta$$  \hspace{1cm} (1.5)

where the rolling resistance coefficient of a pneumatic tire $C_r$, in a dry surface is calculated as [16],

$$C_r = 0.005 + \frac{1}{p_t} (0.01 + 0.0095 \left( \frac{v}{100} \right)^2)$$  \hspace{1cm} (1.6)

where $p_t$ is the tyre pressure in [bar], and $v$ is the vehicle speed in [km/h].

1.5 Research scope

Vehicle fuel economy relies on many different variables, such as vehicle’s weight, aerodynamics, engine, driveline, rolling resistance, transmissions and accessory load.

As shown in figure 1.9, an internal combustion engine loses energy through process inefficiencies, thermal loses, and friction loses. A vehicle’s drivetrain loses energy mainly by friction in the gears and bearings. Only about $13 - 25\%$ of the fuel
chemical energy is actually used to move the vehicle at a required speed together with running useful ancillary equipment, like the air conditioning [16].

For both urban and highway driving, the mechanical energy that does make its way through the driveline to turn the wheels is consumed by three greatnesses: aerodynamic drag, rolling resistance, and braking. Braking consumes momentum from the vehicle, which must be replenished by acceleration [16].

Braking is a major consumer of mechanical energy during urban driving. In contrast, aerodynamic drag consumes relatively more energy during highway driving since this resistive force rises with vehicle speed.

Improvements at the component level include the use of higher efficiency lubricants. Historically, reducing oil viscosity has resulted in improved fuel economy. However, lower viscosities can lead to reduced lubricants films, which may fail to hold up under higher temperatures and heavy loading associated with axle operations [16].

The rear axle is one of the main sources of power loss in the transmission, and hence represents an area where lubricant improvements can have a significant impact on the overall vehicle fuel economy. The fluid development challenge is to bridge the gap between fuel economy and operating temperature control [7].

1.6 Axle efficiency test rigs

After a research of the state of the art, namely, in SAE (Society of Automotive Engineers) technical papers, it was possible to verify some topologies of the axle efficiency test rigs utilized in several studies.

In 1980 a test stand was developed in order to evaluate the axle efficiency. Figure 1.10 shows a layout diagram of the equipment utilized in a axle box test bench. In there, an automotive engine, equipped with a three mode proportional electronic
speed control governor, is used as the main power source. The power from the engine is transferred to the rear axle pinion through a four-speed manual transmission and a driveshaft equipped with a rotary transformer, strain gage type, torque sensor.

Figure 1.10: Axle box test bench diagram - differential gears locked

The test axle’s differential gears are locked in order to allow all power to be transferred through one axle shaft.

A second torque sensor is mounted on this shaft which is coupled to a dynamometer equipped with an automatic load controller [17].

Figure 1.11 presents an outline of an axle efficiency test rig, were the axle is equipped with a thermocouple mounted into the housing. The test rig is also equipped with torques transducers that are capable of providing accurately measures, as well as load cells which measure the force applied into the driveshafts by the dynamometer [18].

Figure 1.11: Axle box test bench diagram - differential gears unlocked
1 Introduction

1.7 Thesis outline

Chapter 1: Introduction has presented a brief introduction about automobile industry and its impact in $CO_2$ emissions. To measure these emissions, the laboratory tests were also presented. It is also presented the state of the art of rear axle transmissions test benches.

Chapter 2: Powertrain characterization according to the automobile industry study made in previous chapter, an BMW 318d was taken as an input data of the present study and its powertrain was characterized. The power and torque requirements are defined on the basis of the laboratory test as well as speed, torque and temperature data accuracy according to CETRIB specifications.

Chapter 3: Power architecture A numerical comparison between the electrical power input when using test bench with breaking architecture or a drive system with energy recovery test bench is made. Also two solutions of drive systems with energy recovery: electrical drive systems and hydraulic secondary control drive systems are discussed. The power flow between several conditions of rear axle transmissions (acceleration, deceleration and curve) is defined.

Chapter 4: Components selection In order to accomplish the accuracy requirements imposed by CETRIB and the power and torque needful for executing the tests, in this chapter all the components needed to achieve the main goal are selected: electric motors (motor modules, line module and control unit), torque transducers, couplings, a PLC and analog modules for doing the data acquisition.

Chapter 5: Mechanical design Is presented an overview of the final mechanical solution design for integrate the components selected in previous chapter. Its assembly is described as well as the development of some parts designed. The mechanical solution is validated taking into account the main efforts applied into it.

Chapter 6: Automation project design Two types of control architecture are discussed, both based on LabView software.

Chapter 7: Conclusion and future works presents the conclusions of this thesis as well as purposes future works.
As said before, the BMW 318d Saloon is used as a reference for this project. In this section the engine and the driveline (gearbox and differential) will be characterized to predict power and torque requirements at the wheels.

2.1 Types of power transmission systems

The power transmission system or powertrain may have three different configurations: front-wheel drive, rear-wheel drive or four-wheel drive, figure 2.1 [19, 20].

As shown in figure 2.2 the vehicle is composed by: the engine, the clutch, the gearbox, the propeller shaft, the universal joints, the differential, the drive shafts and the tyres.

The major advantage of four-wheel to rear-wheel drive is that it spreads the road loads of the vehicle across all four tyres. In a rear-wheel drive vehicle, the rear wheels do the driving while the front wheels are used for the steering duties [21]. The BMW 318d Saloon is equipped with a rear-wheel-drive power transmission configuration [6].
2 Powertrain characterization

Figure 2.1: Main types of power transmission systems

2.2 Engine

Torque and power are different measurements of the engine performance, with torque referring to the engine’s ability to create turning power and power being the ability of the engine to do work over a period of time [21].

The Diesel engine of the vehicle considered in this work has 2.0 liter capacity and 4 cylinders [6].

The engine torque over the engine’s speed range, as presented in figure 2.3, is variable. It delivers a maximum torque of 320 Nm between 1750 up to 2550 rpm [6].

The power delivered by the engine can be calculated in the basis of the indicated power or brake power. The indicated power is the power actually produced in the cylinder. The brake power is the power measured at the crankshaft.

The curve commonly starts to increase with the crankshaft speed and about the seven-eighths of the speed range reaches the maximum power value, and starts to
2.2 Engine

decrease, as shown in figure 2.3. The decline in power at the upper speed range is mainly due to the short time available for inducing fresh charge into the cylinders at very high speeds.

The engine torque and power are detailed in table 2.1 [6].

![Figure 2.3: Torque and power characteristic curves of BMW 18d](image)

**Table 2.1: BMW 318d engine characteristics**

<table>
<thead>
<tr>
<th>Engine</th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Cylinders/valves</td>
<td>-/-</td>
<td>4/4</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Capacity</td>
<td>cm³</td>
<td>1995</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Stroke/bore</td>
<td>mm/mm</td>
<td>90/84</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Max output power</td>
<td>kW</td>
<td>105</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Max torque</td>
<td>Nm/rpm</td>
<td>320</td>
<td>1750-2500</td>
<td></td>
</tr>
<tr>
<td>Compression ratio</td>
<td>: 1</td>
<td>16.5</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
2.3 Driveline

The driveline is composed by the gearbox, the propeller shaft and joints, the differential and joints, the drive axes and the tyres. They take the engine power and deliver it to the wheels [21].

2.3.1 Gearbox

The internal combustion engine has many advantages: like high power to weight ratio, reasonable good efficiency and energy storage. However, this kind of engine has also some disadvantages:

- no torque generation at zero speed;
- maximum power only on a small speed range at a certain speed also, figure 2.3;
- fuel consumption is dependent on operating conditions (torque, speed and temperature) on the working point of the engine.

At the maximum engine power and vehicle road speed, the ideal traction hyperbola, $F_{Z,Aid}$ and the effective traction hyperbola, $F_{Z,Aid}$ can be calculated as:

$$F_{Z,Aid} = \frac{P_{max}}{v}$$

$$F_{Z, Ae} = \frac{P_{max}}{v} \eta_{tot}$$

![Diagram showing ideal and effective traction hyperbola](image)

Figure 2.4: Ideal and effective traction hyperbola; A typical traction profile available in a internal combustion engine.
2.3 Driveline

The maximum traction between tyres and road is limited by the friction limit.
As shown in figure 2.4, the shaded area cannot be used without a gearbox. This
output converter adapts the characteristic of the combustion engine as closely as
possible to the ideal of the traction hyperbola.
The gearbox and the gear ratios values adapt engine traction curve to the corre-
spondent traction as presented in figure 2.5 [15].

Figure 2.5: Traction diagram of a 4-speed gearbox.

There are three main groups of gearboxes: manual, automatic, which includes
the dual clutch (DCT) and lastly, the continuously variable transmissions (CVT).
Manual gearboxes require the driver to be continuously selecting the most appro-
priate gear ratio. The automatic gearboxes are able to make the selection in an
automatic way. The driver just has to select which direction to follow [13]. DCT
is an automated manual transmission which uses two separate clutches, one of each
odd and even gear sets [22]. CVT have now become more popular because provide
improved economy and lower emissions. This type of gearbox don’t have steps in
between gear changes [13].
The BMW 318d under analysis is equipped with a 6-speed manual gearbox. The
correspondent gear ratios are shown in table 2.2 [6].
2 Powertrain characterization

Table 2.2: BMW 318d Saloon Gearbox ratios

<table>
<thead>
<tr>
<th>Gearbox ratios</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>I</td>
<td>4.002</td>
</tr>
<tr>
<td>II</td>
<td>2.109</td>
</tr>
<tr>
<td>III</td>
<td>1.396</td>
</tr>
<tr>
<td>IV</td>
<td>1.000</td>
</tr>
<tr>
<td>V</td>
<td>0.781</td>
</tr>
<tr>
<td>VI</td>
<td>0.668</td>
</tr>
</tbody>
</table>

Using the torque characteristic shown on figure 2.3 and dividing each point’s for the gear ratios (table 2.2), the traction curve available in BMW 318d series can be defined as presented in figure 2.6.

Figure 2.6: Traction effort available on BMW 318d

2.3.2 Final drive

The final-drive gears, also designated as axle box or differential, are used in the transmission system for redirecting the drive from the gearbox or propeller shaft through 90° and to provide a permanent gear reduction between the engine and the
driving road-wheels. The final-drive gears may be directly or indirectly driven from the output gearing of the gearbox \[19, 20\].

Indirectly driven final drives are used at the rear of the vehicle. The final drive gears may be bevel or worm gears.

