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**Test cell layout optimization  
kinematic and dynamic simulation**

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# Ottimizzazione del layout della cella prova, simulazione cinematica e dinamica

## Sommario

Il presente elaborato ha lo scopo di proporre un metodo automatico di selezione del sistema motore-motoriduttore al fine di massimizzarne le performance dinamiche. Questo, con particolare focus sull'ottimizzazione del layout di una cella di prova per macchine agricole heavy-duty. É stato inoltre creato un modello parametrico del sistema completo della cella con l'obbiettivo di ridurre tempi e costi di integrazione dei componenti, oltre che di creare un ambiente di simulazione privilegiato per futuri test MIL.

Sono stati utilizzati differenti tools: Matlab per l'automazione e generalizzazione di criteri per la selezione dei sistemi motore-motoriduttore presenti in letteratura, mentre per la modellazione diretta di componenti fisici e per la creazione di una rappresentazione virtuale di una cella prova, sono stati utilizzati Simulink/Simscape.

Attraverso il metodo automatico sviluppato con Matlab é stato possibile ridurre tempi e costi di progettazione del sistema motore-motoriduttore, confrontando simultaneamente diverse soluzioni in commercio.

I modelli realizzati con Simulink/Simscape sono stati validati tramite dati raccolti sperimentalmente, mostrando una correlazione robusta ed un errore NMRSE di 0.0057 e 0.3098 rispettivamente nei casi di controllo di velocità e di coppia.

L'ambiente SW costituito dal tool di ottimizzazione e dal modello di sala consentirà l'analisi anticipata di eventuali problematiche di integrazione delle trasmissioni selezionate.

# Test cell layout optimization kinematic and dynamic simulation

## Abstract

The purpose of this elaborate is to propose an automatic method of selecting of the motor-reducer system in order to maximize its dynamic performance. This, with special focus on 'optimizing the layout of a test cell for heavy-duty agricultural machinery. A parametric model of the complete cell system was also created with the goal of reducing time and cost of component integration, as well as creating a preferred simulation environment for future MIL testing.

Different tools were used: Matlab for automation and generalization of criteria for the selection of motor-gearbox systems found in the literature, while Simulink/Simscape were used for direct modeling of physical components and for creating a virtual representation of a test cell.

Through the automatic method developed with Matlab was able to reduce the design time and cost of the motor-reducer system by simultaneously comparing several commercially available solutions. The models made with Simulink/Simscape were validated through experimentally collected data, showing robust correlation and NMRSE error of 0.0057 and 0.3098 in the speed and torque control cases, respectively. The SW environment consisting of the optimization tool and the room model will allow early analysis of any integration issues of the transmissions selected.

# Contents

<b>Sommario</b>	<b>I</b>
<b>Abstract</b>	<b>II</b>
<b>Contents</b>	<b>III</b>
<b>1 Introduction</b>	<b>2</b>
<b>2 State of the art</b>	<b>4</b>
2.1 Motor . . . . .	5
2.2 Motor control . . . . .	7
2.2.1 Field Oriented Control (FOC) . . . . .	8
2.2.2 Direct Self Control and Direct Torque Control . . . . .	9
2.2.3 V/F Control . . . . .	9
2.3 Gearbox . . . . .	10
2.3.1 Gear . . . . .	11
2.3.2 Hydraulic transmission . . . . .	12
2.4 Load . . . . .	13
2.5 Braking system . . . . .	13
2.5.1 Hydraulic Braking System . . . . .	15
2.5.2 Pneumatic Braking System . . . . .	16
2.6 Axle Load Absorbers Brakes . . . . .	17
2.7 Test cell and testing in agriculture industry . . . . .	17
2.7.1 Driveline test . . . . .	18
2.7.2 DLG Cycle Test . . . . .	19
2.8 Controllers . . . . .	20
2.8.1 PI controller . . . . .	20
2.9 Modeling and simulation of the behavior of a system . . . . .	21
2.9.1 Matlab . . . . .	21
2.9.2 Simscape . . . . .	22

<b>3</b>	<b>Sizing criteria</b>	<b>23</b>
3.1	Sizing criteria for a dynamic load . . . . .	23
3.2	Alpha - Beta criteria . . . . .	25
3.2.1	Customization of alpha-beta criteria . . . . .	28
3.3	Matlab sizing tool . . . . .	29
3.3.1	High Dynamic Load . . . . .	33
<b>4</b>	<b>Model realization</b>	<b>36</b>
4.1	Simscape setup . . . . .	37
4.2	Performance prediction with speed set point . . . . .	38
4.2.1	Electric drive modeling . . . . .	38
4.2.2	Control modeling . . . . .	40
4.2.3	Mechanical System modeling . . . . .	41
4.2.4	Induction Motor modeling . . . . .	42
4.3	Simscape model validation . . . . .	43
4.3.1	Error evaluation, Mean Squared Error . . . . .	45
<b>5</b>	<b>Performance prediction with speed set point and Torque Control</b>	<b>47</b>
5.1	Simscape model . . . . .	48
5.1.1	Electric drive . . . . .	48
5.1.2	Control . . . . .	49
5.1.3	Mechanical System . . . . .	51
5.1.4	Induction Motor . . . . .	53
5.2	Validation and output data of Simscape model with step torque brake .	54
5.2.1	Error evaluation, Mean Squared Error . . . . .	56
5.3	Analysis and considerations on pneumatic brake . . . . .	57
<b>6</b>	<b>Sizing tool and simscape model for motor and gear size for dynamic load</b>	<b>59</b>
6.1	Simscape Model without spring and damper . . . . .	60
6.2	Simscape Model with spring and damper . . . . .	61
	<b>Conclusions</b>	<b>65</b>
	<b>Bibliografia</b>	<b>66</b>

# Chapter 1

## Introduction

The product validation process demonstrates if the end-product satisfies its stakeholder expectations. Thus, after the verification phase which proves that a "product was done right", validation proves whether "the right product was done". This fundamental phase of the standard well known "V-model" includes different processes, methods and tools well discussed in Nasa's site[1] and implies the construction of a dedicated test environment.

Facing the task of building a controlled, repeatable and representative testing environment implies a proper design and integration of different lab resources composing this environment (tooling, acquisition systems, test benches, environmental control, ...) which can also be named as a test cell.

Considering the high variety of testing environments, i.e. test cells, which are tailored and designed to provide a proper product validation for each specific device under test one may catch a glimpse on the main challenges of this master thesis: provide an algorithm to ease and speed up the design phase of a test environment for agricultural-oriented vehicles.

Taking as a main case of study the testing facilities located in the Modena San Matteo CNHi R&D plant, this thesis will address the creation of an automated process to select, combine and simulate components of a 2WD actuation system test cell for tractors prior to its construction. This algorithm will guarantee a significant reduction of time for components selection and also an innovative environment to test a priori new control architectures and benches layout, ensuring a continuous physical to virtual transition for the next years.

Due to the complexity of the problem and the system specificity very low contributions are present in literature. So that, a huge task of this master thesis activity has been related to literature research, methodologies generalization and synthesis in order to find the best way to approach the problem described above.

In this thesis is proposed a practical approach for test cells layout optimization, kinematic and dynamic simulation.

- In chapter 2, the state of the art is analyzed and the theory necessary to understand each element treated, observing both the mechanical components and the electrical components of the case study;
- In chapter 3, a focus on the selection and sizing criteria is presented, based on the state of the art, laying the groundwork for the novel algorithm discussed in chapter 6.
- In chapter 4, an ideal Simscape model of a test cell is presented, considering some system simplification as absence of torque control and mechanical system compliance.
- In chapter 5, a real Simscape model of a test cell is presented, improving system dynamic model presented in chapter 4, taking into account the compliance, speed and torque control.
- In chapter 6, a Matlab-based sizing tool for motor and gearbox systems with defined motion law is selected and presented, according to the state of the art discussed in chapter 2 and the synthesis carried;

# Chapter 2

## State of the art

A mechanical system using a single-motor electric drive as a motion generation system can be schematized with functional blocks as shown in the following figure:

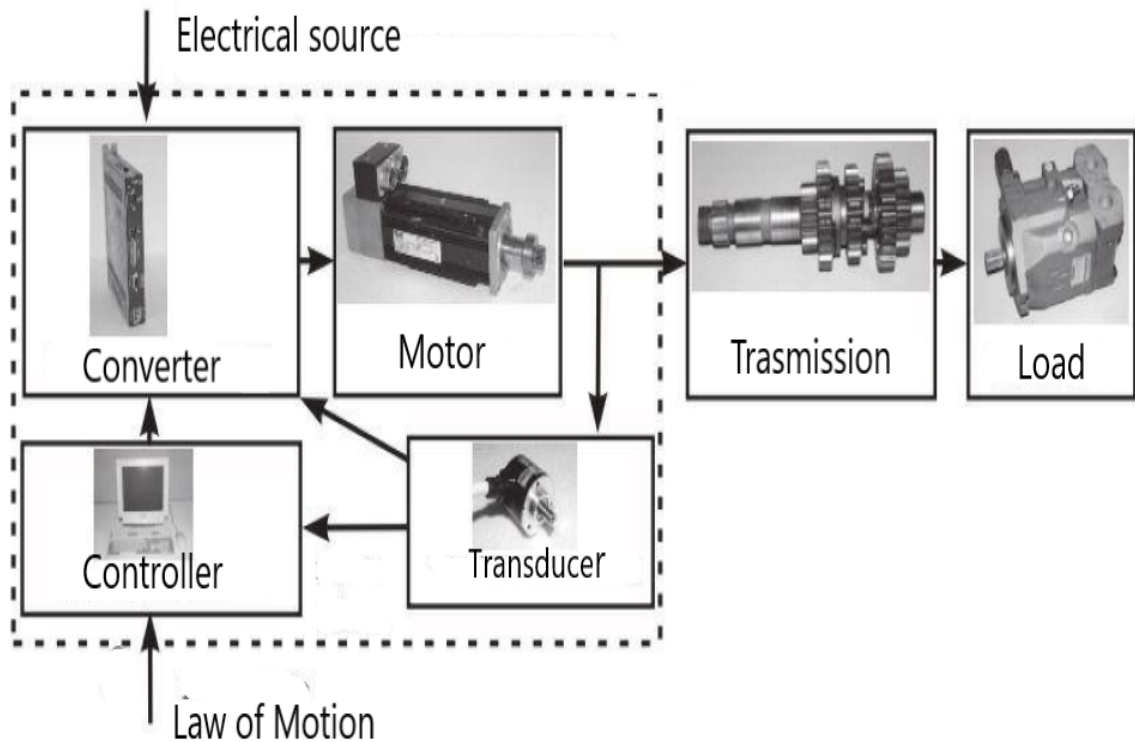


Figure 2.1: Scheme of a system using an electric drive for motion generation and control [2]

The load represents the machine or the device you want to move, which can be connected to the motor directly or through some mechanical transmission device: this often includes a gearbox that reduces the speed and increases the transmitted torque. The electronic converter, which is not always present, is used to adequately power the engine; in this case study it is taken into account but is not deepened.

The transducers measure the actual movement of the motor and providing data to the



controller and converter which compare these quantities with the relative desired values and correct the engine power parameters to ensure that the actual motion respects the as much as possible the desired motion. CEI 301-1 defines electric drive "a system that converts electrical energy into mechanical, with the use of electronic power equipment, in accordance with a control function". The term "electric drive" therefore includes the set of motor, converter, sensors and power electronics, in accordance with a control function. [2]

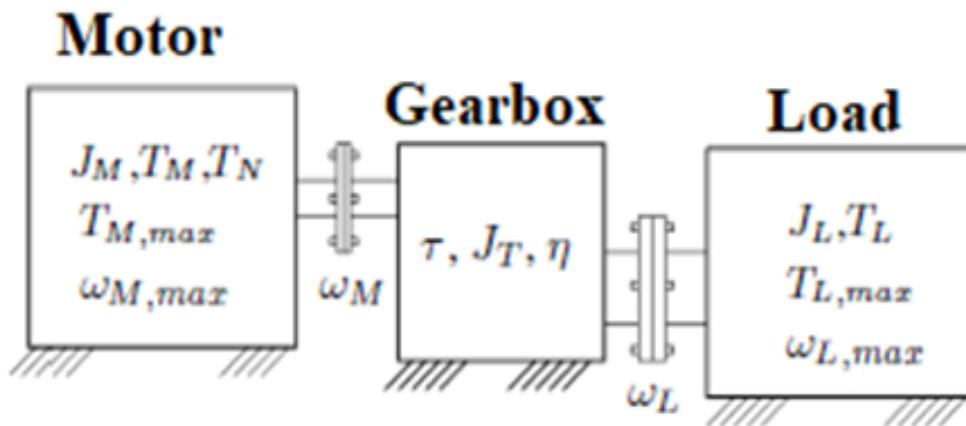


Figure 2.2: Functional scheme of a generic machine [3]

## 2.1 Motor

The engine is a machine capable of transforming a source of energy, which can be in chemical form (in the presence of a fuel), electric or thermal, in a mechanical energy or mechanical work, typically used in the field of application with the purpose of propulsion for vehicles of various types. In this case study reference is made to electric motors, in particular induction electric motors.

Induction machines are also called asynchronous machines are the most used electrical machines in the industry. In fact, induction motors are so common in industry that in many plants no other type of electric machine can be found. Induction motors are available as single-phase or three-phase.

Stator and rotor are the main parts of the induction machine. Stator has three main parts: Outer frame, Stator core and Stator winding. Outer frame or motor housing is the outer body of the motor. Its function is to support the stator core and to protect the inner parts of the machine against mechanical impacts. Stator core is made of thin soft-iron laminations varies from 0.3 to 0.5 mm in thickness stacked and screwed together to minimize hysteresis and eddy current losses.

Stator winding are three phase winding carried on the stator core where the six terminals of the winding are connected in the terminal box of the machine. Therefore, the

three-phase winding could be connected in star or delta.

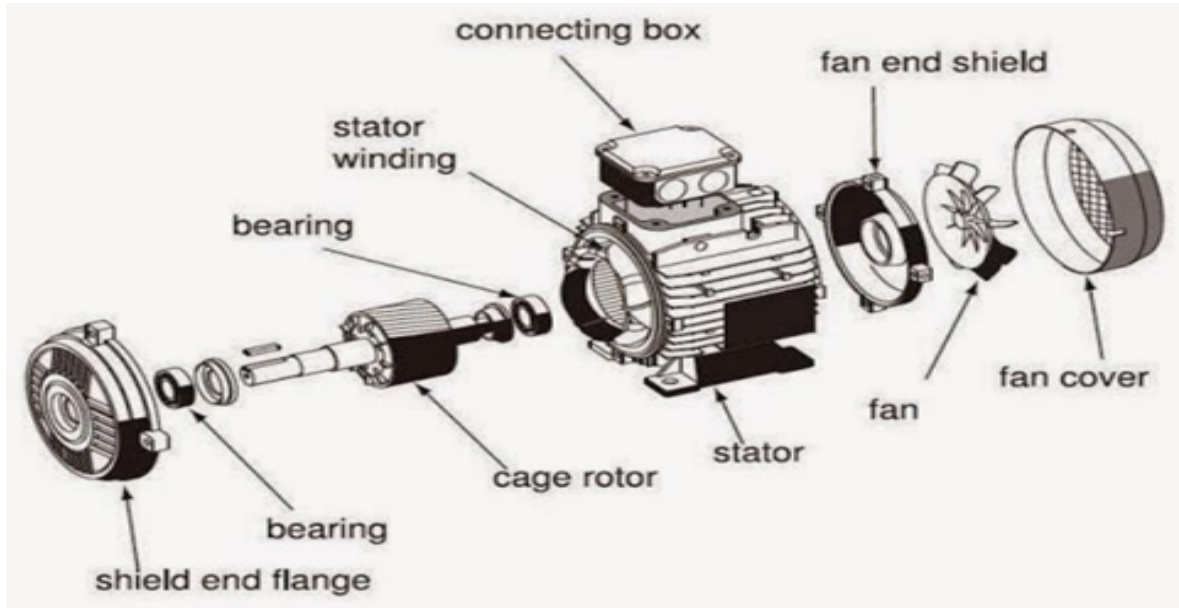


Figure 2.3: An exploded view of squirrel cage induction motor [4].

Rotor is the rotating part of the motor. It has cooling fins. At the back, there is another bearing and a cooling fan affixed to the rotor. The fan is enclosed by a fan cover. There are two types of rotors, Squirrel cage rotor and Phase wound rotor. Most of the induction motors employed in the industry have squirrel cage rotor Due to its simple and robust construction. It consists of a laminated cylindrical core having semi-closed circular slots at the outer periphery. Copper or aluminum bar conductors are placed in these slots and short circuited at each end by copper or aluminum rings. The slots are skewed compared to the shaft to obtain smooth and sufficient torque, to reduce the magnetic locking of the stator and rotor, and to increase the rotor resistance since the length of the rotor bar conductors is incremented. Phase wound rotor consists of stampings as in stator core. A number of semi-closed slots are punched at its outer periphery.

A 3-phase insulated winding is placed in these slots. The rotor is wound for the same number of poles as that of stator. The rotor winding is connected in star and its remaining three terminals are connected to the slip rings. In this case, any external resistance can be added in the rotor circuit. The rotor is also skewed as in squirrel cage rotor.

When 3-phase supply is fed to the stator winding of a 3-phase wound induction motor, a resultant rotating magnetic field at constant angular velocity is produced in the stator core. Let this field is revolving in an anti-clockwise direction at synchronous speed where:

$$n_s = \frac{120 * f}{P} \quad (2.1)$$

Where  $f$  is the frequency of the input electrical power, and  $P$  is the number of poles of the induction machine. The rotating magnetic field is cut by the stationary rotor conductors and an emf is induced in the rotor conductors. As the rotor conductors are short circuited, current flows through them, Furthermore, a resultant field is produced by the rotor current carrying conductors. This field tries to come in line with the stator rotating field, as a result, an electromagnetic torque is developed, and rotor starts rotating in same direction as that of stator rotating field.

The rotor then run at a mechanical speed close to and less than the synchronous speed as it tries to attain synchronous speed but never reach. It is because if the rotor revolve at the synchronous speed then the relative speed between rotating stator field and rotor will be zero, therefore, neither emf will be induced in rotor conductors or current nor rotor field and hence no torque will be produced. Thus, an induction motor can never run at synchronous speed. [4]

When we talk about the speed of an engine without referring to the load conditions, we refer to the nominal speed or the speed at no load, or the speed assumed by the engine in the absence of any type of load. The relationship between torque and speed (mechanical characteristic) can be represented graphically by a curve in the plane engine torque - Motor speed, this mechanical feature allows you to study the behavior of the engine in some conditions regardless of its electrical characteristics.

By varying the value of the parameters for all permissible values, the characteristic curve changes and covers a portion of the engine torque - engine speed plane. The area under consideration corresponds to the possible operating points of the engine, divided into continuous operating zone and intermittent operating zone, that is, the set of points for which the engine can only operate for short periods. The limits of the operating zones depend on several factors of the engine itself and its control systems. [2]

## 2.2 Motor control

Before the invention of variable frequency voltage and current source inverters the induction motor was never thought as continuously variable speed drive. Only some adaptations of the load characteristic were feasible by manipulations of the rotor resistance. The early days of variable speed induction motor drives can be recorded back to the 1960s, supplied by the silicon controlled rectifier (SCR). In that time the principle of speed control was based on steady state considerations of the induction machine. The  $v/f$  control was one outcome and even today it is commonly used for the open-loop speed control of drives with low dynamic requirements. [5]

Though R. H. Park introduced rotating reference frames already in 1929, it took a long time to develop the idea of field-oriented control (FOC) that is based on the fundamental insight that the torque is proportional to the cross product of stator

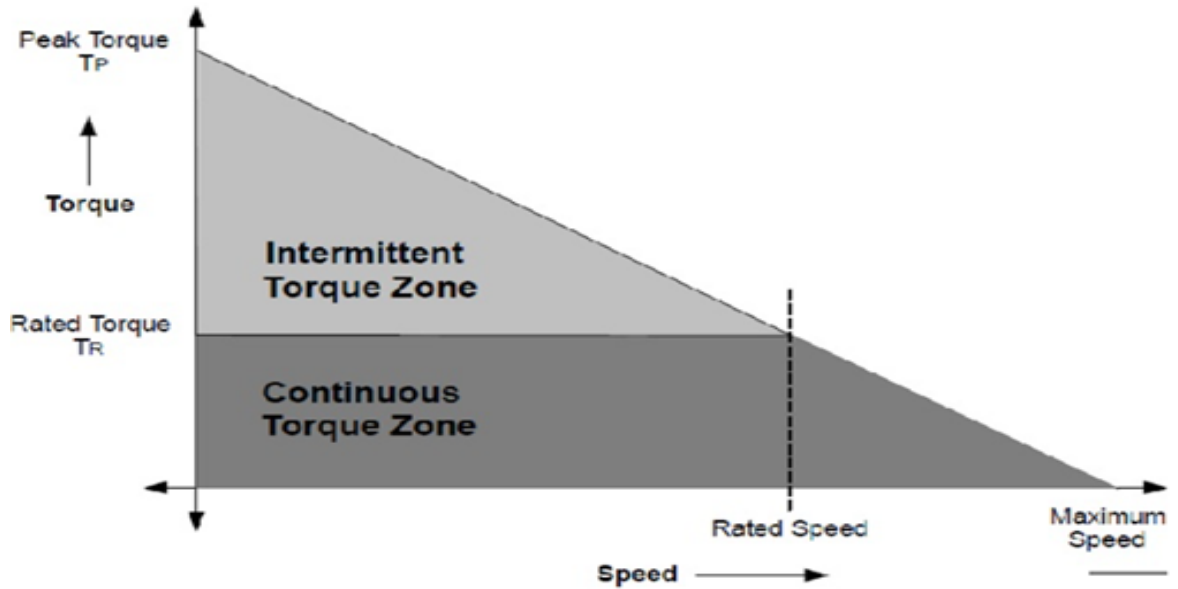


Figure 2.4: Brushless motor operating zone [2].

current and flux. The resulting decoupled control of torque and field excitation is then quite similar to DC motor control.

In 1971, Direct FOC (DFOC) was developed within Siemens by Blaschke. In the mid 1980's, when many researchers worked on improvements of the basic FOC, Depenbrock presented the Direct Self Control (DSC), and Takahashi and Noguchi the Direct Torque Control (DTC). Unlike FOC which includes pulse width-modulated current control loops, DSC and DTC are hysteresis controls working directly with stator flux and torque without having the need for an inner current control loop. [5]

### 2.2.1 Field Oriented Control (FOC)

In the concept of Indirect FOC (IFOC), flux orientation is realized only by means of feed-forward control, typically by a calculation of the slip frequency from the reference values. This approach is simple and well performing for the speed and position control even at low speeds. However, the major drawback is that, the orientation of the control is very sensitive to the rotor resistance, which affects the robustness of the control. To overcome this problem the rotor resistance has to be estimated online.

The other way is the Direct FOG (DFOC). The original approach of Blaschke included flux measurement coils to accomplish the flux orientation. Instead of flux measurement, DFOC includes usually flux observers to enable flux orientation. The different types of flux observers will be treated in the next section.

The concept of field orientation is not restricted to rotor flux orientation but also possible with stator or air-gap flux.

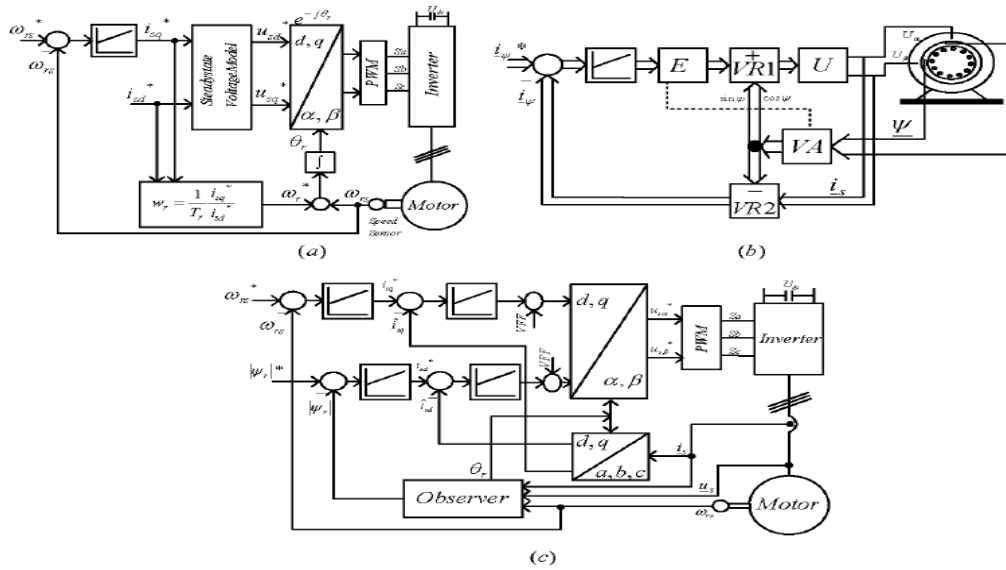


Figure 2.5: Different field-oriented control schemes [5]

## 2.2.2 Direct Self Control and Direct Torque Control

Unlike FOC with stator current as inner control objective, DSC and DTC govern the stator flux by means of hysteresis controls. The overall concept of torque control in DSC and the DTC is the same, block schematic of these controls are shown in the figure2.6.

