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Transmission layout and operation gearshift criteria for electric micro cars

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Affidavit

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Acknowledgement

During the writing of this thesis I have received a great deal of support from many people.

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It would be unfair not to highlight the helpful and sweet support of my journey companion. As well as to acknowledge the many friends that help me get this bread and help me grow every day. Always grateful to have you all.

Abstract

Background and research problem

The gearbox is an example of how much the powertrain has been simplified with the advent of E-mobility. Highly-complex mechatronics transmissions are often cut down to a single-speed gear reducer by virtue of the specific characteristics of the electric motors, although some sports and premium electric cars use them to maximize performance and efficiency.

This thesis conducts an initial layout process for the transmission of electric low-end micro cars, which often lack of a powerful motor and can greatly benefit from a transmission's optimization.

Methods

Several script-based models are programmed to analyze the longitudinal dynamics of a micro electric vehicle (EV).

With a defined list of priorities, assumptions, performance requirements and design limitations, the performance and efficiency are analyzed on real driving cycles. The study explicitly finds an optimized solution for the TERA Ibex transmission on a specific cycle, although the methodology and developed tools can be further re-used to evaluate any other driving cycle, motor or vehicle.

Results and conclusion

The results highlight both the benefits of a dual transmission layout and the improvement potential of the single speed reducers.

The quantitative results reveal that a dual clutch transmission with a high spread between gears and regenerative braking is the ideal layout for both performance and efficiency reasons.

It minimizes energy consumption, performs gear shifts quickly and seamless, achieves a high top speed, a fair acceleration and increases driving safety by granting the climb of steep slopes, as well as any situation that requires a sudden acceleration.

Zusammenfassung

The Masterarbeit beschäftigt sich mit der grundsätzlichen Auslegung eines Zweigang-Schaltgetriebes in einem elektrisch angetriebenen Kleinwagen. Auf Basis einer definierten Liste von Prioritäten, Annahmen, Leistungsanforderungen und Designbeschränkungen werden Fahrleistung und Antriebsstrangeffizienz anhand realer Fahrzyklen analysiert. Die Arbeit findet explizit eine optimierte Lösung für das TERA Ibex-Getriebe in einem bestimmten Fahrzyklus, wobei die Methodik und die entwickelten Werkzeuge weiter verwendet werden können, um verschiedene andere Fahrzykluse, Motoren oder Fahrzeuge zu evaluieren.

Die Ergebnisse verdeutlichen sowohl die Vorteile einer Doppelgetriebeanordnung als auch das Verbesserungspotenzial von Ein-Gang-Getrieben.

Die quantitativen Ergebnisse zeigen, dass für den vorliegenden Anwendungsfall in einem Kleinfahrzeug, ein Doppelkupplungsgetriebe mit einer hohen Spanne zwischen den Gängen und regenerativer Bremsung sowohl aus Leistungs- als auch aus Effizienzgründen die ideale Auslegung ist.

Es minimiert den Energieverbrauch, arbeitet schnell und nahtlos, erreicht eine hohe Höchstgeschwindigkeit, eine ordentliche Beschleunigung und erhöht die Sicherheit, indem es das Befahren von steilen Hängen oder plötzliche Beschleunigungsvorgänge, zulässt.

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1. Topic Introduction

1.1. TERA Ibex

The Ibex is an ambitious project of “Team Eco Racing Austria”, a team of motivated students who share the vision of a sustainable and efficient mobility, and that originated within the Graz University of Technology.

Designed, simulated and manufactured from scratch and entirely by students of the TU Graz, it aims to meet high standards and satisfy the needs of the general user. It focuses on the efficiency without forgetting about top speed, long driving range, comfort, space for payload and an attractive design. The final goal is to obtain the official permission to be driven on Austrian roads.

The design and optimization of Ibex's powertrain was the spark that lighted the idea of this Master's thesis, and it will serve as a concrete real example of micro car to be optimized.

Figure 1.1.: TERA Ibex

Table 1.1.: Summary of technical specifications of Ibex

Rated Power	12 Kw
Rated Torque	38 Nm
Type of Engine	3-Phase PMSM
Drive Wheel	Rear Wheel
Battery Capacity	20 kWh
Estimated Range	200 km
Curb Weight	450 kg
Battery Weight	200 kg
Seats	2

The electric motor is a prototype donated by the Magna Steyr AG [1]. It was originally developed as a secondary electric motor that would aid the rear axle of a hybrid vehicle, working also as a generator by means of capturing the kinetic energy during braking. Using it as the only power source for the Ibex makes it an interesting challenge for reaching the goals of vehicle dynamics.

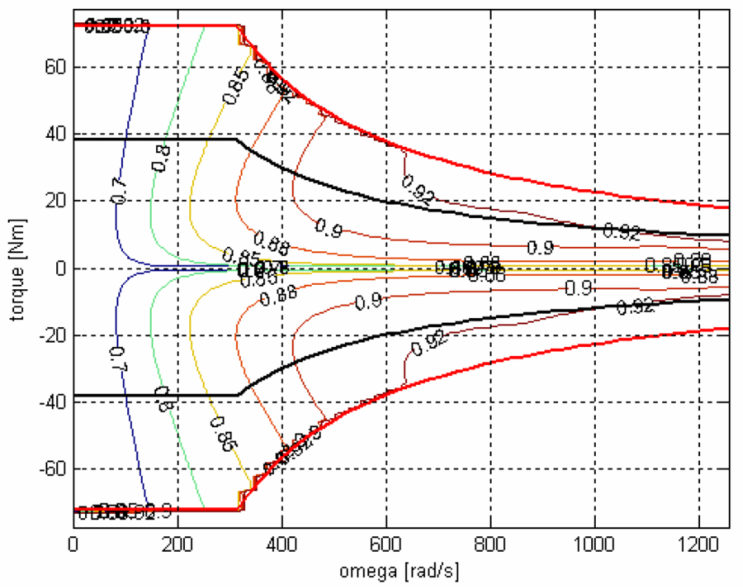


Figure 1.2.: Ibex motor: nominal (black) and maximum (red) torque curve. The contours represent the efficiency ranges.

1.2. Definition of Micro Car

The term "micro car" refers to a very small vehicle. Micro cars originated in the years following World War II, when motorcycles transport was commonly used [2]. This type of cars became popular in the mid-40s [3], when tens of independent manufacturers started small productions of little tri- and quadricycles that were more affordable, lighter and in some cases more convenient than full-sized cars, doing still the job of transporting people and goods. Also known as bubble cars for their look, they were equipped in the post-war era with very little petrol engines of one or two cylinders, with one or two doors (sometimes none), often lacked a reverse gear and were often less than three meters long. Several micro car trucks able to deliver payload across narrow streets were also developed, and the term micro car kept scattering with each unconventional new small vehicle design.

In some jurisdictions these cars were treated as motorcycles in order to attain several advantages: less taxes, softer requirements for the driver in terms of age and/or driving license type, etc. Most countries arranged laws to draw the limits between a motorcycle, a car and these halfway vehicles, framing them in different classifications according to its maximum dimensions, weight, power, propulsion type or maximum speed.

The ultimate goal of the TERA Ibex is to get the official permission to drive on the Austrian roads and therefore the European requisites for its classification must be met, which are regulated by the UNECE, that writes standards valid in Europe, Canada, Russia and US among others.

According to UNECE and as for 2018 [4], the main categories of vehicles are:

- Category M: vehicles carrying passengers
- Category N: vehicles carrying goods
- Category L: 2- and 3-wheel vehicles and quadricycles
- Category T: agricultural and forestry tractors and their trailers.

The EU implemented a new authorization system for L-category vehicles (2-, 3- and 4-wheel vehicles such as motorcycles, mopeds, quads, and minicars). New requirements established under Regulation 168/2013 ensure that future L-category vehicles will pollute less and meet the corresponding safety standards. The old Directive 2002/24/EC was revoked at the end of 2015 and has been replaced by Regulation 168/2013. The EU type-approval regulation set out in these regulations should become a worldwide standard.

According to the European regulatory classification [5], its dimensions, power, weight seat number, steering system and type of propulsion, the TERA Ibex qualifies as a heavy quadricycle on-road, more specifically as a L7e-A2.

Table 1.2.: L7e-A2 specifications

Admitted Propulsions	ICE, Electric Engine, Compressed Air, Hybrid
Terrain	On road
Steering	Only with steering wheel
Number of Seats	1 + Driver
Maximum Curb Weight	$\leq$ 450 kg
Maximum Speed	Not restricted
Maximum Length	$\leq$ 4000 mm
Maximum Width	$\leq$ 2000 mm
Maximum Height	$\leq$ 2500 mm
Maximum Rated Power	$\leq$ 15 kW
OBD Systems	OBD Stage I and II
European Driving License	B1, B

1.3. State of the Art: Electric Micro Cars

According to the previous definition, a market analysis of some of the most representative electric driven micro cars is performed. They are described within the technical specifications and then compared with the rest to get a ground understanding, find common patterns and extract a more comprehensive overview of both this segment as well as the the development targets (see section 2.2) of the TERA car.

1.3.1. Renault - Twizy 80

One of the pioneers and best sellers of their time. Produced by the French company Renault in Spain, this platform has also been branded and sold as “Nissan NMC”. Its 1+1 seat structure has been also converted into a one-seater with increased cargo space for delivery purposes [6].

Figure 1.3.: Renault Twizzy 80 [7]

Rated Power	13 Kw
Top Speed	80 km/h
Curb weight	474 kg
Type of Engine	Induction Motor
Transmission	Single Speed
Vehicle Class	L7e
Seats	2
Release Year	2012

Table 1.3.: Technical specs [7] of Twizy 80

1.3.2. Open Motors - TABBY EVO

TABBY EVO is a hardware open-source hardware platform for electric vehicles. It comes without a bodywork (see figure 1.4) and as an open source framework, it can be used for the creation and optimization of EV: it can be used to bootstrap businesses (electric vehicle start-ups), to create own vehicles, for education purposes, etc.

Figure 1.4.: Tabby Evo [8]

Rated Power	19 Kw
Top Speed	100 km/h
Curb weight without bodywork	380 kg
Type of Engine	Induction Motor
Transmission	Single Speed
Vehicle Class	L7e
Seats	2
Release Year	2016

Table 1.4.: Technical specs [8] of Tabby EVO

1.3.3. Tazzari - Zero City

The Tazzari Zero is a battery electric L7e heavy quadricycle produced by the Tazzari Group, in Imola, Italy. The color of the chassis, the roof and the rims can be independently chosen among 20 colors. The interiors are also offered in 16 different options. The company also offers a lighter and a heavier quadricycle (vehicle class L6e and M1 respectively) and different car body styles, as roadsters.

Figure 1.5.: Tazzari Zero City [9]

Rated Power	15 Kw
Top Speed	90 km/h
Curb weight	450 kg
Type of Engine	3-Phase Asynchronous
Transmission	Single Speed
Vehicle Class	L7e
Seats	2
Release Year	2017

Table 1.5.: Technical specs [9] of Zero City

1.3.4. Eli – Zero+

Eli EVs is an American-Chinese company that designs in California but produces in Beijing. The concept behind this model is to be an uncomplicated, friendlier, more engaging, less intrusive and more agile vehicle for everyday use [10].

Figure 1.6.: Eli - Zero+ Evo [11]

Rated Power	5 Kw
Top Speed	40 km/h
Curb weight	350 kg
Type of Engine	Unspecified
Transmission	Single Speed
Vehicle Class	L6e
Seats	2
Release Year	2020

Table 1.6.: Technical specs [11] of Eli Zero+

1.3.5. Global Electric Motorcars - GEM e2

The American company Global Electric Motorcars (GEM) has produced low-speed, neighborhood electric vehicles since 1998 [12]. These vehicles have car-like safety features, including automotive-glass windshields, suspension, and roofs that meet SAE crush standards. The GEM e2 has a top speed of 40 km/h and is street legal in nearly all US states on public roads posted at 35 mph (56 km/h) or less.

Figure 1.7.: GEM e2 [13]

Rated Power	5 Kw
Top Speed	40 km/h
Curb weight	540 kg
Type of Engine	Induction Motor
Transmission	Single Speed
Vehicle Class	L7e
Seats	2
Release Year	2015

Table 1.7.: Technical specs [13] of GEM e2

1.3.6. Micro – Microlino

An electrified attempt to update the iconic BMW Isetta [14] bubble car, it promises[15] the ideal mix between a motorbike and a car. Designed in Switzerland. After building the first prototypes in China, received 500 reservations in just 2 days, and that kicked off the series production [15].

Figure 1.8.: Microlino [15]

Rated Power	11 Kw
Top Speed	90 km/h
Curb weight	450 kg
Type of Engine	Induction Motor
Transmission	Single Speed
Vehicle Class	L7e
Seats	2
Release Year	2019

Table 1.8.: Technical specs [15] of Microlino

1.3.7. Smart EQ fortwo

An all-electric version of the original smart fortwo, the Smart Fortwo electric drive (later rebranded to EQ [16]), began development in 2006 [17]. The new smart generation introduced is characterised by an evolutionary development of the typical smart silhouette [16], and that includes all the comfort and driving performance of the combustion engine version.

As Tazzari (see section 1.3.3), smart offers the EQ fortwo with a cabrio car body style and a high degree of customization, including several colors for the roof, body and wheel rims.

Figure 1.9.: Smart EQ Fortwo [18]

Rated Power	41 Kw
Top Speed	130 km/h
Curb weight	1080 kg
Type of Engine	3-Phase PMSM
Transmission	Single Speed
Vehicle Category	M1
Seats	2
Release Year	2020

Table 1.9.: Technical specs [18] of smart EQ fortwo

1.3.8. Mini Electric

The MINI Electric (referred to in some markets as the Mini Cooper SE [19]) is an all-electric Mini launched in 2020. The drive train utilizes technology developed for the earlier BMW i3 [20] and should not be confused with the 2010 Mini E which was a limited production vehicle used as a technology and market test platform. [21]

Figure 1.10.: Smart EQ Fortwo [22]

Rated Power	135 Kw
Top Speed	150 km/h
Curb weight	1440 kg
Type of Engine	3-Phase PMSM
Transmission	Single Speed
Vehicle Category	M1
Seats	4
Release Year	2020

Table 1.10.: Technical specs [22] of Mini Electric

1.3.9. TU München – Visio.M (Prototype)

The Visio.M venture was born in the Institute of Automotive Engineering of the Technical University of Munich and funded by the German Federal Ministry for Education and Research with a total of 7.1 million euro [23].

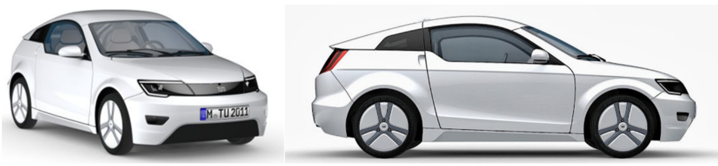


Figure 1.11.: Visio.M Prototype [24]

Rated Power	15 Kw
Top Speed	120 km/h
Curb weight	450 kg
Type of Engine	Induction Motor
Transmission	Single Speed
Vehicle Category	L7e
Seats	2
Release Year	2014

Table 1.11.: Technical specs [24] of Visio.M

Well known companies of the German automotive industry, together with scientists from the TU Muenchen, explored how the price and safety of small, efficient electric vehicles can be brought to a level enabling them to achieve a significant share of the mass market [25]. The gearbox is a single speed with torque vectoring, which doses the torque supplied to each wheel of the rear axle as required [24].

1.3.10. State of the Art: Comparison

The previously listed vehicles belong to a narrow segment of electric cars, yet they serve for different purposes and therefore have fairly characteristics.

In order to find a common ground in the disparity of their specifications, the power-to-weight ratio is chosen and used to understand better the expected design requirements for the TERA Ibex.

$$\boxed{PW_r = \frac{P_{Nom}}{W_v}} \tag{1.1}$$

PW_r: Power-to-weight ratio
P_{Nom}: Nominal motor power
W_v: Vehicle mass incl. battery

Most of the listed cars have a PW_r ranging from 20 to 40, except for two outliers above the upper limit and two outliers below the low end of the range. Above the range, the Tabby Evo and the Mini Electric are found. The former (PW_r=50) is a light platform without bodywork, and must be considered as an unfinished car. The later (PW_r=82,3) is a M1-classified car with four seats and a higher-end motor rated power. Below the 20-40 PW_r range, GEM e2 and Eli Zero+ are found, reaching a PW_r equal to 9,3 and 10,2 respectively, and should be considered as “neighborhood vehicles” rather than fully capable urban cars.