The geometry of bevel gears can be approximated by two friction cones: the crown wheel and the pinion, figure 2.7a. There are three main types of bevel gears: spur, spiral and hypoid.

The spur bevel gears are cheap to produce and mechanically more efficient. However, are noisy when meshed. The spiral bevel gears (or helical bevel gears) are used to reduce the noise generated by the spur bevel gears, figure 2.7b.

In passenger cars, hypoid gears are usually used. In this case the bevel drive pinion engages above the axial center of the crown gear, as shown in figure 2.7c. This offset is always placed below to give a lower propeller shaft and a reduction in the tunnel height. This offset makes the diameter of the bevel drive pinion larger, and the crown gear can be smaller for the same load than in helical bevel gears. The increased sliding friction between the tooth flanks contributes substantially to reducing noise \[15\].

Worm gears are now rarely used, mainly because of their extremely low efficiency.

### 2.3.3 The need for final-drive differential gearing

A given turning circle of a vehicle may be defined as the average circumferential distance moved by the vehicle with the driver’s steering-wheel held at some point to one side of the central straight-ahead position. In actual fact, the paths followed by the inner and outer road-wheels will be slightly different.

Figure 2.8 shows that the outer wheels will have a larger turning radius than the inner wheels about some common center. This means that the outer wheels will travel a greater distance than the inner ones. To make this happen at the same time, the outer wheels have to rotate faster than the inner ones.

The differential mechanism included in the axle box or differential allows the wheels to rotate at different speeds, but still maintains the same torque into both wheels \[19\].
2 Powertrain characterization

(a) Straight bevel gears
(b) Spiral bevel gears
(c) Hypoid bevel gears

Figure 2.7: Types of bevel gears

Turning radius, $r$

\[ \omega \]

\[ v \]

\[ v_{\text{in}} \]

\[ v_{\text{out}} \]

Track, $t$

\[ \delta \]

Figure 2.8: The need for differential
There are different types of differentials:

- open differentials;
- lock differentials;
- limited slip differentials;
- torsen limited slip differentials.

The most common type is the open differential because it’s the most inexpensive, has the light weight and provides lower maintenance.

The main disadvantage of the open differential is that the engine power follows the path of the least resistance, which means that in case of one wheel has lower traction than the other, for example, when slips in a ice pavement, the other wheel will not have torque enough to move the vehicle forward. The other types of differentials are used in order to solve this problem.

A locking differential may provide increased traction compared to a standard/open differential by restricting each of the two wheels on an axle to the same rotational speed without regard to available traction or differences in resistance seen at each wheel. The LSD (limited slip differential), as its name suggests allows for some difference in angular velocity of the output shafts, but imposes a mechanical limit on this disparity.

The BMW 318d under analysis has an open differential and the corresponding final drive ratio, $i$, is 3.231 [6]. The traction curve at the wheels is presented below in figure 2.9.
Knowing the vehicle velocity, $v$, the minimum turning radius, $r_{\text{min}}$, and the wheel track, i.e., the distance between two road wheels on the same axle, $t$, it is possible to evaluate the maximum speed difference between the outer and the inner wheels.

According to figure 2.8, the vehicle speed is defined as,

$$ v = w \tau $$

Thus,

$$ v_{\text{out}} = w r_{\text{out}} $$

$$ v_{\text{in}} = w r_{\text{in}} $$

Where $r_{\text{out}}$ and $r_{\text{min}}$ are defined by,

$$ r_{\text{out}} = r + t/2 $$

$$ r_{\text{in}} = r - t/2 $$

Defining $\varepsilon$ as,

$$ \varepsilon = \frac{v_{\text{out}} - v_{\text{in}}}{v_{\text{out}}} \times 100 $$
2.4 Real BMW E46 differential

\[ \varepsilon = \frac{t}{\frac{r_{\text{min}} + t}{2}} \]  

(2.5)

Using equation (2.4) with \( r_{\text{min}} = 11.3/2m \) and \( t = 1.582m \),

\[ \varepsilon = 24.567\% \]

So, in the case of BMW 318d, the wider gap between the outer and inner wheels rotations speeds has a value of 24.567%, i.e., the inner wheels rotates 24.567% less than the outer wheels.

### 2.4 Real BMW E46 differential

As the goal of the present study is to design a rear axle transmission test rig, having a real rear differential is a crucial point because with it, the main dimensions may be known (in order to design a mechanical structure) as well as its inertia.

The college had a BMW E46 (produced between 1998 to 2006) of a 320d differential, and based on the purpose exposed above, it was disassembled, see figure 2.10, and its inertia was calculated.

The references of each differential component are presented on table 2.3.

<table>
<thead>
<tr>
<th>Component</th>
<th>Brand</th>
<th>Ref.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Drive shafts bearings</td>
<td>FAG</td>
<td>H01298_T29</td>
</tr>
<tr>
<td>Propeller shaft bearings next to the pinion</td>
<td>KOYO</td>
<td>HM 801310</td>
</tr>
<tr>
<td></td>
<td>KOYO</td>
<td>HM 88542</td>
</tr>
<tr>
<td>Drive shafts seals</td>
<td>BMW</td>
<td>1214 323 CFW 2 44-90.1 10/15</td>
</tr>
<tr>
<td>Propeller shaft seals</td>
<td>BMW</td>
<td>1214 666 ACM RECHTS KACO DSS 45 10/14 RTD F01 101</td>
</tr>
<tr>
<td>Crown wheel</td>
<td>H47</td>
<td>YKB 946 0622</td>
</tr>
</tbody>
</table>

The differential pinion has, in this case, 14 teeth and the crown wheel 43. So this particular final drive ratio \( i \) is \( i = 43/14, i = 3.07 \).

Measuring the dimensions and mass of the all moving components, the correspondent polar mass moment of inertia in \( x \) (axis of the propeller shaft) and \( y \) axis (axis of the drive shafts), \( J_x \) and \( J_y \), respectively, was calculated. The index of the terms presented in equations 2.6 and 2.8 are referred in figure 2.11.

\[ J_x = J_1 + \frac{J_3 + J_4 + J_5 + 2 \times (J_6 + J_7)}{i^2} \]  

(2.6)

\[ J_x = 0.032kgm^2 \]  

(2.7)
assum ing that the drive shafts will rotate at exactly the same speed $J_y$ was calculated using equation \(2.8\).

\[
J_y = J_2 + J_3 + J_5 + J_6 + J_4 + J_1 \times i^2 \tag{2.8}
\]

Substituting the values in equation \(2.8\),

\[
J_y = 0.054 kgm^2
\]

2.5 Requirements definition

2.5.1 Tyre dimensions

Tyres have standardized dimensions that are presented in figure \(2.12\) [21].
The vehicle under analysis has a 225/50 R17 tyre. Which means:

- 225 is the tyre width in \(mm\);
- 50 is the height-to-width ratio as a percentage;
- R is the construction type (in this case, radial);
- 17 is the rim diameter in \(in\) (\(1in = 25.4\,mm = 25.4 \times 10^{-3}\,m\)).

The outer diameter of the tyre, \(D_{out}\), is,
2.5.2 Power and torque requirements

Knowing the tyre dimensions, assuming a speed range from 20\(km/h\) to 140\(km/h\), and a 5\% steepness of the path, using equations (1.2), (1.3), (1.4) and (1.5) it is possible to map the torque and power requirements for the test rig.

The torque that has to be applied in the propeller shaft, \(T_1\) is defined as,

\[
T_1 = \frac{F_x D_{out}}{2i}
\]

assuming an efficiency equal to 100\% the correspondent torque in each drive shaft, \(T_2\) and \(T_3\) is calculated as,

\[
T_2 = T_3 = \frac{F_x D_{out}}{2}
\]

or

\[
T_2 = T_3 = \frac{T_1}{2} \times i
\]

To simulate the speed range selected with regard to the tire size, the angular velocity (when traveling through a straight line), \(n_2\) and \(n_3\), of the drive shafts must be calculated as,

\[
n_2 = n_3 = \frac{60v}{\pi D_{out}}
\]

and the correspondent angular velocity of the propeller shaft is,

\[
n_1 = n_2 \times i = n_3 \times i
\]

The power needed in the propeller shaft, \(P_{M1}\) is calculated as,

\[
P_{M1} = T_1 \times \frac{2\pi n_1}{60}
\]

The correspondent power in each drive shaft, \(P_{M2}\) and \(P_{M3}\) is defined as (when traveling through a straight line),
\[ P_{M2} = T_2 \times \frac{2\pi n_2}{60} = T_3 \times \frac{2\pi n_3}{60} \] (2.14)

Figure 2.13 shows the values mentioned above. However, when turning the power requirement for the outer wheel can be 25\% higher than the inner.
2 Powertrain characterization

Figure 2.13: Power and torque requirements for propeller shaft and drive shafts
As can be observed, the maximum torque for the drive shafts is about 270 Nm and for the propeller shaft, 170 Nm at 3700 rpm and 1200 rpm, respectively.

The maximum power for the drive shafts is about 35 kW and 65 kW at 3700 rpm and 1200 rpm, respectively also.

2.6 Measuring system

To quantify the power losses in the differential, the test bench must have tachometers in order to know the speed in each axle, \( n_1, n_2 \) and \( n_3 \). Moreover, a torque transducer has to be allocated in each axle (propeller and drive shafts) to measure \( T_1, T_2 \) and \( T_3 \).

Thus, the power loss, \( P_{\text{loss}} \) can be calculated as:

\[
P_{\text{loss}} = P_{\text{in}} - P_{\text{out}}
\]  

(2.15)

where in case of accelerating,

\[
P_{\text{in}} = T_1 \times \frac{\pi n_1}{30}
\]  

(2.16)

and

\[
P_{\text{out}} = T_2 \times \frac{\pi n_2}{30} + T_3 \times \frac{\pi n_3}{30}
\]  

(2.17)

in case of decelerating,

\[
P_{\text{in}} = T_2 \times \frac{\pi n_2}{30} + T_3 \times \frac{\pi n_3}{30}
\]  

(2.18)

and

\[
P_{\text{out}} = T_1 \times \frac{\pi n_1}{30}
\]  

(2.19)

As shown before the speed range that the tachometers should cover is around 0 to 3700 rpm with a maximum uncertainty of less than ±1 rpm.

The torque range that the torque transducers should cover is around 0 to 300 Nm with a maximum uncertainty of less than ±0.5 Nm.