The difference between the techniques is that DSC performs a hexagonal flux trajectory while that of DTC is circular. The DSC was developed for high power and traction application. Both possess high torque dynamics compared to FOC. However, both the control techniques have the inherent drawbacks of variable switching frequency and higher torque ripple. Since then it has been continuously worked by researchers to overcome these inherent drawbacks. These problems opened various roots for researchers to work on different kinds of strategies to avoid the variable switching frequency, but sticking to the fundamental concept of torque control. In order to realize the control, either DSC or DTC, flux and torque estimates have to be provided by a flux observer, quite similar to FOC. Although, mostly the so-called voltage model is mentioned in combination with DTC this is not mandatory, as the control will also run with other types of observers. [5]

## 2.2.3 V/F Control

The torque produced by induction motors is created by interaction of stator and rotor magnetic fields. If the frequency is reduced by a certain value, then magnetic saturation occurs due to the large magnetic flux causing failure of the motor. Because of this, the voltage and frequency should be changed proportionally resulting constant

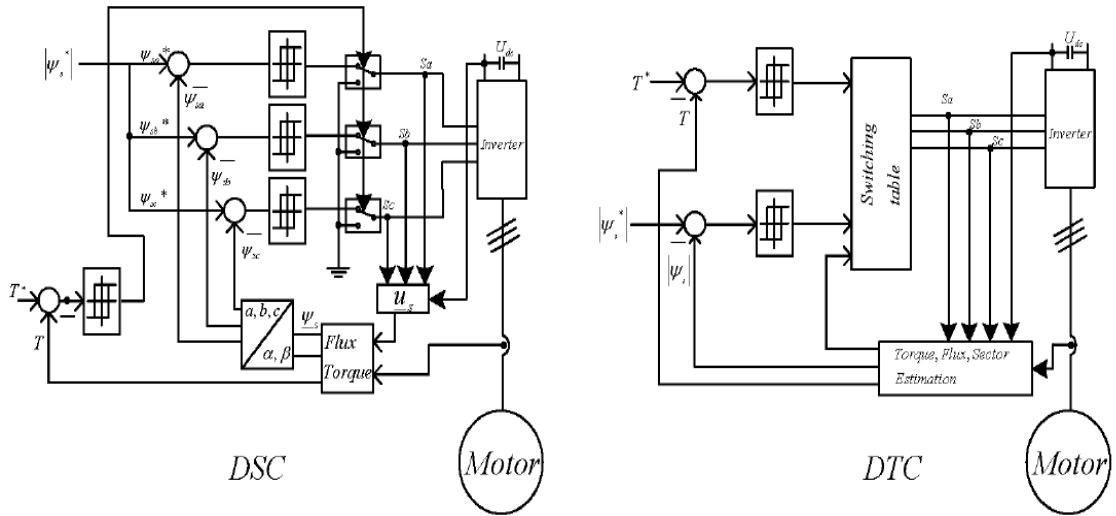


Figure 2.6: DSC and DTC block schematics [5]

voltage and frequency ratio. This is known as constant V/f control of induction motors. In this control method the speed of the induction motor is controlled with the stator voltage and frequency by maintaining the airgap flux at the desired steady state value. For constant V/f control, the values of the voltage and frequency need to be stored. Normally, look up tables are used for storing voltage and frequency values. There should be a voltage value for every frequency value. If small changes are needed either the look up table to be too long or interpolation between the values should be used. This need large storage and too much time for the computations. [6]

## 2.3 Gearbox

The main function of mechanical transmission of torque and speed is to suit the work scheme of the drive component to the necessities given by the performance study of the driven component. In vehicles driven by internal combustion engines, a transmission system is designed so that the relation between drive and driven systems can be changed during its operation. This thought can be applied to all kind of motorization types or source of torque and power. Although the terminology transmission is used in such systems, transmitting is not its primary function. Converting torque and power to adequate the momentary needs for the operation is its main role.

A transmission system is a vital part on the development of driven mechanical systems. Applied to guarantee the continuous optimum torque required to a determined function, transmission systems have evolved in the last decades to many different construction types and mechanical principles.

For each application there are different system options, and the selection of a determi-

nate type can be an engineer's choice or may be the consequence of references inherited to the project. This restricts the options to select an ideal transmission system, which can lead to undesired results and to the use of old solutions, thus propagating old technologies. A good example is the manual transmission system used in the automotive industry and the bicycle gear set. Both systems are more than five decades old, used in large scale and have received very few improvements over the years.

There are different types of transmission, are divided by the structure and how they works.

### 2.3.1 Gear

Torque transmission using gears is given by the coupling of two or more gears connected directly by their teeth. According to the Fundamental Law of Coupling, the velocity ratio of the gears has to be constant during the coupling. These sets are governed by the rate of torque in respect to the input and output velocity or by the number of teeth from the drive and driven gears. Manual transmission will be referred to gear transmissions that need the user's intervention to operate. Single rate transmission boxes are frequently used in operation machines in industry as on drive motors, mechanical reduction in press and in naval operations. These reductions can be fixed or with the ability to decouple during operation.

A multi ratio gearbox can be differentiated in three types; synchronized, non-synchronized and preselected.

- Synchronized: the name of these transmissions is due to the synchronizer that has the function of adding a determined gear pair to transmit torque. The most common gearbox has three shafts, being the entrance, the secondary and the exit shaft. The gears of the secondary shaft are fixed, while the gears of the entrance and exit shaft are subjected to the action of the synchronizer;
- Non-synchronized: in those gearboxes the pairs are not previously coupled. The coupling is given with the movement of the secondary shaft;
- Preselected: here the shift lever is used to pre-select the gear to be used, a separate control is then used to engage the gear. This system avoids the need for the user to control a manual clutch, resulting a faster shifting and being able to operate high input power with a lighter mechanism

This transmission shifts gear via a pre-established control that doesn't need a direct action of the user. For this work, automatic transmission will be referred to gear transmissions. In the automatic-manual transmission, the selection is given by a pre-defined control or by the direct action of the user. In this case, the user can choose if he wants to control the shifting or not. Generally both actions operate in parallel, although a

direct action of the control unit is needed to guarantee the integrity of the operation. In the planetary transmission the torque is transmitted by two loading paths. Due to its compact size and reliability, this system is largely used in automatic transmissions for cars, planes and helicopters.

Planetary gear trains have great advantages since they divide the load between the gears due to their concentric entrance and exit shafts. This results in the radial resultant of the forces being zero at the entrance and exit shaft, which guarantees a reduced package and weight when compared to gear boxes with parallel shafts. This coaxial aspect is important since it allows high speed shifting without decoupling the gears. The smaller package results in smaller and more rigid components, lesser noise and vibration and a higher efficiency.

Since the planetary transmission transmits only torque, the bearings' project is simplified, allowing, in some cases, a nearby coupling to the drive and driven components.

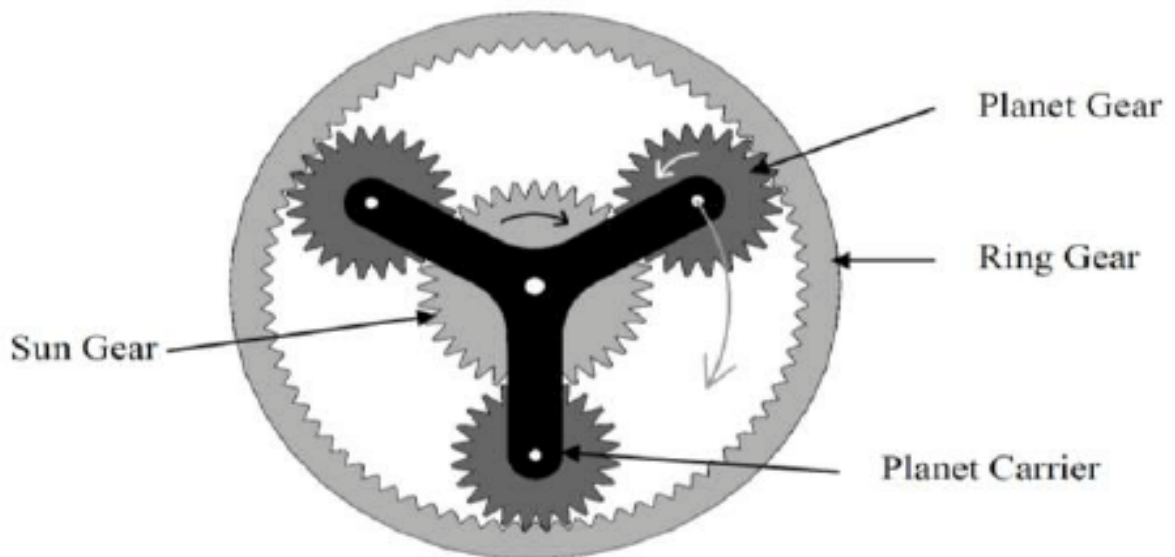


Figure 2.7: Planetary transmission [7].

### 2.3.2 Hydraulic transmission

In a hydrostatic transmission the mechanical energy of the drive shaft is transformed in pressure energy in an almost incompressible fluid and is then converted into mechanical energy on the driven shaft. The flow, produced with a determined entrance velocity, is variable and depends on the selected displacement.

The characteristic feature of this transmission is that energy is transferred through fluid pressure and flow rate, thus through kinematic parameters. It means that the static pressure is way higher than the dynamic pressure corresponding to the fluid's high velocities. An advantage is that with a simple adjustment on the input flow, the mean velocity can be continuously selected. This means that the output velocity can be



maintained independent of the output torque. The limitations of these transmissions are vast because of their operational method regarding the conversions from rotational energy into pressure energy and vice-versa. [8]

## 2.4 Load

The term load refers to a general user who absorbs energy to perform a specific task, in this case study a user is considered to absorb mechanical energy to perform a certain action. It is of interest to describe the load through its mechanical characteristic, that is the link between the resistant speed and its relative resistant torque. This curve is useful for studying loads that work predominantly at constant speed.

For the choice of the signs of the resistant torque and the resistant speed an opposite convention to that of the motor is used; choosing this convention defines the correct behavior of load when it is in the first and third quadrant of the graph that relates torque and load speed. The load characteristic curves may vary depending on the working conditions, it is also possible to define for the loads an operating field called the place of the loads. Place of the loads means the set of possible operating conditions under which the load is expected to be present. The place of the loads can be of two types [2]:

- Place of static loads, loads that operate for long periods at substantially constant speed (constant or slowly variable speeds and loads), even if the speed or load occasionally varies.

The place of the loads is represented in the  $C_r, \omega_r$  plane, delimited by lines, which in general may have a different trend from that of the characteristic curves;

of dynamic loads, as opposed to static load, there are resistant torque variations involving inertia actions occurring in the acceleration and deceleration phases.

Especially for dynamic loads, but also for static loads, the load location can easily affect all four quadrants.

In the specific case of place of dynamic loads, an additional point, called RMS (Root Mean Square), is introduced, characterized by the quadratic mean values of torque and load speed, is used to perform a thermal verification during the sizing of the motor gear motor system. [2]

## 2.5 Braking system

A Braking system is designed to control and halt the moving automobile vehicle or industrial applications, to pull off this, various components within the brake system should convert the moving object from kinetic energy into heat energy, this conversion

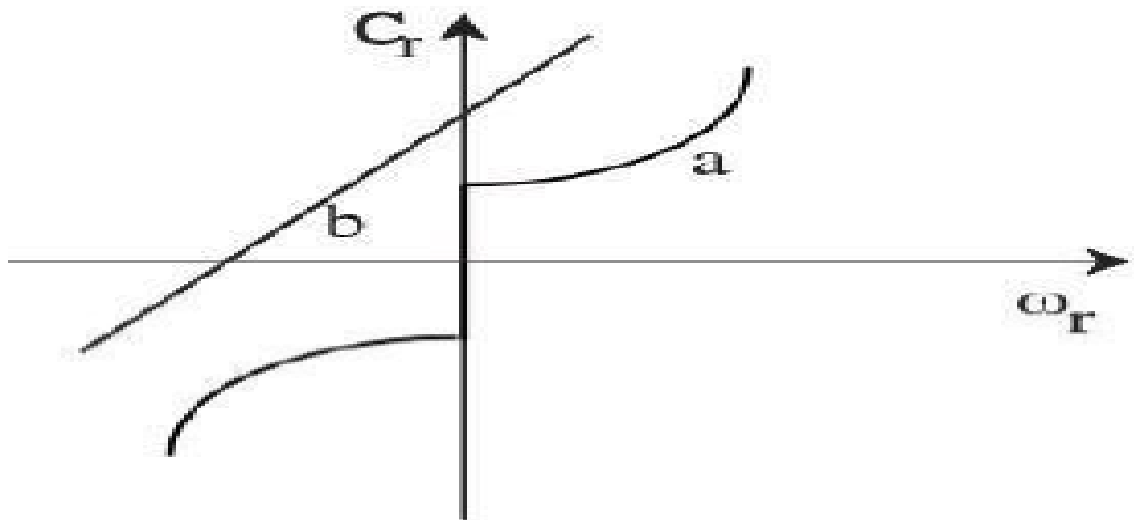


Figure 2.8: Characteristic curve of a purely passive load (a) and a load not purely passive (b) [2].

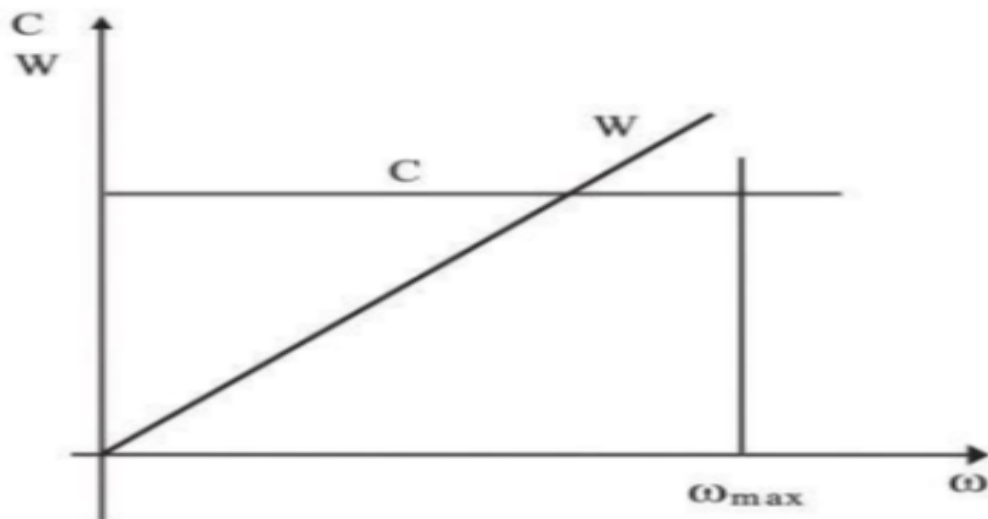


Figure 2.9: Load location at constant torque, torque curves C and power W [2].

of force is carried by friction, which reduces the speed and also helps in de-accelerate the vehicle. This Frictional force is generated by the Frictional contact of bodies between the shoe and moving drum or disc of an automobile braking system.

Friction opposes the body that is in contact with each other by relative motion, so when it's used energy to counter the friction it produces heat, and components such as pads, liners, drums, and discs get heated during the process and act to slow down the moving body.

The frictional force depends upon the surface nature materials, to achieve the braking effect of a moving body it depends on the speed of relative motion. Hence to experience



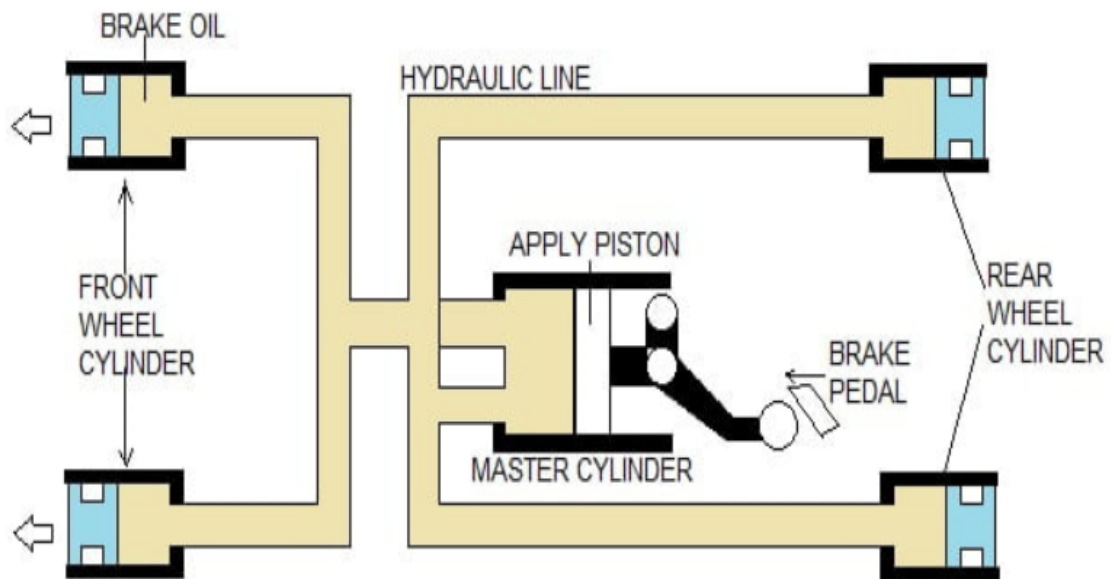


Figure 2.11: Hydraulic brake system [9]

## 2.5.2 Pneumatic Braking System

It maintains a higher level of pressure inside it, and the pressure in line starts dropping the moment when the driver applies the brake causing a result in slowing down. It operates on compressed air helps in charging up the entire braking system to maintain optimum pressure. The charged-up lines make sure for the brakes to be dis-engaged, the very moment the driver applies the force onto the pedal the air pressure drops.

Due to this sudden drop of pressure, the control reservoir, and gets disconnected causing the auxiliary cylinder to be connected to the brake cylinder. Compressed air from the auxiliary cylinder gets to the brake cylinder causing it to be more engaged. This is how the pneumatic brake works. The Pneumatic offers the following advantages:

- Pneumatic brake equipment is much more reliable than hydraulic brakes;
- If there is a leakage in the pneumatic brake system, there is less wastage as compared to leakage in the hydraulic brake system;
- It is very powerful with a single pedal that can work the entire pneumatic brake equipment which makes everything easy to control and there are fewer chances of mistakes;
- Their systems are clean as the compressed air used in the operation can be directly dumped into the atmosphere. [9]

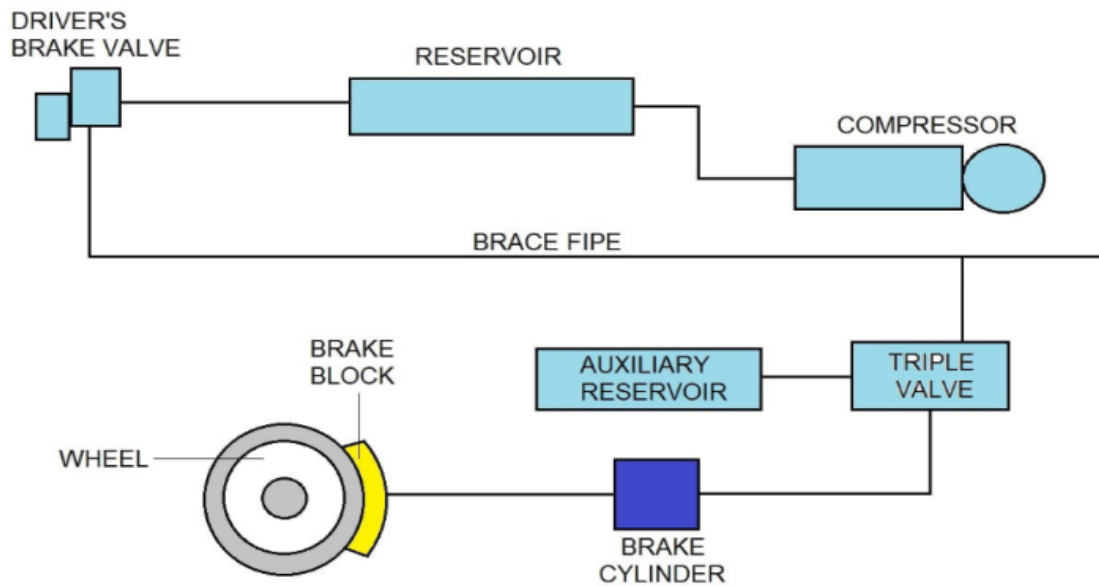


Figure 2.12: Pneumatic brake system [9].

## 2.6 Axle Load Absorbers Brakes

Axle Load Absorbers (ALA) brakes, alternatively called IHCs, are hydraulics devices that allow the application of a torsional load through the friction developed by a system of brake discs and associated counter-discs, controlled by an external pressure. A hollow toothed shaft, inside the brake itself, houses the adapter in a splined housing. Externally mounted on this shaft are several friction discs in an oil bath, which being compressed by a hollow piston, develop the braking torque to be transmitted to the load. The latter, concentric to the shaft itself, is controlled by an external control applicable through the appropriate port.

Through the pressure port, it is possible to adjust the force with which the hollow piston brings the friction plates to compression, overcoming the back pressure exerted by the cooling oil. Through the service oil inlet port, the fluid is pushed inside the channels of the hollow gear shaft, and redirected between the various friction discs and counterdiscs through holes arranged perpendicular to the channel. The oil heated by thermal convection on the discs is then recovered from the return port, which is external to the first one.

## 2.7 Test cell and testing in agriculture industry

A test bench or testing workbench is an environment used to verify the correctness or soundness of a design or model. In the context of software or firmware or hardware engineering, a test bench is an environment in which the product under development is tested with the aid of software and hardware tools.

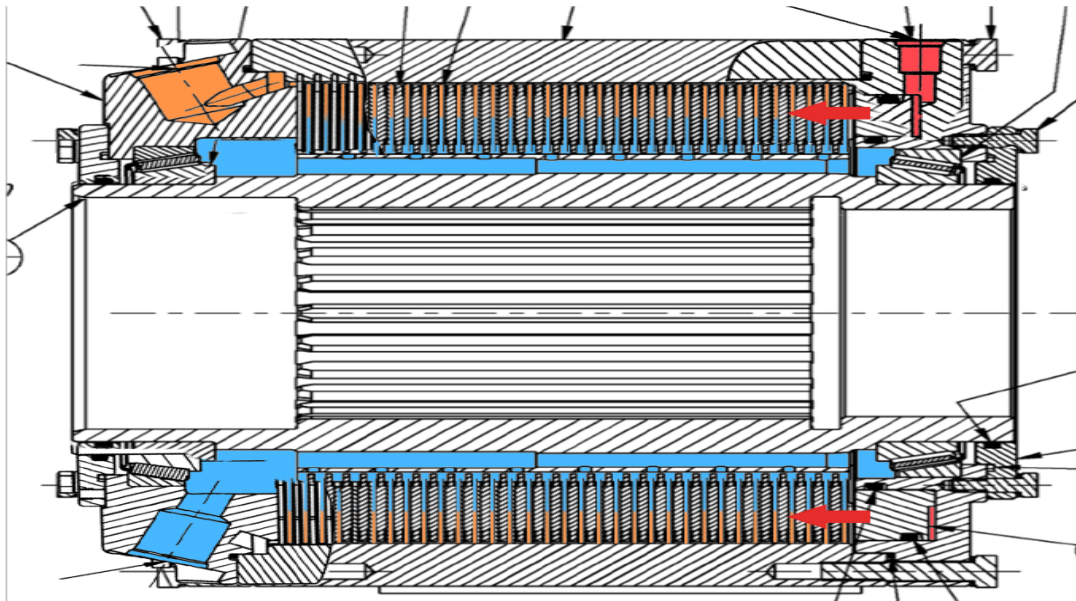


Figure 2.13: Approximate oil flow inside the brake. In red is the line of pressure, in blue the inlet cooling line and in orange the outlet.