Vehicle	Power [W]	Weight* [kg]	PWr [-]	Top Speed [km/h]
Tabby EVO	19000	380	50,0	100
Twizy 80	13000	474	27,4	80
Zero City	15000	450	33,3	90
GEM e2	5000	540	9,3	40
Eli Zero+	5000	490	10,2	40
Microlino	11000	450	24,4	90
Mini Electric	135000	1640	82,3	130
EQ fortwo	41000	1080	38,0	130
Visio.M	15000	530	28,3	120
TERA Ibex	12000	650	18,5	120

Table 1.12.: Specification comparison of L7e and M1 electric micro cars.
$*$Weight includes the battery mass

After plotting the data from table 1.12 in a visual fashion (see figure 1.12), it becomes clear than the Ibex has not only one of the lowest motor power absolutely, but also relatively to the others if comparing the power-to-weight ratios of the list. Although it has one of the lowest power-to weight ratios (PW_r=18,5) of the list, yet it aims to reach one of the highest top speeds of all (see section 2.2).

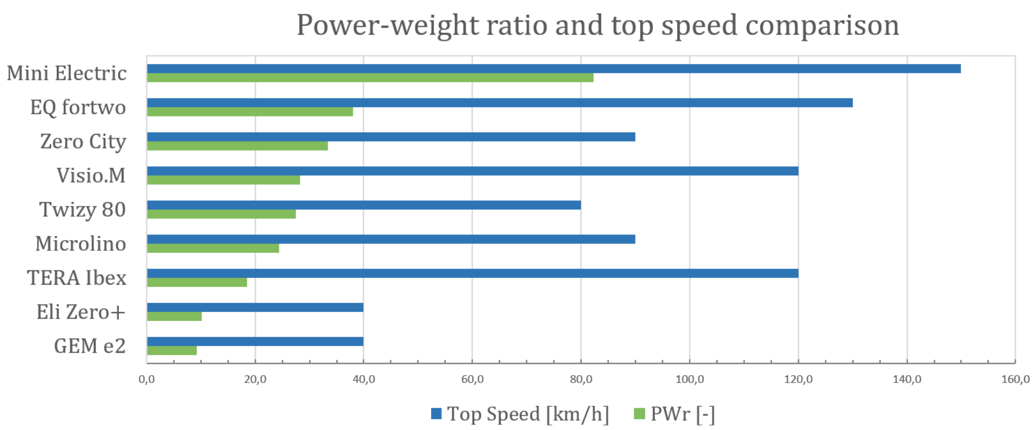


Figure 1.12.: Power-weight ratio and top speed comparison [table 1.12]

1.4. Automotive Transmissions

The purpose of any automotive transmission is to provide a vehicle with distributed propulsion. That is often achieved by a gear train between the power unit (be it an ICE, electric motor or a hybrid combination) and the drive wheels. The propulsion should be smooth, efficient and reliable.

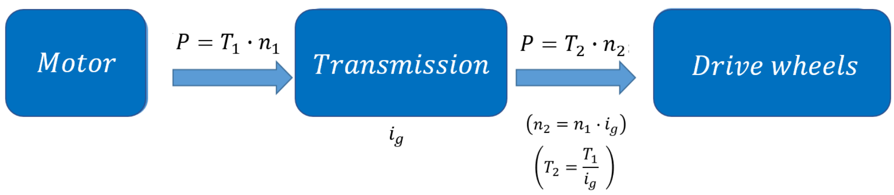
A gearbox is a mechanical system that transmits power converting torque and speed from the power source (engine) to the driven wheels. It usually has different speed ratios to maximize the benefits of the mechanical advantage.

The gearbox reduces the speed of the engine or electric motor thus achieving a torque increase on the wheels while keeping the transmitted power constant.

$$\boxed{P = \mathcal{T}\ n} \tag{1.2}$$

P: Power
$\mathcal{T}$: Torque
n: Rotational speed

Figure 1.13.: Power torque and speed distribution



(1.3)

T_1:	Motor torque	T_2:	Driveshaft torque
n_1:	Motor speed	n_2:	Driveshaft speed
		i_g:	Total gear ratio

Selection of transmission ratios is the crucial factor that determines the conversion of the engine map into the traction force available on the wheel. Provision of a sufficiently high traction force is necessary to overcome the situation-dependent driving resistance that occurs [26]. In this regard, there are few requirements groups that must always be satisfied:

1. The vehicle must overcome the driving resistance at a desired maximum velocity on level ground (without acceleration). As a rule this requirement determines the lowest gear ratio (highest gear).
2. The vehicle must have sufficient gradability as well as acceleration capacity from standstill.
3. The engine operating points must be advantageously selected depending on the driving situation, in order to design the driving power values in the best manner possible, and on the other hand to keep energy consumption as low as possible.

1.4.1. Transmission Classification

The automotive transmissions can be classified as either having a fixed gear translation, which remains constant and cannot be changed (single speed translation) or having the ability to shift between different amount of gears (shiftable) as indicated in the sketch of figure 1.14.

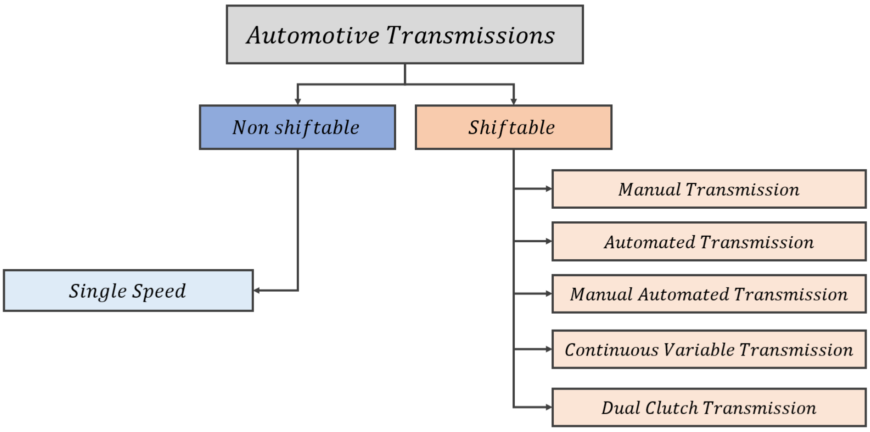


Figure 1.14.: Classification of the main automotive transmissions

1.4.2. Single Speed Layout

According to a state of the art research, the single speed (SS) it is the most representative solution found in electric cars: a reduction gearbox with only one gear reduction. Its advantages are the robustness, simplicity, low weight, low cost maintenance, and often a good compromise for performance and efficiency. The trade-off are that the maximum speed, the acceleration or the efficiency compromise one another. As the electric vehicles can spin the motor backwards to go in reverse, there is no need of a reverse gear.

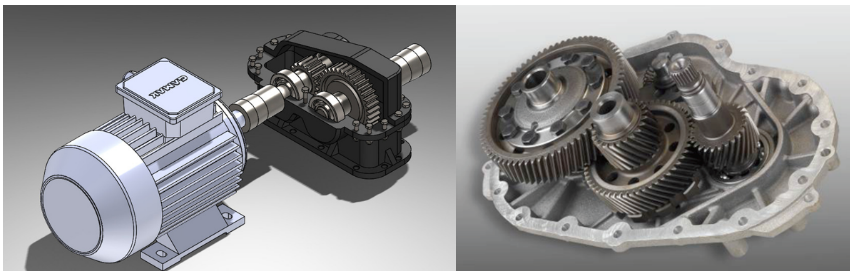


Figure 1.15.: Single stage reducer [27]

1.4.3. Shiftable Transmissions

The need for a multi-gear transmission in an automobile is classically a consequence of the characteristics of the ICE. These engines typically operate over a range of 600 to about 7000 rpm (though this varies, and is typically less for diesel engines), while the car's wheels rotate between 0 RPM and around 1800 RPM. That is a very narrow range and the transmission must provide enough speed, torque and maintain a good fuel efficiency.

The multi-gear transmissions for EV is usually limited to extreme performance sports cars (acceleration/top-speed), heavy load/steep grades trucking, and off-road applications. Various investigations show that multiple ratios (specially two) provided by a transmission help to improve efficiency and driving range in EVs [28]. For this reason, this paper will investigate in detail dual-speed transmissions.

1.4.4. Clutches

A clutch is a component that is used in various designs, virtually in all shiftable transmission; they connect two shafts together in such a manner that torque can be transmitted. In this process, different physical principles are employed and a distinction is made between shiftable (e.g friction clutches) and nonshiftable clutches (e.g. elastic clutches) [29].

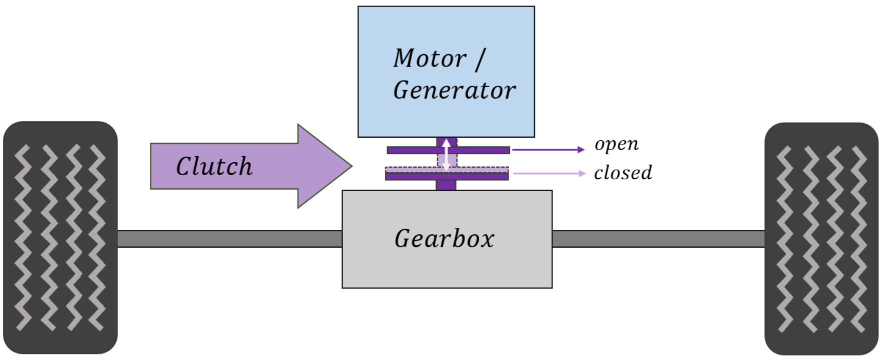


Figure 1.16.: Sketch of an ideal single dry-clutch friction disc

Structure of Dry Clutches

Dry Friction clutches are characterised by a high level of efficiency (no drag torque) as well as by a small moment of inertia, whereby the moment of inertia can increase considerably with additional demands regarding damping or starts [29]. They consists of at least one friction system that includes two friction plates pressed against each other. Depending on the amount of clutch plates, single-plate and multi-plate clutches are distinguished .

A single-plate dry clutch of diaphragm spring design for passenger car applications consists of the following assembly groups:

- A clutch plate almost exclusively with integrated torsional vibration damper, which is moveable axially on the transmission input shaft
- A pressure plate assembly mounted to the flywheel
- A clutch actuation with release device, which transfers the release travel from the non-rotating actuation elements to the pressure plate assembly by means of a release bearing and a sliding sleeve [29]

Structure of Wet Clutches

The wet friction clutches; their advantages include small mass, small moment of inertia, good control as well as a high power/weight ratio and large torque capacity. They are suited to vehicles with little installation space and high torque. Wet clutches are generally built in a multi-plate design, immersed by transmission oil in an oil-tight housing [29]. In contrast to dry clutches, wet clutches have a higher order of complexity because of the oil supply. In the other hand, the oil provides a high heat capacity and higher wear resistance.

Structure of Dual Clutches

Dual clutches are two clutches actuated independently of each other that each serve one of two independent sub-gearboxes of a dual clutch transmission [29], making powershifts possible.

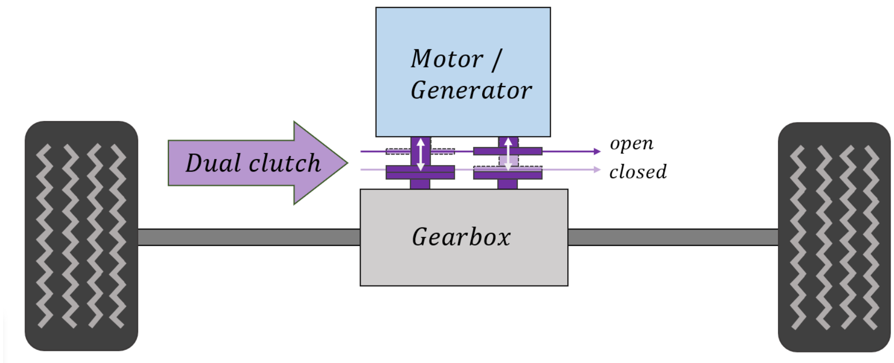


Figure 1.17.: Sketch of a simple DCTs

There are also hydrodynamic torque converters and magnetic clutches.

1.4.5. Gearshifting Strategies

Based on the motor characteristic, an excellent launch performance can be achieved or the fuel economy can be optimized. This trade-off leads to very different designs and strategies. A driving and shifting comfort as felt by the driver stands in direct correlation with the characteristic of the transmission output torque and thus with vehicle acceleration during shifting.

Grade	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**
Attribute evaluation (gearshift)	extremely heavy jerk	heavy jerk	jerk	very obvious	obvious	well perceptible	perceptible	slightly perceptible	barely noticeable	not noticeable
Customer satisfaction	very dissatisfied				slightly dissatisfied	pretty satisfied		very satisfied	outstandingly satisfied	

Figure 1.18.: Shifting rating according to ATZ [29]

The quality of the gearshift is documented with an ATZ grade [26]. In modern powershift transmissions, assessments lower than 7 are only acceptable in exceptional situations (e.g. cold start). The appraisal of what makes a good gearshift is subjective and depends on the desired vehicle image. The target definition ranges from sporty, i.e. gearshifts have to be clearly perceptible, to comfort oriented, i.e. not noticeable. Criteria for judging the shifting process are:

- shifting comfort (jerk, noise, frequency)
- spontaneity (deadtime, shifting duration, acceleration and
- careful calibration [29]

Accordingly, strategies and algorithms are part of the management system. Usually, the different electronic control units are arranged in a hierarchy, this means that the management coordinates and controls other control units like transmission, engine, battery, and brake systems, depending on the operational modes [26].

In this investigation, the comfort does not play any role, and it is the energy efficiency what has the maximum priority.

1.4.6. Manual Transmissions

In the multi-gear transmissions group, some are manual-operated, in which the driver is responsible to change between gears by means of a shifter knob or gear stick. Manual transmissions (MTs) have in many markets (among them Europe and Japan) the highest market share [26].

In the multi-gear transmissions group, these are the manual-operated, which means that the driver is responsible to change between gears by means of a shifter knob or gear stick that is used in combination of the clutch. Manual transmissions come with different numbers of gear ratios, and depending on the power train configuration, different layouts and numbers of shafts are used [26].

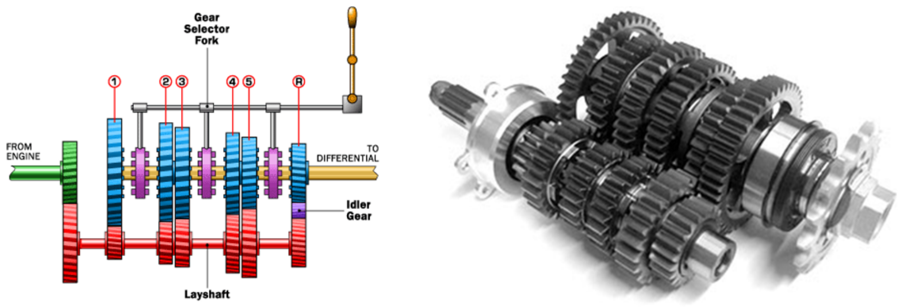


Figure 1.19.: Multi-speed manual transmission [30]

1.4.7. Automated Transmission

Automatic transmissions with various gear ratios consisting of a torque converter with a rear-mounted planetary-type gearbox are known as conventional automatic transmissions (ATs) [29]. ATs or more exactly stepped ATs provide, full powershift functionality. The driver does not need to change gear ratios or actuate any device; this is all handled by the transmission and its control; the torque hands over between different ratios without interrupting the power flow to the output shaft. [26].

An effective coupling of planetary gear sets allows automatic transmissions with a large number of gears. However, the selection of ratios is restricted, since the individual gearwheels are used for several gears. The individual planetary gear sets are arranged in a row like discs.