Finally, temperature sensors to monitoring the temperature have to be allocated in several positions:

- Lubricant temperature at pinion gear;
- Case wall;
- Lubricant temperature;
- Ambient temperature.
2 Powertrain characterization

The requirements imposed by CETRIB was that the range of these measurements should be between 0 °C to 200 °C with a maximum uncertainty of less than ±1 °C. These torque and speed transducers will be selected in chapter 4 and the temperature sensors in chapter 6.
The global economy’s growth results in a continuously expanding demand for energy. However, this causes an increasing rise in \( CO_2 \) emissions, resulting in climate change, whose progress and effects are not fully foreseeable at present. To preserve the future of our planet, it is necessary to eliminate the consequences of climate change \[23\].

Furthermore, as energy prices rise, increasing this cost factor will have a negative impact on production costs in the industrial sector.

Energy-efficiency is one of the solutions for industries who want to be competitive in the future. They can help to reduce manufacture costs, improve the return of their investment and lower \( CO_2 \) emissions \[24\].

The differential should be tested in several road conditions: different directions of travel (move back, forth and curve) and when simulating this kind of situations, the acceleration or deceleration modes should be defined. The power flux in this two last mentioned situations, changes.

In this chapter, the drive systems possibilities to execute these tests will be discussed.

### 3.1 Brake test bench vs test bench with energy recovery

Three machines (M1, M2 and M3), are necessary to simulate the propeller shaft (output of the driveline) and the wheels, figure \[3.1\].

The machine that plays the role of gearbox, M1, must be able to produce the correspondent speed and torque provided from it. Furthermore, machines M2 and M3, should simulate the load actually borne by the wheels of the vehicle.

These three machines must have the possibility to be speed or torque controlled.

To simulate corners, M2 and M3, which play the role of wheels must be speed controlled. Imposing this parameter, turns to the right side or to the left side can be simulated, by means of giving more or less speed to the driveshafts. A straight trajectory can be simulated, imposing the same speed in M2 and M3. On the other
3 Power architecture

![Power Transmission Scheme](image)

Figure 3.1: Power transmission’s scheme

hand, machine M1, should be torque controlled.

As described in figure 2.13 presented on chapter 2.5, the maximum power required for the propeller shaft is $63.5 \text{kW}$.

If this power was consumed by the two brake dynamometers represented by machines M2 and M3, all of this power could not be reused and will be converted into heat, figure 3.2. The numbers presented in this figure are examples of possible inefficiencies [25].

![Brake Test Bench](image)

Figure 3.2: Brake test bench

Moreover, as the tests have a duration of 12 hours and nowadays the kWh price is around $0.3095 \text{€/kWh}$ this solution is economically untenable [26]. Furthermore, implementing the solution with two, M2 and M3, dynamometers the differential deceleration test can not be executed. In this case, the load should be imposed by M1 and because of that the dynamometer should be allocated in there.

In order to provide a more economically viable and sustainable solution, a search of different systems types that can provide energy-saving in industrial sectors was made. Using this type of solution, the drive system only require to get 30% of
the required power from the electric network, figure 3.3. Once again, the numbers referred to in figure 3.3 are purely qualitative, since they are not based in the application of the present study.

![Test bench with energy recovery](image)

In this test rig situation, the secondary control drive systems and the electrical drive systems can be noted as the most relevant.

### 3.2 Drive systems with energy recovery

#### 3.2.1 Hydraulic secondary control drive systems

The secondary control drive systems have been developed and are marketed by Rexroth Bosch Group. The secondary control has high dynamics for setting up closed-loop speed, position or torque controls with energy recovery.

This kind of systems includes at least one hydraulic pump - the primary unit, an hydraulic accumulator and a consumer - the secondary unit that work at one supply network with constant pressure, figure 3.4. These machines are called axial piston variable pumps A4VSO. They are designed with a swash plate for hydrostatic drives. Its flow is proportional to the input drive speed and the plate displacement.

The hydraulic accumulator stores the energy fed by the consumer when working in reversed direction. When working under load, this accumulated energy is used to satisfy the consumption peaks.

To control the axial piston units there are digital controllers called HNC100-SEK. They are suitable for the closed-loop speed and torque control as well as the open-loop torque control.
3.2.2 Electrical drive systems

As in secondary control systems, the electrical drive systems have a primary unit, an accumulator and consumers too.

The power supply is the power unit, the accumulator role is performed by the DC capacitors and the consumers are the motors working in motoring mode; when working in regenerative mode, the energy produced in each drive/motor module can be fed back to the DC link instead of being converted into heat in a brake resistor.

In this case, instead of taking energy from the grid there is a direct energy equalisation between several different drives in an interlinked system, figure 3.5. [23]
3.2 Drive systems with energy recovery

Figure 3.5: Energy equalisation between several drives

Siemens is a brand that has drives capable of providing this technology.

The Simotion Motion Control system provided by Siemens is combined with the Sinamics S120 drive systems and Simotics motors to create energy-efficient solutions which help to achieve considerable reductions in energy consumption. To allow efficient energy processes a previous energy management has to be done. This process involves three phases: identification, evaluation and implementing.

Identification provides the recognition of energy flows. After identification, the energy consumption has to be calculated - evaluation. Implementing involves the selection of the common DC link, which will save the energy, as well as power supply which will feed the power losses in the process.

The Sinamics S120 drive system is a modular system for high-performance single-axis and multi-axis drives for a very broad range of applications. These units are capable of controlling in vector, servo and V/f modes. They also perform the speed and torque control functions.

The Sinamics S120 drive system is composed by power units and control units. The control unit is selected according to the number of drives to be controlled and the required performance level. On the other hand, the power unit must be rated to satisfy the energy requirements of the application.

A motor module integrates a voltage-source DC link and an inverter for supplying a motor. They are designed for multi-axis drive systems and are interconnected by a common DC busbar. Since they have a common DC link they can exchange energy with one another as shown in figure 3.5.

The voltage-source DC link is supplied with main voltage by a line module. Siemens provides three types of line modules: the basic, the smart and active line modules. The first is not capable of feeding regenerative energy back into the supply system. The smart line modules can supply energy and return regenerative energy. However, in case of power failure, a braking module and braking resistor are required if the drives have to be decelerated in a controlled manner. The active line modules have the same characteristics mentioned in the smart line modules but, they can
generate a regulated DC voltage which remains constant despite fluctuations in the line voltage is shown in figure 3.6 [30].

![Sinamics S120 drive system overview](image)

Figure 3.6: Sinamics S120 drive system overview

### 3.3 Power flow

The power flux is not defined until now, i.e., the machines that are working in motoring or regenerative mode have to change their function between each other, depending on what’s being tested: acceleration or deceleration.

When testing the differential in acceleration mode, M1 has to work as motoring and the machines M2 and M3 in regenerative mode, see figure 3.7a.

In the other hand, in deceleration test, M2 and M3 should work in motoring mode and M1 as regenerative mode, figure 3.7b.

### 3.4 Electrical input power estimation

In order to select the most appropriated drive system solution for this application, an estimation of the electrical input power for both cases was made.

#### 3.4.1 Hydraulic secondary control drive systems

Considering the differential in an acceleration test mode (with M2 and M3 at the same speed) and its efficiency equal to 100%, as described in previous section, M1
3.4 Electrical input power estimation

![Diagram of electrical machines operating modes in acceleration and desacceleration]

Figure 3.7: Electrical machines operating modes in acceleration and desacceleration

is working as motoring mode, M2 and M3 as regeneration mode.

In order to know the power that has to be supplied in this situation, the efficiency of the units for these power ranges must be known.

The efficiency of the primary and secondary unit can be calculated using the following equation,

\[ \eta_t = \frac{q_v P}{P_{q_{v,\text{max}}} \cdot 600} \]  

(3.1)

where \( q_v \) is the secondary unit flow in [L/min], \( P_{q_{v,\text{max}}} \) is the drive power with maximum flow in [kW] and \( p \) is the operating pressure in [bar].

Using an A4VSO variable pump size 40, which characteristic is compatible with the requirements of machines M2 and M3 and is presented in figure 3.8, the total efficiency \( \eta_t \) can be calculated using equation (3.1) \[31\].
3 Power architecture

![Diagram of drive power and flow characteristic of an A4VSO size 40 control unit]

Figure 3.8: Drive power and flow characteristic of an A4VSO size 40 control unit

In view of the most critical situation and looking at figure 3.9 considering,

- $P_{\text{supp}}$ as power supplied;
- $P_{M1}$ as power required by M1;
- $P_{M2}$ as power regenerated by M2;
- $P_{M3}$ as power regenerated by M3.

The conversion’s efficiency of electrical energy into mechanical energy $\eta_2$ it’s around 98% [28]. Thus, the power that has to be supplied if a secondary control test bench was adopted, is calculated as,

$$P_{\text{suppl}} = \frac{P_{\text{in}}}{\eta_t \times \eta_2}$$ (3.2)

$$P_{\text{in}} = P_1 - (P_2 + P_3)$$ (3.3)

$P_{M1}$ is the power required by M1 in the critical situation which it’s equal to 63.5kW. Thus, the power $P_{M2} = P_{M3}$ is 31.75kW.