The software may need to be modified slightly in some cases to work with the test bench but careful coding can ensure that the changes can be undone easily and without introducing bugs.

In the automotive agriculture industry, there are different types of tests that are performed on the complete vehicle or on individual components, the tests performed are divided into:

- Structural test;
- Driveline test;
- Brakes test;
- HVAC test;
- Veichle test;
- Hydraulic components test.

Of interest among the various types of tests is driveline testing, one of the main elements in the automotive field.

### 2.7.1 Driveline test

Tests performed on the driveline can be divided into:

- Durability, to test the driveline, the test cell is constructed in such a way that a variable load is applied to the wheel hubs; the applied load is a function of the different sizes of driveline being tested and the type of actual operation being emulated. The control of powertrain gear change occurs according to load conditions;
- Software tuning, the goal of the test is to fine-tune the vehicle logics and parameterize the controllers, the test cell is constructed in such a way that a variable load is applied to the wheel hubs;
- Dynamic test, the test focus is the braking analysis in case of dynamic loading condition. The test cell controls the engine speed emulating the missing ECUs in the system. The dynamic loading is applied by means of calibrated flywheels spinning at a certain speed;
- EOL Test (End Of Line), the test focus is a general checking of the correct driveline behaviors. The test cell controls the gear-shifting and the rotation in order to validate the gears assembly inside the housing.

### 2.7.2 DLG Cycle Test

The DLG Test Centre, that sets the benchmark for tested agricultural technology and farm inputs and is the leading provider of testing and certification services for independent technology tests, has developed a new test the "DLG - PowerMix™". This test has been created as a new measurement of tractor power and is an essential step closer to more precision in agriculture testing. The "DLG-PowerMix™" has for the first time combined measurements of all three major power strains: - drawbar pull, hydraulic power output and PTO power. Through this test we have aimed to create a better system of evaluation for the testing of tractors, creating data which we believe is of much greater value to the farmer.

The "DLG - Power Mix" takes measurements through dynamic and cyclic loading of the tractor enabling the integration of part loads.

The test also realistically illustrates the effects of shifting and load changing, with the interaction between these systems collated and recorded. Acceleration and deceleration phases can be integrated in to the test cycles. Measurements of the tractors cooling capacity is part of a total sum as is the hydraulic efficiency of the coupled implement. The "DLG-PowerMix™" also allows for the first time to test driving and control strategies, this is because the dynamic and cyclic loading of the tractor control system has been made clear. Fuel consumption data is collated under many different operating conditions. In the test an efficient machine must deliver good values of fuel consumption in the total operational field in order to achieve good overall results.

A well adjusted torque characteristic is also necessary to fully satisfy the test program. [10]

## 2.8 Controllers

Controllers are widely used in many industries today, as they help to regulate processes that would otherwise be unpredictable or not give the response needed. There are many different types of controllers today and research is continuing in finding better and more improved designs. The two main types of controllers are linear and non-linear. Most systems have a least a small degree of non-linearity to them. Many are extremely non-linear. For this reason, nonlinear controllers often can provide better results. However, the downside to these is that the math and theory needed to begin designing them is extremely advanced. Linear controllers on the other hand can be modeled relatively easily and often can provide very good results. [11]

### 2.8.1 PI controller

Proportional Integral controller sometimes also known as proportional plus integral (PI) controllers. It is a type of controller formed by combining proportional and integral control action. Thus it is named as PI controller. In the proportional-integral controller, the control action of both proportional, as well as the integral controller, is utilized.

This combination of two different controllers produces a more efficient controller which eliminates the disadvantages associated with each one of them. In this case, the control signal shows proportionality with both the error signal as well as with integral of the error signal. Mathematical representation of proportional plus integral controller is given as: [12]

$$m(t) = K_p e(t) + K_I \int e(t) \quad (2.2)$$

The figure 2.14 represents the block diagram of the system with PI controller. It is to be noted here that one can use integral controllers separately without combining it with proportional controllers. However, generally proportional plus integral controllers are used that combinely overcome the disadvantage of integral controllers. A major disadvantage which is associated with the integral controller is that these are quite unstable.

The reason behind this is that integral controllers show somewhat slow response towards the produced error.



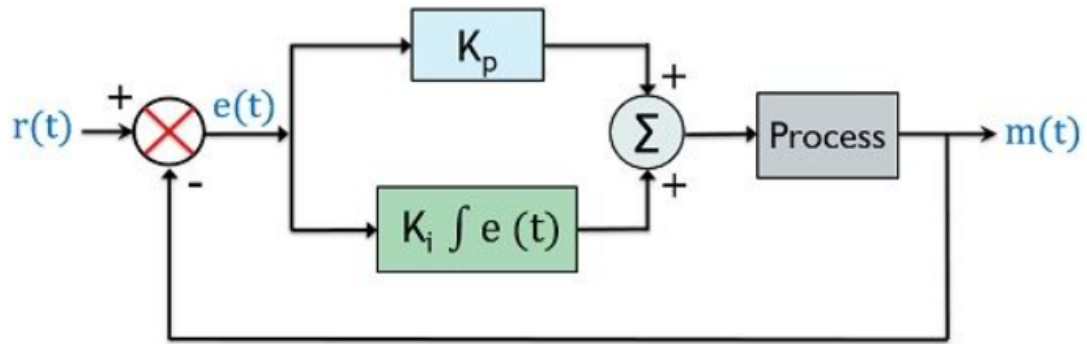


Figure 2.14: PI controller closed loop [12].

## 2.9 Modeling and simulation of the behavior of a system

Modeling is a way to create a virtual representation of a real-world system including software and hardware. Simulation is used to evaluate a new project, diagnose the problems of an existing project and test a system under conditions that are difficult to reproduce in a real system. If the components of the system are driven by mathematical relationships, it is possible to model and simulate a virtual representation of the system under various conditions, in order to examine its behavior.

Creating models and simulating them is important for testing conditions that can be difficult to reproduce exclusively with hardware prototypes, especially in the early stages of the design process, when hardware may not be available. Modeling and simulation activities can improve the quality of the system design process from the earliest stages, thus reducing the number of errors encountered in later stages.

This results in a significant reduction in development time and costs. MATLAB, Simulink, Stateflow and Simscape are tools for designing systems, simulating their behaviour in an environment and automatically generating distributable code on hardware.

Common representations of system models include block diagrams, diagrams, and state diagrams. Thanks to these representations it is possible to model mechatronic systems, control software, signal processing algorithms and communication systems. [13]

### 2.9.1 Matlab

MATLAB combines an optimized desktop environment for iterative analysis and design processes with a programming language that expresses mathematical operations with matrices and arrays directly. MATLAB has professionally developed toolboxes, rigorously tested and fully documented.

MATLAB applications allow you to see how different algorithms work with data. It-

eration is possible until you find the desired results, then automatically generates a MATLAB program to play or automate your work.

## 2.9.2 Simscape

Simscape lets you quickly create models of physical systems in the Simulink environment. With Simscape, you can create models of physical components based on physical links that are directly integrated with block diagrams and other modeling paradigms. System models, such as electric motors, bridge rectifiers, hydraulic actuators, and refrigeration systems, can be modeled by assembling the key components in a schematic. Simscape complementary products provide more complex components and analysis capabilities.

Simscape allows you to develop control systems and test system-wide performance. Custom component templates can be created with the MATLAB-based Simscape language, which allows text-based authoring of physical modeling components, domains, and libraries. You can parameterize your own models using MATLAB variables and expressions, as well as design control systems for your physical system in Simulink. Simscape supports C code generation to deploy models in other simulation environments, including Hardware-In-the-Loop (HIL) systems.

# Chapter 3

## Sizing criteria

The correct sizing of the motor reducer system allows to move the load according to the project specifications and to avoid under-sizing or oversize. Depending on the type of load, that is static load or dynamic load, there are different methods for the correct sizing, the only thing in common among all methodologies is the iterative process to arrive at the solution. In many applications the load works at a constant speed with a constant torque, identifying the best gearbox means choosing the gearbox that allows to use smaller motor.

In the case of constant speed loads the simplest choice is to choose an engine that corresponds roughly to an ideal speed generator. Once the heaviest working condition of the load (with maximum resistant torque) has been identified, the resistant power of the load to be supplied is defined. The engine is then chosen so that the rated power, or the continuous power, of the engine is greater than the resistant one considering the performance of the transmission; Finally, the required gear ratio is calculated through the ratio of the load speed to that of the engine.

Depending on the available motors and gearboxes, several suitable motor and gearbox groups can be identified, the final choice must be made on economic aspects or on technical aspects such as starting time. [2]

### 3.1 Sizing criteria for a dynamic load

The verification operation is to check that the operating range of the motor covers that of the load for both steady-state and transient conditions. It is necessary to verify:

- For each operating condition the torque-velocity point is within the intermittent operating range of the motor;
- The root mean square values of torque and speed are within the continuous operating range.

The first check ensures that the motor can generate the motion required by the law of motion; the second check allows an assessment of whether this developed motion produces overheating of the motor. The root mean square values representing the torque and thermal equivalent speed are calculated as given below:

$$C_{m,RMS} = \sqrt{\frac{1}{T} \int_0^T C_m^2 dt} \quad \omega_{m,RMS} = \sqrt{\frac{1}{T} \int_0^T \omega_m^2 dt} \quad (3.1)$$

Using the representation of dynamic locus of loads presented in figure 2.4 it's possible to show, in the following figure, four examples of motor verification, in which only case C is found to be suitable since it meets the selection criteria, this is because the RMS point lies within the continuous operating range of the motor and the load location lies within the intermittent operating range of the motor.

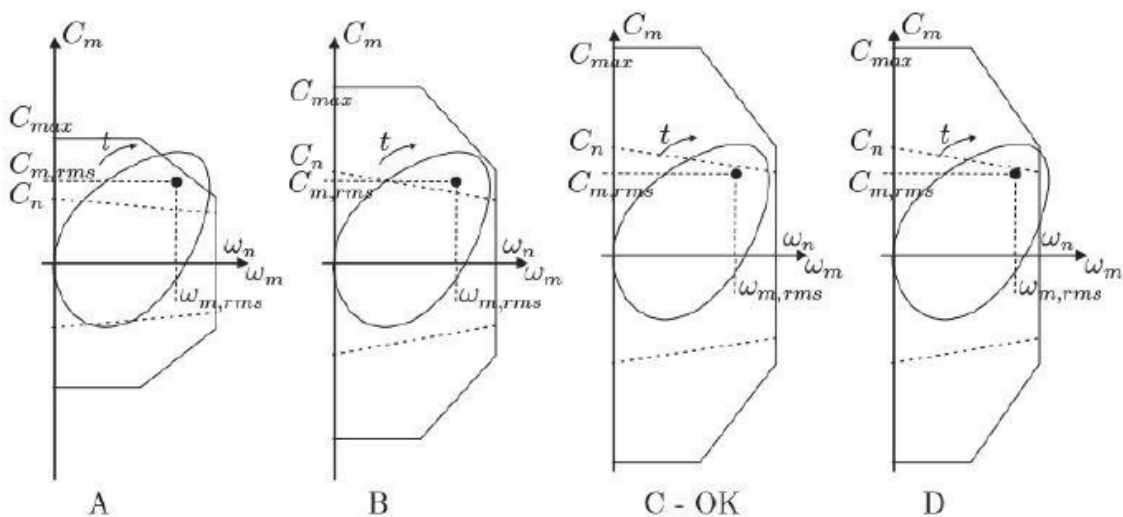


Figure 3.1: Examples verification of motors for dynamic loads [14].

In cases of highly dynamic cycles ( highly variable torque and speed ) for the selection of the motor-gearbox assembly, it is not meaningful to refer to the rated power of the motor. In fact, the inertia actions that are generated on the motor itself can be a significant percentage compared to the load actions, furthermore, using gear ratios that reduce the torques required by the load will increase the acceleration of the motor and thus the inertia actions.

For a guideline choice of size, reference is made to an equivalent power calculated for the load, given by the product of the mean square torque (sum of resisting and inertial torque) for the maximum speed while, for the motor, the product between the nominal (continuous) torque and the maximum speed is considered:

$$(C_r + J_r \dot{\omega}_r)_{rms} * \omega_{r,max} < C_n \omega_{m,max} \quad (3.2)$$

As it is a complex procedure to analyze peak torque, it is convenient to make an initial selection of motors and gearboxes through thermal torque. For this reason, a criterion is used that allows an initial filtering of gearmotors, this criterion is called  $\alpha$  -  $\beta$ . [2]

## 3.2 Alpha - Beta criteria

As reported in the paper "Evaluation of motor-reducer coupling in high demanding industrial applications" proposed by Giberti the alpha - beta criteria is based on the definition of two coefficients, the alpha coefficient (acceleration factor), which depends only on the engine parameters while the beta coefficient is based only on the load parameters (load factor). In order to use this criterion it is necessary to define two design constraints of the system:

- Limit on maximum achievable speed, Since each motor has a maximum achievable speed  $\omega_{L,max}$ , it should be:

$$\omega_{L,max} \leq \tau \omega_{M,max} \quad (3.3)$$

where  $\omega_{L,max}$  is the maximum speed achieved by the load. Considering a specific motor, the speed reducer must satisfy the following constrain:

$$\tau \geq \tau_{M,lim} = \frac{\omega_{L,max}}{\omega_{M,max}} \quad (3.4)$$

where  $\tau_{M,lim}$  is defined, for each motor, as the ratio between the maximum speed achieved by the load and the one achievable by the motor. For a specific motor,  $\tau_{M,lim}$  is the minimum transmission ratio value that can be employed to drive the given load;

- Limit on motor rated torque, the motor torque  $T_M$  can be written as:

$$T_M = \frac{\tau T_L^*}{\eta} + J_M \frac{\dot{\omega}_L}{\tau} \quad (3.5)$$

where:

$$T_L^* = T_L + J_L \dot{\omega}_L \quad (3.6)$$

is the generalized resistant torque at the load shaft.

When selecting the motor-reducer unit, the transmission ratio  $\tau$  and the transmission mechanical efficiency  $\eta$  and the motor inertia  $J_M$  are still unknown.

Equation 3.5 highlights the dependence of the motor torque on this variables, while from 3.6 it's possible to observe that all the terms related to the load are known.

The root mean square torque is obtained from:

$$T_{M,rms}^2 = \int_0^{t_a} \frac{1}{t_a} \left( \frac{\tau T_L^*}{\eta} + J_M \frac{\dot{\omega}_L}{\tau} \right)^2 dt = \int_0^{t_a} \frac{T_M^2}{t_a} dt \quad (3.7)$$

By developing the calculations, considering  $T_{M,N}$  positive for definition, it's possible express the torque motor rms as:

$$\frac{T_{M,rms}^2}{J_M} \geq \frac{\tau^2 T_{L,rms}^{*2}}{J_M \eta^2} + J_M \frac{\dot{\omega}_{L,rms}^2}{\tau^2} + 2 \frac{(T_L^* \dot{\omega}_L)_{mean}}{\eta} \quad (3.8)$$

Then it is possible to introduce the two coefficients, the acceleration coefficient is defined as follows:

$$\alpha = \frac{T_{M,N}^2}{J_M} \quad (3.9)$$

Where  $T_{M,N}$  is motor nominal torque and  $J_M$  is the motor inertia. The load factor is defined as follows:

$$\beta = 2[\dot{\omega}_{L,rms}^2 T_{L,rms}^* + (T_L^* \dot{\omega}_L)_{mean}] \quad (3.10)$$

Where  $\dot{\omega}_{L,rms}^2$  is rms angular acceleration of the load.

The coefficient  $\alpha$  is exclusively defined by parameters related to the motor and therefore it doesn't depend on the machine task: it can be calculated for each motor using the information collected in the manufacturer catalogues. Moreover it could be reported on them, to provide a classification of the commercial motors on the basis of this standard. Otherwise, the coefficient  $\beta$  depends only on the working conditions (applied load and law of motion) and it's a measure that defines the power required by the system. Substituting  $\alpha$  and  $\beta$  in inequality 3.8 is obtained:

$$\alpha \geq \frac{\beta}{\eta} + \left[ \frac{T_{L,rms}^* \tau}{\eta \sqrt{J_M}} - \dot{\omega}_{L,rms} \frac{\sqrt{J_M}}{\tau} \right]^2 \quad (3.11)$$

Since the term in brackets is always positive, or null, the load factor  $\beta$  represents the minimum value of the right hand side of eq. 3.11. It means that the motor accelerating factor  $\alpha$  must be sufficiently greater than the load factor  $\beta$ . A motor must be rejected if  $\alpha < \beta$ , while if  $\alpha \geq \beta$  the motor can have enough rated torque if  $t$  is chosen properly. The preliminary motor choice is conducted comparing only the values  $\alpha$  and  $\beta$ ; these values are easily calculated knowing the mechanical properties of the motor and the load features. In the Fig. 3.2 the accelerating factors  $\alpha_i$  of the considered motors are compared with the load factor  $\beta$ . Let's first consider an ideal transmission with

mechanical efficiency equal to one. Solving inequality 3.11 it's possible to find out, for each motor, a range of transmission ratios bounded by  $\tau_{min}$  and  $\tau_{max}$  eq. 3.12 and 3.13, which satisfy the limit on the motor rated torque.

$$\tau_{min} \leq \tau \leq \tau_{max} \quad (3.12)$$

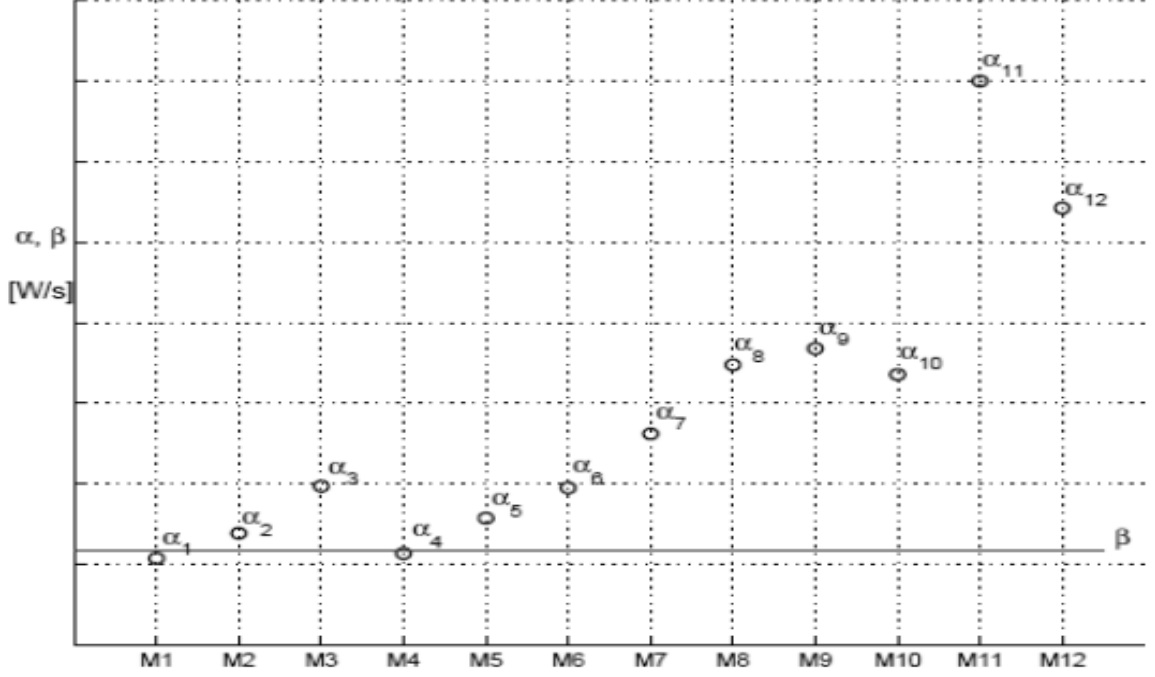


Figure 3.2: Alfa and Beta comparison [14].

$$\tau_{min}, \tau_{max} = \sqrt{J_M} \frac{\sqrt{\alpha - \beta + 4 * \dot{\omega}_{L,rms}^2 T_{L,rms}^{*2} \pm \sqrt{\alpha - \beta}}}{2T_{L,rms}^{*2}} \quad (3.13)$$

Having a database containing the commercial transmissions, it's sufficient to bring each of them on the graph, representing each transmission with an horizontal line whose value on y-axis coincides with the gear ratio. If an horizontal line intersects the vertical line of a motor, this indicates that the motor can supply the required rated torque and maximum speed if that specific speed reducer is selected. By decreasing the performance of the transmission, and then with increasing of power dissipation, it is possible that the power output from a given motor is no more adequate to meet the demands of the load. Therefore, a motor which is able to perform the task planned in ideal conditions ( $\eta=1$ ), when coupled to a transmission characterized by a poor performance, should be discarded. Referring to eq. 3.11, we observe that known  $\beta$ , for each motor exist a minimum transmission mechanical efficiency value below which the condition expressed in 3.3 it is no more guaranteed. This value is called the mechanical efficiency limit and it is defined as:

$$\eta_{lim} = \frac{\beta}{\alpha} \quad (3.14)$$

A real mechanical reducer is also characterized by its inertia  $J_T$  which inevitably affects the total inertia of the system. The generalized resistant torque is:

$$T_L^* = T_L + (J_L + J_T)\dot{\omega}_L \quad (3.15)$$

that contributes to modify the load factor. This variation makes the system different from the one previously analyzed with the result that inequality 3.11 can be no more satisfied. In Fig. 3.3 the diagram is reported for a specific motor: the highlighted area is a graphical representation of constraints expressed in equations 3.4, 3.9.

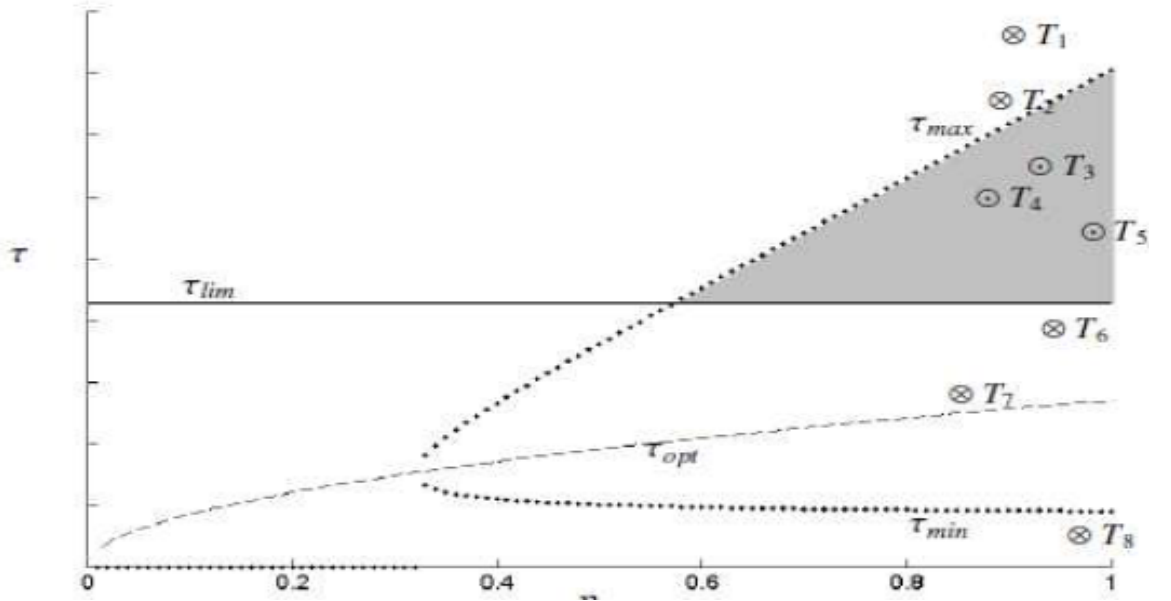


Figure 3.3: Area of acceptable motor-reducer coupling for a given transmission inertia [14].

Catalogs of commercial transmission report for each product, its transmission ratio  $\tau_i$  and its mechanical efficiency value  $\eta_i$  experimentally measured by the manufacturer. Transmission candidates for the application can be putted on the chart  $\eta - \tau$ , each defined by its coordinates  $(\tau_i, \eta_i)$ . For each selectable motor it is clear that the suitable transmissions are the only one inside the defined area. The example in Fig. 3.3 shows that a transmission (i.e. T2), even if it would be acceptable by traditional criterion of selection that do not consider the influence of real transmissions, has to be discarded [14].