To provide multiple ratios most modern ATs use planetary gear sets. They offer a high power density for the transmission's power transfer, but limit the freedom designing and adjusting the individual ratios. Moreover, more planetary gear sets also means a greater transmission length size, as visible in the center of figure 1.20, with the planetary gear sets coloured in in orange and fuchsia.

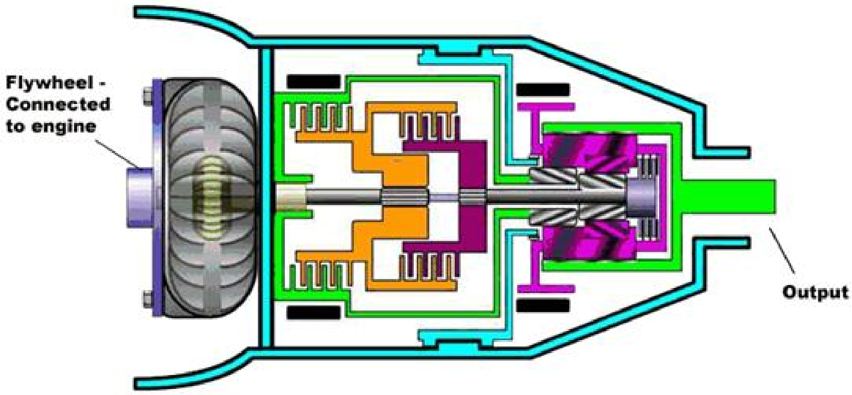


Figure 1.20.: Automatic transmission (planetary gear sets) [31]

1.4.8. Automated Manual Transmissions

When manual transmissions for passenger cars began to be automated, the term “Semi-automatic transmission” was used. The term referred to the two operations “Engaging the clutch/Moving-off” and “Changing gear” [29], and are also called automated manual transmissions (AMTs).

AMTs are mainly used in commercial vehicles as well as A- and B-segment passenger- and sports cars. The high efficiency of MT remains and they are as easy to operate as automatic transmissions. For applications where primary objectives are fuel economy and easy operation, AMTs are being considered [26]. The biggest difference to automatic powershift transmissions for the user is the less comfortable gearshifting, which is subject to power interruption according to its design principles, as occurs with MTs.

1.4.9. Continuously Variable Transmission

The continuously variable transmissions used in mass-produced passenger cars are almost exclusively pulley transmissions [29]. The central component of the pulley transmission is the variator. The variator provides the ratios with the ability to change them continuously. Both: torque converters and clutches are used as launch devices for CVTs. Usually, a planetary gear set provides a reverse gear [26].

The power is transmitted frictionally through the chain or belt, which runs between two axially adjustable taper discs. Through the axial adjustment of the taper discs, the chain runs on variable diameters, infinitely varying the ratio (see figure 1.21). This allows for a maximum efficiency of the motor and continuous acceleration, which is good for fuel economy.

The torque-related pressure of the taper discs on the chain requires a lot of attention, since excessive pressure reduces the efficiency of the chain, leading to increased power consumption, power loss by the contact pressure pump and above all an increased stress on the transmission. It is also essential to prevent the chain slipping, since this would inevitably lead to destruction of the transmission. This makes the design and reliability of the contact pressure pump, and its control, a critical factor in these continuously variable transmissions [29].

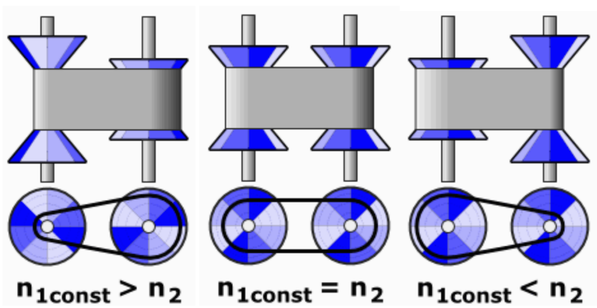


Figure 1.21.: CVT sketch [32]

In order to increase the overall gear ratio of the continuously variable transmission beyond the normal 6.0 to 6.5 of the variator [29], selector gearboxes with spur gears or planetary gears are front- or rear-mounted.

1.4.10. Dual Clutch Transmission

Dual clutches (see last part of section 1.4.4) are not only naming the transmission technology, they are mandatory to provide powershift capabilities. They operate as launch devices and also finalize the ratio changes in the transmission. Dual clutches are connecting to concentric input shafts and each clutch serves a part of the transmission mostly distinguished by even and odd gears [26]. Many applications use a single dual mass flywheel between engines crankshaft and the dual clutch to reduce the torsional vibrations from a combustion engine operation into the gearbox.

Dual clutch transmissions (DCTs) combine efficiency, sportiness, and comfort. The overall transmission efficiency is influenced by many parameters like friction, drag torques, and power demands of the actuation and cooling system [33].

DCTs combine the benefits of MTs and ATs, as they achieve:

- Dynamic driving without manual shifting
- Fuel economy comparable the MTs
- Powershifts like in ATs
- Individual gear ratio flexibility
- Start–stop and hybrid ready [26]

1.4.11. Transmission of Electric Vehicles

Some high-end, sports EVs producers have developed shiftable transmissions in order to maximize the performance while increasing the range. Two good examples of that are the Porsche Taycan 2020 and the Rimac Concept One.

Porsche Taycan

The Taycan's rear axle uses a two-speed gearbox for improved acceleration and range. The Porsche's gearbox has a single planetary gearset and two clutches that handle the ratio swap or decouple the rear motor altogether, allowing for efficient running using only front-axle power. The gearing step is large, with the second-gear ratio roughly half of the first and the shift point happening around 80 km/h. In most driving, the Taycan runs in top gear. First-gear starts happen in Sport mode [34].

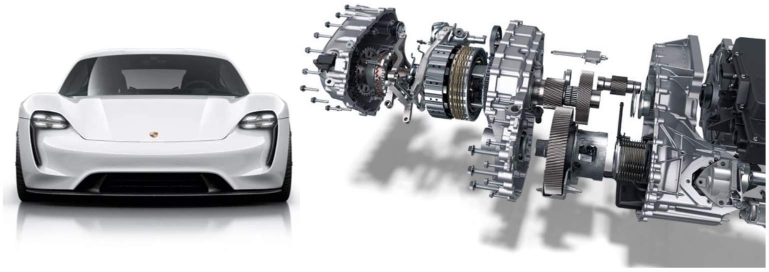


Figure 1.22.: Porsche Taycan Turbo 4S [35]

Rimac Concept One

Rimac Automobili's unique powertrain is divided into four sub-systems. Each system consists of a separate electric powertrain with an independent inverter, motor and gearbox for each wheel of the car. This allows to control each wheel independently, hundreds of times each second, in both directions [36].

Figure 1.23.: Rimac Automobili, Concept One [36]

While the front motors feature single-speed gearboxes, the rear motors carry a two-speed double-clutch gearbox on each side.

Tesla Model S Dual Motor

In dual-motor versions of the Model S, Tesla has long used a setup in which the motors were geared differently, prioritizing the use of the front or rear axle depending of the performance required and accomplishing some of the same high-speed efficiency goals. The original Tesla Roadster was supposed to use a two-speed transmission, but a fixed reduction-gear setup was substituted in after the unit wasn't lasting in development cars, as it wasn't able to handle the electric motor's tremendous torque output [37], and at the end had to be called for revision and locked into one of the gears mechanically.

Transmission Comparison

Rating the transmissions for potential use in the target vehicle of this thesis with a scale such as a triple "x" mark is the best and the single-x mark is the worst, the different transmissions are evaluated.

	SS	MT	AT	AMT	CVT	DCT
Quickness	-	x	xxx	xx	xxx	xxx
Efficiency	xx	xxx	x	xxx	xxx	xxx
Space demand	xxx	xxx	x	xx	x	xx
Complexity	xxx	xx	xx	x	x	xx
Performance	x	xx	xx	xx	xx	xxx
Cost	xxx	xxx	x	x	x	x
Maintenance	xxx	xxx	xx	xx	x	xx
gear spread	-	xxx	xxx	xxx	xx	xxx

Figure 1.24.: EV transmission comparison

Considering the specific case of the TERA Ibex, the ATs and the CVTs are discarded for their maintenance requirements, its powertrain space demand or because the design complexities already listed above.

2. Framework and Methodology

2.1. Scope of Work

The goal of this paper is to explore different transmission layouts for electric micro cars and specifically to find the most suitable and efficient constellation for the TERA Ibex.

Various mathematical script-based models in Octave are written from scratch to describe the vehicle, motor and different driving cycles, which are then used by a bigger simulation that is able to calculate and compare the energy consumption and clarify the influences and relationships of its components, hence providing a mean to optimize the number of gears, their gear ratios and to provide a prediction about its driving dynamics.

At the end, the outcome delivers a practical solution for the Ibex as well as a comprehensive simulation tool, at hand for others to analyze other micro cars with different motors or other driving scenarios.

2.2. Performance Requirements

The main optimization driver for the present investigations of the transmission is the energy efficiency measured as energy consumption in a driving cycle. But there are some requirements from the TERA team that must be complied, which guide the design process.

1. Maximum speed
   - It must reach a speed of 120 km/h on a flat road
2. Gradability

- It must be able to climb slopes with a 30% inclination

3. Acceleration

   - Ability to accelerate from 0 to 50 km/h in under 10 seconds
   - Ability to accelerate from 0 to 100 km/h in under 25 seconds

2.3. Design Constraints

The design constraints are a blend of dimensional, regulatory, legal, and electrical boundary conditions. The assumed and approximated data or coefficients are also defined in this section.

- Legal European requirements of size L7e-A2 (see table 1.2)

Table 2.1.: Powertrain boundaries

Type of Electric Motor	PMSM
Operating/Max Power	12/23 kW
Operating/Max Torque	38/72 Nm
Operating/Max Speed	3000/12000 rpm
Rotor Inertia	$\leq$ 0,0047 kgm^2
DC Link Voltage	300 V
Peak Current	160 A
Continuous Dissipated Power	1600 W
Cooling	Liquid
Type of Battery	Lithium Ion Cells
Battery Capacity	20 kWh
Drive Wheel	Rear Wheel
Type of Suspension	Double Wishbone Arm

Table 2.2.: Dimensional Constraints

Diameter of Output Engine Shaft	32 mm
Maximum Length of the GB	3200 mm
Maximum Width of the GB	2100 mm
Length Ibex	3700 mm
Width Ibex	1600 mm
Height Ibex	1200 mm
Rear Wheel Radius	273 mm
Wheelbase	2310 mm
Outer motor shaft diameter	27 mm
Motor coupling diameter	34 mm
Transmission max height	∼555 mm
Transmission max width	∼450 mm
Transmission max depth	∼430 mm

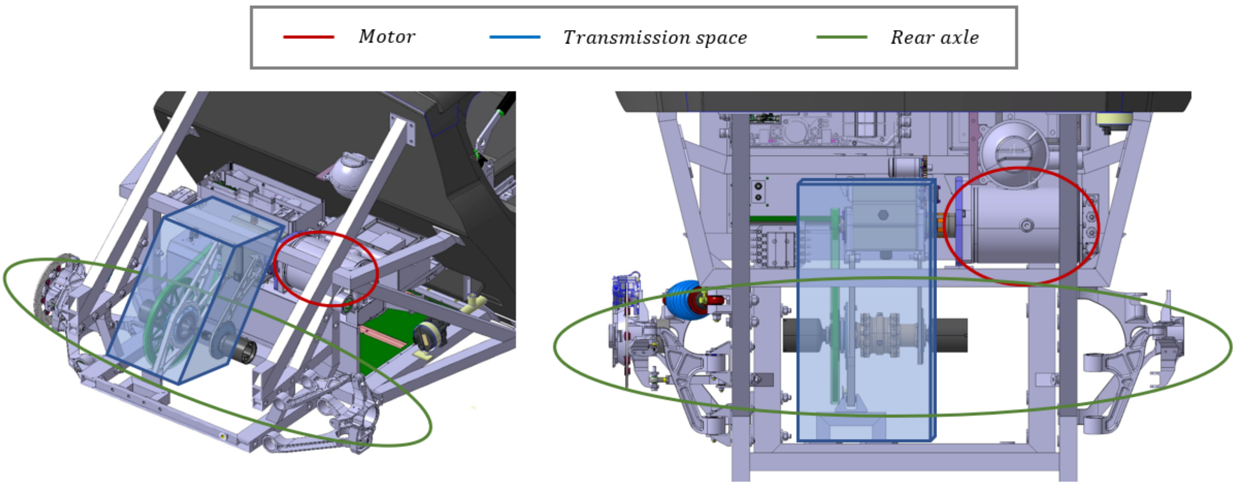


Figure 2.1.: Maximum transmission space allowance TERA Ibex

Table 2.3.: Approximated Parameters

Cross Sectional Area Ibex	1,92 m^2
Air Density	1,25 kg/m^3
Rolling Resistance Coefficient	0,005 [-]
Gear Ratio efficiency	0,97 [-]
Reg. braking as % of the braking force	80%
Regeneration losses by the battery	6%
Regeneration losses by the inverter	6%
Regeneration losses by the tire adhesion	2%

3. Implementation

One of the few reasons why many electric cars can avoid multi-speed gearboxes is given by the nature of the electric motor: they do not stall, have a wide operation range and deliver full torque from zero RPM.

Electric motors however do not generate the maximum torque over the whole operating range; it drops after a certain engine speed. And although they achieve more than double the efficiency “tank to wheel” compared with their internal combustion counterparts [38], the efficiency varies with the torque and speed.

3.1. Resistance Forces

The transmission must ensure that the torque and speed is sufficient as well as effectively transmitted. To do so, and before considering the torque and speed distribution, the longitudinal direction forces are analyzed. Other directions are neglected as they almost only influence steering and suspension. The result will prove if performance requirements can be matched with the given specifications of vehicle and motor. These forces will act as building blocks for the power consumption calculation in section 3.5 on page 45.

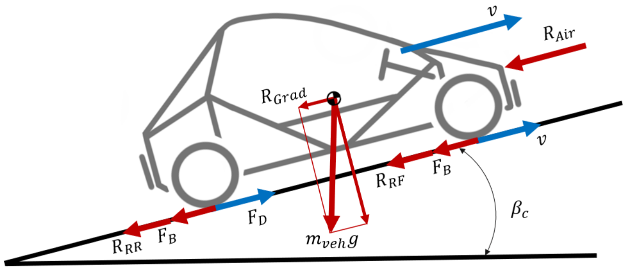


Figure 3.1.: Sketch of several longitudinal forces acting on a vehicle

3.1.1. Rolling Resistance Forces

The rolling resistance force is an interaction between the ground and the wheel, which is independent of speed assuming a ground surface that is completely flat and rigid. The forces on each wheel vary with the weight distribution, but in the present consideration it will be modelled as a single force considering the vehicle total mass.

$$\boxed{R_{roll} \approx c_{roll}\ m_{veh}\ g\ cos\beta_c\ sign(v)} \tag{3.1}$$

R_{roll}:	Rolling resistance	g:	Gravity
c_{roll}:	R_{roll} coefficient	β_c:	Angle
m_{veh}:	Vehicle mass	v:	Vehicle speed

3.1.2. Drag or Aerodynamic Forces

The aerodynamic forces are mostly present when the vehicle gains speed. It is caused by the air, forced to flow around the vehicle, causing a higher pressure at the front and a lower pressure at the back. It occurs all around the vehicle, although modelled as a single force.

$$\boxed{R_{air} = \frac{1}{2}\, c_{air}\, A_{veh}\, \rho_{air}\, v\, |v|} \tag{3.2}$$

R_{air}:	Aerodynamic resistance	A_{veh}:	Cross-sectional area
c_{air}:	Aerodynamic coefficient	ρ_{air}:	Air density

These forces are present as long as the vehicle is driving. By coding the equations and using the speed as input variable it is possible to loop and extract the values over the vehicle speed.