Using equation (3.1) and looking at figure 3.8 $\eta_t$ is calculated. The efficiency of machine M1 was considered the same of M2 and M3.

$$\eta_t = \frac{100 \times 170}{31.75 \cdot 600} = 0.89$$
3.4 Electrical input power estimation

Figure 3.9: Secondary control drive systems power supplied estimation scheme

\[ P_1 = \frac{P_{M1}}{\eta_t} \]

\[ P_1 = \frac{P_{M1}}{0.89} \]

\[ P_1 = 71.35 \text{ kW} \]

\[ P_2 = P_{M2} \times \eta_t \]

\[ P_2 = 28.3 \text{ kW} \]

\[ P_3 = P_{M3} \times \eta_t \]

\[ P_{M3} = P_{M2} \Rightarrow P_3 = P_2 \]

\[ P_{in} = 14.8 \text{ kW} \]

Thus, the power that as to be supplied in this application adopting a secondary control drive system is equal to,
3 Power architecture

\[ P_{suppl} = \frac{14.8}{\eta_t \times \eta_2} \quad (3.7) \]
\[ P_{suppl} = 17kW \]

3.4.2 Electrical drive systems

Considering the situation mentioned in previous subsection and same variables, an estimate of the power supplied can be done too, figure 3.10. The efficiency of the line modules, \( \eta_3 \) is equal to 95\% and the drive + motor ensemble efficiency, \( \eta_4 \), it’s around to 90\% \[30\]. Thus,

\[ P_{suppl} = \frac{P_{in}}{\eta_3} \quad (3.8) \]
\[ P_{in} = P_1 - (P_2 + P_3) \quad (3.9) \]
\[ P_1 = \frac{P_{M1}}{\eta_4} \quad (3.10) \]
\[ P_1 = \frac{63.5}{0.90} \]
\[ P_1 = 70.56kW \]
\[ P_2 = P_{M2} \times \eta_4 \quad (3.11) \]

Figure 3.10: Electrical drive systems power supplied estimation scheme
### 3.4 Electrical input power estimation

\[ P_2 = 31.75 \times 0.9 \]
\[ P_2 = 28.6kW \]

\[ P_3 = P_{M3} \times \eta_4 \] (3.12)

\[ P_{M3} = P_{M2} \Rightarrow P_3 = P_2 \]
\[ P_{in} = P_1 - 2 \times P_2 \]
\[ P_{in} = 13.41kW \]

\[ P_{suppl} = \frac{P_{in}}{\eta_3} \]
\[ P_{suppl} = 14.12kW \]

The power that has to be provided by the power supply in case of using a hydraulic secondary control drive system is \( P_{suppl,sec} = 17kW \). On the other hand, when using an electrical drive system this value decreases 20%.

Thus, the energy-efficient system chosen for this test bench was the electrical drive systems since they are modular, more compact than the hydraulic secondary control systems and even more important, the electrical machines and the drives of this systems are more efficient than the hydraulic machines which provides less power consumption.
Being the type of drive systems selected, in this chapter, the selection of torque transducers, couplings, electric motors and the correspondent drives and line module will be described.

As shown in section 2.5.2, increasing the drive shafts speed, a greater power and torque are necessary to allow to reproduce the real driving conditions.

To select all the components mentioned above the critical situation: maximum speed, presented on figure 2.13 will be considered.

Those values are resumed in table 4.1.

<p>| | | | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>n\text{\textsubscript{\textit{max}}}</td>
<td>rpm</td>
<td>T@n\text{\textsubscript{\textit{max}}}</td>
<td>P@n\text{\textsubscript{\textit{max}}}</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Nm</td>
<td>kW</td>
</tr>
<tr>
<td>M1</td>
<td>3660</td>
<td>170</td>
<td>63.5</td>
</tr>
<tr>
<td>M2</td>
<td>1300</td>
<td>270</td>
<td>36.20</td>
</tr>
<tr>
<td>M3</td>
<td>1300</td>
<td>270</td>
<td>36.20</td>
</tr>
</tbody>
</table>

4.1 Torque transducers

The differential has a torque efficiency ranging from 95 to 97%, thus, a high accuracy is required in order to minimize the errors in the measurements which may lead to the wrong conclusion that the differential has an efficiency equal to 100%. To measure the propeller and the drive shaft torques an HBM T40B torque transducer with a 500 Nm rated torque was chosen, figure 4.1.
4 Components selection

Figure 4.1: HBM T40B torque transducers

The main characteristics of this HBM torque transducers are presented in table 4.2.

Table 4.2: T40B HBM torque transducers characteristics

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>HBM T40B</td>
<td>500 Nm</td>
</tr>
<tr>
<td>Accuracy class</td>
<td>0.05</td>
</tr>
<tr>
<td>Nominal output signal</td>
<td></td>
</tr>
<tr>
<td>Voltage output</td>
<td>+10V</td>
</tr>
<tr>
<td>Nominal rotational</td>
<td>20 000 rpm</td>
</tr>
<tr>
<td>Mass moment of inertia</td>
<td>0.0039 kgm$^2$</td>
</tr>
</tbody>
</table>

4.2 Electric motors

M1, M2 and M3 should have a power and torque characteristic increased by the speed. As the tests will be performed on a steady state, the most critical situation will be maximum speed, and so is considered for dimensioning.

The electric motors available in Motion Control systems provided by Siemens are the Simotics servomotors and main motors, which are available in synchronous or induction/asynchronous versions.

The maximum power provided by Simotics servomotors is $34.2 \text{ kW}$, that does not satisfy the expected needs.

In turn, the asynchronous main motors have power ratings ranging from $2.8 – 1230 \text{ kW}$, higher than the synchronous that presents a power range from $15 – 310 \text{ kW}$.

However, the synchronous motors are more compact than induction, have a low rotor moment of inertia and therefore maximum dynamic response. The characteris-
tic speed/power curves of these motors are shown in figure [4.2]. Here, the theoretical curve of the maximum permissible overload capacity is shown as a limit in the characteristic diagrams and corresponds to value $P_{max}$.

(a) 1PH8 asynchronous motors

(b) 1PH8 synchronous motors

Figure 4.2: Typical speed/power diagram for 1PH8 motors

The selected synchronous motors, figure [4.3], and their main characteristics are presented in table [4.3] as well as their power and torque characteristics curves, figures [4.4] and [4.5]. The cooling system of these motors is forced ventilation.

Figure 4.3: 1PH8 synchronous motors, forced ventilation
Table 4.3: Motors for M1, M2 and M3 specifications

<table>
<thead>
<tr>
<th>Shaft height</th>
<th>$P_{\text{rated}}$</th>
<th>$M_{\text{rated}}$</th>
<th>$I_{\text{rated}}$</th>
<th>$V_{\text{rated}}$</th>
<th>$I_{\text{rated}}$</th>
<th>$n_{\text{max}_{\text{inv}}}$</th>
<th>$J$</th>
<th>Weight</th>
</tr>
</thead>
<tbody>
<tr>
<td>M1 1PH8137</td>
<td>132</td>
<td>66.7</td>
<td>177</td>
<td>100</td>
<td>440</td>
<td>120</td>
<td>4500</td>
<td>0.0885</td>
</tr>
<tr>
<td>M2, M3 1PH8184</td>
<td>180</td>
<td>38</td>
<td>454</td>
<td>80</td>
<td>316</td>
<td>53</td>
<td>1450</td>
<td>0.46</td>
</tr>
</tbody>
</table>

Figure 4.4: Power and torque characteristics curves of 1PH8137
4.2 Electric motors

4.2.1 Motor modules

Three motor modules are needed for supplying the three motors selected in the previous section. As referred in section 3.2.2, the motor modules are designed for multi-axis drive systems and are interconnected by means of a shared DC busbar. A connection example of a single motor module in booksize format with a current ranging from 45 to 200 A is presented in figure 4.6 [30].

The main characteristics of the selected motor modules are presented in table 4.4. They were selected based on each motor rated current and power.

4.2.2 Line module

The voltage-source DC link is supplied with main line voltage by a Line Module. This component generates a DC voltage and supplies motor modules with energy via the voltage source DC link [30].

In order to select a suitable power source, the worst situation was considered, i.e., when M2 and M3 are working in motoring mode and M1 is not regenerating energy, figure 4.7.
When testing the differential at the maximum speed/load conditions, the power required in M2 and M3, $P_{M2}$ and $P_{M3}$, respectively is equal to $31.75\, \text{kW}$.

Thus,

$$P_2 = \frac{P_{M2}}{\eta_4}$$

$$P_3 = \frac{P_{M3}}{\eta_4}$$

$$P_2 = P_3 = 35.3\, \text{kW}$$

As M1 is not regenerating energy, the power that has to be supplied by the line module, $P_{in}$, is calculated as,

$$P_{in} = P_2 + P_3$$  \hspace{1cm} (4.1)$$

$$P_{in} = 70.6\, \text{kW}$$  \hspace{1cm} (4.2)$$

Looking at equation (4.2), the maximum power that can be required for the test bench is $70.6\, \text{kW}$. Thus, the rated power of the line module has to be, $P_{in} \geq 70.6\, \text{kW}$.

An active line module was selected, figure 4.8. This component generates a reg-
4.2 Electric motors

Table 4.4: Technical specifications of motor modules

<table>
<thead>
<tr>
<th>Ref.</th>
<th>M1</th>
<th>M2, M3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type of cooling</td>
<td>Internal air cooling</td>
<td>Internal air cooling</td>
</tr>
<tr>
<td>Output rated current</td>
<td>A 132</td>
<td>A 85</td>
</tr>
<tr>
<td>Type rating based on $I_{\text{rated}}$</td>
<td>kW 71</td>
<td>kW 46</td>
</tr>
<tr>
<td>Current carrying capacity</td>
<td>A 200</td>
<td>A 200</td>
</tr>
<tr>
<td>DC link busbars</td>
<td>A 200</td>
<td>A 200</td>
</tr>
</tbody>
</table>

Figure 4.7: Line module selection scheme

ulated DC voltage which remains constant despite fluctuations in the line voltage. The main characteristics of the selected component are presented in table 4.5

Table 4.5: Active line module in booksize format

<table>
<thead>
<tr>
<th>Ref.</th>
<th>6SL3130-7TE28-0AA3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cooling</td>
<td>Internal air cooling</td>
</tr>
<tr>
<td>Rated power @ 380V AC</td>
<td>kW 80</td>
</tr>
<tr>
<td>DC link current</td>
<td>A 134</td>
</tr>
<tr>
<td>Input current</td>
<td>A 128</td>
</tr>
<tr>
<td>Current carrying capacity</td>
<td>A 200</td>
</tr>
<tr>
<td>DC link busbars</td>
<td>A 200</td>
</tr>
</tbody>
</table>

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4 Components selection

4.2.3 Control unit

The control unit is selected according to the number of drives to be controlled and the required performance level.

A CU320-2 control unit was selected because it was designed to control multiple drives. In this application, the closed-loop control types that have to be available are the speed and torque control. Using synchronous motors, the servo control is the only way to control them ways.

This control unit is capable of controlling up to 6 drives in servo control function.

An overview of the control unit, line module and motor modules is presented in figure 4.9.
4.3 Couplings

Flexible couplings are needed to connect the electric motors to the torque transducers. With them, misalignments can be compensated.

Thus, three ROBA®-DS couplings for HBM torque transducers marketed by mayr® were selected, figure 4.10. These shaft couplings are backlash-free and torsionally rigid. They compensate for axial, radial and angular shaft misalignments.[33].

There are different construction designs available:
4 Components selection

- Shrink disk hub external;
- Shrink disk hub internal;
- Sandwich construction.