### 3.2.1 Customization of alpha-beta criteria

In order to use the alpha-beta criterion for the pre-selection of the gearbox motor system, it is necessary to change the operating range of the motor, starting with an



operating range of a brushless motor as shown in Figure 2.4 and defining an operating range for an induction motor. This change is related to the scope of application of the algorithm, this criterion arises for a robotic field of application in which the brushless motor turns out to be the most popular; in industrial testing the loads turn out to be larger and therefore require higher torques. For this reason the most popular motors in this field turn out to be induction motors.

In the specific case of high-performance induction motors, the intermittent zone and the continuous zone turn out to be superimposable for this reason the generated operating range will consist of a single operating zone. By modifying the alpha-beta criterion and considering the peculiarities of such induction motors, it is possible to apply this algorithm to the present case study.

### 3.3 Matlab sizing tool

The first step is to implement the alpha-beta criterion within the Matlab software in order to perform proper filtering for motor and gearbox selection according to a known load.

The introductory part consists of pre-selecting the motors and gearboxes that it is necessary to check, specifying the main characteristics of each element. Following the definition of the systems to be tested, a correct law of motion must be set in order to make a meaningful selection. The law used is a parametric ramp, allowing cyclically increasing load severity. The following figure shows the law of motion used at the first iteration of the algorithm:

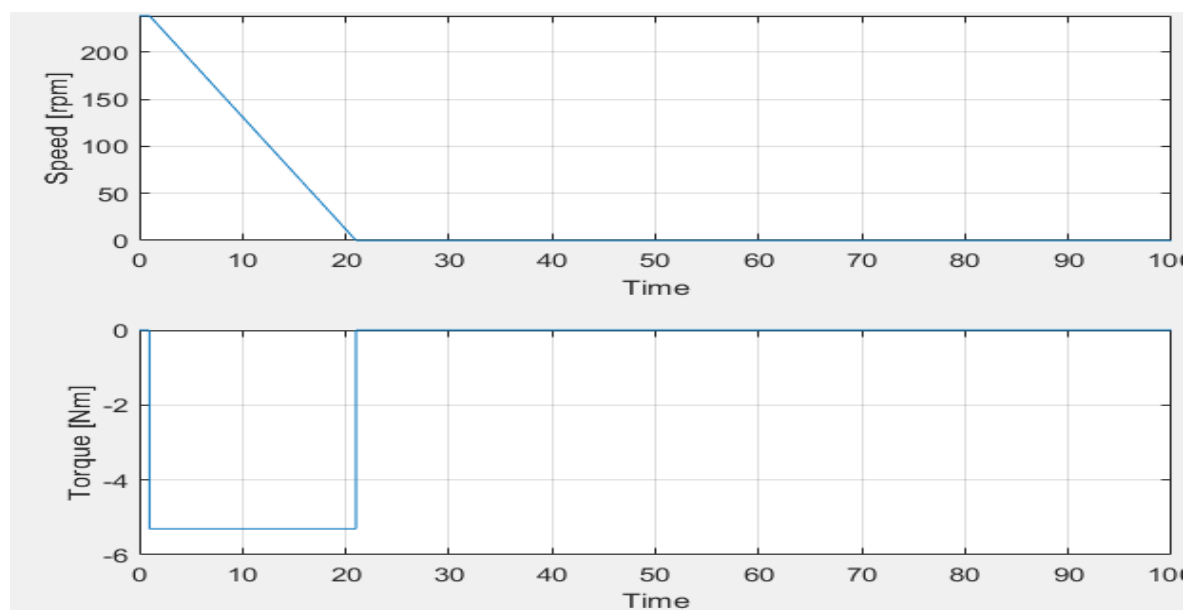


Figure 3.4: Ramp law of motion.

In order to parametrically define the law of motion, it is necessary to define a max-

imum velocity, which is present in the initial section at constant speed, and a descent time; this allows us to derive deceleration, i.e., the critical aspect of the dynamics of the system.

As for the estimated torque, it is necessary to know the inertia to be moved; in order to obtain this figure, considerations had to be made regarding the weights and wheel diameters of the vehicle to be moved. In order to calculate the inertia of the load, it is necessary to distribute the weights between the front and rear axes so that the inertia of a rotating body can be calculated through a balance of the kinetic energy of the system:

$$\begin{cases} T = \frac{1}{2}Mv^2 + \frac{1}{2}mv^2 \\ T = \frac{1}{2}J_{r1}\omega_{r1}^2 + \frac{1}{2}J_{r2}\omega_{r2}^2 \end{cases} \quad (3.16)$$

Knowing that angular velocity is defined as:

$$\omega = \frac{v}{r} \quad (3.17)$$

Through these considerations, the definitions of inertias for the front and rear of the vehicle can be obtained; the results are shown as follows:

$$\begin{cases} J_{r1} = Mr_1^2 \\ J_{r2} = mr_2^2 \end{cases} \quad (3.18)$$

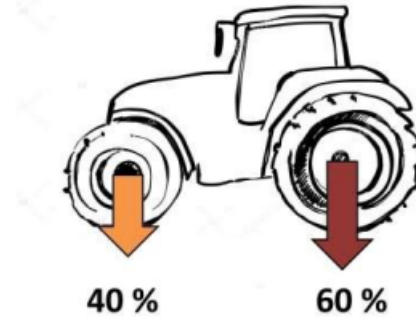


Figure 3.5: Weight distribution at the wheels.

After calculating the inertia of the system, the torque required to be able to carry out the predetermined law of motion can be derived by calculating it as follows:

$$Torque = J_{load} * Acceleration \quad (3.19)$$

Following the definition of the dynamic quantities of the system through the formulas in chapter 3, it is necessary to define the characterizing  $\alpha$  coefficients of each motor. Following the calculation of the individual  $\alpha$  values, for each engine, the  $\alpha$  value can

be compared with the load  $\beta$  value, verifying that it meets the criterion; the result for one engine is shown in the following graph:

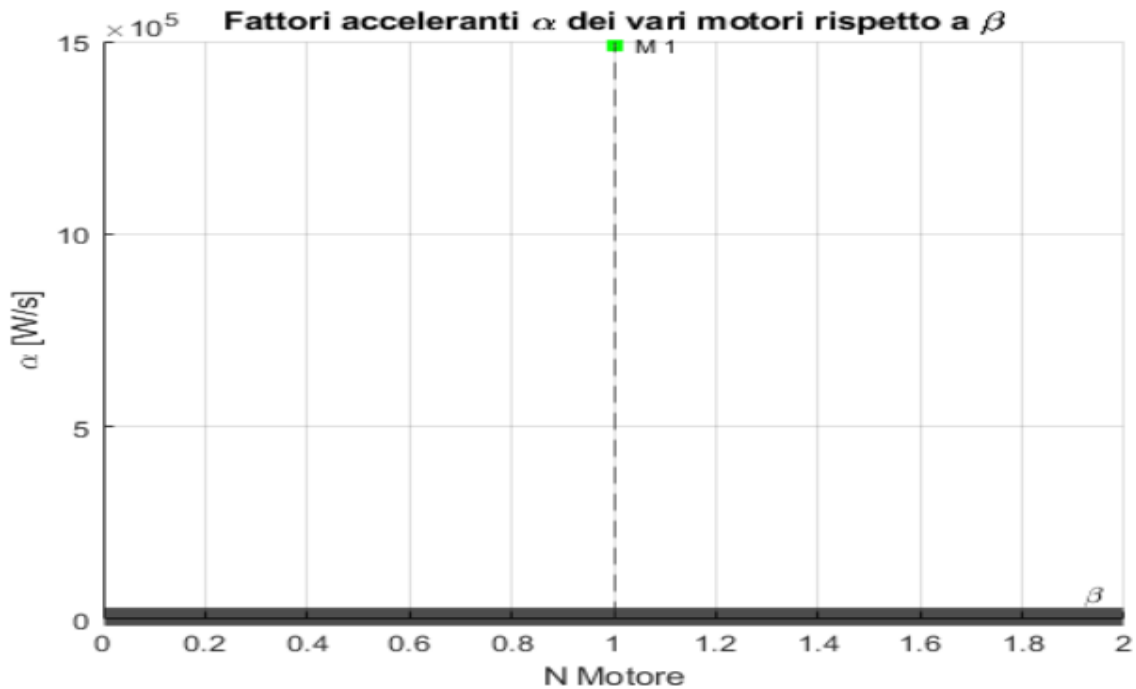


Figure 3.6:  $\alpha$ - $\beta$  graph.

Following the graphic representation of the  $\alpha$ - $\beta$  diagram, below is the code necessary for the realization of this graph with the relative controls:

Through the graph, it can be seen that the accelerating factor, of the engine under consideration, is greater than the load coefficient. By associating the different pre-selected gearboxes, with the  $i$ -th motor, it is possible to check whether the gear ratio meets the limits of the  $\tau_{max}$  and  $\tau_{min}$  criteria. The comparison, between the  $\tau$  associated with the  $i$ -th engine and the  $\tau_{max}$  and  $\tau_{min}$ , is shown within the following graph:

The graph shows engines that do not meet the kinematic  $\tau$ , with blue circles, while engines that meet the requirements are shown with black squares. Next, the operating range of the  $i$ -th motor is represented, and compared with the load profile associated with the  $n$ -th gearbox. In the graphs in the figure 3.10, it is possible to observe how the place of the loads is represented by the green color when the operating field of the  $i$ -th motor, coupled with the  $n$ -th gearbox meets the requirements, on the contrary, if the requirements are not met the place of the loads is red.

In order to make the results more understandable, at the end of the iterations the software produces an Excel file in which are reported the data of the motor and the gearbox that meet the imposed load. To obtain the graph shown in figure 3.8 you must use the code shown in figure 3.9.

```

Motor_Inertia = readmatrix('Motori.xlsx'); %read file txt
Motor_Low_Alpha = 1;
Motor_Inertia=Motor_Inertia(2:end , :);
Total_motor =size(Motor_Inertia,1); %motor count
figure
yline(beta,'-k','\beta ', 'LineWidth',8);
hold on
grid on
title('Coefficient \alpha Vs \beta')
xlabel('N° Motor');
xline(1:Total_motor,'--');
ylabel('\alpha [W/s]');
for Motor_selected = 1:Total_motor %per tutti i motori nel file xlsx
    Nominal_motor_torque(Motor_selected) = Motor_Inertia(Motor_selected,3); %Nominal torque [Nm]
    Nominal_motor_speed(Motor_selected) = Motor_Inertia(Motor_selected,6); %Nominal speed [rpm]
    Motor_Inertia(Motor_selected) = Motor_Inertia(Motor_selected,2); % Motor inertia[kgm^2]
    Alpha_Motor(Motor_selected)= Nominal_motor_torque(Motor_selected)^2/Motor_Inertia(Motor_selected); %[w/s]
    Max_Motor_Speed(Motor_selected)=Motor_Inertia(Motor_selected,10); %[rpm]
    Kinematic_Trasmission_Value(Motor_selected) = Max_speed/Max_Motor_Speed(Motor_selected);
    Eta_limit(Motor_selected) = beta/Alpha_Motor(Motor_selected);
    if Alpha_Motor(Motor_selected) < beta
        LowAlpha_Motor(Motor_Low_Alpha) = Motor_selected;
        Motor_Low_Alpha = Motor_Low_Alpha+1;
        plot(Motor_selected,Alpha_Motor(Motor_selected),'ro', 'LineWidth',2, 'MarkerSize',4)
    else
        plot(Motor_selected,Alpha_Motor(Motor_selected),'gs', 'LineWidth',2, 'MarkerSize',4)
    end
    text(Motor_selected,Alpha_Motor(Motor_selected),[' M ',num2str(Motor_selected)], 'FontSize',8);
end
end

```

Figure 3.7: Matlab code for  $\alpha$ - $\beta$  graph.

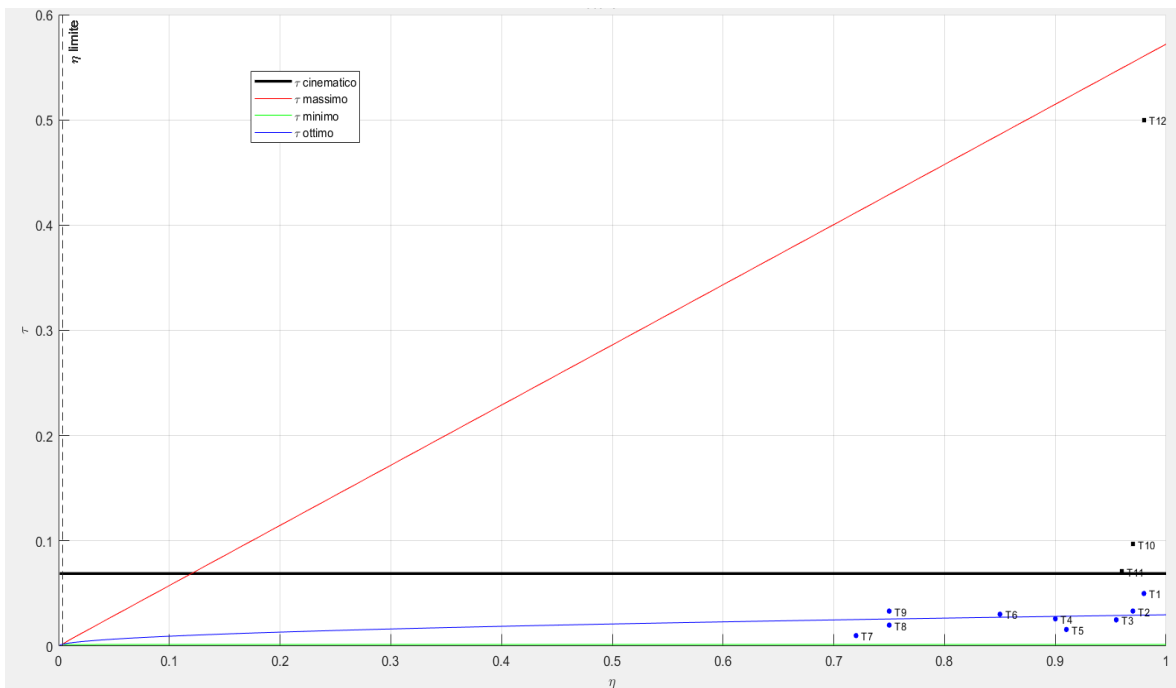


Figure 3.8:  $\tau$  comparison graph, with the same representation shown in figure 3.3.

```

figure
hold on
xlabel('\eta');
ylabel('\tau');
for Reducer_cont = 1:Reducer_number %for each reducer coupled with each motor

[TL_star,Load_Speed2,TL_rms_star,wl_rms,Derivate_wl_rms,beta2] = coppia_trasmissione_r_rad(Average_Speed,Acceleration,TL_2_load,Trasmissione_inertia(Reducer_cont)); %load info
%Tn_step=tau*tl_star/eta
[Torque_motor_temp,Motor_Speed_temp,Motor_torque_temp_rms,Motor_Speed_temp_rms] = coppia_trasmissione_n_rad(Average_Speed,TL_star,Direct_eta(Reducer_cont),Retrograde_Eta(Reducer_number),Tau_reducer(Reducer_cont));
%Tn_step=jm*acc/tau
[Motor_Torque_T(Reducer_cont,:),Motor_Speed_T(Reducer_cont,:),Torque_Motor_rms_T(Reducer_cont),Motor_Speed_rms_T(Reducer_cont)] = coppia_motore_rad(Average_Speed,Acceleration,Torque_motor_temp,Motor_Inertia(Motor_Number),Direct_eta(Reducer_cont)

for i = 1:length(eta)
    Top_Number_1(i) = sqrt(Alpha_Motor(Motor_Number)-beta2/eta(i)+4*Derivate_wl_rms*TL_rms_star/eta(i));
    Top_Number_2(i) = sqrt(Alpha_Motor(Motor_Number)-beta2/eta(i));
    Denominator = 2*TL_rms_star;
    tau_max(Reducer_cont,i) = abs(eta(i)*sqrt(Motor_Inertia(Motor_Number))*(Top_Number_1(i) + Top_Number_2(i))/Denominator);
    tau_min(Reducer_cont,i) = abs(eta(i)*sqrt(Motor_Inertia(Motor_Number))*(Top_Number_1(i) - Top_Number_2(i))/Denominator);
    tau_opt=sqrt((Motor_Inertia(Motor_Number)*Load_Acceleration_rms_M0_T)/TL_rms_star*eta);
    tau_kinematic(Motor_Number,i) = Kinematic_Trasmissione_Value(Motor_Number);
    if eta(i)-Direct_eta(Reducer_cont)<=15
        t_ed(Reducer_cont) = tau_max(Reducer_cont,i);
        t_ed_min(Reducer_cont) = tau_min(Reducer_cont,i);
    end
end
eta_limit = beta2/Alpha_Motor(Motor_Number); %eta limit
eta_limit_array(Motor_Number,Reducer_cont) = eta_limit; %for each motor eta limit
matrix_tau_opt(Reducer_cont,:) = tau_opt; %save tau optimal
index = find(eta==eta_limit);
indice_limit = find(tau_max(Reducer_cont,:)>Kinematic_Trasmissione_Value(Motor_Number));
maximum = max(max(Tau_reducer(Reducer_cont)),tau_max(Reducer_cont,end));
xline(eta_limit, '-', '\eta limite', 'HandleVisibility', 'off');
plot(eta,tau_kinematic(Motor_Number,:), 'k-', 'linewidth', 2)
plot(eta,tau_max(Reducer_cont,:), 'r')
plot(eta,tau_min(Reducer_cont,:), 'g')
plot(eta,matrix_tau_opt(Reducer_cont,:), 'b')
title(['Motor n°: ',num2str(Motor_Number)]);
grid on
end

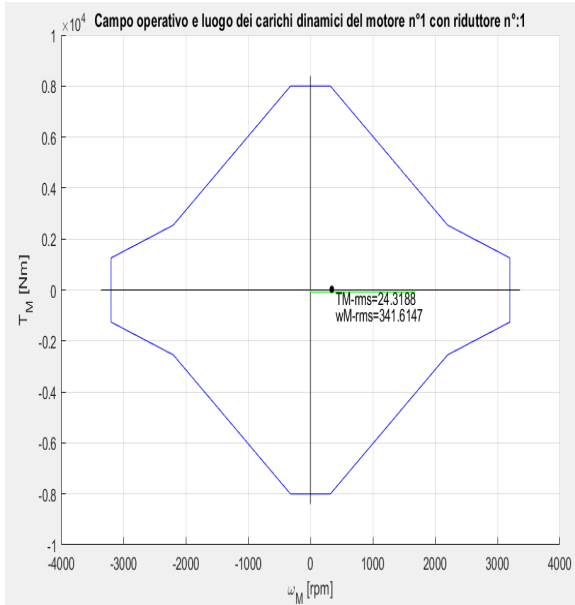
```

Figure 3.9: Matlab code for  $\tau$  comparison graph

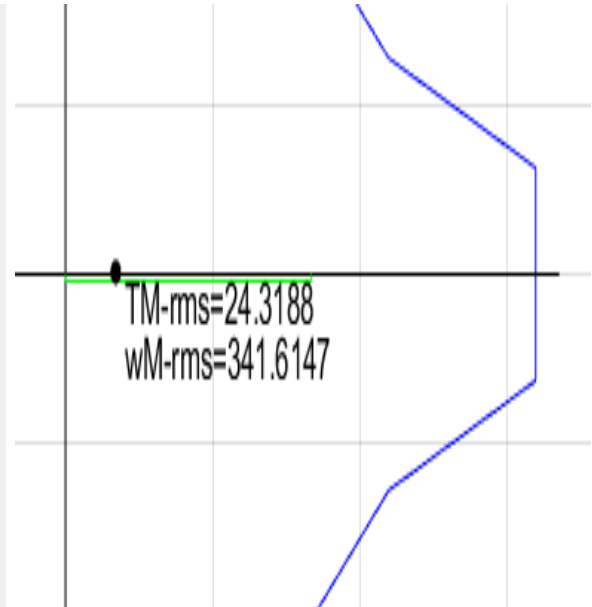
### 3.3.1 High Dynamic Load

The tool can also be used for a load with high dynamics, the case study is a typical maneuver in agriculture, namely shuttling.

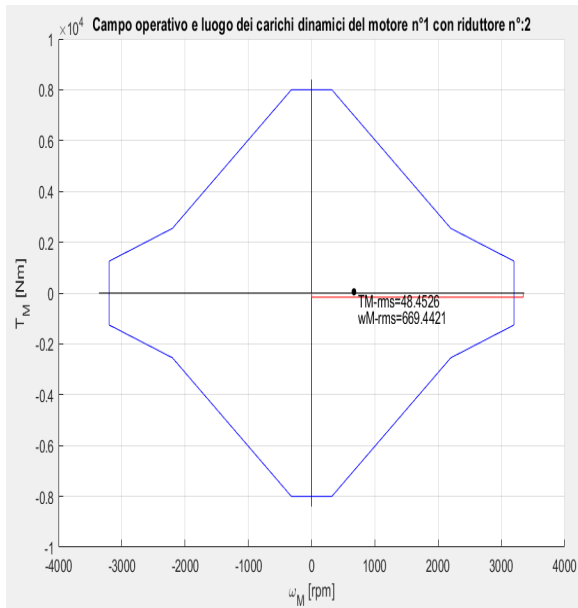
In the figure 3.11 is show the trend of the speed and torque during the maneuver (green line). Using the same procedures, by performing a first phase of filtering of the engines and gearboxes through the  $\alpha$  and  $\beta$  criteria, it is possible to identify the place of the dynamic load and compare it with the operating field of the  $i$ -th engine coupled with the  $n$ -gear. Below is a coupling of an engine and a gearbox that can meet the requirements of shuttling maneuver:



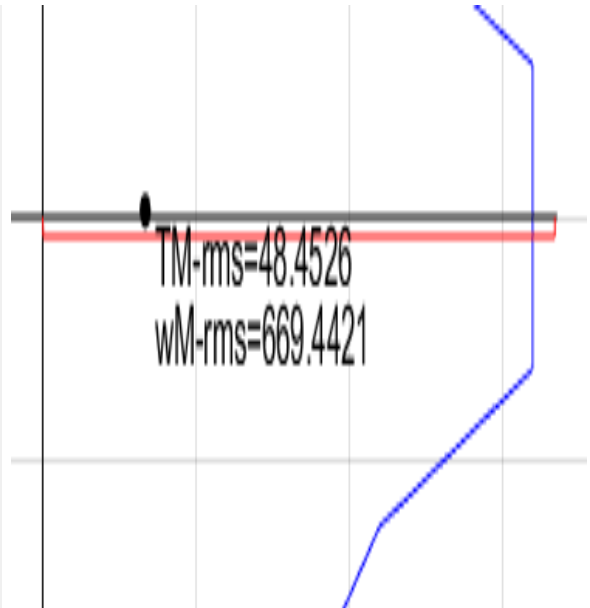
((a))



((b)) Zoom of load range



((c))



((d)) Zoom of load range

Figure 3.10: Place of loads satisfied by the operating range of the motor coupled with the gearbox (a) and not satisfied (c)

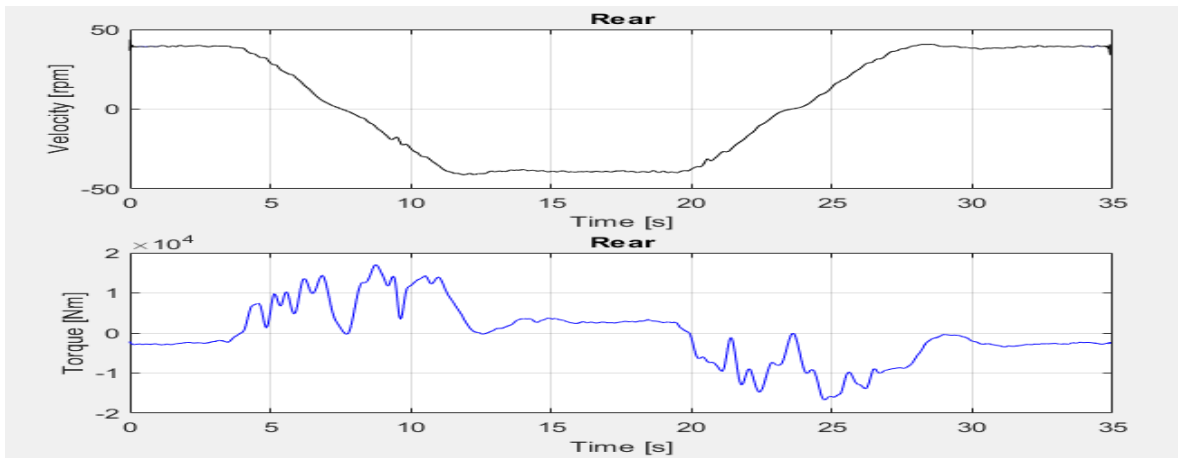


Figure 3.11: Speed and Torque of Shuttling maneuver.