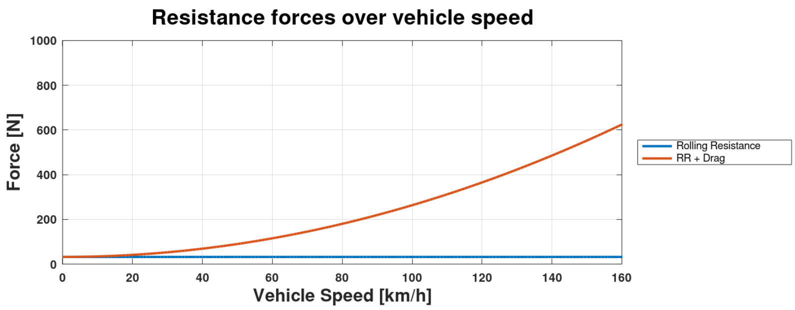


Figure 3.2.: Rolling and drag resistance forces for Ibex specs (table 1.1)

3.1.3. Acceleration Forces and Gradability

In addition to the driving resistance occurring in steady state motion, inertial forces also occur during acceleration and braking. The total mass of the vehicle and the inertial mass of its corresponding rotating parts influence the resistance to acceleration [29]. The acceleration resistance is simplified in this model using a rotational inertia coefficient that express the proportional mass rotating.

$$\boxed{m_* = m_{veh} + \frac{\sum I_{Red}}{r_w^2}} \tag{3.3}$$

$$\boxed{I_{Red} = I_{Axis} + i_{Axis}^2\, (I_{Trans} + i_{Trans}^2\, I_{Eng})} \tag{3.4}$$

I_{Red}:	Moment inertia at driving axle	m_*:	Generalized vehicle mass
I_{Axis}, I_{Trans} :	Axis, transm. moment of inertia	r_w^2:	Rolling resistance torque
i_{Axis}, i_{Trans}:	Axis, transm. gear ratio	R_{grad}:	Climbing resistance

A car's gradability is its ability to climb slopes. The climbing resistance is the proportion of the overall road resistance that is influenced by the road slope. The downward force works against the direction of the travel of the vehicle when the vehicle moves upwards on the inclined plane and when driving downhill it works in the direction of the movement of the vehicle. It is proportional to the gross weight of the vehicle and the inclination angle of the inclined plane on which the vehicle is moving.

$$\boxed{R_{grad} = m_{veh}\ g\ sin\beta_c\ sign(v)} \tag{3.5}$$

$$\boxed{F_{accel} = m_*\ a} \tag{3.6}$$

a: Vehicle acceleration

3.2. Driving Force

The driving force, also known as traction or net force, is the force required to generate or terminate motion between the vehicle and the road. As the description intuitively suggest, it is the force that counterbalances all the acceleration and resistance forces. The numerical solution is obtained solving the simplified equation for longitudinal motion under steady state conditions that neglects tire [39], which also better describes the acceleration force components.

$$\boxed{F_D = F_{accel} + R_{roll} + R_{air} + R_{grad}} \tag{3.7}$$

F_D: Driving force

Equation 3.7 is the key to assess the power required by the vehicle in different scenarios: transient speeds, driving cycles, extreme slopes and any kind of drivetrain-performance requirements compliance. It will be also the foundation to forecast the (instantaneous or accumulated) energy consumption in all those cases.

The driving force can also be a function of the vehicle speed and its rated power. This linear relationship is defined at equation 3.8, and by solving the driving power it shows that such peak power can be calculated and also explained as the the maximum driving force that the motor can deliver at a certain speed.

$$\boxed{F_D = \frac{P_D}{v}} \tag{3.8}$$

P_D: Driving power

As an insightful and visual use case example: for an estimation of the vehicle maximum speed on a horizontal road, the climbing resistance and the acceleration resistance are set to zero. The traction force demand as a function of the rolling resistance, the aerodynamic drag and the power train efficiency [39] can be calculated and matched with the motor power and the traction force it ideally supplies.

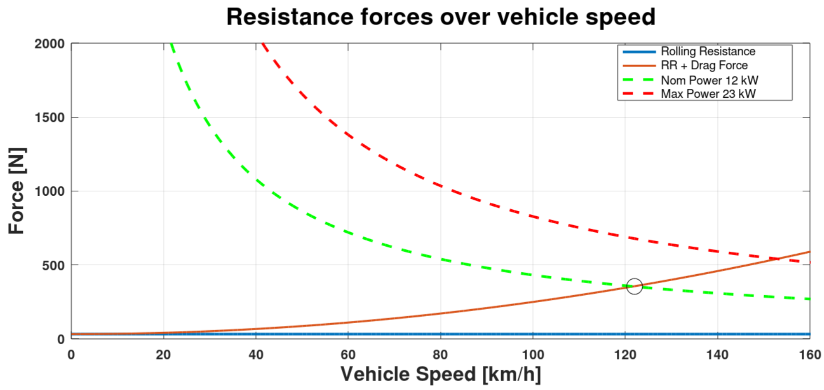


Figure 3.3.: Motor power curves and resistance forces

In the figure 3.3, the equations 3.7 and 3.8 are used to calculate the driving force (in this case that is solely the sum of the rolling resistance and aerodynamic drag forces) and both the nominal and maximum power, and plot them over the speed. The maximum speed on a nominal power operation is found and circled around 123 km/h, where the two curves meet.

3.3. Driving Cycles

A driving cycle is a series of data points representing the speed of a vehicle versus time. Driving cycles are produced by policy makers in different countries to estimate the performance of vehicles in various ways, as for instance fuel consumption, electric vehicle autonomy and polluting emissions. For a particular type of vehicle, fuel consumption and pollutant emission rates are mainly a function of the vehicle's use (journey type, frequency, etc.), and of the vehicle's operating conditions (speed, engine speed, rates of acceleration, temperature conditions, et), and depend on both the traffic conditions and the individual behaviour of the driver. Thus, a realistic assessment of emissions, pollution reduction methods and the effectiveness of emission control technologies cannot be carried out without taking into account the actual operating conditions of the vehicles [40].

Different types of investigations require different driving cycles, therefore it is not possible to have one standard cycle for all purposes. There exists an infinite number of possible driving cycles as there are infinite numbers of possible types of road environments and traffic situations to describe.

It is possible to generate a driving cycle by collecting data points out on a real-life scenario with a specific journey, as well as it is possible to create it artificially to simulate a standard situation.

3.3.1. WLTP

The Worldwide Harmonised Light Vehicles Test Procedure (WLTP) is a global and coordinated standard for determining the amount of particles, gas emissions, and fuel consumption of traditional, hybrid cars and the energy consumption of battery electric vehicles.

It was launched in 2017, replacing the previous and obsolete NEDC cycle. Unlike the NEDC, it features higher speeds, as well as realistic, non-uniform accelerations. WLTP is the standard fuel economy and emission test by the UNECE members (EU-28, Norway, Iceland, Switzerland/Liechtenstein, Turkey and Israel), along with India, South Korea and Japan.

There are three different WLTC test cycles [41], depending on the vehicle class defined by the power/weight ratio PWr in W/kg (rated engine power / kerb weight):

- Class 1 – Low power vehicles with $PWr \leq 22$
- Class 2 – Vehicles with $22 < PWr \leq 34$
- Class 3 – High-power vehicles with $PWr > 34$

The TERA Ibex scores a 18,46 PWr and therefore falls in the WLTC Class 1, the lowest power-to-mass ratio.

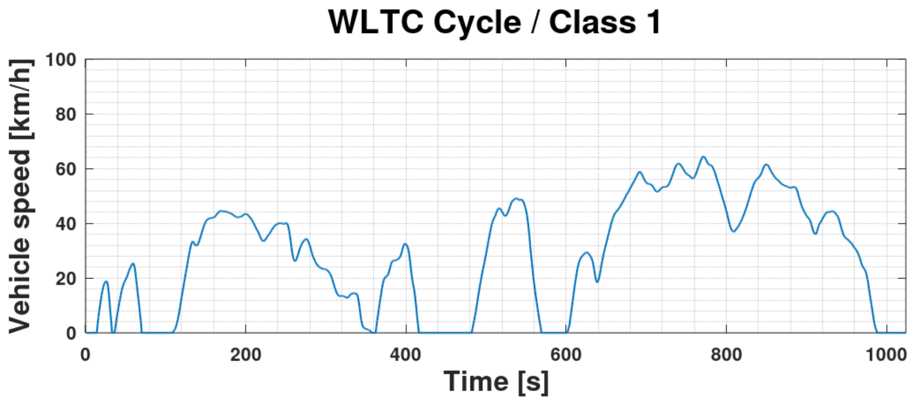


Figure 3.4.: Speed over time, WLTC Class 1 [42]

	Low	Medium	Total
Phase distribution [s]	0-589	590-1022	0-1022
Duration [s]	589	433	1022
Stop duration [s]	155	48	203
Distance[m]	3324	4767	8091
Stop proportion [%]	26,3%	11,1%	19,9%
Max. speed [km/h]	49,1	64,4	64,4
Average speed [km/h]	27,6	44,6	35,6

Table 3.1.: WLTC Class 1 Test cycle data [42]

The WLTC Class 1 test cycle, seen in figure 3.4, has low-speed section followed by another medium-speed phase, as defined in table 3.1. The maximum speed is below 70 km/h; however, it has steep accelerations, which makes arguably a good representation of a conventional urban driving pattern.

3.3.2. Driving Force Demand

Although the driving cycle consists of just speed and time, it is possible to derive the acceleration and with it obtain the required driving forces for the vehicle using the resistance forces defined in chapter 3.1 and 3.2 and plotting them over the vehicle speed. The positive forces must be provided by the motor, while the negative must be counteracted by regular brakes (dissipating this kinetic energy as heat) or by the motor acting as a generator (recovering part of the energy back into the batteries).

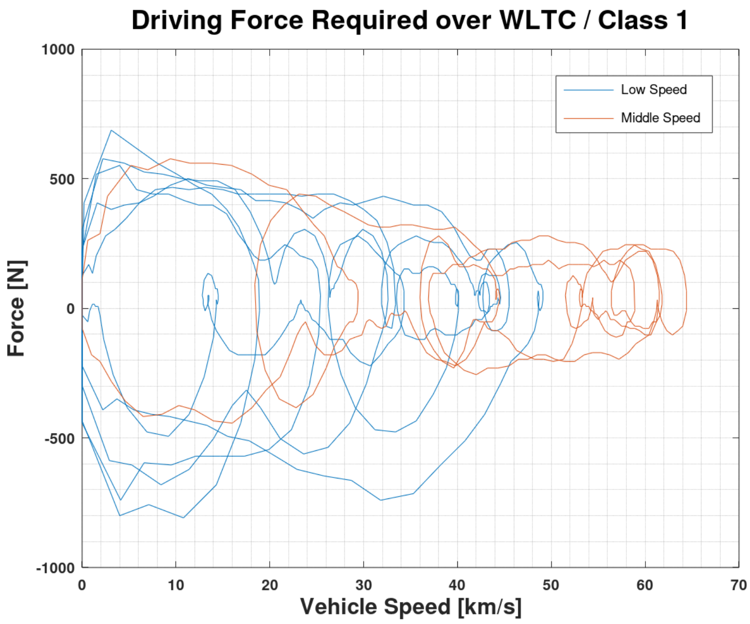


Figure 3.5.: Driving force over Ibex speed, WLTC Class 1 (see table 3.1)

Because the electric motor of the TERA Ibex is a given component and cannot be changed, it is useful to do a quick check of how the vehicle in general and the motor in particular can handle the driving force demand of the WLTC Class 1 cycle. That is easily made by taking the positive values of such driving forces from the figure 3.5 and plotting them along the nominal and maximum power curves.

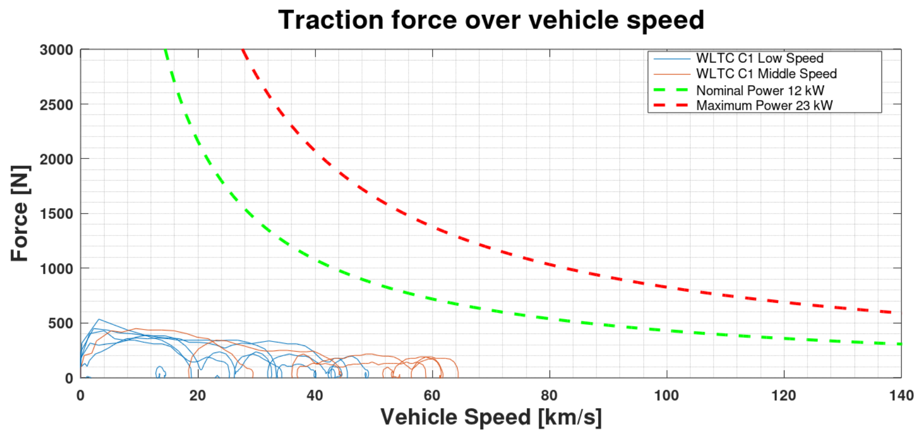


Figure 3.6.: Motor power and driving force required for WLTC Class 1

The required forces and the power curves are plotted in the figure 3.6), where it is visible that the former are well under the latter. Although no gear ratio (GR) is considered and consequently the motor efficiencies are set aside, it provides a high degree of confidence that the motor will not be a limiting factor in order to complete this cycle.

3.4. Electric Motor

The majority of the machines currently used in vehicles are permanent magnet machines or induction machines[43]. PMSM tend to be more expensive than AC induction motors and have been known to be also more difficult to start up. However, the advantages of permanent magnet motors include higher efficiencies, smaller sizes and PMSMs' ability to maintain full torque at low speeds [44].

Permanent Magnet Synchronous Motors (PMSM) have the advantages of small size, light weight, high efficiency and high power density. The electric-driving system based on PMSM has excellent comprehensive performance, such as a high efficiency, high torque/current ratio, strong ability of flux-weakening speed-expansion, low vibration, low noise and fast dynamic response [45]. This is true for the TERA Ibex, although the current state of the art for electric motors has vastly improved most of its characteristics, including the power density, thermal management and a smaller dependency of non-rare earth magnets.

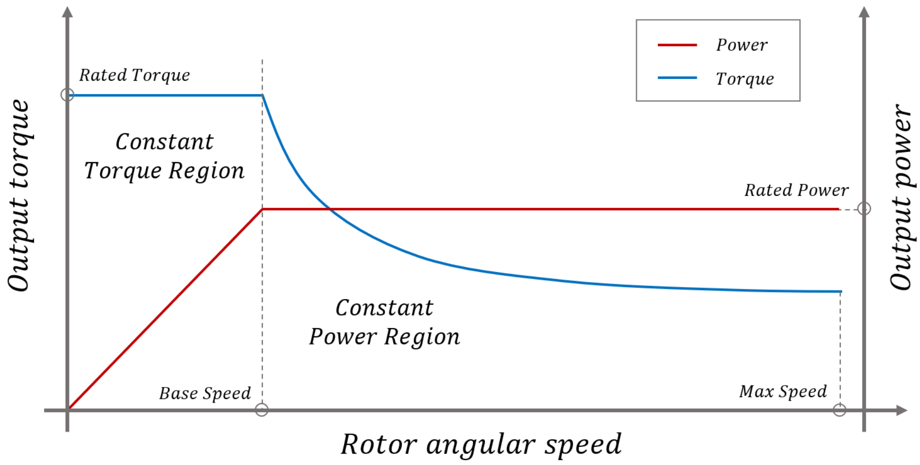


Figure 3.7.: Torque-power characteristics of PMSM

To fundamentally characterize the motor, the torque and motor speed must be narrowed down to its operating limits, including the energy efficiency for each point. The ideal setup is a comprehensive data map with multiple data points. Usually those data are provided by the manufacturer, but in this specific case the coarse picture presented in figure 1.2 was the only documentation available.

The torque-speed curves were accurately mapped by two means: by modelling the curves with peak torque and a two-step function formula, as well as by a power interpolation function (see figure 3.8).