The preferred type of construction is the shrink disk hub external, shown in figure 4.11 because it provides a compact and low mass moment of inertia design, a quick installation and external clamping and the shortest design [33].

![Figure 4.11: Preferred type of construction (shrink disk hub external)](image)

The ROBA®-DS size for torque transducers selected in previous section was 64 for 500 Nm T40B torque ranging, respectively.

The technical data of this couplings is presented on table 4.6.

<table>
<thead>
<tr>
<th>ROBA-DS size</th>
<th>64</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nominal torque</td>
<td>Nm</td>
</tr>
<tr>
<td>Peak torque</td>
<td>Nm</td>
</tr>
<tr>
<td>Outer diameter</td>
<td>mm</td>
</tr>
<tr>
<td>Minimum hub bore</td>
<td>mm</td>
</tr>
<tr>
<td>Maximum hub bore</td>
<td>mm</td>
</tr>
<tr>
<td>Maximum speed</td>
<td>rpm</td>
</tr>
<tr>
<td>Permitted misalignments</td>
<td>angular</td>
</tr>
<tr>
<td></td>
<td>axial</td>
</tr>
<tr>
<td></td>
<td>radial</td>
</tr>
<tr>
<td>Mass moments of inertia</td>
<td>Shrink disk hub, clamping external</td>
</tr>
<tr>
<td></td>
<td>Connection plate</td>
</tr>
</tbody>
</table>

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4.4 System inertia

To obtain a robust speed control of the electrical machines by the drives, it is necessary to ensure that the relation between the mass moment of inertia exerted by the motor load and its own mass moment of inertia is smaller than 3:1. In this context, similarly to what was done in section 2.4, the correspondent mass moment of inertia seen by each electric machine, $J_{1,eq}$, $J_{2,eq}$, and $J_{3,eq}$ respectively, was calculated, see figure 2.11.

$$J_{1,eq} = J_{ROBA} + J_{T,500} + J_{Part1} + \frac{J_{M2} + J_{M3}}{i^2}$$ (4.3)

Substituting the values in equation (4.3),

$$J_{1,eq} = 0.133811 kgm^2 < 3 \times J_{M1}$$

Assuming that the electrical machine $M_2$ will rotates at exactly the same speed as electrical machine $M_3$, $J_{2,eq} = J_{3,eq}$ was calculated using equation (4.4).

$$J_{2,eq} = J_{ROBA} + J_{T,500} + J_{Part2} + (J_{Part1} + J_{T,500} + J_{ROBA}) \times i^2 + J_{M1} \times i^2$$ (4.4)

$$J_{2,eq} = J_{3,eq}$$

Substituting the values in equation (4.4),

$$J_{2,eq} = 1.094 kgm^2 < 3 \times J_{M2}$$
In this chapter, the mechanical design will be described according to the following order: first of all, will be presented the test bench specifications and a summary of the Bosch Rexroth structural profiles; an overview of the mechanical structure will be presented; next, as the structure have 3 main parts, each assembly will be explained in section 5.3; in section 5.4 will be presented the final mechanical solution implemented.

5.1 Test bench structural specifications

To bring together all the components mentioned in previous chapter, a mechanical structure was designed. Additional parts were also designed to fix the electric motors, the torque transducers and the differential to the structure.

The structure of the test bench was designed following several specifications:

- Versatile connection elements;
- Rapid manufacturing acquisition;
- Few parts obtained by machining processes;
- Modular structure;
- Reusable and durable.

Rexroth Bosch Group offers many types of structural profiles complying with these criteria allowing high static and dynamic loads.

The profiles are slotted with 6 mm, 8 mm and 10 mm modular width. All of them within one profile range have the same slot dimensions and spacing to ensure perfect coordination during construction, figure 5.1 [34].

The modular dimensions available for 10 mm width slot are: 40 mm, 45 mm, 50 mm and 60 mm, figure 5.2. The 10 mm slotted profiles are suitable for applications
with high loads. Due to its very sturdy design and consequently high degree of robustness, this slot allows maximum strength connections and because of that was the slot dimension selected.

![Bosch profiles with different slot dimensions](image)

Figure 5.1: Bosch profiles with different slot dimensions

The main dimensions of the electric motors M1, M2 and M3, figure 5.3 are presented in table 5.1 [30].

![Modular dimensions for profile with slot 10mm with](image)

Figure 5.2: Modular dimensions for profile with slot 10mm with

<table>
<thead>
<tr>
<th>M1</th>
<th>M2, M3</th>
</tr>
</thead>
<tbody>
<tr>
<td>1PH8137</td>
<td>1PH8184</td>
</tr>
<tr>
<td>D [mm]</td>
<td>48</td>
</tr>
<tr>
<td>L [mm]</td>
<td>679</td>
</tr>
<tr>
<td>H [mm]</td>
<td>317.1</td>
</tr>
<tr>
<td>A [mm]</td>
<td>260</td>
</tr>
<tr>
<td>D [mm]</td>
<td>65</td>
</tr>
<tr>
<td>L [mm]</td>
<td>1047</td>
</tr>
<tr>
<td>H [mm]</td>
<td>490</td>
</tr>
<tr>
<td>A [mm]</td>
<td>364</td>
</tr>
</tbody>
</table>

The main dimensions of the electric motors M1, M2 and M3, figure 5.3 are presented in table 5.1 [30].

![M1, M2 and M3 principal dimensions](image)

Table 5.1: M1, M2 and M3 principal dimensions

<table>
<thead>
<tr>
<th>M1</th>
<th>M2, M3</th>
</tr>
</thead>
<tbody>
<tr>
<td>1PH8137</td>
<td>1PH8184</td>
</tr>
<tr>
<td>D [mm]</td>
<td>48</td>
</tr>
<tr>
<td>L [mm]</td>
<td>679</td>
</tr>
<tr>
<td>H [mm]</td>
<td>317.1</td>
</tr>
<tr>
<td>A [mm]</td>
<td>260</td>
</tr>
<tr>
<td>D [mm]</td>
<td>65</td>
</tr>
<tr>
<td>L [mm]</td>
<td>1047</td>
</tr>
<tr>
<td>H [mm]</td>
<td>490</td>
</tr>
<tr>
<td>A [mm]</td>
<td>364</td>
</tr>
</tbody>
</table>

The 90 mm was the height dimension chosen for this application (with a 10 mm width slot) because the 90 mm height dimension profiles offers the largest width. Therefore, as the electric motors are very wide (A = 364mm), they can seat in the profiles without an exaggerated number of connections.

The material of these structural profiles is AlMgSi 0.5 F25, which corresponds a 6000 series aluminum alloy, the 6063 T6. This material is alloyed with magnesium and silicon and is easy to machine and weldable.

The main properties of this material are presented in table 5.2.
5.1 Test bench structural specifications

Figure 5.3: M1, M2 and M3 main dimensions

Table 5.2: 6064 Aluminum alloy material properties

<table>
<thead>
<tr>
<th>Property</th>
<th>Symbol</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Elastic Modulus</td>
<td>$E$</td>
<td>Pa</td>
<td>$6.9 \times 10^9$</td>
</tr>
<tr>
<td>Poisson’s ratio</td>
<td>$\nu$</td>
<td></td>
<td>0.33</td>
</tr>
<tr>
<td>Shear Modulus</td>
<td>$G$</td>
<td>Pa</td>
<td>$2.58 \times 10^{10}$</td>
</tr>
<tr>
<td>Mass density</td>
<td>$\rho$</td>
<td>kg/m$^3$</td>
<td>2700</td>
</tr>
<tr>
<td>Tensile strength</td>
<td>$\sigma_r$</td>
<td>Pa</td>
<td>$17 \times 10^7$</td>
</tr>
<tr>
<td>Yield strength</td>
<td>$\sigma_y$</td>
<td>Pa</td>
<td>$9 \times 10^7$</td>
</tr>
<tr>
<td>Thermal conductivity</td>
<td>$k$</td>
<td>$J/kg^{*}C$</td>
<td>900</td>
</tr>
</tbody>
</table>

The profiles that were used are shown in figure 5.4 and their technical data are presented in table 5.3.
5 Mechanical design

(a) (b) (c)

Figure 5.4: Rexroth profile with 90mm of modular dimension with 10mm slot

<table>
<thead>
<tr>
<th>Profile surface [cm²]</th>
<th>Moment of Inertia [cm⁴]</th>
<th>Torsion Index [cm⁴]</th>
<th>Mass [kg/m]</th>
</tr>
</thead>
<tbody>
<tr>
<td>90 × 90</td>
<td>38.4</td>
<td>299.8</td>
<td>118.60</td>
</tr>
<tr>
<td>90 × 180</td>
<td>63.6</td>
<td>2138.3</td>
<td>429.00</td>
</tr>
<tr>
<td>90 × 360</td>
<td>90.2</td>
<td>14 065.0</td>
<td>937.10</td>
</tr>
</tbody>
</table>

Table 5.3: Profiles technical data

5.2 Structure overview

An overview of the mechanical structure of this test bench is presented in figure 5.5.

As shown, the structure is mainly composed by three parts:

- Part 1 - in orange;
• Part 2 - in green;
• Part 3 - in rose.

Parts 1 is composed by 6 legs of $90 \times 90$ profile and 3 main profiles: two $90 \times 90$ profiles over a $90 \times 360$ profile presented in figure 5.6. All of the mentioned profiles, except the legs, are 1500 mm long.

In a similar manner, part 2 is composed by 2 legs and 4 principal profiles, too: 2 $90 \times 180$ profiles alongside a $90 \times 360$ profile. These three profiles are fix to the above of a $90 \times 90$ profile, figure 5.7. Part 3 is formed by the same components as the part 2 and they are placed in exactly same way and symmetric. As in part 1, all of the profiles have 1500mm long, except the $90 \times 180$ profile, placed in the right side of part 2, that has a 3360mm and the $90 \times 90$ profile ($360 + 2 \times 90 = 540$mm).
5 Mechanical design

5.3 Assembly

In this subsection the assembly of the mechanical structure will be described, as well as will be mentioned all the components required for this.

5.3.1 Feet for support legs

First of all, 10 base plates have to be fixated into the 10 support legs, figure 5.8.

To fasten these components, for each leg, 4 T-bolts M8 × 20 (no. 3 842 528 715) are needed with the correspondent flange nuts (4 × 3 842 345 082). Finally, to fasten the base plates into the floor, 4 floor dowel are required (4 × 3 842 526 560).
5.3.2 Part 1

Assembly of legs and $90 \times 360$

To fasten the legs with $90 \times 360$ profile, 6 90/90 brackets (no. 3 842 523 575) are necessary (with this ordering number the fastening components are already included).