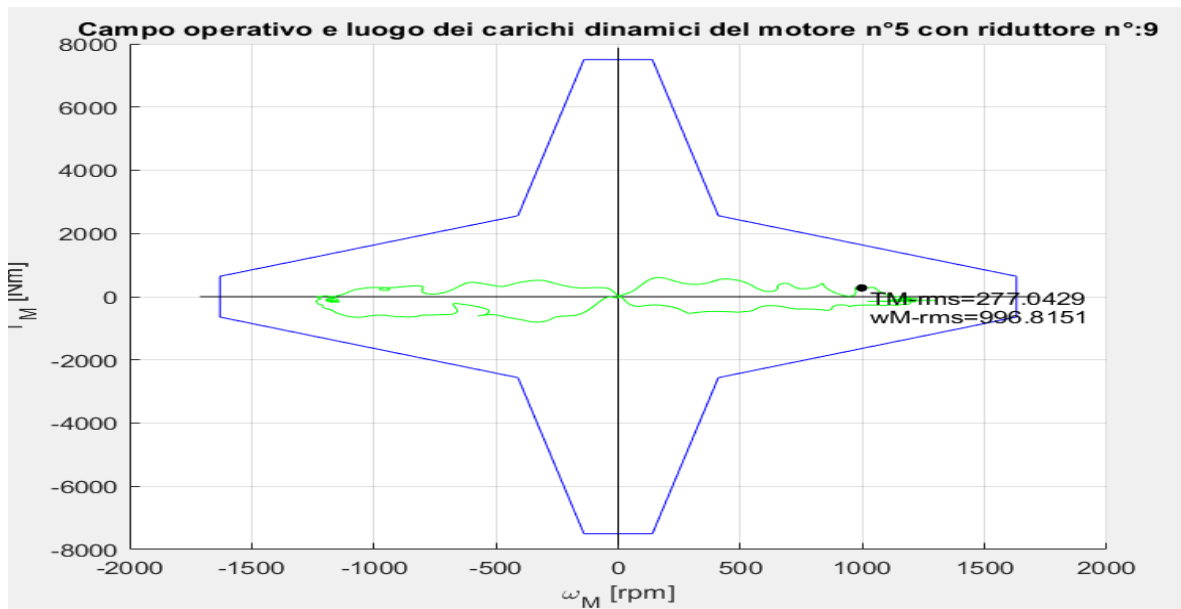


Figure 3.12: Operating range of motor coupling and gearbox applied to a shuttling operation

# Chapter 4

## Model realization

Mechatronics applications are distinguished by controlled motion of mechanical systems coupled to actuators and sensors. Modeling plays a role in understanding how the properties and performance of mechanical components and systems affect the overall mechatronic system design.

Mechanical systems can be conceptualized as rigid and/or elastic bodies that may move relative to one another, depending on how they are interconnected by components such as joints, dampers, and other passive devices. A mechanical component modeled as a point mass or rigid body is readily identified by its velocity, and depending on the number of bodies and complexity of motion there is a need to introduce a coordinate system to formally describe the kinematics.

Through a kinematic analysis, additional (relative) velocities can be identified that indicate the connection with and motion of additional mechanical components such as springs, dampers, and/or actuators.

The interconnection of mechanical components can generally have a dependence on geometry. Indeed, it is dependence of mechanical systems on geometry that complicates analysis in many cases and requires special consideration, especially when handling complex systems.

A preliminary description of a mechanical system should also account for any constraints on the motional states, which may be functions of time or of the states themselves. The dynamics of mechanical systems depends, in many practical cases, on the effect of constraints. Quantifying and accounting for constraints is of paramount importance, especially in multibody dynamics. [15]

In this specific case, the goal is to model a test bench, considering its mechanical and electronic components, in order to verify the dynamic response of the system and be able to compare the results with those obtained from the test bench in order to validate the model developed on Matlab. The generic test bench is shown in the following figure:



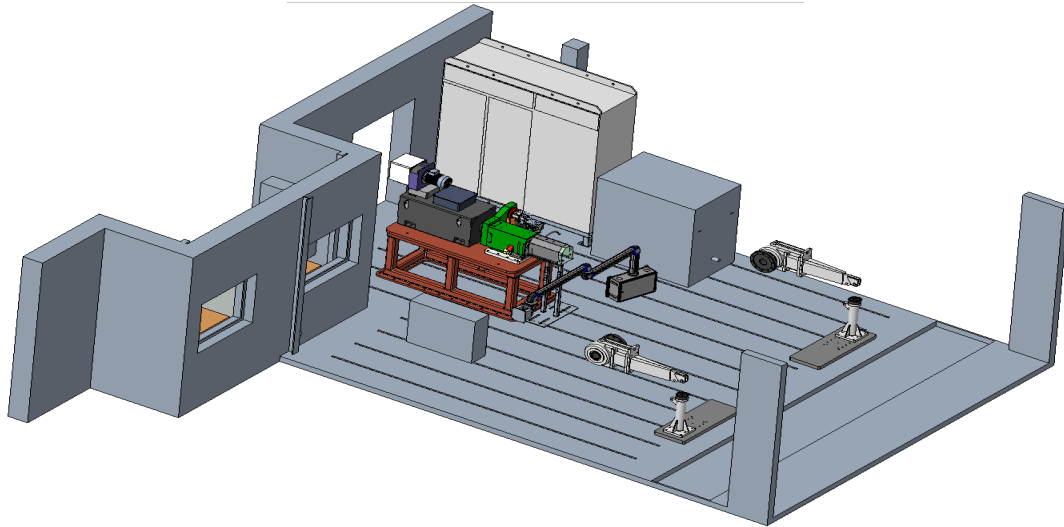


Figure 4.1: Test bench layout.

## 4.1 Simscape setup

Simscape lets you quickly create models of physical systems in the Simulink environment. With Simscape, you can create models of physical components based on physical links that are directly integrated with block diagrams and other modeling paradigms. System models, such as electric motors, bridge rectifiers, hydraulic actuators, and refrigeration systems, can be modeled by assembling the key components in a schematic.

Simscape complementary products provide more complex components and analysis capabilities. Simscape allows you to develop control systems and test system-wide performance. Custom component templates can be created with the MATLAB-based Simscape language, which allows text-based authoring of physical modeling components, domains, and libraries.

It is possible to parameterize models using variables and MATLAB expressions, as well as design control systems for the physical system in Simulink. Simscape supports C code generation to deploy models in other simulation environments, including Hardware-In-the-Loop (HIL) systems.

The design of the test bench model consists of step by step validation of each element introduced. For this reason, the case study is divided into two validations, the first provides a prediction of the performance of the model through a set point speed, while the second provides the prediction of the performance of the model through set point speed and torque.

## 4.2 Performance prediction with speed set point

In this case study the objective is to evaluate the performance of the Simscape model, based on speed control, and compare it with the answers obtained from the test bench. The first phase consists in the realization of the Simscape model containing the electric drive, the control and the mechanical system to operate. Below is the complete Simscape schema:

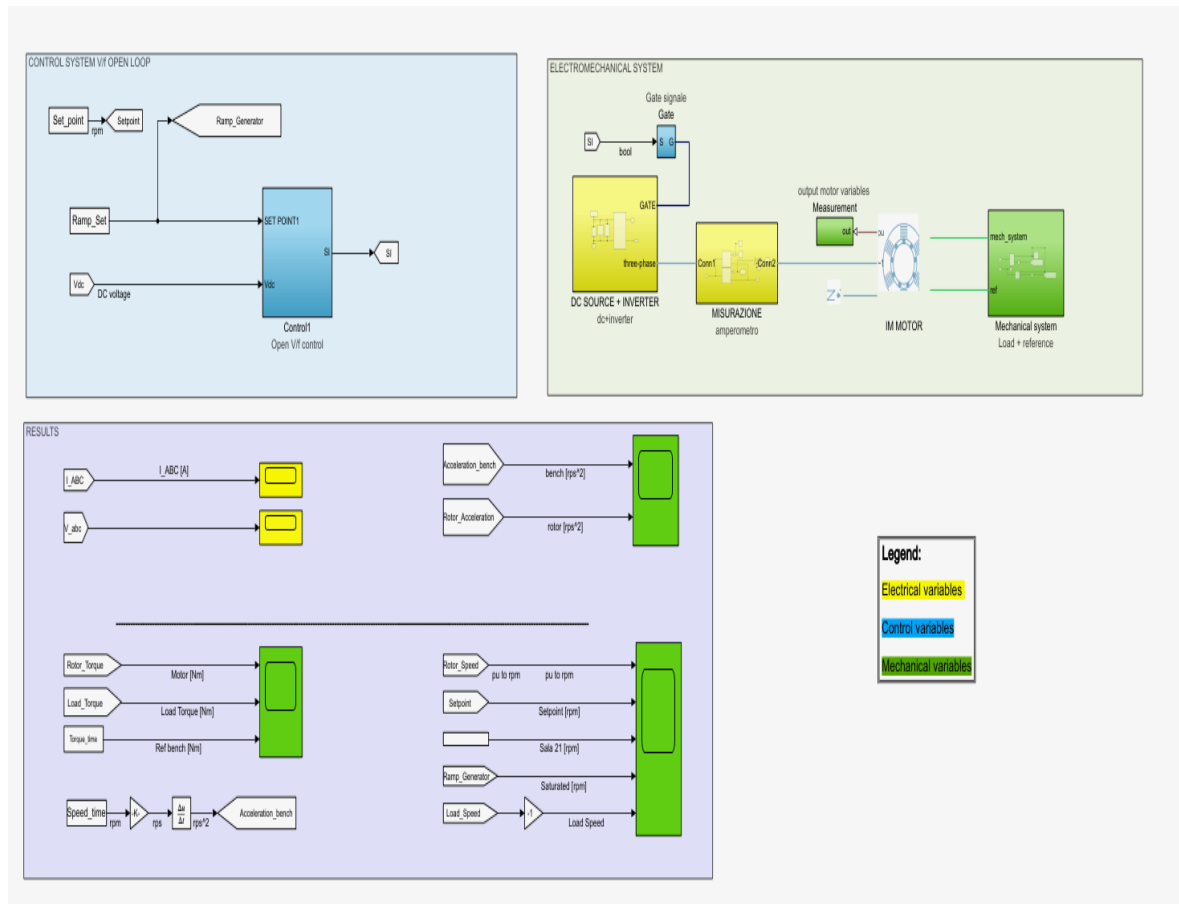


Figure 4.2: Simscape Model for Speed Control

### 4.2.1 Electric drive modeling

Observing the figure 4.2 the yellow blocks correspond to the electrical part of the model, inside these blocks there are sensors for electrical measurements, generators and converters. Inside the first subsystem, renamed "DC SOURCE INVERTER" are inserted the blocks in the figure 4.3

The "DC Voltage Source" block corresponds to an ideal DC voltage generator, this has the task of simulating the DC voltage upstream of the inverter; in parallel a capacitor is positioned to ensure ripple voltages. The "Power Inverter" block represents the inverter

that converts the DC voltage into three-phase alternating voltage, on the line "three - phase" to power the induction motor, the switching of IGBT devices takes place through the control logic v/f through the channel labeled as "GATE".

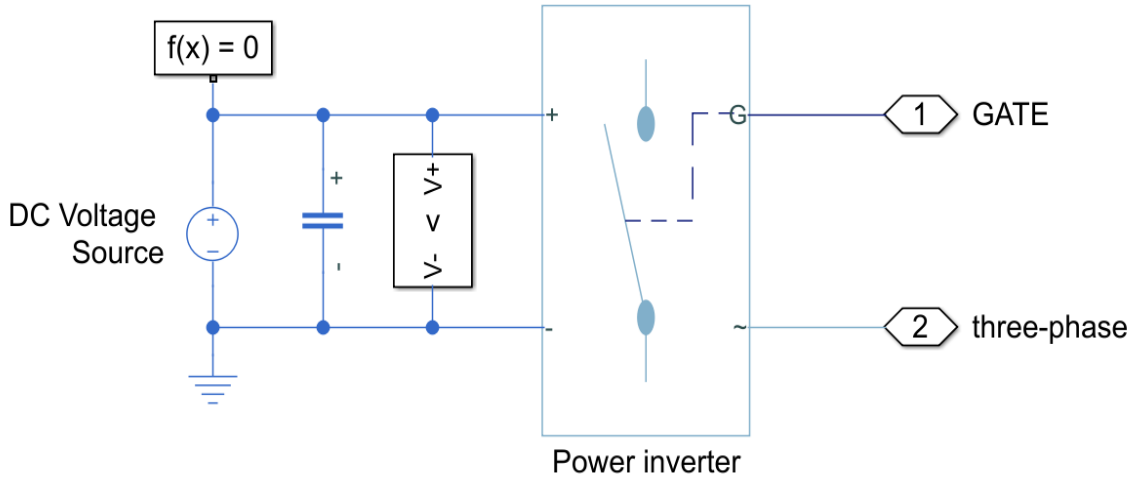


Figure 4.3: DC bus and Inverter

Inside the second subsystem, renamed "MEASUREMENT" is inserted the block in the figure 4.4. The "Current and Voltage Sensor" block has the role of measuring the voltage and the three-phase current, saving the corresponding real-time values within the global variables "V\_abc" and "I\_ABC".

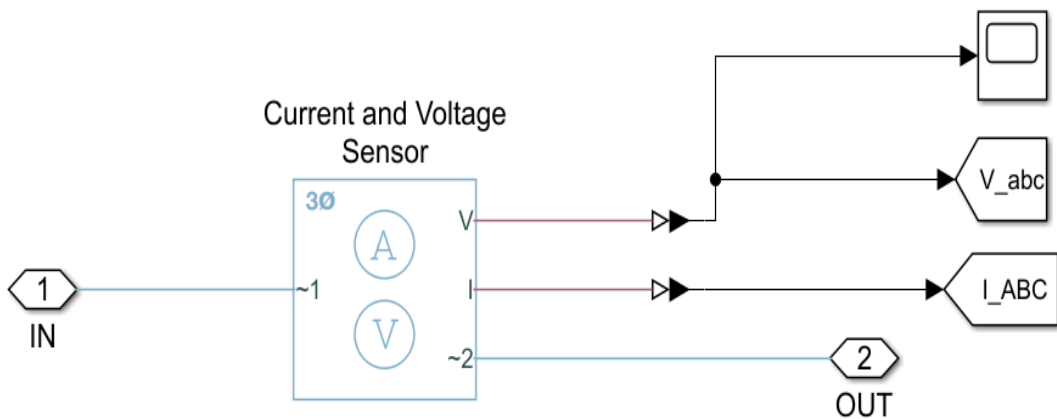


Figure 4.4: Measurement with current and voltage sensor

## 4.2.2 Control modeling

Inside the subsystem "Control", there are the blocks necessary to ensure the control logic Volt-Frequency, in the image 4.5 the blocks are shown: The "Converter" subsystem has the role of converting the reference signal, in rpm, into a frequency signal, then enters the "Induction Machine Scalar Control" block that implements a scalar induction machine, that is V/f or V/Hz.

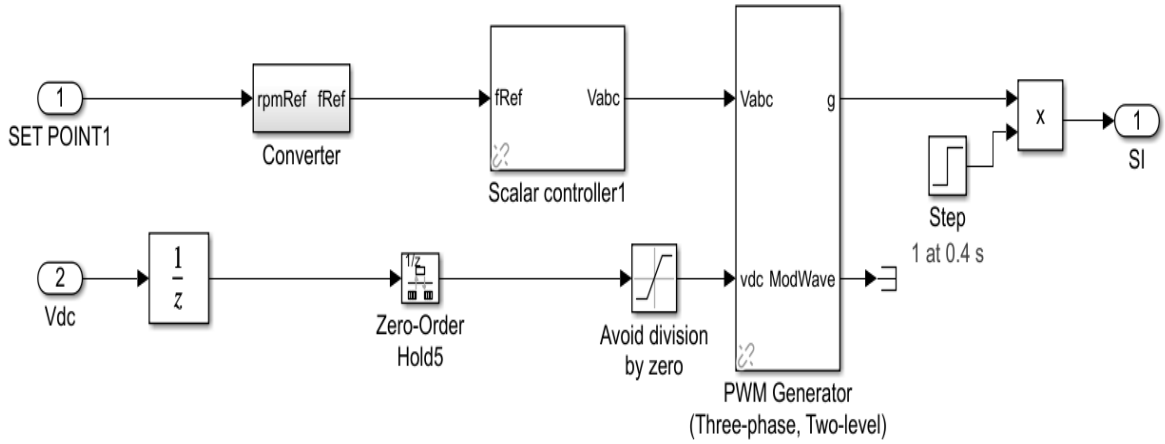


Figure 4.5: V/f control

The Induction Machine Scalar Control block computes the magnitude of the stator voltage based on the reference frequency,  $f_s$  as:

$$V_s = \left( \frac{V_n - V_{min}}{f_n - f_{min}} \right) * f_s \quad (4.1)$$

Where:

- $V_n$  is the rated voltage;
- $V_{min}$  is the minimum voltage;
- $f_n$  is the rated electrical frequency;
- $f_{min}$  is the minimum frequency.

The result obtained by the Induction Machine Scalar Control together with the voltage detection on the DC Bus are used for the generation of the pwm signal, through the block "PWM Generator (Three-phase, Two-level)" to switch the inverter branches. The PWM Generator (Three-phase, Two-level) block controls switching behavior for a three-phase, two-level power converter.

The block:

- Calculates on- and off-gating times based on the block inputs:
  - Three sinusoidal reference voltages, one per phase;
  - A DC-link voltage.
- Uses the gating times to generate six switch-controlling pulses;
- Uses the gating times to generate modulation waveforms.

### 4.2.3 Mechanical System modeling

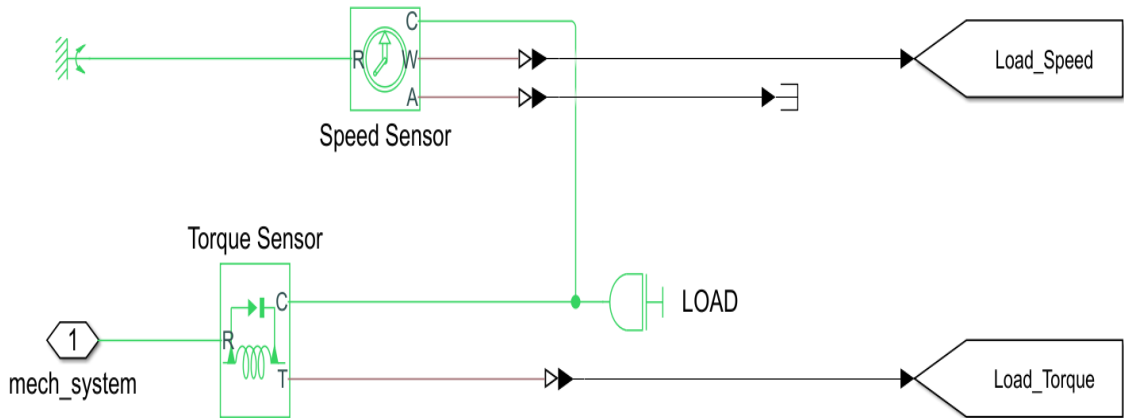


Figure 4.6: Mechanical system

Within the subsystem representing the mechanical system, a direct connection between the induction motor and a known inertia load can be observed. The mechanical system has a torque measurement, acting on the load, and its rotational speed ; the signals are saved, respectively in the *Load\_Speed* and *Torque\_Speed* global variables, for analysis.

The Ideal Torque Sensor block represents a device that converts a variable passing through the sensor into a control signal proportional to the torque. The sensor is ideal because it does not account for inertia, friction, delays, energy consumption, and so on.

The Ideal Rotational Motion Sensor block represents an ideal mechanical rotational motion sensor, that is, a device that converts an across variable measured between two mechanical rotational nodes into a control signal proportional to acceleration, angular velocity, or position (angle); the sensor is ideal since it does not account for inertia, friction, delays, energy consumption, and so on.

The mechanical subsystem is shown in the figure 4.6.

## 4.2.4 Induction Motor modeling

The induction motor with squirrel cage, is modeled through the equivalent circuit present in the data sheet of the engine supplier. The motor input corresponds to the three-phase voltage output from the inverter, while the output corresponds to the torque and the speed delivered by the motor and applied to the mechanical subsystem. The detection of motor quantities is possible through the use of the subsystem "Measurement", of which the components are reported in the figure below:

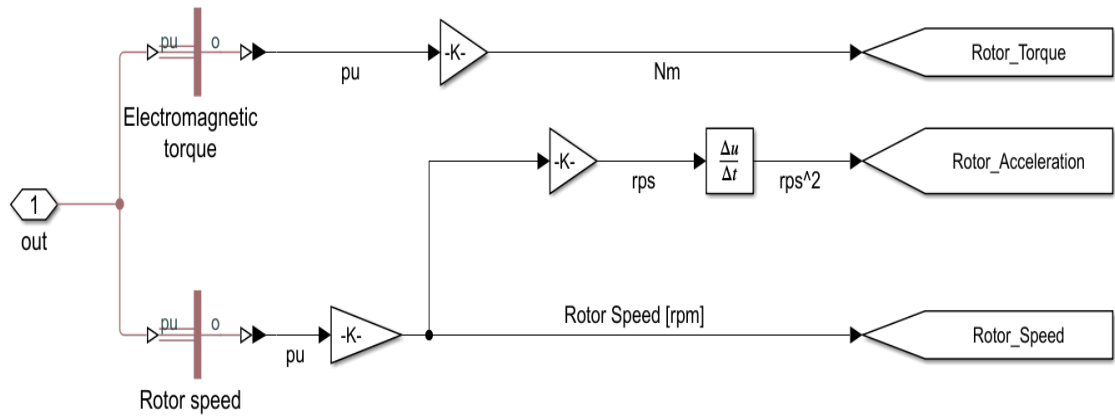


Figure 4.7: Measurement motor signal

Through the "Induction Machine Measurement" block it is possible to describe the specified motor size, expressed in p.u (per-unit). The per-unit system is widely used in the power system industry to express values of voltages, currents, powers, and impedances of various power equipment. It is mainly used for transformers and AC machines.

For a given quantity (voltage, current, power, impedance, torque, etc.) the per-unit value is the value related to a base quantity.

$$Base\_value\_in\_p.u = \frac{Quantity\_expressed\_in\_SI\_units}{Base\_value} \quad (4.2)$$

Generally the following two base values are chosen:

- The base power = nominal power of the equipment;
- The base voltage = nominal voltage of the equipment.

All other base quantities are derived from these two base quantities. Once the base power and the base voltage are chosen, the base current and the base impedance are determined by the natural laws of electrical circuits.

### 4.3 Simscape model validation

In order to make the correlation between the Simscape model and the test bench, it is necessary to perform a test, having known all the operating parameters, and compare the test bench results with the results obtained by the Simscape model.

To perform a correct speed test it is necessary to provide a set point of speed, in this case it corresponds to 1500 rpm. The command provided to the test bench turns out to be a step command, internally the test bench driver, transforms the step command into a ramp command; this step is used to reduce sudden acceleration and deterioration of mechanical components.