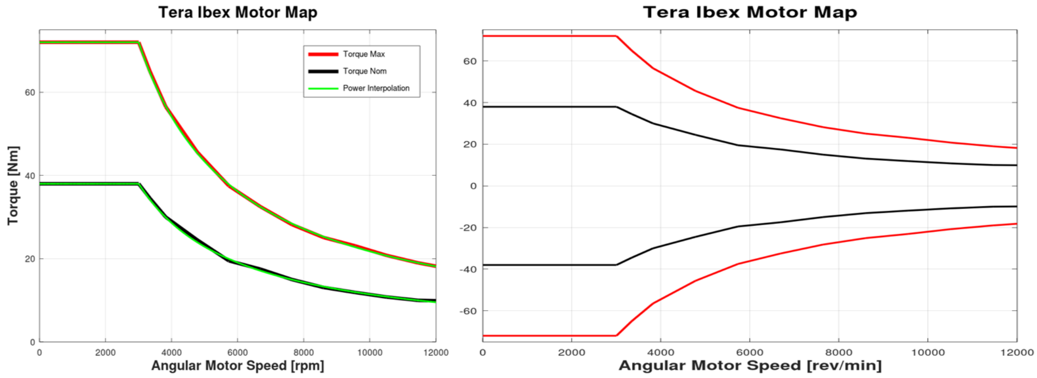


Figure 3.8.: Ibex torque and speed functions

The efficiency values were extracted from the given documentation of figure 1.2, splitting the operating torque and speed in discrete slices of 6 Nm and 155 rad/s each respectively, and storing the corresponding efficiency matrix in a CSV file, arranged to be accessed by the main scripts.

Figure 3.9.: Ibex motor efficiency map

The figure 3.9 display a visual representation of the performance and efficiency values. Using a spreadsheet, the columns represent the angular speed steps, and the row the torque, which can be positive and negative, and in this particular case both positive and negative torque curves are symmetrical. The efficiencies are the values at the cells, and have also a color code for a better look at its distribution. At low speeds (0-155 rad/s) the efficiency is substantially lower (70%) than at higher speeds, where the efficiency reaches 90, 92 and 94%. The marked "sweet spot" is the region where the motor should operate to minimize the energy consumption.

3.5. Power/Energy Consumption

In the chapter 3.1 and 3.2, the driving force and its components across the whole speed range are calculated. Instead of using the speed domain (a ramp from 0 to desired maximum speed), it is possible to feed the equations with driving-cycle speed, hence obtaining the driving forces over the time domain: the instantaneous driving force (F_{Di}).

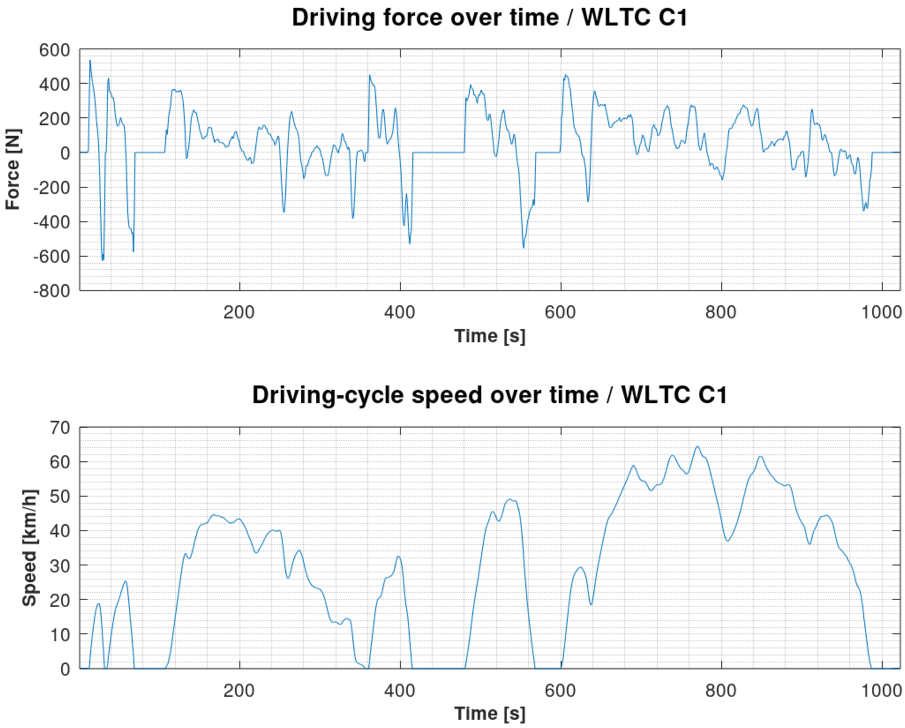


Figure 3.10.: Driving force and cycle speed

The gear reduction is introduced hereafter using the equations 1.3 and the calculated driving force demand F_{Di} is there by reduced using the mentioned gear ratio and also the wheel radius (which means going wheel to motor, opposite direction of that in the sketch of figure 1.13). Likewise, the required linear speed v is then multiplied by the gear ratio and the wheel radius. Consequently, the torque that the motor must supply and the rotational speed that the rotor should be able to reach are calculated and stored in arrays for each time unit of the driving cycle, which in turn is the instantaneous both torque (T_i) and rotational speed (n_i).

In the figure 3.11 the gear ratio of 10:1 was used arbitrarily as example to show the results. The numerical values of both pairs are very different than the ones of 3.10, but the profile of the curve is indisputable the same, because they represent the same magnitude, converted linearly.

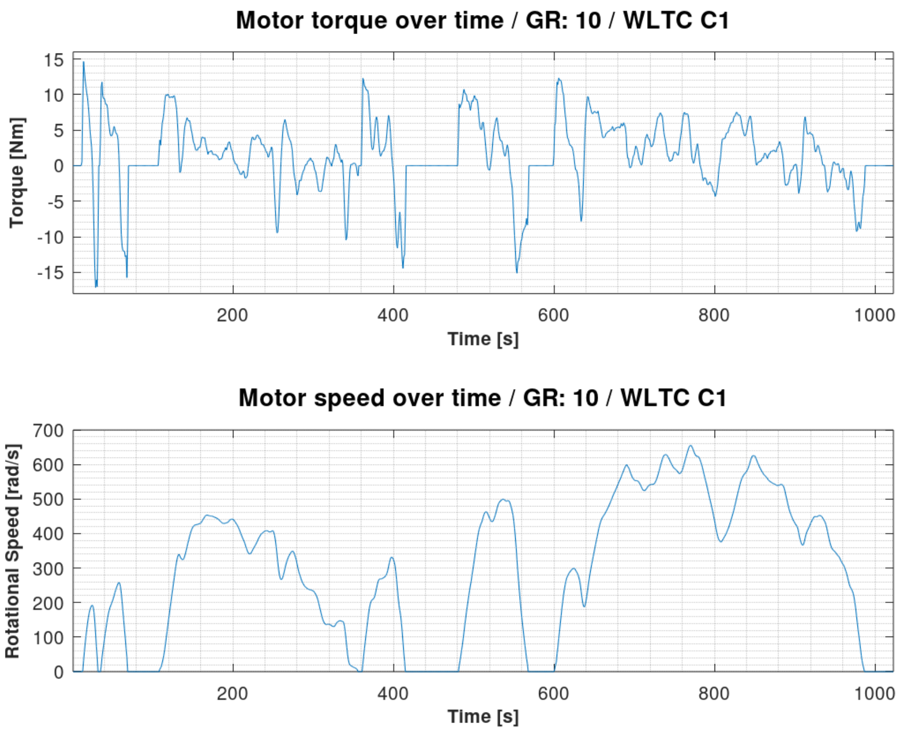


Figure 3.11.: Motor torque and motor speed with GR=10 / WLTC C1

Having calculated the instantaneous torque (T_i) and motor speed (n_i), it is possible to access the motor map and take out the motor efficiency (η) for each time unit: the instantaneous efficiency (η_i). In fact it would be possible to find the efficiencies by hand using the torque and motor speed values of each time unit on figure 3.11 and find its correspondent efficiency in 3.9. This is a simplified consideration due to the limited data available, but should be sufficient for the conceptual layout of the gear ratios performance in the course of the present work

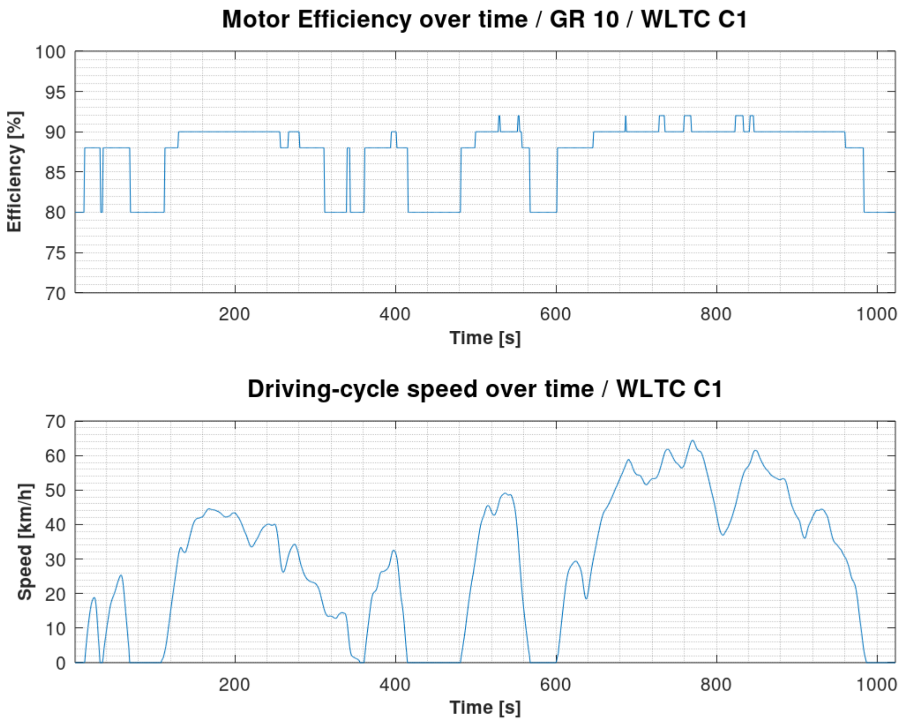


Figure 3.12.: Motor efficiencies and motor speed with GR=10 / WLTC C1

A different way to make visible how the efficiency changes in respect to the speed is by plotting it in a third dimension as represented in figure 3.13, where the relationship of high speeds with high efficiencies is noticeably:

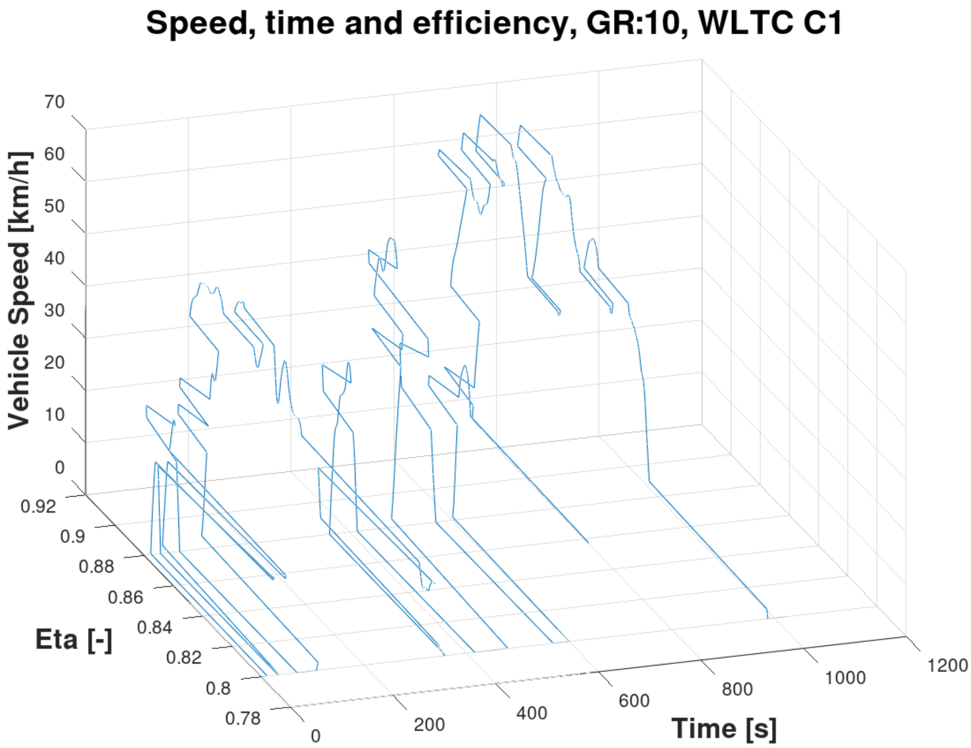


Figure 3.13.: Speed, Time and the instantaneous motor efficiency with GR=10 / WLTC C1

Using F_{Di} and isolating the power in the equation 3.8, the instantaneous driving power (P_{Di}) is then calculated:

$$\boxed{P_{Di} = F_{Di}\; v_i} \tag{3.9}$$

P_{Di}: Instantaneous driving power

The P_{Di} represents the amount of raw power that the motor has to supply to perform the driving cycle accordingly. It can be positive (system requires energy from the battery) or negative (energy has to be drawn from the system, which in sequence it could be stored back in the battery by means of regenerative braking, or just dissipated as heat by the mechanical wheel brakes).

The PMSM motor is relatively very efficient, but it still has inner losses, and that means that it requires more input power than the power it can output. This input power is named the motor power (P_M) and it is the actual amount of energy that the motor withdraws from the battery and electronics.

$$\boxed{P_{Mi} = \frac{P_{Di}}{\eta_i}} \tag{3.10}$$

P_{Mi}: Instantaneous motor power
η_i: Instantaneous motor efficiency

Figure 3.14.: Driving and motor power with GR=10 / WLTC C1

By time-integrating the positive values of instantaneous motor power $P_{Mi(+)}$, the cumulative energy consumption for the driving cycle (E_{dc}) is effectively calculated.

$$\boxed{E_{dc} = \int_0^t P_{Mi(+)}(t)\ dt} \tag{3.11}$$

E_{dc}: Energy consumption for the cycle
$P_{Mi(+)}$: Positive-only values of P_{Mi}
t: Elapsed time of driving cycle

3.6. Regenerative Braking

In EV, regenerative braking is the technology that converts the kinetic energy of the drive wheels by the electric motor (which operates as a generator for this purpose) into electrical energy, that in turns can be stored back in the battery. In this way some of the energy, which is normally lost as frictional heat during braking, is fed in the form of electric energy to the battery and then utilized [46]. At the same time, the motor acting as a generator and its vehicle-braking effect teams up with the wheel brakes, increasing active safety.

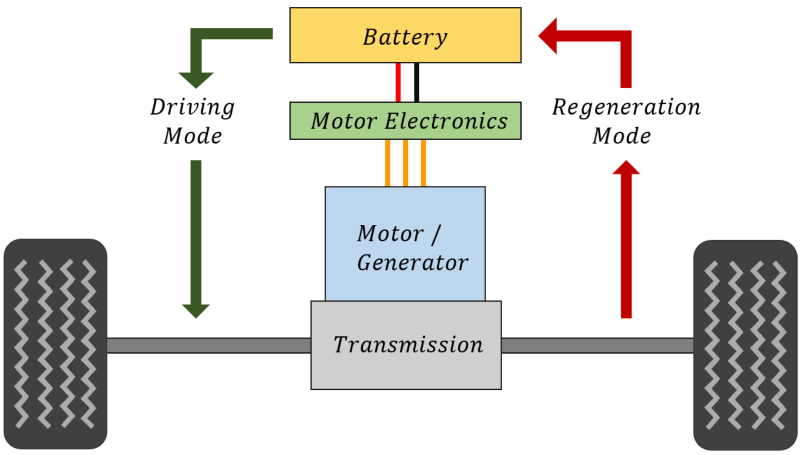


Figure 3.15.: Power flow sketch during regenerative braking

During city driving, the results of a study revealed that approximately 30% of a typical car's power output is lost to the braking process [47]. In another investigation of the effect of regenerative braking, the energy consumption was able to be reduced by up to 25% in short-route driving cycles [48]. For this reason, the regenerative braking influence in an EV must always be considered during its design.

$$\boxed{F_D = F_{Br} + F_{Reg}} \tag{3.12}$$

F_{Br}: Braking force by wheel brakes
F_{Reg}: Braking force by motor-generator

In a research that combined field measurements and exhaustive simulations [49], results indicated that the acting force and generated current show a similar tendency at overall conditions (different velocities, slip ratios and road conditions), and the conclusions will be used to simplify the calculation of the regenerative braking. The efficiency losses taken into account, apart from the motor efficiency, are listed in the table 2.3 and will be used to calculate the total efficiency coefficient for the regeneration braking (η_{Reg}). With it and following the equation 3.13 it is possible to calculate the total energy regeneration for the cycle.

$$\boxed{E_{Reg} = \eta_{Reg} \int_0^t P_{Mi(-)}(t)\ dt} \tag{3.13}$$

E_{Reg}: Regenerated energy for the cycle
$P_{Mi(-)}$: Negative-only values of P_{Mi}
η_{Reg}: Total regeneration efficiency coeff.