![Figure 5.9: Brackets with centering lugs](image)

Connection between $90 \times 360$ and 2 $90 \times 90$ profiles

To provide a connection between these 3 profiles, 16 $\times$ 2 parallel connector are needed. The corresponding ordering number is 3 842 542 736.

An example of this type of connection is presented in figure 5.10.
5.3.3 Part 2 and 3

The assembly of part 2 components and part 3 components is done in the same way because these parts are composed by the same components and because of that, only the assembly of part 2 will be explained.

90 × 180 and 90 × 360 fixation

To provide the connection between the 90 × 180 profile and 90 × 360, 4 90/90 × 2 brackets (no. 3 842 523 575) are required. In addition to this joining, a steel plate was designed, figure 5.11. To fix this plate into both profiles, 8 × 4 T-nuts (no. 3 842 536 603) and 8 × 4 EN ISO 4762 M8 × 25 are necessary.
In order to provide a rigid structure, a $90 \times 90$ profile was selected to connect $90 \times 180$ and $90 \times 360$ profiles at the beginning of part 2 and part 3.

This connection is provided by 4 parallel connectors ($4 \times 3\ 842\ 542\ 736$), using the assembly principle explained in figure 5.12.

Figure 5.12: Parallel connector for a connection between two perpendicular strut profiles

The connection between this module and the legs of part 2 is made using 4 brackets (no. 3 842 523 575).
5.3.4 Connection between part 1, 2 and 3

To fasten parts 2 and 3 to part 1, 6 brackets (figure 5.9) are needed (no. 3 842 523 575), figures 5.13 and 5.14.

Figure 5.13: Brackets for connection between parts 1, 2 and 3 on the upper side

Figure 5.14: Brackets for connection between parts 1, 2 and 3 on the bottom side

5.4 Kinematic chain and assembly

Taking into account the purpose of this test bench - testing rear axles - in this section will be presented the final mechanical solution implemented.

Figure 5.15 presents an overview of the test rig, where all the numbered components are part of the kinematic chain and are referred in table 5.4.
In this section, only the assembly sequence of part 1 will be described because the mounting sequence and the components of all the main structure parts are similar. Finally, the design of the differential support will be described.

### 5.4.1 Electric motor support

The selected mounting position of M1, M2 and M3 electric motors was IMB3, i.e., horizontal shaft and "feet on floor", figure 5.16. With this mounting position the alignment is easily obtained.

To fix the electric motors to a metal plate (that is fasten in Bosch profiles) a foot was designed, figure 5.17.
Figure 5.16: Type of M1, M2 and M3 mounting position

Figure 5.17: Foot for M1

To make the electric motor M1 and feet connection, 4 screws: M12 × 90 EN ISO 4762 - 36N5 and as well as 4 flange nuts DIN 6923-M12-N5 are required.

To allow faster assembly and minimize the risk of damage the Bosch profiles during that operation, a steel plate was also designed, figure 5.18.

Figure 5.18: M1 metal board with T-nut and screws

The feet are fixed to the metal plate, and the last, in time, will fix in Bosch profiles, using 2 × 4 T-nut (no. 3 842 536 603) and 2 × 4 EN ISO 4762 M8 × 50 screws.

To provide an easier alignment between the electric motors and the profiles it was designed a metal plate and two feet. The metal plate is situated between the Bosch profile and the feet. The face that slights on the Bosch profile should have low roughness and the plain tolerance should be around 0.05 mm. In the machining process of this face, a longitudinal rule should be also machined. This rule will guide
the metal board through the Bosch profile. In the opposite side, a parallel slot was
designed to align the feet and the motor itself. All the horizontal faces that are
parallel to the Bosch profile must have plain and parallelism tolerances of 0.05mm
and 0.1mm, respectively. This way, the parallelism between the motor axis and the
horizontal plan are improved. This will be important to do not create external loads
that deteriorate the transducer measurement.

5.4.2 Connection between electric motor and ROBA® coupling

The electric motors M1, M2 and M3 have a feather key as shaft extension.
To connect the motors shafts to the torque transducers it is necessary a coupling,
already mentioned in section 4.3.

5.4.3 Connection between ROBA® coupling and torque
transducer

ROBA couplings are specially designed for T40B HBM torque transducers.
To connect the coupling to the rotor of the transducer $8 \times M10 \times 25$ screws are
needed, figure 5.19.

![Figure 5.19: Connection between Roba coupling and torque transducer](image)

Finally, another component was required to install the stator of the transducer.
This part is fasten in a $45 \times 90$ Bosch (no. 3 842 548 756) profile with 300 mm long
in part 1 and 540 mm long in parts 2 and 3.
5.4.4 Connection between torque transducer and propeller shaft flange

A component was designed to provide the connection between the torque transducer and the propeller shaft, shown in figure 5.20.

Figure 5.20: Modeled part for flanges connection

The guide between the differential flange and the part modeled has a tolerance H6/g5 that provides an accurate mechanical guidance. However, ROBA coupling is able to absorb an angular misalignment of 0.6°. Those two parts are fixed by 4 × M8 and 4 × M6 fasteners.

5.4.5 Differential support

The BMW E46 differential has 3 machined faces for supports: 2 of them have screws M8. To align and provide the differential stabilization 3 shims are needed. To fix the shims, a steel block was designed. These components described were designed and are presented in figure 5.21.

The differential shown in figure 5.21 is a sketch of the real one, with the same outside dimensions.

To provide easier assembly, the steel plate has pockets to allocate the shims. These connection as a H8/g8 tolerance, which gives an assembly that can be done by hand, see the row of figure 5.21. Finally, to fix the steel plate on the Bosch profiles, 4 feet were also modulated.

To fasten the feet to the steel plate, 8 EN ISO 4762 M8 × 35 screws are required and to fasten the feet in bosch profiles, 8 T-nuts (no. 3 842 536 603) and 8 EN ISO 4762 M8 × 35 screws are required, too.
5.5 Mechanical solution validation

Once found an appropriated mechanical solution it is necessary to validate it considering the stresses applying by the component’s supports (motor, torque transducers and differential). The structure will face mainly torsion efforts, i.e., the external loads are twisting moments. As these applied efforts are simple a model was created and the correspondent analytical solution will be presented.

**Straight bars of uniform circular section under pure torsion**

Consider a straight bar with a uniform circular section of an homogeneous isotropic material. Applying a twist couples at its ends in plane normal to its axis, the bar will twists, each section will rotates about its torsional center, figure 5.23.

If $\theta$ were the angle of twist, $M_t$ were the twisting moment, $l$ the length of the member, $J$ the polar moment of the section and $G$ the shear modulus, the $\theta$ can be calculated as,
Bars of noncircular uniform section under pure torsion

Considering a noncircular cross section of a bar, and considering all the assumptions of the previous subsection, \( \theta \) can be expressed as,

\[
\theta = \frac{M_t l}{JG}
\]  

(5.1)

where \( K \) is a factor dependent on the form and dimensions of the cross section. For a circular section \( K \) is the polar moment of inertia \( J \), equation (5.1), for other sections \( K \) is less than \( J \) and may be only a very small fraction of \( J \).

As mentioned before, a model was created in order to know the maximum twisting angle of each basis cross section and these values should be compared to the maximum angle of twisting recommended in the literature for an equivalent material, in order to verify the structural criteria.

As presented in previous sections, the structure is basically composed by three parts, figure 5.24.

To calculate the twisting angle suffered by each cross section part, \( \theta_1 \), \( \theta_2 \) and \( \theta_3 \), respectively, the connection between part 1 and part 2 was considered rigid, and the connection between part 1 and part 3 was considering rigid too.

As shown before, \( K \) is a fraction of \( J \); \( J \), the polar moment of a cross section, is the sum of the moments of inertia \( I_x \) and \( I_y \). Thus, the \( J \) of each profiles was calculated and is presented in table 5.5.

\( \theta_1, \theta_2 \) and \( \theta_3 \) are calculated from the torsion center which coincides with the geometric center, \( G_1 \) and \( G_2 = G_3 \), of each part cross section, figures 5.25 and 5.26.

Using equations (5.3) and (5.4), \( y_{G_1} \) and \( y_{G_2} = y_{G_3} \) were determined.

\[
A_A \cdot y_A + A_B \cdot y_B + A_C \cdot y_C + A_D \cdot y_D = (\sum A_i) \cdot y_{G_1}
\]  

(5.3)
5.5 Mechanical solution validation

Figure 5.24: Main parts of the mechanical structure

Table 5.5: Polar moment of the Bosch profiles section

<table>
<thead>
<tr>
<th>Profile</th>
<th>$J$ (cm$^4$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>90 $\times$ 90</td>
<td>599.6</td>
</tr>
<tr>
<td>90 $\times$ 180</td>
<td>2682.6</td>
</tr>
<tr>
<td>90 $\times$ 360</td>
<td>14,775</td>
</tr>
</tbody>
</table>

$y_{G_1} = 11.07\text{ cm}$

$A_E \cdot y_E + A_F \cdot y_F + A_G \cdot y_G + A_H \cdot y_H = (\sum A_i) \cdot y_{G_2}$  \hspace{1cm} (5.4)

$y_{G_2} = y_{G_3} = 11.76\text{ cm}$

The areas of profiles A, B, C, E, F and G are presented in table 5.3. The area of parts D and H are $A_D = 54 \times 2\text{ cm}^2 = 108\text{ cm}^2$ and $A_G = 36 \times 2\text{ mm}^2 = 72\text{ cm}^2$.

Looking at equation (5.2), it can be noticed that $K$, the fraction of the polar moment of each part cross section, must be calculated at geometric centers. In order to calculate, $K_A^{G_1} = K_B^{G_1}$, $K_C^{G_1}$, $K_D^{G_1}$, $K_E^{G_2} = K_F^{G_2}$, $K_G^{G_2}$ and $K_H^{G_2}$ it was assumed that,

$$\frac{K_i}{J_i} = \frac{K_i^{G_j}}{J_i + A_i \times d_i^2}$$ \hspace{1cm} (5.5)

where $d_i$, is the distance between the geometric center of the total cross section and
the geometric center of each component for the total cross section, \( i = A, B, C...H \) and \( j = 1, 2 \), respectively.