From the practical point of view, the driver receives the step speed set point input and setting the ramp rise time, equal to 12 seconds, creates the signal between 0 rpm and 1500 rpm. The following graph shows the step command and the ramp command realized by the driver:

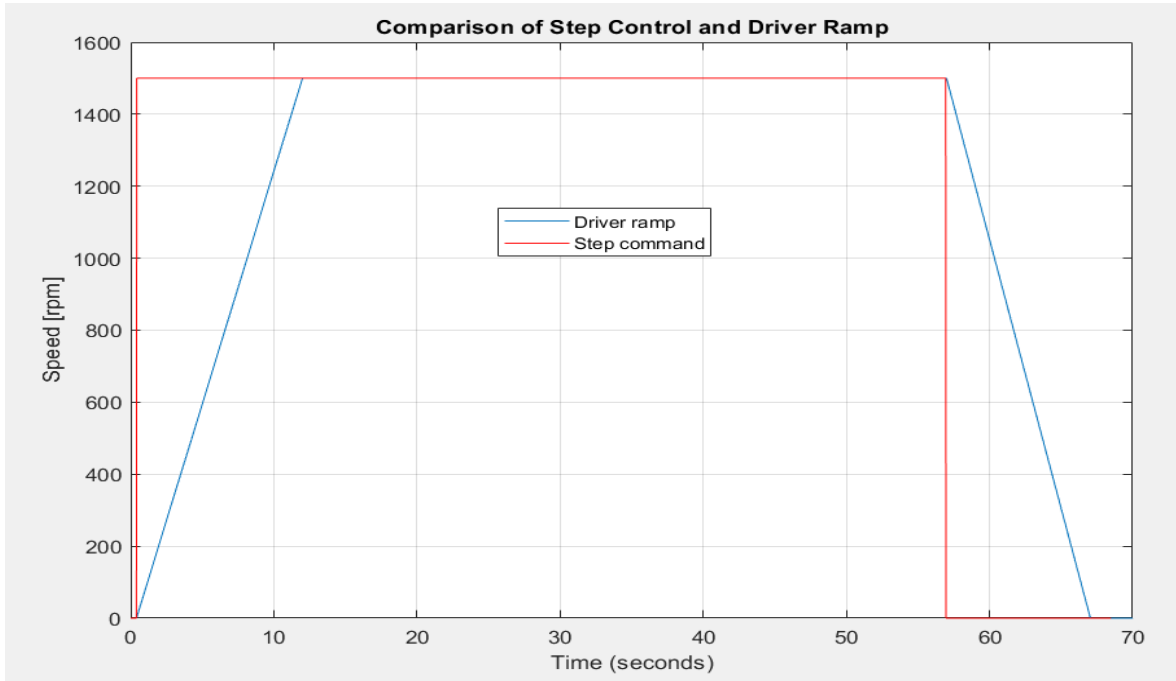


Figure 4.8: Comparison between ideal and real command

By introducing in figure 4.8 the response obtained from the test bench and the response obtained from the Simscape model, it is possible to observe the discrepancies between the two results. Looking at the graph in the figure 4.9 it is possible to observe as the course of the answer obtained from the model Simscape (black line) turns out to be equal to the course of the answer obtained from the test bench (blue line).

Observing ,in particular way the course of the obtained answer of the Simscape model, during the phase of rise, it possible to find an oscillation of the signal; this difference compared to the signal obtained from the test bench is attributable to the internal

control of the driver to avoid critical frequencies. In certain situations, certain engine speeds give rise to mechanical resonances that are annoying to the user and/or harmful to the engine and system.

The test cell driver's GT5000 provides a critical frequency jump function, as shown in the figure 4.10, which reverts the speed reference upstream of the ramp to stand firmly in a band around them. With the two parameters Skip Speed Band 1 and Skip Speed Band 2 you can set the respective half-bands relative to the amplitude of the jump. For example, if the Skip Speed parameter is set to 40 Hz and the Skip Speed Band 1 parameter is set to 5 Hz, the ramp mount reference is set to an increasing input speed reference. Once at 35 Hz, this value shall be maintained until the input reference remains below 45 Hz; when the input reference reaches 45 Hz, the ramp upstream reference shall lead directly to this value.

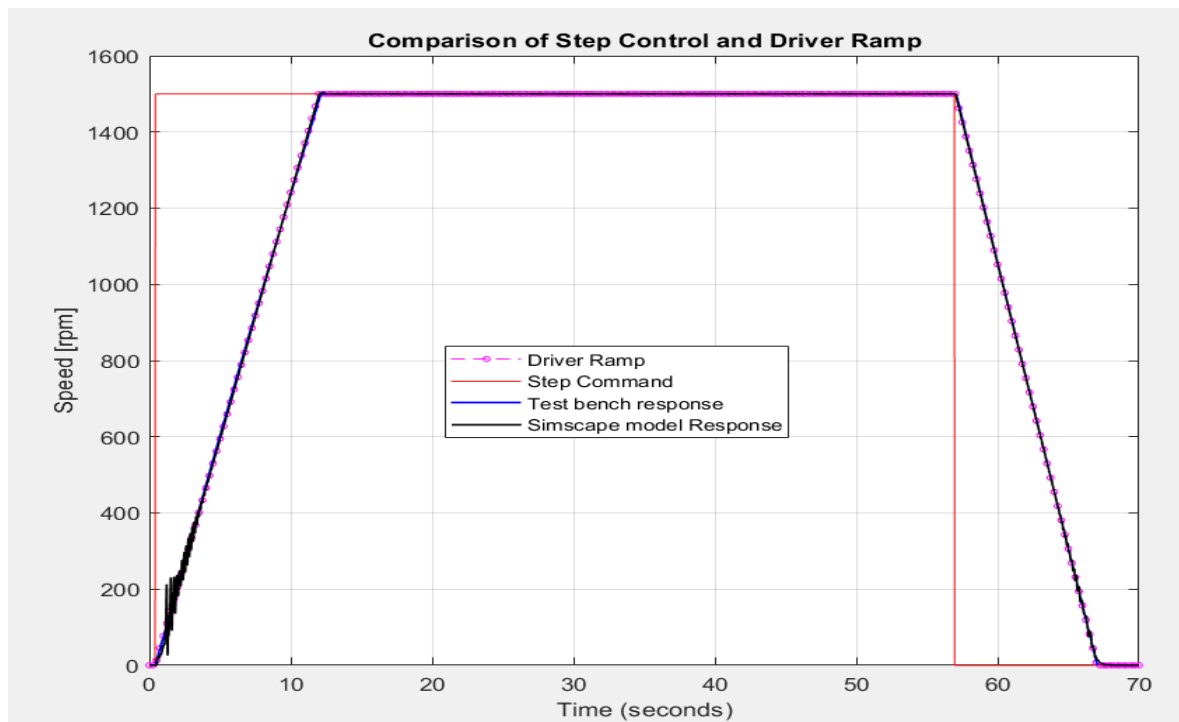


Figure 4.9: Comparison between test bench response and Simscape model response

In case of decreasing reference the situation is symmetrical. The jump of the critical frequency occurs upstream of the ramp, so the downstream reference ramp still passes for 40 Hz but there remains stable.

In case of overlap of the bands are considered the greater of the upper limits and the smaller of the lower ones. The figure 4.10 shows graphically the jump of critical frequencies developed by the test bench driver.



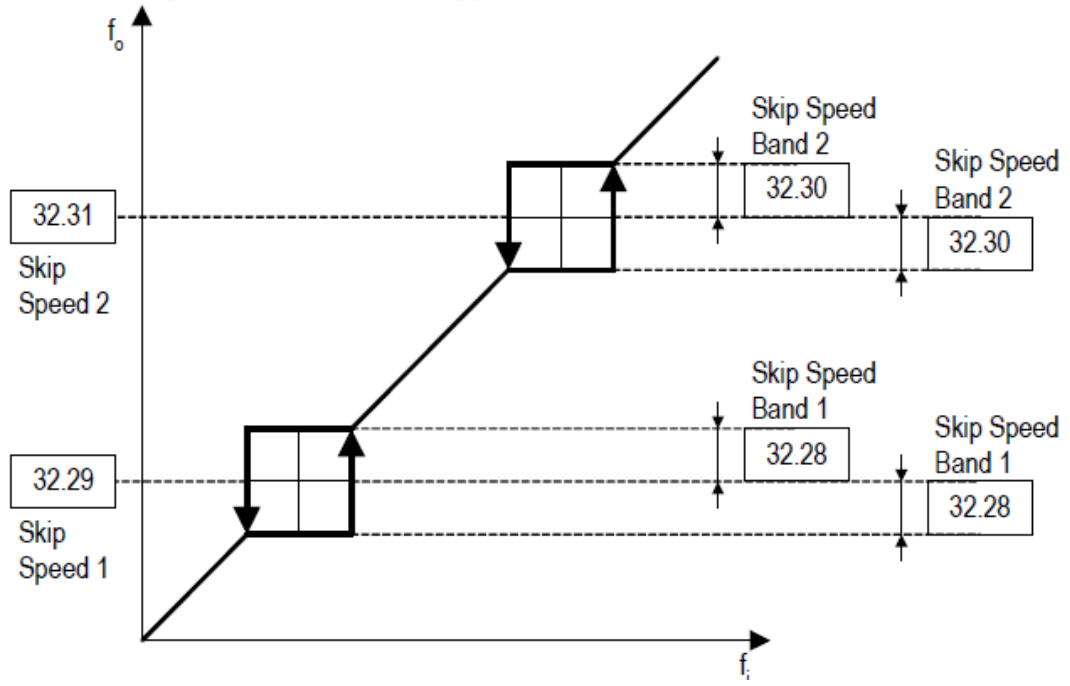


Figure 4.10: GT5000 Driver Critical Frequency Jump Function [17].

### 4.3.1 Error evaluation, Mean Squared Error

The mean square error (MSE) of an estimator  $\hat{\theta}$  of a parameter  $\theta$  is the function of  $\theta$  defined by  $E(\hat{\theta} - \theta)^2$ , and this is denoted as  $MSE_{\hat{\theta}}$ . This is also called the risk function of an estimator, with  $(\hat{\theta} - \theta)^2$  called the quadratic loss function. The expectation is with respect to the random variables  $X_1, \dots, X_n$  since they are the only random components in the expression. Notice that the MSE measures the average squared difference between the estimator  $\hat{\theta}$  and the parameter  $\theta$ , a somewhat reasonable measure of performance for an estimator. In general, any increasing function of the absolute distance  $|\hat{\theta} - \theta|$  would serve to measure the goodness of an estimator mean absolute error,  $E|\hat{\theta} - \theta|$ , is a reasonable alternative. [16]

Following the theoretical definition for the calculation of the error, it is possible to calculate the error between the signal obtained from the test bench and the signal obtained from the Simscape model.

Through the following formula it's possible to define the RMSE value:

$$RMSE = \sqrt{\sum_{i=1}^N (Speed\_Bench_i - Speed\_Model_i)^2} \quad (4.3)$$

The found value of RMSE is equal to 6.9907 rpm. In order to better explain the error between the test bench reference and the model result, the RMSE normalized value can be used, below is the formula used to obtain this value:

$$NRMSE = \frac{RMSE}{\bar{y}} \quad (4.4)$$

Using the RMSE value mentioned above and using the mean speed value obtained from the test bench, 1223.3 rpm, the value of NRMSE is equal to 0.0057.

## Chapter 5

# Performance prediction with speed set point and Torque Control

In the present case study, a different test bench is analysed than the one considered in chapter 4.2. In the test bench under consideration, a torque control is performed in order to make a correlation also with regard to the torque.

The test developed on the test bench consists in maintaining the rotations of the engine at a given value and performing a braking through a pneumatic brake, this in order to evaluate the performance of the brake and engine torques.

The coupling between brake-motor and load consists of a gearbox with double conical pinion, motor and brake are coupled with a unit transmission ratio while, the engine and the load are coupled with a transmission ratio equal to 11.5.

Figure 5.1 show a reducer with double pinion.

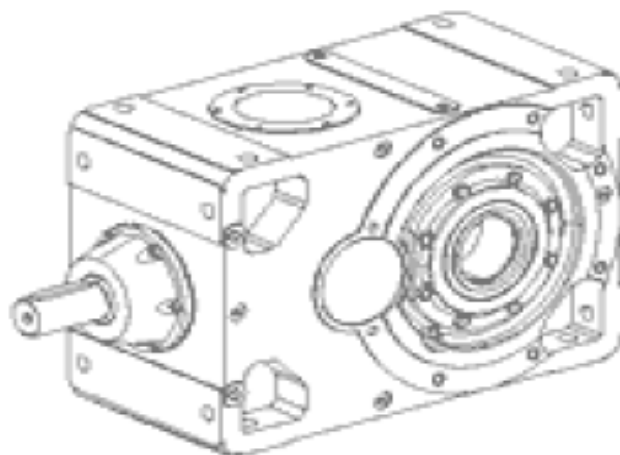


Figure 5.1: Reducer with double pinion [18].

## 5.1 Simscape model

In this case study the objective is to evaluate the performance of the Simscape model, based on Torque control, and compare it with the answers obtained from the test bench. The first phase consists in the realization of the Simscape model containing the electric drive, the control and the mechanical system to operate. Figure 5.2 shown Simscape model of the test bench.

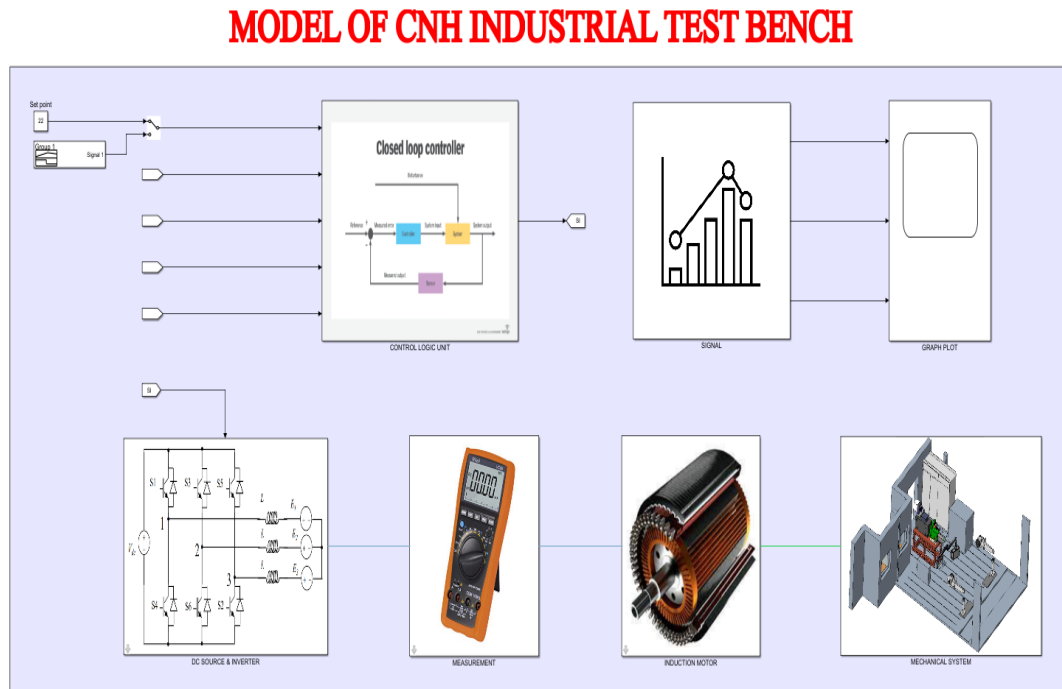


Figure 5.2: Simscape model for Torque Control

### 5.1.1 Electric drive

As for the part of the electricity grid is not changed compared to the previous case, this is because the electricity network to which the test benches are connected is the same for all. The inverter, used for the conversion of the DC voltage to three-phase alternating voltage, is the same, this is because it was chosen a test bench that has the same driver configuration, the adjustment of the latter will have a diversified logic since the specific case has a torque control and not a speed control.

The figure 5.3 shows the subsystem of the electric drive. Inside the second subsystem, renamed "MEASUREMENT" is inserted the blocks in the figure 4.4 . The "Current and Voltage Sensor" block has the role of measuring the voltage and the three-phase current, saving the corresponding real-time values within the global variables "V abc" and "I ABC".



$$Torque\_base = \frac{Nominal\_power * Pole\_Pairs}{Base\_Speed} \quad (5.1)$$

The base speed is calculated as follows:

$$Base\_Speed = 2 * \pi * Natural\_frequency \quad (5.2)$$

After the creation of the torque command the control is actually realized, the subsystem used is shown in the following figure:

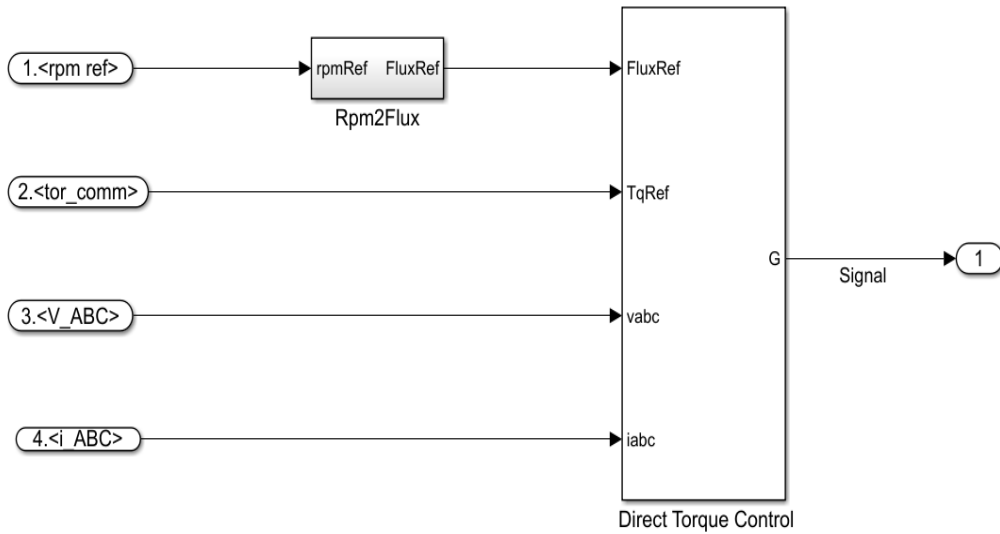


Figure 5.5: Direct Torque Control subsystem

Within the subsystem "Rpm2Flux" the speed set point is converted into a flow signal for the induction motor. The "Direct Torque Control" block is the actual torque control, consisting of four input variables:

- FluxRef, is reference stator flux;
- TrqRef, is reference torque;
- vabc, is stator phase voltages;
- iabc, is stator phase currents.

The block output corresponds to a signal to control the inverter branches. The Induction Machine Direct Torque Control block implements an induction machine direct torque control (DTC) structure.

To estimate the torque and flux, the Induction Machine Direct Torque Control block discretizes the machine voltage equations in the stationary  $\alpha - \beta$  reference frame, using

the Clark transform, which allows a time domain of a three-phase two-component system to be switched to an orthogonal stationary frame, using the backward Euler method.

### 5.1.3 Mechanical System

The following figure shows the mechanical subsystem:

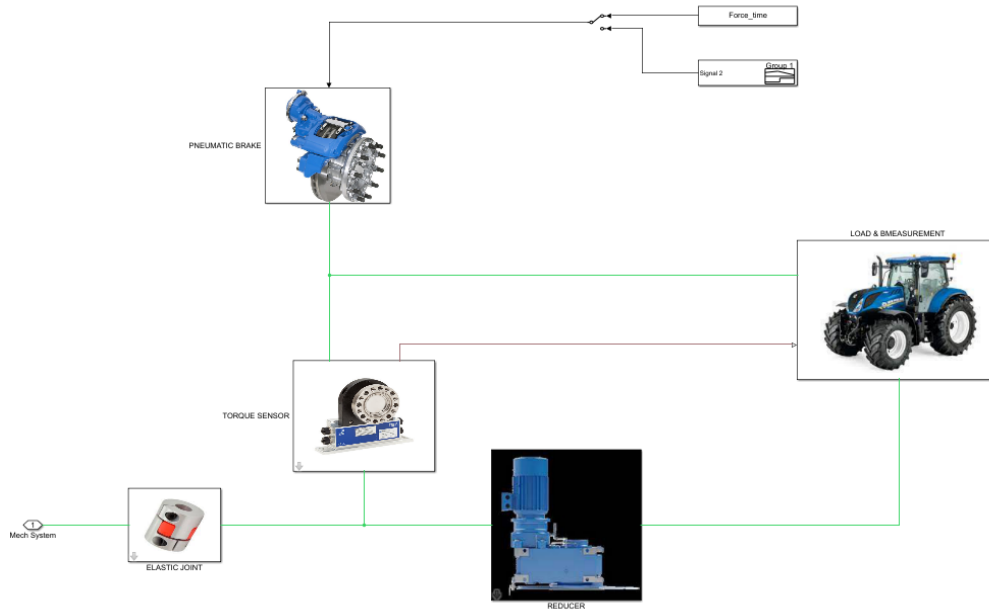


Figure 5.6: Mechanical subsystem

With respect to the case reported in the chapter 4.2.3 the stiffness, damping and inertia of each element are kept in consideration. The shaft on which the motor and the air brake are fitted has a double lamellar joint with its own stiffness and damping coefficient. The subsystem that defines the lamellar joint is shown in the following figure:

The subsystem for simulating brake operation is shown in the following image:

The operation of the "Loaded-Contact Rotational Friction" block is based on the application of an external force, which is the output of the pneumatic system. The Loaded-Contact Rotational Friction block simulates friction between two rotating surfaces loaded with a normal force.

The block is implemented as a structural component based on the Fundamental Friction Clutch block. From the locked state, the two surfaces unlock if the transmitted torque exceeds the static friction, as defined by the static coefficient of friction and current normal force. Torque is transmitted for normal forces larger than the Threshold force parameter.

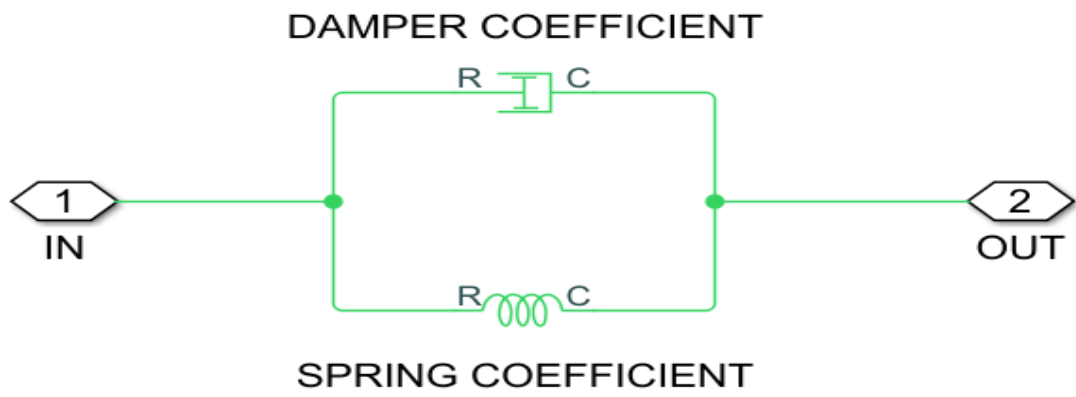


Figure 5.7: Lamellar joint subsystem

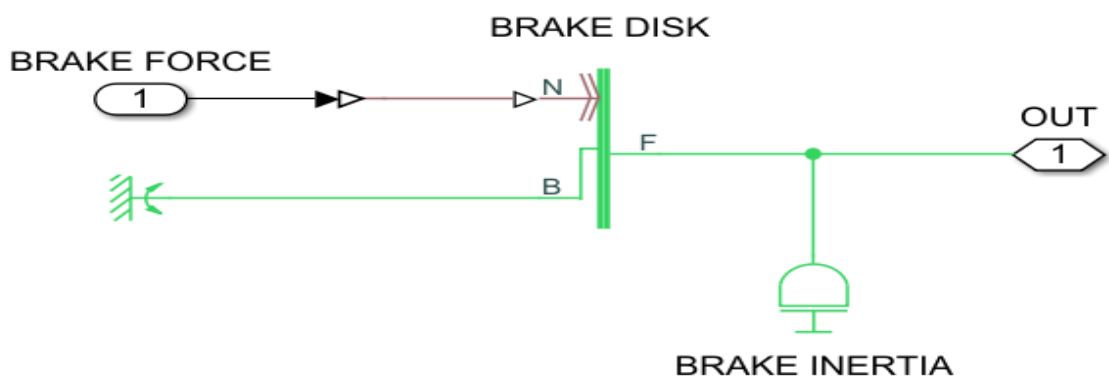


Figure 5.8: Pneumatic brake subsystem



The block simulates friction between two rotating surfaces loaded with a normal force. When the two rotating surfaces are not locked, the transmitted torque is determined with the following equations:

$$\begin{cases} \tau = N * \mu * r_{eff} * sign(\omega) + \tau_{visc} \\ r_{eff} = \frac{2}{3} * \frac{r_o^3 - r_i^3}{r_o^2 - r_i^2} \\ \tau_{visc} = \mu_{visc} * \omega \end{cases} \quad (5.3)$$

Where:

- $\tau$  is the transmitted torque;
- $\mu$  is the friction coefficient;
- $N$  is the normal force;
- $r_{eff}$  is the effective radius;
- $r_o$  is the surface outside radius;
- $r_i$  is the surface inside radius;
- $\omega$  is the relative angular velocity;
- $\tau_{visc}$  is the viscous drag torque;
- $\mu_{visc}$  is the viscous drag torque coefficient.