POTENTIAL ENERGY REGENERATION [kJ]

Single Speed / WLTC 1

Gear Ratio [-]	1	2	3	4	5	6	7	8	9	10
Pot. Reg Brak [kJ]	NaN	NaN	177,99	181,79	185,15	187,45	188,45	189,58	190,22	190,23

Gear Ratio [-]	11	12	13	14	15	16	17	18	19	20
Pot. Reg Brak [kJ]	190,67	191	191,01	191,1	191,16	191,32	191,41	191,82	NaN	NaN

Figure 3.16.: Potential energy regeneration, single speed layout for WLTC C1

Using the single speed layout as a support for understanding, the theoretical potential for energy regeneration by means of regenerative braking would be as shown in figure 3.16. To calculate the actual regenerated energy, the losses during regeneration (see 3.14) must be applied.

$$\boxed{\eta_{Reg} = Reg_{\%}\ \eta_{mot}\ \eta_{bat}\ \eta_{inv}\ \eta_{tire}\ \eta_{gear}} \tag{3.14}$$

$Reg_{\%}$: Regenerated energy as % of the braking force
η_{bat}: Losses due to the battery
η_{inv}: Losses due to the inverter
η_{tire}: Losses due to the tire slippage
η_{tire}: Mechanical losses in the gears

3.7. Single Speed Transmission

With a deterministic approach, discrete and round gear ratios are looped and fed into the model and the power consumption is calculated. By means of integrating this function, and as seen in 3.11 the total energy consumption is also calculated.

At times, certain gear ratios are simply not suitable for the selected driving cycle: the driving cycle requires that the vehicle drives in a certain speed, or with a certain torque that the motor simply cannot achieve after applying the reducing or multiplying factor of the considered gear ratio. The energy consumption is then stored as not valid and marked as "Not a Number" (NaN) consequently.

ENERGY CONSUMPTION [kJ]

Single Speed / WLTC 1

Gear Ratio [-]	1	2	3	4	5	6	7	8	9	10
No Reg Brak [kJ]	NaN	NaN	1018,3	999,18	992,92	987,64	985,63	980,31	978,03	977,01
With Reg Brak [kJ]	NaN	NaN	831,09	815,94	813,16	810,2	809,15	804,91	803,24	802,23

Gear Ratio [-]	11	12	13	14	15	16	17	18	19	20
No Reg Brak [kJ]	976,85	977,27	977,91	977,4	975,82	973,98	971,6	969,72	NaN	NaN
With Reg Brak [kJ]	802,48	803,2	803,85	803,4	801,86	800,17	797,86	796,36	NaN	NaN

Figure 3.17.: Energy consumption, single speed layout for WLTC C1

On a single speed setup, that leaves a cluster of gear ratios that can handle the driving cycle, and the not-suitable are piled on the sides, for having a too high or a too low gear reduction. In the figure 3.17, those are coloured in grey. The suitable ones, in a coloured scale to ease the spot of the most efficiency (green, i.e. lower energy consumption) and the less efficiency (red, i.e. higher energy consumption). Note that the regenerated energy (calculated in the equation 3.13) is subtracted from the energy consumed, making an almost constant 18% energy reduction, in line with the researched literature (see section 3.6).

As said, the most efficient operating range of the motor is found when the rotor is turning at a high speed. Therefore, and not surprisingly, the gear ratio that show the least energy consumption is the one that can complete the driving cycle and has the biggest reduction.

In the case of no regenerative braking, this is true for the gear reduction 18:1 and a cycle energy consumption of 969,72 kJ. In addition and accounting for energy regeneration, the most efficient gear ratio is also the 18:1, with an net energy consumed of 796,36 kJ.

Comparing the energy consumption of both cases, the reduction resultant of using regenerative braking is really significant, revealing that a 18% of the energy consumed during the driving cycle could be stored back in the battery.

3.8. Dual Speed Transmission

On a dual speed transmission with two gear ratios GR_a and GR_b, the fundamental operations are in essence the same, but the means to reach the results are more complex. In each time step of the calculation, there are always two different potential outcomes, compared against each other. The driving force and vehicle speed are translated for every instant into the torque T_{GRa} and motor speed n_{GRa} for a gear ratio GR_a and into the torque T_{GRb} and motor speed n_{GRb} for a gear ratio GR_b.

With those values, and as explained in the section 3.5, it is possible to obtain the instantaneous efficiencies η_{GRa} and η_{GRb}, thus being able to measure the actual motor power, and to that end bein able to select always the gear ratio that ensures:

- The minimum power consumption if P_{Mi} is positive
- The maximum power consumption if P_{Mi} is negative and there is reg. braking
- The minimum power consumption if P_{Mi} is negative and there is no reg. braking
- Maintain the GR that was geared the previous instant if the P_{Mi} is equal for both GR_a and GR_b

By following this criteria, it is possible to store for each pair of gear ratios:

- An array with the instantaneous gear ratio (see figure 3.23).
- An array with the motor power that will be used for the energy calculations

The model is as well responsible for noticing without discarding a gear ratio that in a given instant cannot deliver the torque or motor speed that is demanded by the driving cycle, as long as the second gear ratio can effectively take over the operation.

for every instant the results in which the requirements of the cycle demand certain torque o motor speed that the motor with one of the gear ratios cannot supply.

In my particular calculation, I set the model with a time step of one second and assumed the ability of a gear shift in under a second, i.e. making possible to evaluate and select for the most efficient gear ratio at each moment, and providing a criteria for the gear shift operation.

That, aggravated by the coarseness of the ibex motor efficiency map, makes the simulation model indicate from time to time sudden straight up-shift and right after a down-shift (or vice versa) to the previous gear ratio, that in real life would be better to avoid it. And if this simulation was to be developed further, specifically in a more functional direction, a 2-3 seconds delay signal before the gear shift could be easily implemented. In any case, this singularities happen rarely, happening in the present particular calculation only once in the more than 17 minutes that the WLTC Class 1 lasts.

ENERGY CONSUMPTION [kJ]

Dual Speed / WLTC 1 / No regenerative braking

1	NaN																			
2	NaN	NaN																		
3	NaN	NaN	1018,3																	
4	NaN	NaN	997,4	999,2																
5	NaN	NaN	989,7	989,0	992,9															
6	NaN	NaN	987,1	984,7	985,7	987,6														
7	NaN	NaN	984,8	983,8	983,1	983,9	985,6													
8	NaN	NaN	979,8	979,8	979,0	978,6	979,2	980,3												
9	NaN	NaN	977,9	977,9	977,7	977,0	976,8	977,2	978,0											
10	NaN	NaN	976,8	976,8	977,0	976,4	976,0	976,1	976,2	977,0										
11	NaN	NaN	976,6	976,6	976,7	976,5	976,1	975,3	975,4	976,1	976,9									
12	NaN	NaN	977,2	977,2	977,0	977,2	976,7	975,1	974,5	974,9	975,6	977,3								
13	NaN	NaN	977,8	977,8	977,4	977,1	977,5	975,1	974,4	974,5	975,3	976,7	977,9							
14	NaN	NaN	977,3	977,3	976,8	976,6	977,3	974,7	973,9	974,0	974,8	976,2	977,4	977,4						
15	NaN	NaN	975,7	975,5	975,5	975,1	975,5	973,6	972,6	972,7	973,3	974,5	975,5	975,5	975,8					
16	NaN	NaN	973,9	973,4	973,4	973,4	973,4	973,4	972,1	972,1	972,3	972,8	973,2	973,2	973,4	974,0				
17	NaN	NaN	971,5	971,1	970,9	970,9	970,9	970,9	970,6	970,0	970,1	970,4	970,8	970,7	970,8	971,4	971,6			
18	NaN	NaN	969,6	969,2	969,0	969,0	969,0	969,0	969,0	968,4	968,5	968,5	968,9	968,8	969,0	969,5	969,7	969,7		
19	NaN	NaN	968,9	968,6	968,1	968,1	968,1	968,0	968,1	967,7	967,8	967,9	968,0	967,9	967,8	968,4	968,6	968,6	NaN	
20	NaN	NaN	970,5	970,0	968,1	968,3	968,5	967,8	968,0	968,1	968,3	968,1	968,4	968,3	967,0	967,5	967,6	967,6	NaN	NaN
21	NaN	NaN	971,7	970,5	968,3	967,8	968,4	966,5	966,7	966,8	967,2	967,6	968,2	968,2	966,8	965,7	965,8	965,8	NaN	NaN
22	NaN	NaN	973,2	971,5	969,9	967,9	968,5	966,1	965,9	966,3	966,9	967,6	968,4	968,4	967,0	965,2	964,1	964,1	NaN	NaN
23	NaN	NaN	975,7	973,6	972,6	968,7	969,4	967,0	966,3	967,0	967,7	968,5	969,3	969,2	967,9	966,1	964,0	963,0	NaN	NaN
24	NaN	NaN	975,8	973,8	973,2	968,7	969,3	966,9	966,0	966,7	967,4	968,7	969,3	969,2	967,8	966,0	963,9	962,1	NaN	NaN
25	NaN	NaN	977,2	974,0	973,5	968,9	969,2	966,7	965,8	966,4	967,1	968,4	969,2	969,1	967,6	965,7	963,6	961,8	NaN	NaN
Gear Ratio [-]	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20

Figure 3.18.: Energy consumption, dual speed gearbox layout for WLTC C1, no regenerative braking considered

ENERGY CONSUMPTION [kJ]

Dual Speed / WLTC 1 / Regenerative braking

Gear Ratio [-]	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
1	NaN																			
2	NaN	NaN																		
3	NaN	NaN	893,7																	
4	NaN	NaN	873,1	871,9																
5	NaN	NaN	865,6	862,0	863,3															
6	NaN	NaN	862,8	857,7	856,2	856,4														
7	NaN	NaN	860,5	856,9	853,8	852,8	853,7													
8	NaN	NaN	855,4	852,8	849,7	847,7	847,4	847,6												
9	NaN	NaN	853,4	850,7	848,1	845,9	845,0	844,6	844,9											
10	NaN	NaN	852,3	849,7	847,4	845,3	844,3	843,8	843,4	843,9										
11	NaN	NaN	852,1	849,5	847,1	845,4	844,3	843,0	842,5	843,0	843,4									
12	NaN	NaN	852,6	850,0	847,4	846,0	844,9	842,6	841,5	841,8	842,2	843,6								
13	NaN	NaN	853,3	850,6	847,8	846,0	845,6	842,6	841,4	841,5	842,1	843,2	844,2							
14	NaN	NaN	852,8	850,2	847,4	845,5	845,5	842,3	841,0	841,0	841,6	842,7	843,8	843,6						
15	NaN	NaN	851,2	848,5	846,2	844,1	843,9	841,3	839,8	839,8	840,2	841,1	842,1	842,0	842,0					
16	NaN	NaN	849,3	846,4	844,0	842,4	841,7	840,9	839,2	839,1	839,1	839,4	839,8	839,6	839,6	840,1				
17	NaN	NaN	847,0	844,0	841,6	840,0	839,3	838,5	837,8	837,1	837,0	837,0	837,4	837,2	837,2	837,5	837,6			
18	NaN	NaN	845,1	842,1	839,7	838,0	837,3	836,6	836,1	835,4	835,3	835,1	835,4	835,3	835,3	835,7	835,7	835,4		
19	NaN	NaN	844,4	841,5	838,7	837,1	836,4	835,6	835,2	834,8	834,6	834,5	834,5	834,4	834,2	834,5	834,6	834,4	NaN	
20	NaN	NaN	846,0	843,0	838,7	837,3	836,8	835,5	835,1	835,1	835,1	834,7	835,0	834,9	833,4	833,7	833,7	833,5	NaN	NaN
21	NaN	NaN	847,1	843,4	838,9	836,8	836,7	834,1	833,8	833,8	833,9	834,2	834,8	834,7	833,3	832,0	831,9	831,7	NaN	NaN
22	NaN	NaN	848,7	844,4	840,5	837,0	836,9	833,8	833,1	833,4	833,7	834,3	835,1	835,0	833,5	831,5	830,3	830,0	NaN	NaN
23	NaN	NaN	851,1	846,5	843,1	837,8	837,8	834,6	833,4	834,1	834,5	835,1	835,9	835,7	834,3	832,4	830,2	829,1	NaN	NaN
24	NaN	NaN	851,2	846,7	843,7	837,6	837,6	834,4	833,0	833,7	834,2	835,2	835,7	835,6	834,1	832,2	830,0	828,0	NaN	NaN
25	NaN	NaN	852,6	846,9	844,0	837,9	837,4	834,2	832,9	833,5	833,9	834,9	835,7	835,6	834,0	832,0	829,8	827,8	NaN	NaN

Figure 3.19.: Energy consumption, dual speed gearbox layout for WLTC C1, regenerative braking considered

The results without regenerative braking of the figure 3.18 show the energy consumption for the cycle inside cells, being the two axis of the matrix the gear ratio combinations. Results clearly aligned with those of the single speed section 3.6: the most efficient pair of gear ratios is the one with the highest reduction but that still can manage to finish the driving cycle. In this case this is the combination of the GR_a=18:1 and GR_b=25:1, and the total energy consumption for the cycle is 961,8, which does not bring a relevant energy savings (compared to the 969,72 kJ of the single speed 18:1).

The results having regenerative braking, plotted in 3.19, show a minor advantage in the combination of gears with a medium spread at the long end of the range, i.e. one big and the other one larger. The less energy consumption for the cycle is located for GR_a=25:1 and GR_b=18:1 and lays at 827.8 kJ. That represents a potential over 15% of energy saved (potentially recovery back) of the energy consumption without regenerative braking.

3.9. Performance Requirements Validation

The performance requirements stated in 2.2 represent demanding challenges if fulfilled simultaneously as already demonstrated in the section 1.3.10. In order to comply with the requirements, using the theoretical maximum speed of 123 km/h and solving the gear ratio in the longitudinal dynamic equations 3.7, 3.8 and 3.10, the maximum GR must be 10,29. That means, at least one of the GR must be lower than 10,29 to be able to attain the top speed (over 120 km/h) in a flat road.

Likeways, using the same equations 3.7, 3.8, 3.10, and 3.5 and solving for a 30% slope, it is possible to obtain which is the minimum GR that is able to climb this slope for a low speed but in continuous operation (i.e. not surpassing the nominal power). The solution is 13,75. This means that at least one of the gear ratios of the transmission must be 13,75 or higher.