\( K_D \) and \( K_H \) were calculated using equation \([5.6]\):

\[
K = ab^3 \left[ \frac{16}{3} - 3.36 \frac{b}{a} \left( 1 - \frac{b^4}{12a^4} \right) \right]
\]  

where \( a \) and \( b \) dimensions, figure \([5.27]\) are \( a = 18\text{cm}, b = 1\text{cm} \) in part D and \( a = 27\text{cm}, b = 1\text{cm} \) in part G.

Using equation \([5.5]\), each \( K_i^{G_j} \) was calculated and the correspondent values are presented in table \([5.6]\).

The maximum twisting moment felt by part 1 is \( Mt = 170\text{Nm} \). Using equation \([5.7]\), \( \theta_1 \) was calculated.
5.5 Mechanical solution validation

\[ \theta_1 = \frac{M_t l}{K_{G_1} G} \]  \hspace{1cm} (5.7)

where

\[ K_{G_1} = K_{G_1}^{G_1} + K_{G_1}^{G_1} + K_{G_1}^{G_1} + K_{G_1}^{G_1} \]  \hspace{1cm} (5.8)

\[ \theta_1 = \frac{170 \times 1.5}{K_{G_1} G} \]

\[ \theta_1 = 0.0128^\circ \]

Applying the same principle \( \theta_2 \) was calculated too, but \( M_t \) in time is equal to 270Nm.

\[ \theta_2 = \frac{M_t l}{K_{G_2} G} \]  \hspace{1cm} (5.9)

---

**Figure 5.26: Part 2 cross section**

**Table 5.6: K’s of the component parts cross sections at the center of torsion**

<table>
<thead>
<tr>
<th>( K_{G_1}^{G_1} )</th>
<th>cm(^4)</th>
</tr>
</thead>
<tbody>
<tr>
<td>( K_{A_1}^{G_1} = K_{B_1}^{G_1} )</td>
<td>1525.87</td>
</tr>
<tr>
<td>( K_{C_1}^{G_1} )</td>
<td>1238.64</td>
</tr>
<tr>
<td>( K_{D_1}^{G_1} )</td>
<td>136.841</td>
</tr>
<tr>
<td>( K_{E_1}^{G_1} = K_{F_1}^{G_1} )</td>
<td>5622.24</td>
</tr>
<tr>
<td>( K_{G_1}^{G_1} )</td>
<td>1184.04</td>
</tr>
<tr>
<td>( K_{H_1}^{G_1} )</td>
<td>176.986</td>
</tr>
</tbody>
</table>
where

\[ K_{G_2} = K_{E_2}^G + K_{F_2}^G + K_{G_2}^G + K_{H_2}^G \quad (5.10) \]

\[ \theta_2 = 0.00714^\circ \]

The maximum twisting angles, \( \theta_1 \) and \( \theta_2 \) are both less than 0.15\(^\circ\)/foot, that is equivalent a \( \theta = 0.7^\circ \) for the same length (\( l = 1500\)mm) \[36].
LabView is a graphic programming system from National Instruments that CETRIB uses in all of their bench tests. Thus, this software was selected to make the test rig automation and data acquisition.

LabView programs are called Virtual Instruments (VIs) and comprise two components:

- The front panel that accommodates the user interface;
- The block diagram that shows the graphic program code.

Using a network connection there are two main architectures that can be adopted: use LabView as graphical interface and use a Siemens PLC or using LabView as graphical interface and controller.

In this chapter these possibilities will be discussed.

### 6.1 Speed acquisition

In order to obtain measurements that allow the analysis of the axle efficiency, the velocity of the drive shafts and the propeller shaft has to be known. The electric motors M1, M2 and M3 have two types of encoders: incremental and absolute.

As the variable to control is not position but speed, an absolute encoder is not needed for this application. Thus, an incremental encoder with $22\text{bit}$ was chosen, which means that, if this incremental encoder has $2^{22}$ pulses per revolution (ppr), the resolution of it is equal to,

$$Res_{enc} = \frac{360^\circ}{2^{22}}$$

$$Res_{enc} = 0.00008583^\circ$$

The internal acquisition time of the drive speed, $T$, is 0.667ms, which corresponds to a speed resolution of
\[ Res_{\text{speed}} = \frac{1}{T} \] (6.2)

\[ Res_{\text{speed}} = 0.02\text{rpm} \]

The speed data acquisition should be at \( f = 1 \text{ Hz} \). At the same acquisition frequency, three analog outputs from the torque transducers have to be read and saved, too. As mentioned before, the output range of them is ±10V.

### 6.2 Temperature acquisition

The efficiency of lubricants depends strongly on the friction properties of the lubricant employed. For the design of modern highly efficient lubricants it is thus essential to understand the influence of the components of the lubricating fluid in terms of film formation and consequent friction which strongly depends on lubricant temperatures.

Thus, five temperatures have to be registered at a frequency of \( f = 0.5 \text{ Hz} \).

The maximum permissible error for this measurement is ±1°C.

### 6.3 Measurement of temperature

Temperature is one of the most frequently measured physical quantities. Several techniques can be used in temperature measurement: thermometers whose sensor element is a resistance or measurement based on the use of thermocouples [37].

#### 6.3.1 Resistance thermometers

Resistance thermometers cover a range from −250 to +850°C, with excellent accuracy and very good stability. Moreover, they provide an electric output signal which is a very important feature.

Resistance thermometers are based on sensor elements that present a resistance change with temperature variation. The sensor elements first used were of metallic type and were usually called Resistance Temperature Detectors (RTDs).

The most popular metal alloy for this sensors is platinum. Platinum is a noble metal, non-reactive across a wide temperature range, and presents a very linear response, figure 6.1 [37].

The strong points of RTD’s are:

- Great accuracy;
6.3 Measurement of temperature

- Great stability;
- Good linearity;
- Absolute measurement of temperature (without needing a reference temperature).

On the other hand, the weak points of RTD’s are:

- Expensive;
- Large response time;
- Power supply requirement;
- Self-heating.

6.3.2 Thermocouples

One of the techniques employed in the measurement of temperature is based on the use of thermocouples.

Thermocouples are robust, versatile and wide in range, reading temperatures from −200 to 2000°C. They present an important feature: their output is an electrical voltage [37].
6 Automation project design

Operating principle

The operating principle of thermocouples is based on Seebeck effect, i.e., if two metallic conductors A and B (pure metals or alloys) form an open circuit with two AB junctions \((Y_1, Y_2)\) at different temperatures \(T_1\) and \(T_2\), a thermal electromotive force, \(\varepsilon\), will appear on their terminals. The value of this greatness depends on the materials A and B and on the junction temperatures \(T_1\) and \(T_2\), figure 6.2 [37].

![Figure 6.2: Seebeck effect](image)

This set of two associated material A and B with two junctions is called thermocouple [37].

If one of the junctions is maintained at a known constant temperature \(T_1\), the other junction temperature may be calculated by measuring the thermoelectric force value generated by the thermocouple. A thermocouple therefore allows differential measurement of temperature [37].

Types of thermocouples

Combinations of different materials A and B correspond to different thermocouples. Some material combinations have been defined as standard and named T, J, K, E, etc.

For each type of standard thermocouple there is a standard table that gives the temperature of the “hot junction” (or measuring junction) corresponding to the thermoelectric electromotive force produced by the thermocouple, assuming the “cold junction” is at a reference temperature, \(0^\circ C\) [37].

This type of temperature transducers produces a rather low signal, in general not exceeding some dozen of mV for the extreme temperatures of the range. Table 6.1 presents several standard thermocouples, indicating for each one the respective operation range and the corresponding electromotive force amplitude [37].

The relation between temperature and the thermocouple thermoelectric electromotive force is called the Seebeck coefficient or sensitivity. This greatness throughout the respective measurement ranges is, frequently, a highly nonlinear function.
6.3 Measurement of temperature

<table>
<thead>
<tr>
<th>Thermocouple</th>
<th>°C</th>
<th>ε [mV]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Copper/Constantan (T-type)</td>
<td>164 to 350</td>
<td>−5.602 to 17.816</td>
</tr>
<tr>
<td>Iron/Constantan (J-type)</td>
<td>0 to 749</td>
<td>0 to 42.283</td>
</tr>
<tr>
<td>Cromel/Constantan (E-type)</td>
<td>164 to 885</td>
<td>−8.824 to 68.783</td>
</tr>
<tr>
<td>Cromel/Alumel (K-type)</td>
<td>164 to 1250</td>
<td>−5.973 to 50.633</td>
</tr>
</tbody>
</table>

The K-type thermocouple is an exception to this characteristic. This thermocouple type presents a nearly constant sensitivity, around $40 \mu V/°C$ between 0 and 1000°C, corresponding to a nearly linear relation between temperature and thermoelectric electromotive force, figure 6.3 [37].

The strong points of thermocouples are:

- Strong points;
- Self-powered;
- Wide variety;
- Inexpensive;
- Wide temperature range: $-200°C$ to $2000°C$;
- Robust and simple;
- No self-heating problems.
On the other hand, the weak points are:

- Non-linear;
- Requires a well known temperature reference;
- Low level voltages.

As the maximum permissible error of temperature measurement is $\pm 1^\circ C$ the thermocouple that was chosen was the T-type, Class 1. The main characteristics of these sensors are presented in table 6.2, the reference temperature is $0^\circ C$ [38].

Table 6.2: Tolerance classes for thermocouples according to the IEC 60 584-2:1995

<table>
<thead>
<tr>
<th>Thermocouple</th>
<th>Class 1</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature</td>
<td>$-40^\circ C \leq T \leq 125^\circ C$</td>
</tr>
<tr>
<td>Tolerance</td>
<td>$\pm 0.5$</td>
</tr>
<tr>
<td>Temperature</td>
<td>$125^\circ C \leq T \leq 350^\circ C$</td>
</tr>
<tr>
<td>Tolerance</td>
<td>$\pm 0.004 \cdot T$</td>
</tr>
</tbody>
</table>

6.4 PROFIBUS vs PROFINET

To make the communication between the field devices and the controllers it has be found an architecture. There are two possibilities available for making this connection: using a data acquisition board (DAQ) or using a network communication.

As the brand selected for drive system was SIEMENS and it already has standard solutions using a network connection, the network communication was the architecture selected.

The two main networks that Siemens offers to make the communication between the field devices and the controllers are PROFIBUS and PROFINET.