Torque and speed measurement is achieved through the use of "Ideal Torque Sensor" and "Ideal Rotational Motion Sensor" sensors.

#### 5.1.4 Induction Motor

The induction motor with squirrel cage, is modeled through the equivalent circuit present in the data sheet of the engine supplier. The motor input corresponds to the three-phase voltage output from the inverter, while the output corresponds to the torque and the speed delivered by the motor and applied to the mechanical subsystem. Through the "Induction Machine Measurement" block it is possible to measure the specified motor size, expressed in p.u (per-unit).

The per-unit system is widely used in the power system industry to express values of voltages, currents, powers, and impedances of various power equipment.

## 5.2 Validation and output data of Simscape model with step torque brake

In order to make the correlation between the Simscape model and the test bench, it is necessary to perform a test, having known all the operating parameters, and compare the test bench results with the results obtained by the Simscape model. In this case, the test is performed by maintaining a constant rotation speed of 22 rpm, applying a braking torque to the system, through the pneumatic brake, observing the course of the engine speed and comparing it with the speed measured in the physical test bench test.

The braking torque, in the four braking turns, increases as the test time increases; an initial braking percentage of 10% of the braking capacity of the brake is applied up to 40%. The figure 5.9 shows the braking torque applied to the mechanical system. Keeping the speed constant and applying the braking torque, it is possible to observe the trend of the engine torque. The induction motor, through the control logic, delivers the necessary torque to be able to maintain the required speed, in the figure 5.10 is represented the trend of the engine torque as a function of the applied braking torque. Observing the trend of the engine torque, with regard to the brakes with 30% and 40% brake capacity, it is possible to find the same torque value delivered by the engine. This phenomenon is attributable to the limit of the engine, as it is not able to develop a torque that exceeds the braking torque applied by the brake.

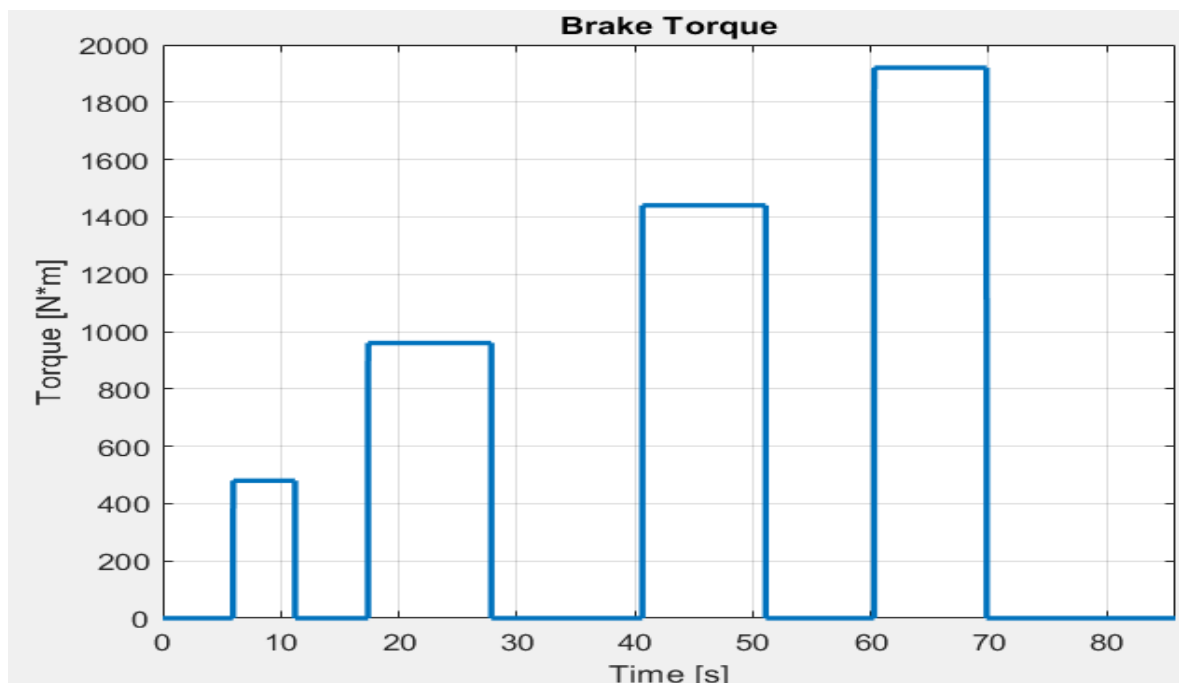


Figure 5.9: Brake torque applied to mechanical system

Because of the mechanical limitation of the engine, observing the figure 5.10 it is

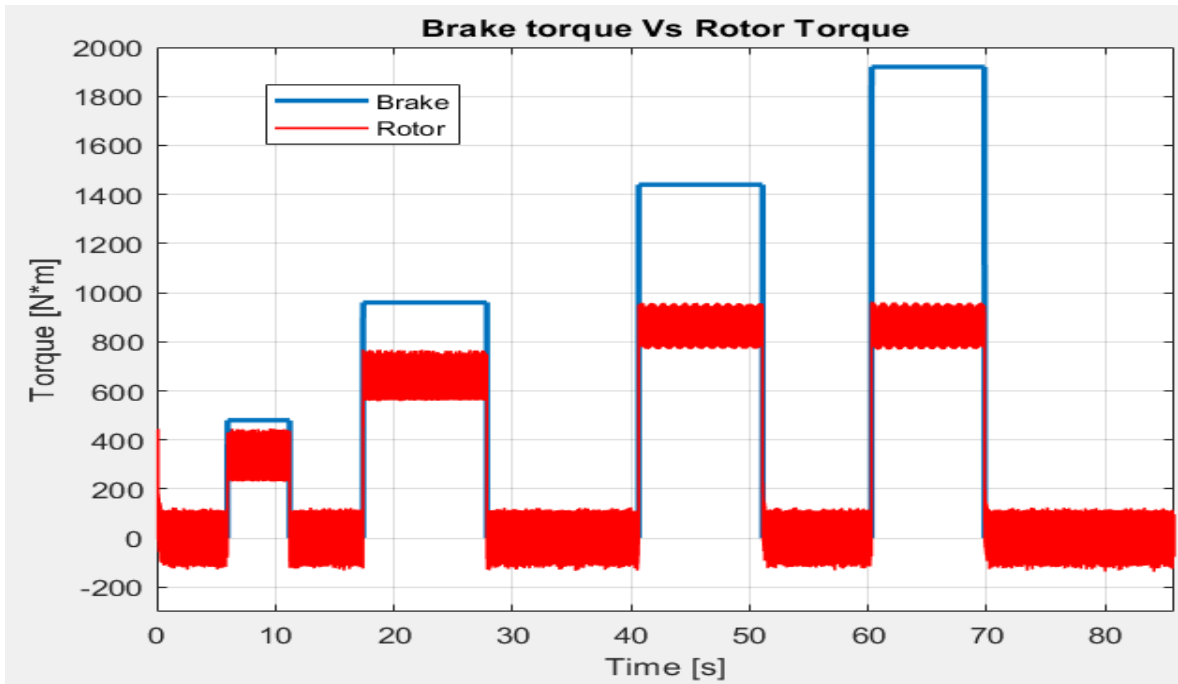


Figure 5.10: Motor torque output as a function of braking torque

possible to predict the speed trend; in fact, since the braking torque is greater than the driving torque, the mechanical system undergoes a reduction in speed. The figure below shows the set point of speed, equal to 22 rpm, and the trend of the engine speed:

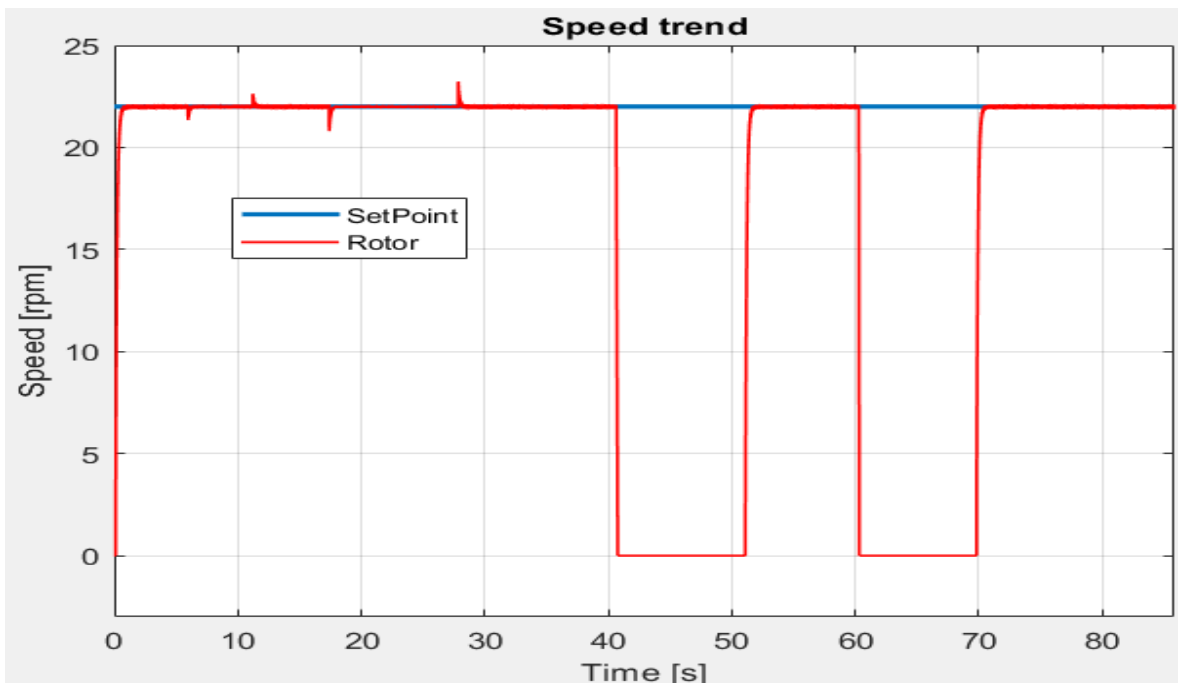


Figure 5.11: Rotor speed versus speed set point

Comparing the graphs in the figures 5.10 and 5.11 , it is possible to observe how at

the braking steps, of 30% and 40%, the rotor slows down until it stops, this is because the braking torque is greater than the rotor torque and therefore the engine fails to meet the speed set point.

As for the braking steps of 10% and 20% the engine manages to counteract the braking torque by detecting, in the speed graph, the oscillations related to the application and the instantaneous release of the braking torque. The result obtained by the Simscape model is compared with the values recorded by the test bench, for this reason the speed graph shows the real trend of the measured speed, in figure 5.12 the comparison graph is shown.

The speed trend in the real case is not instantaneous, this is due to the fact that the brake pressure control is not instantaneous. As for the oscillations in the deceleration phase, they are related to the presence of wear in the brake discs and to the presence of a valve controller used for brake regulation.

The valve controller has a threshold value by default, this results in an intermittent and not continuous adjustment, by setting the minimum value of threshold the response of the valve is more immediate and therefore a more rigid control on the pressure.

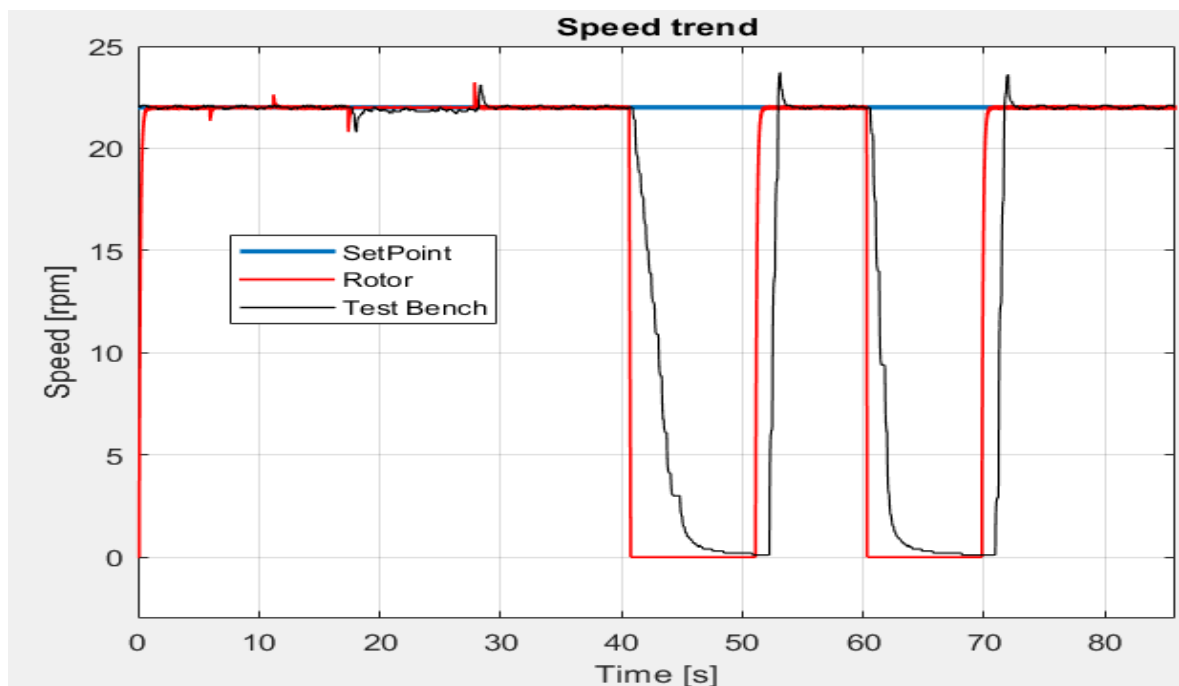


Figure 5.12: Comparison between real performance and Simscape model

### 5.2.1 Error evaluation, Mean Squared Error

It is possible to calculate the error between the signal obtained from the test bench and the signal obtained from the Simscape model; through the formula 4.3 it's possible to define the RMSE value. The value of RMSE is equal to 5.2872 rpm. In order to better

explain the error between the test bench reference and the model result, the RMSE normalized value can be use, using the formula 4.4. Using the RMSE value mentioned above and using the mean speed value obtained from the test bench, 17.0654 rpm, the value of NRMSE is equal to 0.3098.

### 5.3 Analysis and considerations on pneumatic brake

The trend of the brake speed, shown in figure 5.12, leads to an inconsistency between the Simscape model and the results obtained from the test cell; in order to understand what the oscillations, as reported in figure 5.12 (black line), depend on it would be necessary to detect the trend of useful pressure of the brake.

In order to understand the root cause of this behavior a tuning of the valve control is developed in order to analyse a possible change in the brake speed. The valve controller consists of a proportional control characterized by a threshold value beyond which applies the regulation, the control block on the valve data sheet and reported in the figure 5.13.

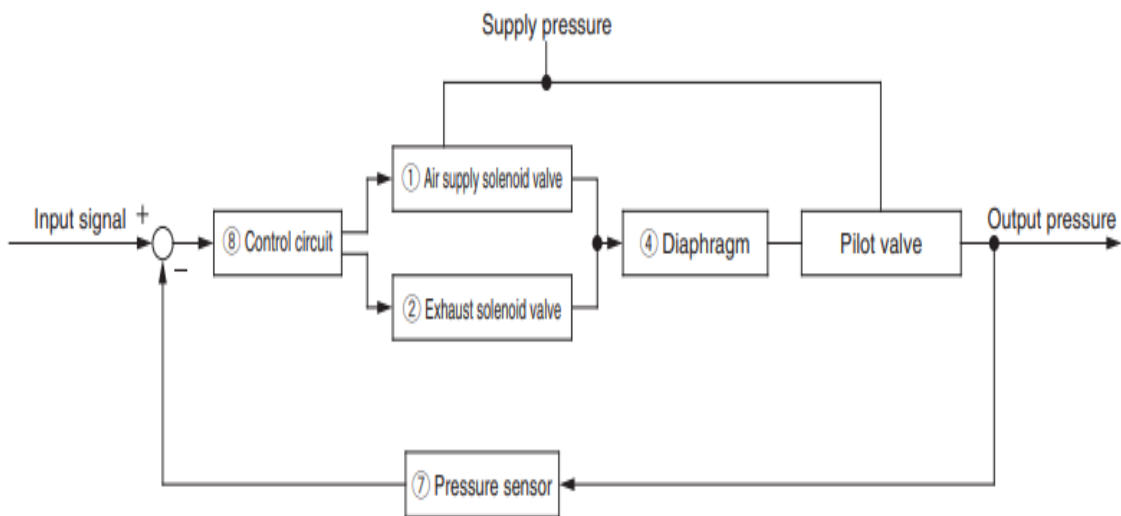


Figure 5.13: Valve's control scheme [19].

By default the proportional parameter and the sensitivity parameter are set to intermediate values, in order to evaluate the behavior of the valve four tests are performed by modifying these parameters.

The tests are performed using the same parameters set in chapter 5.2 , the results obtained experimentally are shown in the graph in figure 5.14.

Observing the speed of the brake you can see how increasing the maximum proportional value and sensitivity (orange line) the brake response is faster, on the contrary

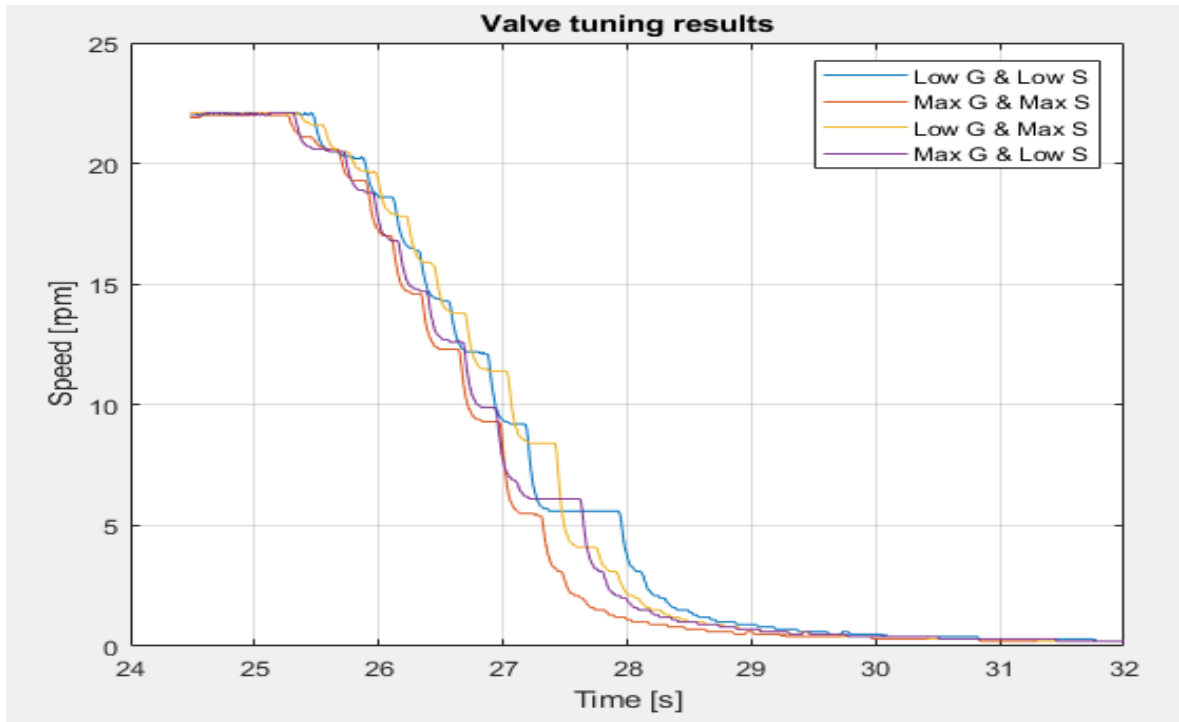


Figure 5.14: Response graph for tuning valve

with the minimum values (blue line) the answer is the slowest. Comparing the trend of the brake speed, in the case of parameter tuning, with the case with the default parameters, black line in the figure 5.12, it is possible to say that the valve controller is not responsible for the oscillatory trend of the brake speed.

This behavior is therefore attributable to the structural characteristic and the wear of the brake, these are specific aspects related to the braking system structure and will be subject of future studies.

# Chapter 6

## Sizing tool and Simscape model for motor and gear size for dynamic load

In this chapter we use the Matlab application, deepened in the chapter 3, and the complete Simscape model, developed in the chapter 5 , in order to be able to define a motor and gearbox system capable of satisfying a certain dynamic load. First of all it is necessary to create the initial database containing the data of the motors and gearboxes useful for using the Matlab sizing tool. To create the motor database are required the following data:

- Motor inertia;
- Characteristic torque and speed curve;
- Maximum Speed;
- Maximum Torque.

To create the reducer database are required the following data:

- Gear ratio;
- Direct and retrograde efficiency coefficient;
- Reducer inertia.

Then it is necessary to define the law of motion that must be realized, as the first test is used the law of motion shown in the figure 3.11. Populating the two databases with a motor and a reducer present in a test bench, using the data sheets of the suppliers to obtain the necessary data, it is possible to observe that such coupling, for

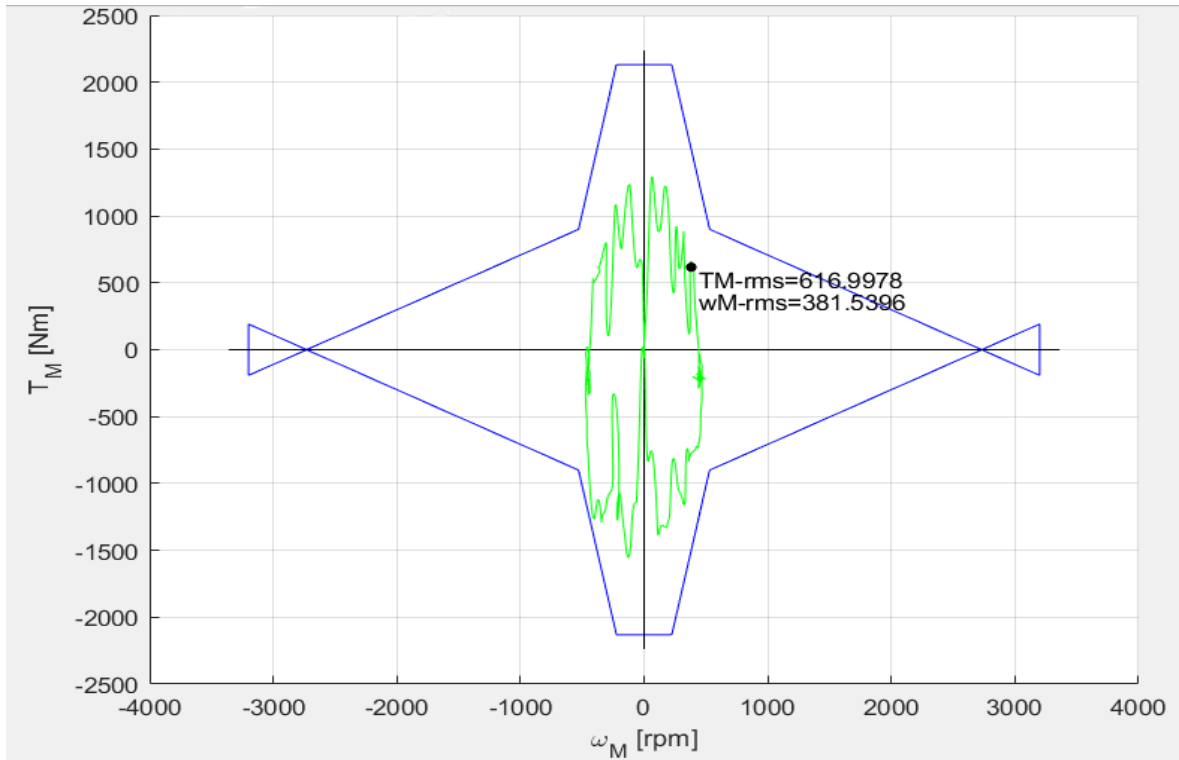


Figure 6.1: Engine operating range and location of dynamic loads

the sizing tool of Matlab, is able to satisfy such law of motion. The result obtained is shown in the figure 6.1

Looking at the graph shown in figure 6.1 according to the Matlab sizing tool, the motor and reducer system can meet the requirements of the law of motion for the load. After analyzing the result obtained through Matlab, it is necessary to introduce the system characteristics to the Simscape model, introducing the speed set point and the torque reference.