That narrows the gear ratios map to a much smaller matrix, see figure 3.20:

Gear Ratio [-]	1	2	3	4	5	6	7	8	9	10
13	NaN	NaN	853,3	850,6	847,8	846,0	845,6	842,6	841,4	841,5
14	NaN	NaN	852,8	850,2	847,4	845,5	845,5	842,3	841,0	841,0
15	NaN	NaN	851,2	848,5	846,2	844,1	843,9	841,3	839,8	839,8
16	NaN	NaN	849,3	846,4	844,0	842,4	841,7	840,9	839,2	839,1
17	NaN	NaN	847,0	844,0	841,6	840,0	839,3	838,5	837,8	837,1
18	NaN	NaN	845,1	842,1	839,7	838,0	837,3	836,6	836,1	835,4
19	NaN	NaN	844,4	841,5	838,7	837,1	836,4	835,6	835,2	834,8
20	NaN	NaN	846,0	843,0	838,7	837,3	836,8	835,5	835,1	835,1
21	NaN	NaN	847,1	843,4	838,9	836,8	836,7	834,1	833,8	833,8
22	NaN	NaN	848,7	844,4	840,5	837,0	836,9	833,8	833,1	833,4
23	NaN	NaN	851,1	846,5	843,1	837,8	837,8	834,6	833,4	834,1
24	NaN	NaN	851,2	846,7	843,7	837,6	837,6	834,4	833,0	833,7
25	NaN	NaN	852,6	846,9	844,0	837,9	837,4	834,2	832,9	833,5

Figure 3.20.: Narrowed range for energy consumption map, considered regenerative braking

The most efficient of the map is the one that full fills the requirements with the higher gear reduction, which makes the motor turn faster, thus using the most efficient operating range. In this case is the pair GR_a=10 and GR_a=25, which accounts for an energy consumption of 833,5 kJ while ensuring the top speed and the high gradability. To visualize it:

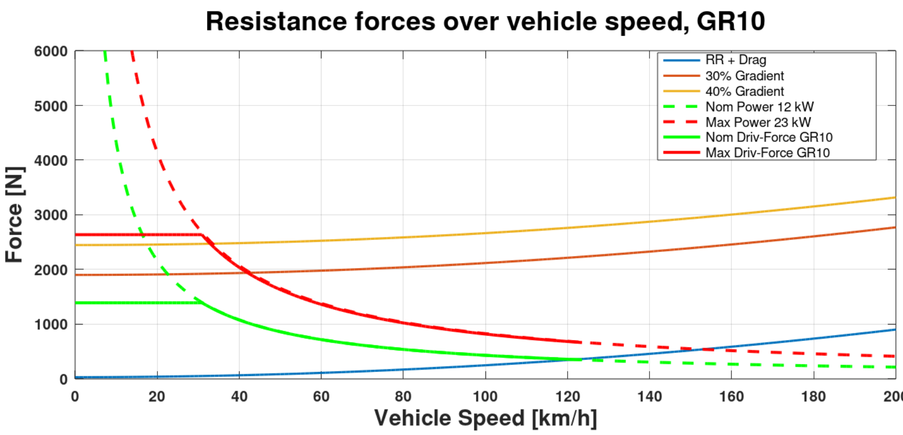


Figure 3.21.: Performance of GR = 10:1 in WTLTC C1

In the figure 3.21 the force curves of a total GR=10:1 are represented. The Ibex can reach the maximum speed (limited by the motor power curves) of 123 km/h for the nominal power.

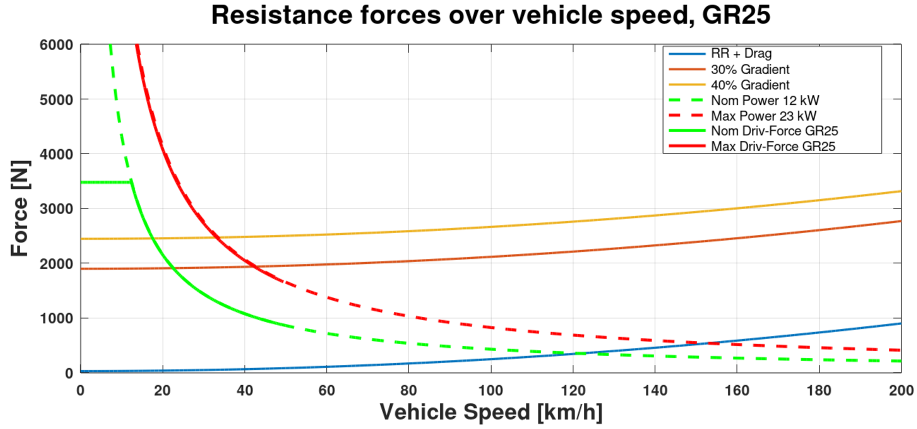


Figure 3.22.: Performance of GR = 25:1 in WTLTC C1

On the other hand, in the figure 3.22, the curves for the much bigger reduction GR=25:1 are shown. It illustrates that it can operate in slopes with a 30% inclination up to 23 km/h continuously and up to 43 km/h for a short periods of time.

- Time required for the 0-50 km/h = 13,4≤ 22
- Time required for the 0-100 km/h = 31,3 ≤ 22

Integrating the acceleration curves over the time (using the equation 3.6 and the code A.4), the time required to attain the desired speed is obtained by counting the accumulated increments of speed over time and adding those speeds or those time units.

It is fundamental to bear in mind the equation 3.10, because if the power can be regenerated, the higher the efficiency η, the higher the energy regenerated for the same amount of braking power. For that, at least one gear has to have a big reduction in the Ibex's transmission.

Plotting the stored array of gear ratios yields the optimal gearshift strategy:

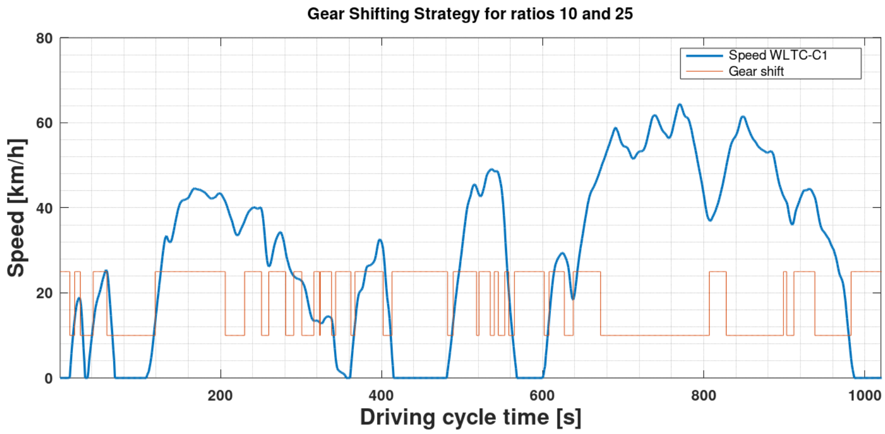


Figure 3.23.: Gearshift operation for the pair GR10 and GR25

The curve only moves between the two upper and lower bands of 10 and 25, always following the best combination of low positive instantaneous consumption and high negative instantaneous braking power (for energy regenerative purposes). It illustrates the sensitivity of the calculation. The lower gear is 20 times selected in more than 20 minutes of urban start-stop cycle, which for some transmissions could be effortless handled as for others like the MTs, the algorithm could be easily optimized to allow for some delay to avoid power interruptions.

4. Conclusions

The development of this research conducts the modelling of a micro EV, specifically the TERA Ibex, its motor, its powertrain behavior and connects it all with a driving scenario, in order to gather quantitative data that can help taking an informed decision in order to optimize the efficiency of its powertrain by designing an optimal gearbox.

The results reveal the tremendous advantage of regenerative braking for the urban cycle WLTC Class 1, achieving between a 15% and 18% of potential net energy savings.

The results also stand out the benefits of a dual-speed gearbox specially for the vehicles with a very low (< 25) power-to-weight ratio.

The ideal transmission for the Ibex is a quick and responsive dual-speed gearbox, with a wide gear spread between the two gears that ensures the maximum top speed, the highest climbing abilities, a correct acceleration and simply the most optimized energy efficiency. There is an ideal layout that fits that description, combining a high efficiency, and a quick and seamless gear shifts; the dual clutch transmission. For that reason, is the layout proposed to be further investigated. The second alternative would be an automated manual transmission.

However, in order to take a definite decision, the single situation has to be further evaluated from more perspectives. The choice of whether to install a premium efficiency transmission strongly depends on both operating conditions, as well as the life cycle costs associated with the investment [50].

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Glossary

e-mobility Electro mobility (or e-Mobility) represents the concept of using electric powertrain technologies, in-vehicle information, and communication technologies and connected infrastructures to enable the electric propulsion of vehicles and fleets.. v

L7e-A2 A heavy on road vehicle with four wheels, maximum 2 non-straddle seats and whose maximum continuous rated power does not exceed 15 kW (Microcars). 4

Magna Steyr AG Magna Steyr AG & Co KG is an automobile manufacturer based in Graz, Austria, where its primary manufacturing plant is also located. It is a subsidiary of Canadian-based Magna International and was previously part of the Steyr-Daimler-Puch conglomerate. 2

Octave Octave is a high-level programming language mainly used for computing numerical. It was developed by John W. Eaton. It was initially released in the year 1980. It was written in C, C++, and Fortran.. 29

SAE SAE International, previously known as the Society of Automotive Engineers, is a U.S.-based, globally active professional association and standards developing organization for engineering professionals in various industries. Principal emphasis is placed on global transport industries such as aerospace, automotive, and commercial vehicles. Accordingly, the name SAE International was established to reflect the broader emphasis on mobility.. 8

Acronyms

AMT Automated Manual Transmissions. 23
AT Automatic Transmissions. 22, 27
ATZ Motortechnische Zeitschrift Magazin. 20

CSV Comma-Separated Values. 44
CVT Continuous Variable Transmission. 27

DCT Dual Clutch Transmissions. xi, 20, 25

EC European Commission. 3
EU European Union. 4
EV Electric Vehicle. v, 5, 7, 18, 25, 50, 63

GB Gearbox. 31
GEM Global Electric Motorcars. 8
GR Gear Ratio or Gear Reduction. xii, 42, 54, 59, 60

ICE Internal Combustion Engine. 15, 18

MT Manual Transmissions. 21, 23, 61

NaN Not a Number. 53
NEDC New European Driving Cycle. 39

OBD On-board Diagnostics. 4

PMSM Permanent Magnet Synchronous Motor. xii, 11, 12, 30, 42, 43, 49
PWr Power to Weight ratio. 39

RPM Revolutions Per Minute. 18, 33

TERA Team Eco Racing Austria. 1, 39, 41, 42, 63

TU Technical University. 13

UNECE United Nations Economic Commission for Europe. 3, 39
US United States of America. 3, 8

WLTC Worldwide Harmonised Light-duty vehicles Test Cycles. 39–41
WLTP Worldwide Harmonised Light Vehicles Test Procedure. 39

List of Symbols

P: Power [W]
PW_r: Power-to-weight ratio [-]
$\mathcal{T}$: Torque [Nm]
n: Rotational speed [rpm, rad/s]
T_1: Motor torque [Nm]
T_2: Driveshaft torque [Nm]
n_1: Motor speed [rpm, rad/s]
n_2: Driveshaft speed [rpm, rad/s]
i_g: Total gear ratio [-]
R_{roll}: Rolling resistance [N]
g: Gravity [m/s^2]
c_{roll}: R_{roll} coefficient [-]
β_c: Angle [°, rad]
m_{veh}: Vehicle mass [kg]
v: Vehicle speed [km/h, m/s]
R_{air}: Aerodynamic resistance [N]
A_{veh}: Cross-sectional area [m^2]
c_{air}: Aerodynamic coefficient
ρ_{air}: Air density [g/cm^3]
I_{Red} Moment inertia at driving axle [kgm^2]
m_*: Generalized vehicle mass [kg]
I_{Axis}, I_{Trans} : Axis, transm. moment of inertia [kgm^2]
r_w^2: Rolling resistance torque [kg]
i_{Axis}, i_{Trans}: Axis, transm. gear ratio [-]
R_{grad}: Climbing resistance [N]
V: Volage [V]
A: Current [A]
F_{accel}: Acceleration force [N]
F_D: Driving force [N]

P_D: Driving Power [W]
E_{dc}: Energy consumption [J, kJ]
$P_{Mi(+)}$: Positive-only values of power motor [J, kJ]
E_{dc}: Energy consumed during the cycle [J, kJ]
F_{Br}: Braking force [N]
F_{Reg}: Regenerative force [N]
η_{Reg}: Total regeneration efficiency coeff [%]
$Reg_{\%}$: Regenerated energy as % of the braking force [-]
η_{bat}: Losses due to the battery [-]
η_{inv}: Losses due to the inverter [-]
η_{tire}: Losses due to the tire slippage [-]
η_{tire}: Mechanical losses in the gears [-]

Appendix A.

Script Code

A.1. Longitudinal dynamics

```
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%% Vehicle Dynamics TERA Ibex     %%
%% Juan Campos Alonso             %%
%% 07/09/2018                     %%
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

% General Data
clear
radiustire  = 0.273;     % [m]
grlweight  = 650;        % [kg]
maxspeed    = 130;       % kph
maxslope    = 0.3;       % 30% [-]

% Motor Data
motor_power_n       = 12;    % kW
motor_power_peak    = 23;    % kW
motor_torque_peak   = 72;    % Nm
motor_speed_n       = 3000;  % rpm
motor_speed_peak    = 12000; % rpm

% Approximated Parameters
drag_c          = 0.235;  % [-]
airflow_surf    = 1.92;  % m^2 (calculated as height*width, in reality it is smaller)
air_density     = 1.25;  % kg/m^3
```

```
gravity          = 9.81;  % m/s^2
rolling_res_c    = 0.005; % [-]

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% FORCE CALCULATION for Steady Max Speed and no slope %%

slopeA  = 0;
speed  = maxspeed;
weight = grlweight*1.02; % mass increase due to rotating ...
    inertia
accel  = 0;

force_accel      = weight*accel;
force_rr         = weight*rolling_res_c*gravity;
force_drag       = ...
    0.5*air_density*drag_c*airflow_surf*(speed*1000/3600)**2;
force_climbing   = gravity*weight*sin(deg2rad(slopeA));
force_tot        = force_accel + force_rr + force_drag + ...
    force_climbing;

power_tot        = (force_tot)*speed*1000/3600;


x = linspace(0, 200, 201);

y1 = [force_rr]+0*x; % Traction Force due to RR
y2 = 0.5*air_density*drag_c*airflow_surf*(x/3.6).^2; % ...
    Traction force due to Drag
y3 = weight*0.3;
y4 = weight*0.76;
y5 = weight*1;
y6 = weight*3;
yc1 = gravity*weight*sin(atan(0.1)); % 10% gradient ...
    climbing force
yc2 = gravity*weight*sin(atan(0.20)); % 20% gradient ...
    climbing force
yc3 = gravity*weight*sin(atan(0.3)); % 30% gradient ...
    climbing force
P1 = 12000./(x/3.6); % Force over speed for a P = const = ...
    12000 W
P2 = 23000./(x/3.6); % Force over speed for a P = const = ...
    23000 W

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
```

```
% PLOT "LIBRARY" FOR ALL FORCE AND POWER CURVES %%%
figure 1
clf
hold on
%plot (x,y1,"linewidth",2) % RR
%plot (x,y1+y2,"linewidth",1.5) % RR+Drag
%%plot (x,y1+y2+y3,"linewidth",2) % Acceleration 0,3 m/s^2
%%plot (x,y1+y2+y4,"linewidth",2) % Acceleration 0,76 m/s^2
%plot (x,y1+y2+y5,"linewidth",1.5) % Acceleration 1 m/s^2
%plot (x,y1+y2+yc1,"linewidth",2) % Gradient climbing force 10%
%plot (x,y1+y2+yc2,"linewidth",1.5) % Gradient climbing ...
    force 15%
%plot (x,y1+y2+yc3,"linewidth",1.5) % Gradient climbing ...
    force 30%
%%plot (x,y1+y2+y6,"linewidth",2) % Acceleration 3 m/s^2
plot (x,P1,'--',"linewidth",2,"color", "g") % 12kW
plot (x,P2,'--',"linewidth",2,"color", "r") % 23kW
%plot(122,354.91,'o','MarkerSize',12,'MarkerEdgeColor','k') ...
    % Circle marking Max Speed for 12 kW
grid %minor
axis([0 160 0 3000])
set(gca,'XTick',0:20:200)
ylabel('Force [N]', "fontweight", "bold", "fontsize",15)
xlabel('Vehicle Speed [km/h]', "fontweight",  "bold", ...
    "fontsize",15)
title(sprintf('Resistance forces over vehicle speed'), ...
    "fontsize", 18)
legend('Rolling Resistance', 'RR + Drag','12 kW', '23 kW');
%daspect ([2 1 1])
box on
hold off
pbaspect([2.5 1 1])
h=get (gcf, "currentaxes");
set(h,"fontweight","bold")
set(gcf, 'PaperPosition', [0 0 10 4]); %Position plot at ...
    left hand  corner with width 5 and height 5.
nameplot = 'RrollR_DragR2';
%saveas(gcf, nameplot, 'png') %Save figure
```