PROFIBUS DP (Decentralized Peripherals) is a network that is made up of two types of devices connected to the bus: master devices and slave devices. It is a bidirectional network, meaning that one device, a master, sends a request to a slave, and the slave responds to that request. With PROFIBUS each port has to be individually configured, in the other hand [39].

Based on Industrial Ethernet, PROFINET enables direct communication of field devices (IO Devices) with controllers (IO Controllers).

Nowadays, PROFINET is the industrial network that provides higher transmission speeds and shorter port configuration times. In PROFINET the IP address can be automatically assigned. Base on these advantages, the communication protocol that was selected was the PROFINET [30].
6.5 LabView as PROFINET controller

In order to integrate real hardware (Sinamics S120 converters), with the measurement system, it is necessary that this hardware is directly connected to LabView, figure 6.4.

PROFINET is available for establishing a connection between Sinamics drivers, PROFINET I/O devices and LabView via a standard Ethernet interface.

The application comprises a PROFINET communication driver for controllers, a DLL (dynamic link library), which is integrated in LabView and which is executed there, i.e., it serves as the interface between the LabView user program and the PROFINET IO devices. It provides functions, which control the communication and allow cyclic IO data to be transferred. This provides functions for communication with PROFINET I/O devices directly from LabView.

This solution has many advantages such as:

- Shorter time and lower costs when implementing the solution;
- Simple integration of the communication driver;
- Simple control of SINAMICS drives from LabView;
- Integration of additional PROFINET IO devices in LabView;
• Connection via standard Ethernet interface.

However, as a limit: using this architecture, high demands on real-time communication are not assured and therefore, this is not a viable option [40].

6.6 Controller-based

Siemens provides basic and advanced controllers. The S7-1200 is a family of basic controllers with compact design for simple automation tasks [41].

The S7-1500 is a advanced controller projected for complex plants with high demands in terms of speed and deterministic response. Furthermore, this controllers don’t have integral I/O, they have to be purchased separately, which configures to it a final price much higher than the basic controllers.

This application doesn’t require high demands in terms of speed and deterministic response, thus, the most appropriated family of controllers is the basic one [41].

After contact Siemens, the basic controller more appropriated for this application is the S7-1214, figure 6.6.

This component has 16 inputs:

• 14 digital;
6.6 Controller-based

- 2 analog ranging from 0 to 10V.

The outputs available are:

- 10 digital.

The HMI role keeps to be played by LabView, which continues to communicate with the other field devices by PROFINET.

6.6.1 Analog module for torque transducers

As the output signal of the torque transducers ranging between $-10V$ and $10V$, the analog input integrated on the PLC S7-1214 cannot be used. Thus, one I/O module was selected to read the values provided by the three HBM T40B which reference is 6ES7231-4HD32-0XB0. The characteristics of these I/O module are presented on table 6.3 [41].

<table>
<thead>
<tr>
<th>Analog inputs</th>
<th>4; current or voltage differential inputs</th>
</tr>
</thead>
<tbody>
<tr>
<td>Input ranges</td>
<td></td>
</tr>
<tr>
<td>Voltage</td>
<td>10; 5; 2.5</td>
</tr>
<tr>
<td>Current</td>
<td>4 to 20mA; 0 to 20mA</td>
</tr>
<tr>
<td>Analog value creation</td>
<td>12 bit; +sign</td>
</tr>
<tr>
<td>Resolution with overrange</td>
<td>12 bit; +sign</td>
</tr>
</tbody>
</table>

As referred before, the output signal of the torque transducers is $\pm 10V$, thus, using this voltage input range, the resolution of torque measure will be,

$$R_{torque} = \frac{500Nm}{2^{12}}$$

$$R_{torque} = 0.1221Nm$$ (6.3)

The torque resolution is 12 times smaller than the accuracy of the transducer which doesn’t commits the measure according to its accuracy class.

6.6.2 Analog module for thermocouples

Another I/O module was selected to read the values provide by the five thermocouples which reference is 6ES7231-5QF32-0XB0, figure 6.7 [41].

The characteristics of these I/O module are presented on table 6.3.
Table 6.4: 6ES7231-5QF32-0XB0 main characteristics

<table>
<thead>
<tr>
<th>Characteristics</th>
<th>Details</th>
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<tr>
<td>Analog inputs</td>
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<tr>
<td>Technical unit for temperature</td>
<td>Degrees celsius/Degrees fahrenheit</td>
</tr>
<tr>
<td>measurement adjustable</td>
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<tr>
<td>Input ranges</td>
<td>10; 5; 2.5</td>
</tr>
<tr>
<td>Thermocouple</td>
<td>Type J, K, T, E, R, S, N, C, TXK/XK(L); voltage range: ±80mV</td>
</tr>
<tr>
<td>Resolution with overrange</td>
<td>15 bit; +sign</td>
</tr>
</tbody>
</table>

The resolution of the analog module is:

\[
R_{\text{voltage}} = \frac{80 \text{mV}}{2^{15}}
\]

\[
R_{\text{voltage}} = 0.00244 \text{mV}
\]  \hspace{1cm} (6.4)

By looking at figure 6.3 and the temperature measurement range (20° C - 120° C), the worst sensitivity was considering to calculate the correspondent temperature resolution.

\[
S_{0@20^\circ C} \approx 40 \mu V/^\circ C
\]
\[ R_{\text{temperature}} = \frac{R_{\text{voltage}}}{S_{0{20}^\circ C}} \]

\[ R_{\text{temperature}} = 0.052^\circ C \]  \hspace{1cm} (6.5)

Comparing the value obtained in equation (6.5) and the maximum error provided by the thermocouple, the analogue module is adequate for this application.
7.1 Conclusions

The aim of this project was to design the mechanical and automatic project for a rear axle transmission test rig.

After doing the review of the state of the art, it was noted that instead of using a brake test bench where all the power provided for the electric machines, working as motor modes, would be consumed by the two brake dynamometers and could not be reused, it would be interesting to use a drive system where energy recovery was available. Using this architecture, the drive system just has to get the energy from the network who lost in system power losses.

Two different drive systems with energy recovery were compared. As the electrical drive system has machines that have an efficiency around 93%-95% and hydraulic machines around 89%, when doing a test with positive torque flux, for example, it can be saved 20% in system losses, when using an electrical drive system. Due to that, the hydraulic secondary control drive system was stayed out.

After searching the laboratory tests for measuring the vehicle’s $CO_2$ emissions and taking as a start-point the BMW 318d, the power and torque requirements for the test bench drive system were defined. For the machine that plays the role of gearbox, M1, it is necessary to have 170Nm at 3660rpm ($P_{M1} = 63.5$ kW). For the machines that play the role of the wheels’s load, M2 and M3, it is necessary to have 270Nm at $1300$ rpm ($P_{M2,M3} = 37$ kW). With these requirements, the electric machines were chosen as well as the correspondents drives and a compatible power supply.

In order to test the several situations whereby the differential is normally subjected, the power flow of the test bench for simulating acceleration, braking and curves was also defined.

To provide an accurate measurement of the torque applied in each differential shaft, three torque transducers were selected, as well as couplings that ensures a correct connection between the electric motors and the transducers. With ROBA couplings the misalignment and assembly errors can be compensated.
7 Conclusions and future work

The final kinematic chain solution chosen permits the torque losses measurements in the differential transmission. With the mechanical structure all of the kinematic chain components can be positioned. The correspondent assembly and exploded drawings are presented in the present report.

Finally, a PLC and the correspondent necessary analog modules was chosen as a command solution, using LabView as a graphical interface. The analog modules selected for measuring the output of the torque transducers has 13-bits which provides a compatible resolution according to the torque transducers accuracy class. Additionally, thermocouples where selected which provide a $\pm 1^\circ$ of maximum error. The analog module for measuring these data where selected and the correspondent resolution is compatible with the maximum permissible error.

7.2 Future works

As future works, it will be interesting to design a control system for controlling the differential temperatures, using a fan and a heater.

To provide the electric power, to do the signal conditioning, protection and cutting functions an electric box should be also designed.

Finally, in order to join all the information and making the command function, the PLC has to be programmed as well as LabView interface.
References


References


SECTION A-A
SCALE 1 : 4
### Tolerances

**ISO 2768-m**

<table>
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<tr>
<th>Tolerance Type</th>
<th>ISO 2768-m</th>
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<th>Tol. Angular</th>
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<td>±0.2</td>
<td>±0.3</td>
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</tbody>
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### Surface Roughness

- Ra 1.2
- 0.05

### Section A-A

**Scale 1:2**

### Measurements

- 20.50
- 22.50
- 45
- 210
- 9
- 12.67 ±0.05

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**Designation / Designation:**

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**Data / Date Revisão / Revision:**

1 5 6 4 3 2

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**Nome / Name:**

- [Name]
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**Scale 1:4**

**SECTION A-A**

**ISO 2768-m**

**Tolerances**

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- Number: 1
- Name: SOLIDWORKS Educational Product. For Instructional Use Only
- Revision: 1
- Sheet: A4
- Date: 1/1
All edges chamfered with 0.2x45°
### Tolerâncias / Tolerances

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<thead>
<tr>
<th>ISO 2768-m</th>
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</table>

### Dimensões nominais / Nominal dimensions
- \( \Phi 130 \) 
- \( \Phi 75 \) H6 
- \( \Phi 66.5 \) g5 
- \( 103.2 \) 
- \( 0.20 \times 45° \) (all edges)

### Rugosidades / Surface roughness
- Ra 1.2

### Desenho de definição / Technical drawing

---

### Desenho de detalhe

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### Material / Material
- Acoplamento dif trans out

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### Data / Date
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### Nome / Name
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### Tolerâncias / Tolerances

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<tr>
<td>( \Phi 0.2 )</td>
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<td>( \Phi 0.3 )</td>
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### Tolerâncias angulares / Tol. of angle

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### Data / Date
- Revisão / Revision
- Nome / Name
- Scale / Scale
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### Shrink disk hub, clamping internal

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### Module 1, connection plate*

* Does not correspond to the former HBM ID. number 1-4411._ _ _ _ (see page 11)

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### Module 1, sleeve

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Dimensions of T40B 500 Nm - 1 kNm without rotational speed measurement

Dimensions in mm (1 mm = 0.03937 inches)
Dimensions without tolerances, per DIN ISO 2768-mk
Dimensions of T40B 500 Nm - 1 kNm without rotational speed measurement, continued

Dimensions in mm (1 mm = 0.03937 inches)
Dimensions without tolerances, per DIN ISO 2768-mk

Partial sections cut A-A