## 6.1 Simscape Model without spring and damper

By developing the simulation of the Simscape model is possible observe the results obtained. The first analysis refers to the trend of the load torque, compared with the real torque, and the trend of the engine torque. The result obtained is shown in the figure 6.2. Observing the evolution of the Simscape model torque, compared to the real torque, it is possible to say that the model follows the real reference; as for the motor torque, it can be stated that it follows the same trend of the load less the gear ratio and the efficiency of the reducer.

After analyzing the trends of the torque it is necessary to observe the speed trends, the latter are shown in the figure 6.3. By observing the speed, we can say that the engine meets the load requirements, except for the initial induction engine start-up phase.



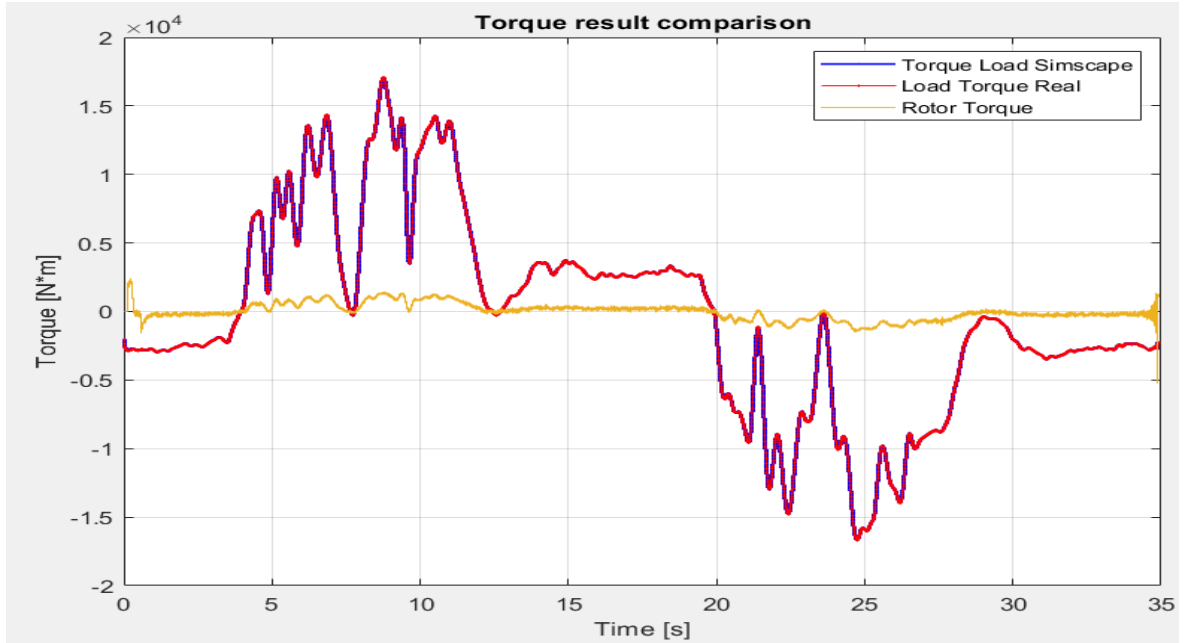


Figure 6.2: Trend torque of load and motor in Simscape compared to real load torque

The speed of the motor turns out to have the same course of the speed of the load, finding a difference of values legacies to the relationship of transmission and the efficiency of the reducer.

In conclusion, in the case study it is possible to say that the sizing tool of Matlab and the Simscape model have the same results. This result is linked to the initial hypotheses, that is, both for the Matlab sizing tool and for the Simscape model, stiffness and damping are not taken into account.

## 6.2 Simscape Model with spring and damper

In the present simulation the same law of motion is used, that is the maneuver of shuttling brought in figure 3.11 but introducing the stiffness of the shafts and stiffness and damping of the elastic joint present in the real system.

The first test is on the torque, in the figure 6.4 the trends of the torque are reported.

Observing the trend of the torque it is possible to observe how the torque required by the load is maintained, but in order to observe how the torque of the engine has changed it is necessary to compare it with the torque of the case without spring and damper. In the figure 6.5 the results are shown.

Observing the comparison between the engine torques of the two cases it is possible to say that the trend of the engine torque with the stiffness and damping is different from the previous case. It is possible to observe a longer time to stabilize the system and there are peaks of higher values, in absolute value, than the case with stiffness; this phenomenon is linked to the non-ideality of the case study.

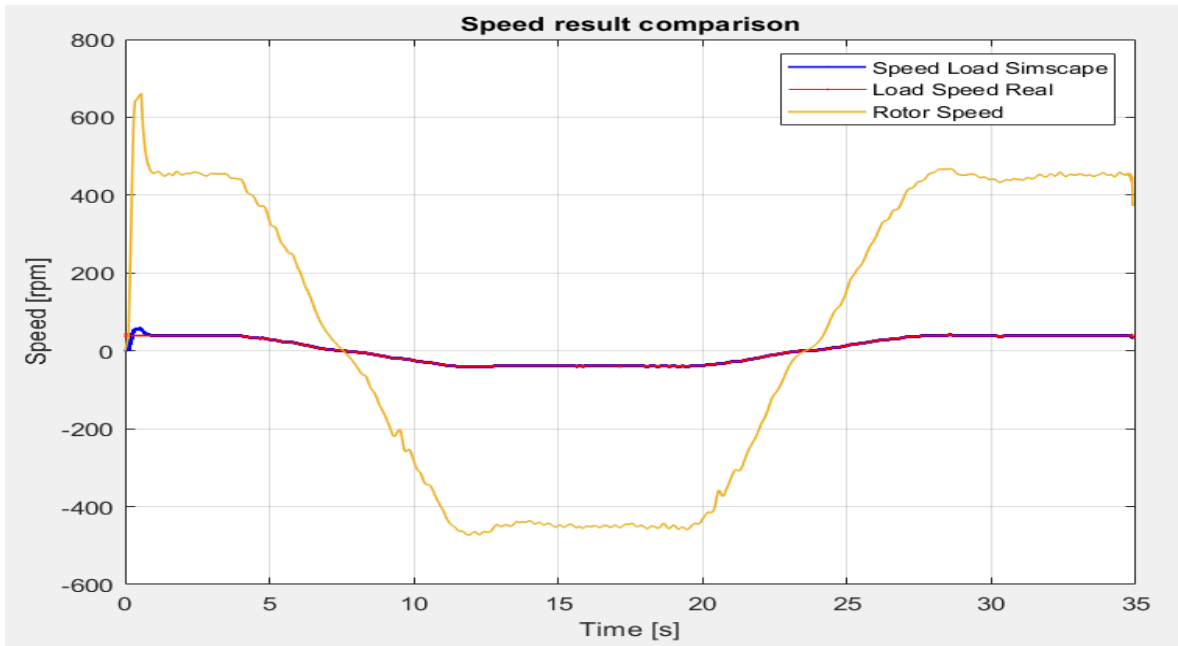


Figure 6.3: Trend speed of load and motor in Simscape compared to real load speed

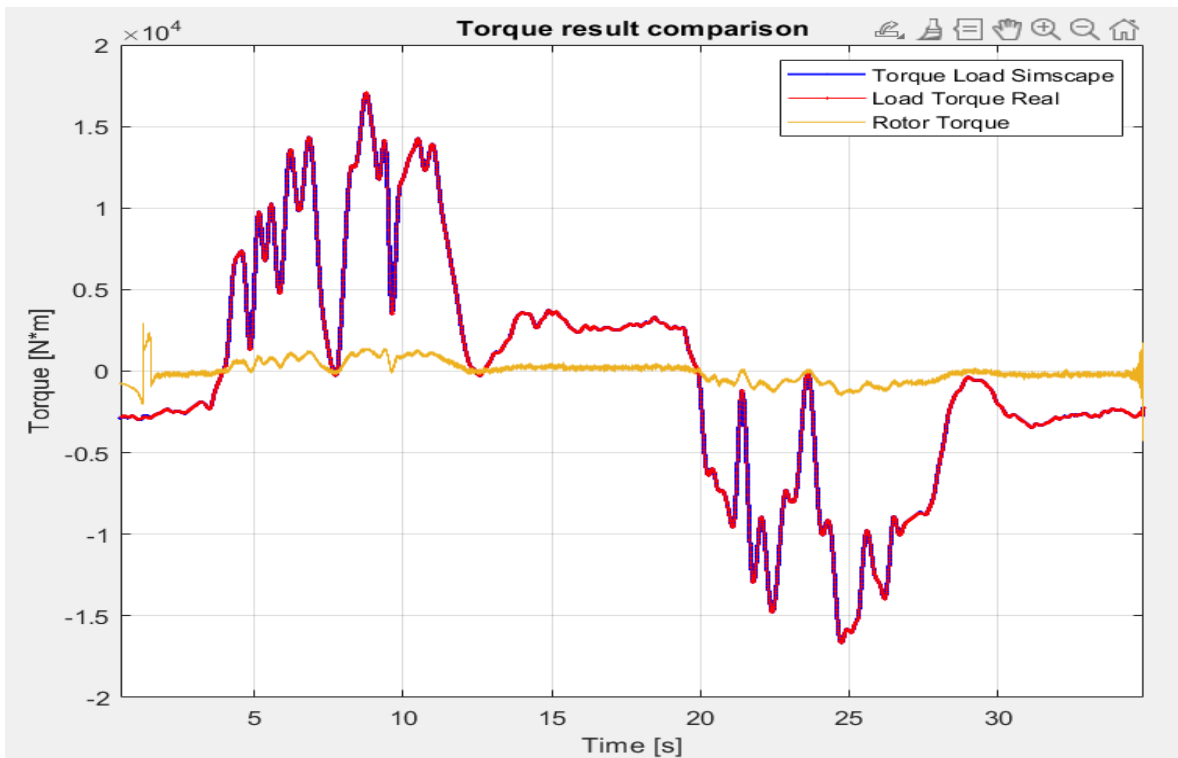


Figure 6.4: Trend torque of load and motor in Simscape compared to real load torque with spring and damper

Then it is necessary to observe the speed trend for the case that has stiffness and damping, the result is shown in the figure 6.6.

Observing the trend of the speeds it is possible to say that the law of motion is respected less than the initial stretch in which the engine undergoes a deceleration, this phenomenon is linked to the irregularities of the case.

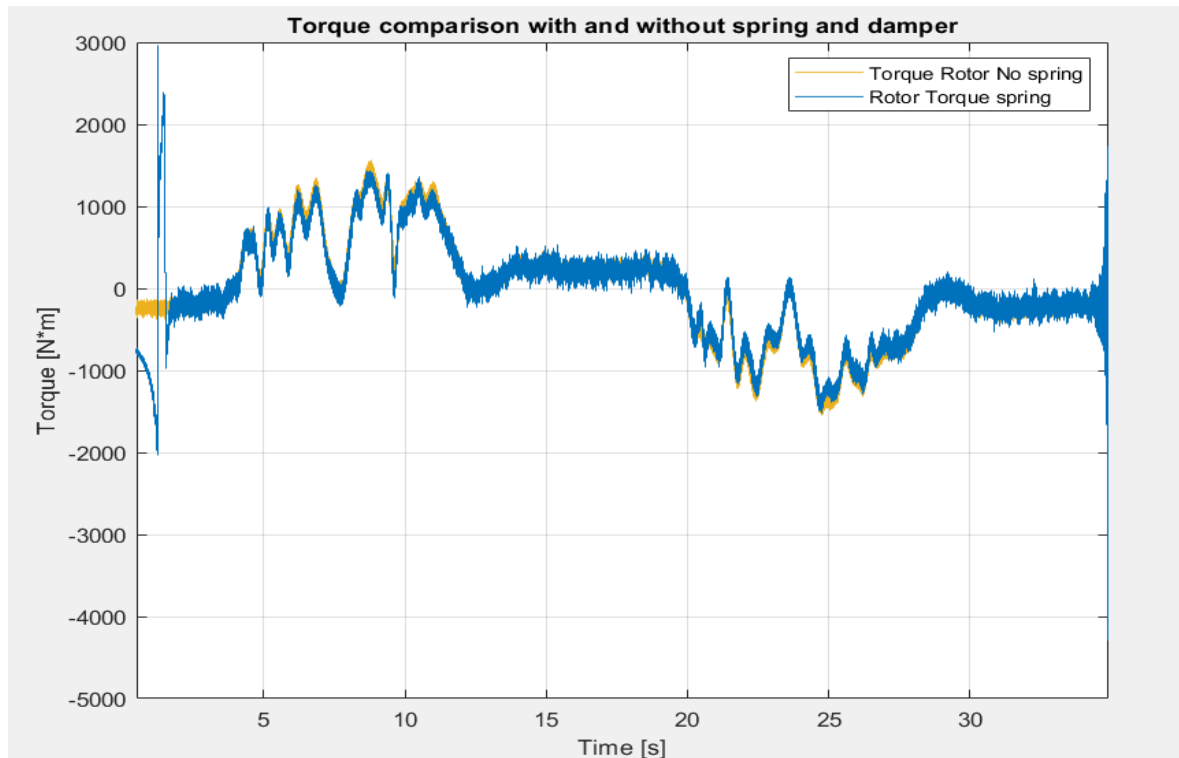


Figure 6.5: Torque trend comparison between ideal and real case

In order to highlight the variation of the speeds with respect to the case without stiffness and damping, the two speeds are compared and reported in the graph inside the figure 6.7.

In both cases it can be stated from the speed trend that the speed trend is respected less than an initial transitional period linked to the non-ideality of the case study. In conclusion it is possible to say that the selected motor and reducer system, considering stiffness and real damping of the system, unless a start-up time and achievement of regime is able to meet the law of motion imposed.

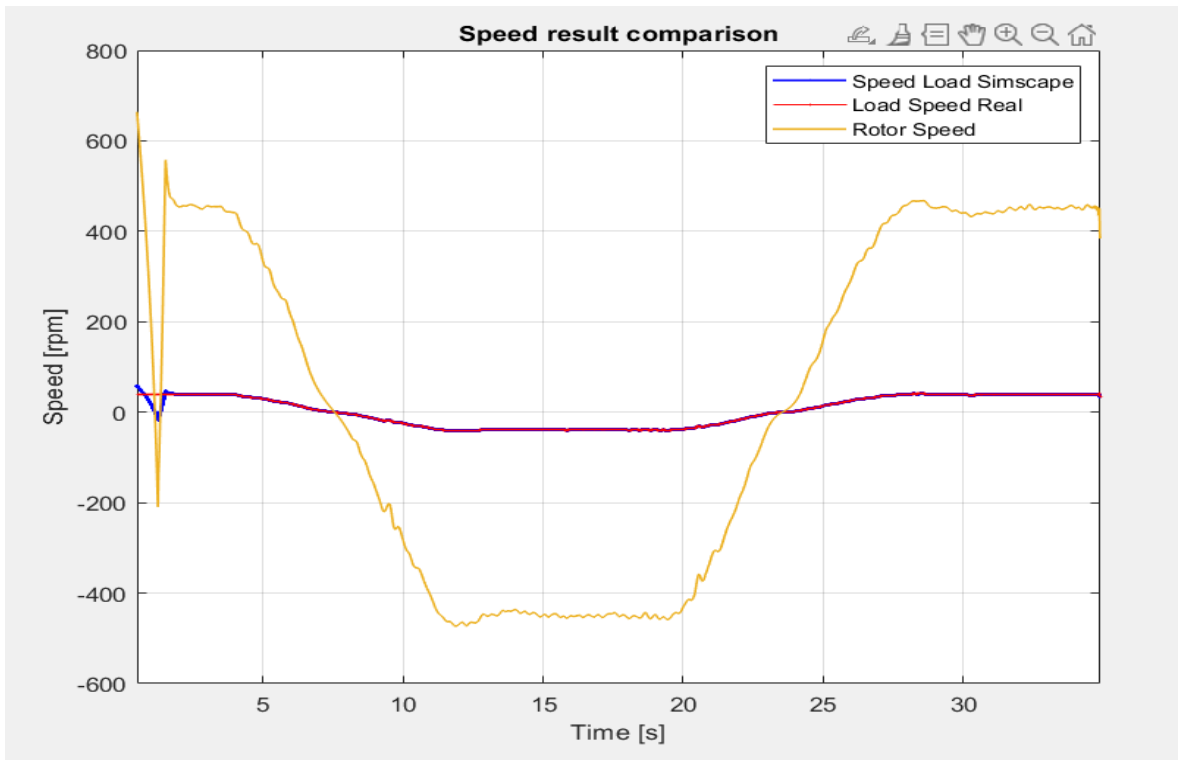


Figure 6.6: Speed trend comparison

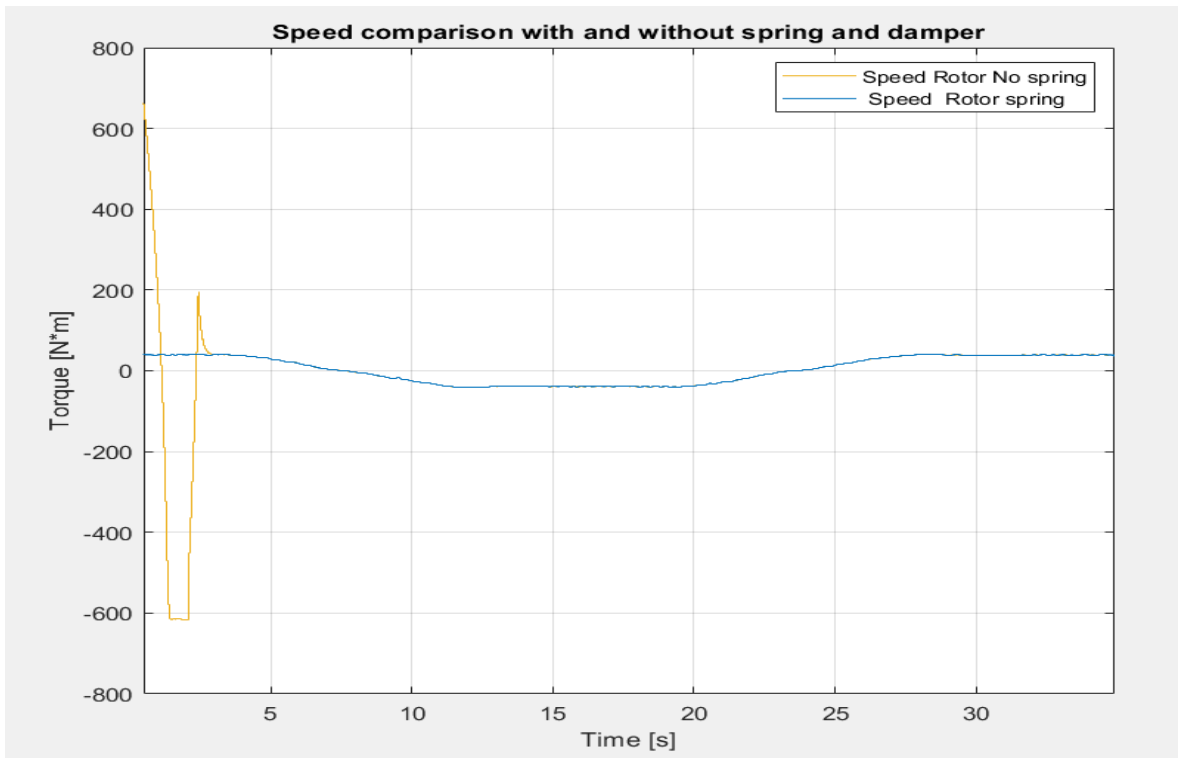


Figure 6.7: Speed trend comparison between ideal and real case

# Conclusions

In conclusion, in the present case study, the theoretical and practical aspects of properly sizing a reducer motor system according to the type of load applied were covered. Specifically, the following topics were covered:

- Pre-selection of the motor reducer system through an existing criterion in the state of the art at the time of writing this paper, with related modification of the existing selection criterion in order to apply it to the case study;
- Mathematical modeling of the components that make up the test room environment, with related theoretical insights into their operation and control;
- Definition of control techniques for the various components handled and tuning techniques on the present controllers;
- Correlation of created mathematical models with experimentally obtained results through error prediction techniques.

The end result of the present study is the realization of an algorithm for selecting a gear motor according to the chosen dynamic load demand; this allows the performance of such a system to be known before it is purchased and implemented , so that time and costs can be saved. The mathematical model was correlated with real data obtained experimentally, with negligible margin of error for the purpose of performance prediction estimation.

The planned improvements to the algorithm turn out to be the implementation of an additional algorithm for dynamic emulation of mechanical loads in test rooms to be able to emulate real loads obtained through experimental detection. A further improvement appears to be the deployment of such an algorithm in a Real-Time environment such as LabView or Veristand, possibly in a custom device in order to be able to use such prediction and performance tuning in real time in the test cell.

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# List of Figures

2.1	Scheme of a system using an electric drive for motion generation and control [2] . . . . .	4
2.2	Functional scheme of a generic machine [3] . . . . .	5
2.3	An exploded view of squirrel cage induction motor [4]. . . . .	6
2.4	Brushless motor operating zone [2]. . . . .	8
2.5	Different field-orientated control schemes [5] . . . . .	9
2.6	DSC and DTC block schematics [5] . . . . .	10
2.7	Planetary transmission [7]. . . . .	12
2.8	Characteristic curve of a purely passive load (a) and a load not purely passive (b) [2]. . . . .	14
2.9	Load location at constant torque, torque curves C and power W [2]. . . . .	14
2.10	Location of the dynamic loads of a servo system [2]. . . . .	15
2.11	Hydraulic brake system [9] . . . . .	16
2.12	Pneumatic brake system [9]. . . . .	17
2.13	Approximate oil flow inside the brake. In red is the line of pressure, in blue the inlet cooling line and in orange the outlet. . . . .	18
2.14	PI controller closed loop [12]. . . . .	21
3.1	Examples verification of motors for dynamic loads [14]. . . . .	24
3.2	Alfa and Beta comparison [14]. . . . .	27
3.3	Area of acceptable motor-reducer coupling for a given transmission inertia [14]. . . . .	28
3.4	Ramp law of motion. . . . .	29
3.5	Weight distribution at the wheels. . . . .	30
3.6	$\alpha$ - $\beta$ graph. . . . .	31
3.7	Matlab code for $\alpha$ - $\beta$ graph. . . . .	32
3.8	$\tau$ comparison graph, with the same representation shown in figure 3.3. . . . .	32
3.9	Matlab code for $\tau$ comparison graph . . . . .	33
3.10	Place of loads satisfied by the operating range of the motor coupled with the gearbox (a) and not satisfied (c) . . . . .	34
3.11	Speed and Torque of Shuttling maneuver. . . . .	35



3.12	Operating range of motor coupling and gearbox applied to a shuttling operation . . . . .	35
4.1	Test bench layout. . . . .	37
4.2	Simscape Model for Speed Control . . . . .	38
4.3	DC bus and Inverter . . . . .	39
4.4	Measurement with current and voltage sensor . . . . .	39
4.5	V/f control . . . . .	40
4.6	Mechanical system . . . . .	41
4.7	Measurement motor signal . . . . .	42
4.8	Comparison between ideal and real command . . . . .	43
4.9	Comparison between test bench response and Simscape model response	44
4.10	GT5000 Driver Critical Frequency Jump Function [17]. . . . .	45
5.1	Reducer with double pinion [18]. . . . .	47
5.2	Simscape model for Torque Control . . . . .	48
5.3	Electric Drive . . . . .	49
5.4	Control subsystem . . . . .	49
5.5	Direct Torque Control subsystem . . . . .	50
5.6	Mechanical subsystem . . . . .	51
5.7	Lamellar joint subsystem . . . . .	52
5.8	Pneumatic brake subsystem . . . . .	52
5.9	Brake torque applied to mechanical system . . . . .	54
5.10	Motor torque output as a function of braking torque . . . . .	55
5.11	Rotor speed versus speed set point . . . . .	55
5.12	Comparison between real performance and Simscape model . . . . .	56
5.13	Valve's control scheme [19]. . . . .	57
5.14	Response graph for tuning valve . . . . .	58
6.1	Engine operating range and location of dynamic loads . . . . .	60
6.2	Trend torque of load and motor in Simscape compared to real load torque	61
6.3	Trend speed of load and motor in Simscape compared to real load speed	62
6.4	Trend torque of load and motor in Simscape compared to real load torque with spring and damper . . . . .	62
6.5	Torque trend comparison between ideal and real case . . . . .	63
6.6	Speed trend comparison . . . . .	64
6.7	Speed trend comparison between ideal and real case . . . . .	64