A.2. Driving Cycle Processing

```

%%%%%%%%% WLTC CLASS 1 DIAGRAM %%%%%%%
filename = 'WLTC_class_1.txt';
delimiterIn = ' ';
headerlinesIn = 1;
Cycle = importdata(filename,delimiterIn,headerlinesIn);

figure 1 % Cycle Diagram
plot(Cycle.data(:,3))
axis([0 1500 0 100])
grid minor
ylabel('Vehicle speed [km/h]', "fontweight", "bold", ...
    "fontsize",15)
xlabel('Time [s]', "fontsize",15)
title(sprintf('WLTC Cycle / Class 1'), "fontsize", 18)
%legend('Rolling Resistance', 'RR + Drag', '1 m/s^2', '3 ...
    m/s^2', '12 kW', '30 kW');
hold off
h=get (gcf, "currentaxes");
set(h,"fontweight","bold")
print -dpdf -color WLTC-Class1.pdf
print -dpng -color WLTC-Class1.png
%saveas(gcf,'WLTC-Class1-saveas.png')


%%%%%%% DRIVING FORCE REQUIRED FOR THE CYCLE %%%%%%%

DrivForce = Speed = [];

for i=1:1023%size (Cycle.data(:,3))(1)
    DrivForce(i) =  weight*Cycle.data(i,4) + ...
        weight*rolling_res_c*gravity; + ...
        0.5*air_density*drag_c*airflow_surf*(Cycle.data(i,3)*1000/3600)**2;
    Speed(i)=Cycle.data(i,3);
end



figure 2 % Required Driving Force Diagram
plot (Speed,DrivForce)
```

```
%%plot (Speed,DrivForce, 'o')
axis([0 70 -1000 1000])
grid minor
ylabel('Force [N]', "fontweight", "bold", "fontsize",15)
xlabel('Vehicle Speed [km/s]', "fontsize",15)
title(sprintf('Driving Force Required over WLTC / Class ...
    1'), "fontsize", 18)
%legend('Rolling Resistance', 'RR + Drag', '1 m/s^2', '3 ...
    m/s^2', '12 kW', '30 kW');
hold off
h=get (gcf, "currentaxes");
set(h,"fontweight","bold")
%saveas(gcf,'Driving_Force_Required_WLTC-C1.png')
%%print -dpdf -color Driving_Force_Required_WLTC-C1.pdf
%%print -dpng -color Driving_Force_Required_WLTC-C1.png

figure 3 % Required Driving Force Diagram EXTRACTING THE ...
    LOW AND MIDDLE SPEED
plot (Speed(2:589),DrivForce(2:589))
hold on
plot (Speed(590:1024),DrivForce(590:1024))
%%plot (Speed,DrivForce, 'o')
axis([0 70 -1000 1000])
grid minor
ylabel('Force [N]', "fontweight", "bold", "fontsize",15)
xlabel('Vehicle Speed [km/s]', "fontsize",15)
title(sprintf('Driving Force Required over WLTC / Class ...
    1'), "fontsize", 18)
legend('Low Speed', 'Middle Speed', '1 m/s^2', '3 m/s^2', ...
    '12 kW', '30 kW');
hold off
h=get (gcf, "currentaxes");
set(h,"fontweight","bold")
%saveas(gcf,'Driving_Force_Required_WLTC-C1.png')
%%print -dpdf -color ...
    Driving_Force_Required_WLTC-C1-Low-Middle.pdf
%%print -dpng -color ...
    Driving_Force_Required_WLTC-C1-Low-Middle.png

figure 4 % Required Driving Force of Cycle with its ...
    accelerations combined with middle and low speed
%plot (x,y1,"linewidth",2)
%plot (x,y1+y2,"linewidth",2)
%%plot (x,y1+y2+y4,"linewidth",2)
```

```
%plot (x,y1+y2+y5,"linewidth",2)
%plot (x,y1+y2+y6,"linewidth",2)
% Start of the mid-low speed cycle thing
plot (Speed(2:589),DrivForce(2:589))
hold on
plot (Speed(590:1023),DrivForce(590:1023))
plot (x,P1,'--',"linewidth",2,"color", "g")
plot (x,P2,'--',"linewidth",2,"color", "r")
grid minor
axis([0 140 0 2000])
ylabel('Force [N]', "fontweight", "bold", "fontsize",15)
xlabel('Vehicle Speed [km/s]', "fontsize",15)
title(sprintf('Traction force over vehicle speed'), ...
    "fontsize", 18)
box on
hold off
pbaspect([2 1 1])
h=get (gcf, "currentaxes");
set(h,"fontweight","bold")
set(gcf, 'PaperPosition', [0 0 10 4]); %Position plotat ...
    left hand  corner with width 5 and height 5.
legend('WLTC C1 Low Speed', 'WLTC C1 Middle Speed', ...
    'Nominal Power 12 kW', 'Maximum Power 23 kW');
saveas(gcf,'WLTCForce_Speed_POWERS_.png')
%%print -dpdf -color acc0c76_2.pdf
%%print -dpng -color REQUIRED_MIX_CYCLE_Mid-Low.png
```

A.3. Energy Consumption Analysis

A.3.1. Single Speed Layout

```
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%% Energy Consumption TERA Ibex   %%
%% Single Speed Transmission      %%
%% Juan Campos Alonso             %%
%% 13/07/2020                     %%
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

%%%%%%% TERA Ibex Data Import %%%%%%%
load TERA_Ibex.txt
%%%%%%% Efficiency Map Import %%%%%%%
filename = 'Tera_Ibex_Motor_Map.csv';
delimiterIn = ',';
headerlinesIn = 1;
Effs = importdata(filename,delimiterIn,headerlinesIn);
%%%%%% WLTC CLASS 1/2/3 Import %%%%%%
filename = 'WLTC_class_1.txt';
delimiterIn = ' ';
headerlinesIn = 1;
Cycle = importdata(filename,delimiterIn,headerlinesIn);

%%%%%%% DRIVING FORCE REQUIRED FOR THE CYCLE %%%%%%%
GearEff = 0.98; % [-] Gearbox Efficiency
DrivForce = Speed = DrivForcePos = [];

Theta = []; % Instantaneous motor torque, angular speed and ...
    efficiency
GR_Energy = EnergyC = Powermot = Powermot_pos = ...
    Powermot_neg = Torquemot = Torquewhe = Wmot = Wwhe = [];
linestxt = size(Cycle.data(:,3))(1);

g = 1
for gg= 1:30 %%0.5:0.1:39.5 Incremental steps in the gear ...
    ratio variation
    GearR = gg;
    DragR = RollR = AccR = MTorque = MSpeed = [];
    Powerdriv = PowerdrivPos = T = W = [];
    for i=1:linestxt
```

```
        Speed(i)  = Cycle.data(i,3);
        AccR(i)   = weight*Cycle.data(i,4);
        RollR(i)  = weight*rolling_res_c*gravity*(Speed(i)>0);
        DragR(i)  = ...
            0.5*air_density*drag_c*airflow_surf*(Cycle.data(i,3)/3.6)**2;
        DrivForce(i) = AccR(i) + RollR(i) + DragR(i);
        Powerdriv(i) = DrivForce(i)*Speed(i)/3.6;
        T(i)  = DrivForce(i)*radiustire/GearR;
        Torquemot(g,i)  =  DrivForce(i)*radiustire/GearR;
        Torquewhe(g,i)  =  DrivForce(i)*radiustire;
        W(i)  = Speed(i)/3.6*GearR/radiustire;
        Wmot(g,i)  =  Speed(i)/3.6/radiustire*GearR;
        Wwhe(g,i)  =  Speed(i)/3.6/radiustire;
        MTorque(i)  = floor(abs(T(i))/6.01 + 1); % This is ...
            not a torque, just an index for Theta
        MSpeed(i)  = floor(W(i)/155.01 + 1); % This is not a ...
            speed, just an index for Theta
        if MTorque(i)  == 0
          Powermot(g,i)  = 0;
          Powermot_pos(g,i)  = 0;
          Powermot_neg(g,i)  = 0;
        elseif abs(MTorque(i)) < 13 && MSpeed(i) < 9
          MTindex = abs(MTorque(i));
          Theta(g,i) = Effs.data(MTindex,MSpeed(i));
            if Theta(g,i) == 0 % Zero efficiency means out ...
                of operating range
              Theta(g,i) = NaN;
              Powermot(g,i)      = NaN;
              Powermot_pos(g,i)  = NaN;
              Powermot_neg(g,i)  = NaN;
            else
              Powermot(g,i)  = Powerdriv(i)/Theta(g,i)/GearEff;
              if Powermot(g,i) > 0 % storing only-positive ...
                  power consumption
                Powermot_pos(g,i)  = Powermot(g,i);
                Powermot_neg(g,i)  = 0;
              else % storing negative power consumption for ...
                  regenerative braking
                Powermot_pos(g,i)  = 0;
                Powermot_neg(g,i)  = Powermot(g,i);
              end
            end
        else
          Powermot(g,i)      = NaN;
```

```
            Powermot_pos(g,i) = NaN;
            Powermot_neg(g,i) = NaN;
        end
    end
    disp ('Gear ratio:'), disp(GearR), disp(gg);
    disp ('Energy consumption over cycle [kWh]:');
    disp (sum(Powermot(g,:))/1000/3600); % [kWh] Display ...
        power driving demand
    disp ('Energy consumption over cycle [kJ]:');
    disp (sum(Powermot(g,:))/1000); % [kJ] Display power ...
        driving demand
    disp(' ')
    GR_Energy(g) = sum(Powermot(g,:))/1000; % Vector ...
        storing the different total-energy-consumption for ...
        differen gear reductions.
    GR_Energy_pos(g) = sum(Powermot_pos(g,:))/1000; % Total ...
        positive power consumption
    GR_Energy_neg(g) = sum(Powermot_neg(g,:)/1000); % Total ...
        negative power consumption
    GR_Energy_neg_abs(g) = ...
        abs(sum(Powermot_neg(g,:))/1000); % Absolute value ...
        of total negative power consumption
    GR_Energy_reg(g) = ...
        (sum(Powermot_pos(g,:))+0.7*sum(Powermot_neg(g,:)))/1000; ...
        % Net Power consumption with 70% of regenerative braking
    g+=1
end
```

A.3.2. Dual Speed Layout and Gearshift Criteria

```
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%% Gear Shifting Strategy Script  %%
%% Juan Campos Alonso             %%
%% 20/09/2020                     %%
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

%%%%%%% Instantaneous Power consumption Import %%%%%%%
clear;
load Powermot_Ibex_data_C1.txt
Powermot2 = [];
GR_Energy2 = GR_Energy_pos2 = GR_Energy_neg2 = ...
    GR_Energy_neg_abs2 = GR_Energy_reg2 = [];
g=1;

for Ga=2% 3:18 % Each gear ratio, in combination of...
  for Gb=6%1:25 % ... each gear ratio
    for i=1:size(Powermot)(2) % each second of the driving ...
        cycle
      if isnan(Powermot(Ga,i))
        if isnan(Powermot(Gb,i))
          g(Ga,Gb,i) = NaN;
          disp ('Unachievable driving cycle for the ...
              selected gear ratios'); % Error message
          disp ('Ga:'), disp(Ga), disp ('Gb:'), disp(Gb);
        else
          g(Ga,Gb,i) = Gb;
        end
      else
        if isnan(Powermot(Gb,i))
          g(Ga,Gb,i) = Ga;
        elseif Powermot(Ga,i) < Powermot(Gb,i)
          g(Ga,Gb,i) = Ga;
        elseif Powermot(Ga,i) > Powermot(Gb,i)
          g(Ga,Gb,i) = Gb;
        else
          if i == 1
          g(Ga,Gb,i) = min(Ga,Gb);
          else
          g(Ga,Gb,i) = g(Ga,Gb,i-1);
          end
```

```
            end
          end
        Powermot2(Ga,Gb,i) = Powermot(g(Ga,Gb,i),i);
        GR_Energy2(Ga,Gb) = sum(Powermot2(Ga,Gb,:))/1000; % ... Array stores the total—energy—consumption for each ... gear reduction.
        GR_Energy_pos2(Ga,Gb) = ... sum(Powermot2(Ga,Gb,:)(Powermot2(Ga,Gb,:)>0))/1000; ... % Total positive power consumption
        GR_Energy_neg2(Ga,Gb) = ... sum(Powermot2(Ga,Gb,:)(Powermot2(Ga,Gb,:)<0))/1000; ... % Total negative power consumption
        %GR_Energy_neg_abs2(Ga,Gb) = ... abs(sum(Powermot_neg(g,:))/1000); % Absolute value ... of total negative power consumption
        GR_Energy_reg2(Ga,Gb) = ... GR_Energy_pos2(Ga,Gb)+0.7*GR_Energy_neg2(Ga,Gb); % ... Net Power consumption with 70% of regenerative braking
      end
    end
end

for j=1:size(Powermot)(2);
  gear(j) = g(Ga,Gb,j);
end
figure(33)
clf
plot(Speed, "linewidth",1.5)%,'—o')
hold on
stairs(gear)
%plot(Powermot(Ga,:)/150,"color",[.2 .9 .6])%,'—o') % ... smaller gear ratio, soft green
%plot(Powermot(Gb,:)/150,"color",[1 .4 1])%,'—o') % higher ... gear reduction, high accel, intense red or purple
%axis([1 size(Powermot)(2) 0 105])
axis([1 1020 0 80])
ylabel('Speed [km/h]', "fontweight", "bold", "fontsize",15)
xlabel('Driving cycle time [s]', "fontweight", "bold", ... "fontsize",15)
title('Gear Shifting Strategy for ratios 2 and 6')
legend("Speed WLTC—C1",'Gear shift');
grid minor
box on
%pbaspect([2.5 1 1])
```

```
h=get (gcf, "currentaxes");
set(h,"fontweight","bold")
set(gcf, 'PaperPosition', [0 0 10 4]); %Position plotat ...
    left hand  corner with width 5 and height 5.
nameplot = 'Gearshift_Strategy_GR2-GR6_';
saveas(gcf, nameplot, 'png') %Save figure

disp('Done!')
```

A.4. Power interpolation of Ibex motor

```
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%% Motor power interpolation Ibex %%
%% Juan Campos Alonso             %%
%% 11/08/2020                     %%
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

figure 1 % USING GEAR REDUCTION

reduction   = 19 ;        % Gear Reduction []
radiustire  = 0.273;      % [m]

% Power INTERPOLATION
% Force from Torque Max
xxr =  0:184.6119609:22155;
phase1 = (xxr≥0)&(xxr≤3000);
phase2 = xxr>3000;
yf(phase1) = 72;
yf(phase2) = 204032*(xxr(phase2).^(-0.993));
plot(xxr*pi/30*radiustire/reduction*3.6,yf*reduction/radiustire, ...
    "linewidth",2,"color", "r")

hold on
% Force from Torque Nom
xxr =  0:184.6119609:22155;
phase1 = (xxr≥0)&(xxr≤3000);
phase2 = xxr>3000;
yf(phase1) = 38;
yf(phase2) = 112392*(xxr(phase2).^(-0.998));
plot(xxr*pi/30*radiustire/reduction*3.6,yf*reduction/radiustire, ...
    "linewidth",2,"color", "k")
%%axis ([0 12000 0 75])
legend ({"Force Max", "Force Nom"})
title(sprintf('Ibex Motor (tot. gear reduction of 7.5:1)'), ...
    "fontsize", 18)
ylabel('Force [N]', "fontweight", "bold", "fontsize",15)
xlabel('Vehicle Speed [km/h]', "fontweight", "bold", ...
    "fontsize",15)
saveas(gcf,'Tera_Ibex_Motor_Map_N-kph_gr15.png')
```