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Integrated Torque Sensing for Enhanced Safety and Efficiency in Human-Centered Robotics

Master Thesis

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Abstract

Integrated Torque Sensing for Enhanced Safety and Efficiency in Human-Centered Robotics, by Léon Borremans, Brussels, May 2025

This thesis will focus on torque-sensorless control of a newly developed high ratio, backdrivable Wolfrom-based gearbox.

The goal of the master thesis is to investigate the feasibility of controlling an actuator composing of a brushless DC motor and this novel gearbox connected to a single-degree-of-freedom link, only using the measurements from the electric motor. To this end, a MATLAB/Simulink model is developed that, based on the double inertia model, can accurately predict the behavior of this actuator. It is modified such that human interactions with the link can be simulated. As one wants safe control without a torque sensor, these interactions are estimated with a disturbance observer and processed in the control algorithm. This algorithm is applied and validated onto a drivetrain in real-life. Human interactions are performed with a Pilz instrument in accordance with the ISO norm for robot/cobot-human interactions.

Keywords: [Wolfrom-based, backdrivability, double inertia model, disturbance observer, higher-order-sliding-mode observer, impedance control]

Samenvatting

Geïntegreerde Koppelmetering voor Verbeterde Veiligheid en Efficiëntie in Mensgerichte Robotica, door Léon Borremans, Brussel, mei 2025

Deze thesis richt zich op koppelregeling zonder sensor van een nieuw ontwikkelde, hoogvertragende, terugaandrijfbare Wolfrom-overbrenging.

Het doel van de masterproef is om de haalbaarheid te onderzoeken van het aansturen van een actuator bestaande uit een borstelloze DC-motor en deze nieuwe overbrenging, gekoppeld aan een éénvrijheidsgraden-mechanisme, enkel gebruikmakend van de metingen van de elektromotor. Daartoe wordt een MATLAB/Simulink-model ontwikkeld dat, gebaseerd op het dubbele-inertiemodel, het gedrag van deze actuator nauwkeurig kan voorspellen. Het model wordt aangepast zodat menselijke interacties met de schakelarm kunnen worden gesimuleerd. Aangezien men veilige controle zonder koppelmetering wenst, worden deze interacties geschat met een verstoringswaarnemer en verwerkt in het regelalgoritme. Dit algoritme wordt toegepast en gevalideerd op een aandrijflijn in de praktijk. Menselijke interacties worden uitgevoerd met een Pilz-instrument in overeenstemming met de ISO-norm voor interacties tussen robot/cobot en mens.

Trefwoorden: [Wolfrom-gebaseerd, terugaandrijfbaarheid, dubbel inertiemodel, verstoringswaarnemer, hogere-orde-schuifmoduswaarnemer, impedantieregeling]

Résumé

Détection de Couple Intégrée pour une Sécurité et une Efficacité Accrues dans la Robotique Centrée sur l'Humain, par Léon Borremans, Bruxelles, mai 2025

Ce mémoire porte sur le contrôle sans capteur de couple d'une nouvelle boîte de vitesses Wolfrom à haut rapport, réversible.

L'objectif du mémoire de master est d'étudier la faisabilité du contrôle d'un actionneur composé d'un moteur à courant continu sans balais et de cette nouvelle boîte de vitesses, reliés à un maillon à un degré de liberté, en utilisant uniquement les mesures du moteur électrique. À cette fin, un modèle MATLAB/Simulink est développé, basé sur le modèle à double inertie, capable de prédire avec précision le comportement de cet actionneur. Il est modifié de manière à simuler les interactions humaines avec le maillon. Comme on souhaite un contrôle sûr sans capteur de couple, ces interactions sont estimées à l'aide d'un observateur de perturbations et intégrées dans l'algorithme de commande. Cet algorithme est appliqué et validé sur une transmission en conditions réelles. Les interactions humaines sont réalisées à l'aide d'un instrument Pilz conformément à la norme ISO relative aux interactions entre robots/cobots et humains.

Mots-clés : [basé sur Wolfrom, réversibilité, modèle à double inertie, observateur de perturbations, observateur de mode glissant d'ordre supérieur, commande d'impédance]

List of Symbols

Symbol	Description
Abbreviations	
PGT	planetary gear train
DC	direct current
BLDC	brushless DC motor
CAD	Computer-Aided-Design
LTI	linear time invariant
LPF	low pass filter
DOB	disturbance observer
HOSM	higher-order-sliding-mode
MEDOB	multi-encoder disturbance observer
PRMS	Pilz Robot Measurement System
ISO	International Organization for Standardization
Gear Ratio's	
i	total gear ratio
i_w	gear ratio of compound w
i_0	gear ratio of compound 0
i_a	gear ratio of virtual compound a
j	number of circuits inside a PGT
n	number of links inside a PGT
Positions, Speeds and Accelerations	
ω_S	rotational speed signal of sun S (rad/s)
ω_C	rotational speed signal of carrier C (rad/s)
ω_{Ps}	rotational signal speed of planet Ps (rad/s)
ω_{Pa}	rotational signal speed of planet Pa (rad/s)
ω_{Pb}	rotational signal speed of planet Pb (rad/s)
ω_P	rotational signal speed of all planets (rad/s)
ω_{Ra}	rotational signal speed of ring Ra (rad/s)
ω_{Rb}	rotational signal speed of ring Rb (rad/s)
$\theta_M, \dot{\theta}_M = \omega_M, \ddot{\theta}_M$	motor-side position, speed and acceleration signals ($rad, rad/s, rad/s^2$)
$\theta_L, \dot{\theta}_L = \omega_L, \ddot{\theta}_L$	load-side position, speed and acceleration signals ($rad, rad/s, rad/s^2$)
$\bar{\Theta}_L$	load-side reference end-position (rad)
$\bar{\Omega}_L$	load-side reference constant speed (rad/s)
$\bar{\theta}_L, \dot{\bar{\theta}}_L, \ddot{\bar{\theta}}_L$	load-side position, speed and acceleration reference signals ($rad, rad/s, rad/s^2$)
$\tilde{\theta}, \dot{\tilde{\theta}}, \ddot{\tilde{\theta}}$	virtual/error load-side position, speed and acceleration signals ($rad, rad/s, rad/s^2$)
$\theta, \ddot{\theta}$	generic position and acceleration signals ($rad, rad/s^2$)
$v_{rel, max}$	maximum relative speed between two objects (m/s)
Inertia's, Masses and Surfaces	
J	total load-side reflected inertia (m^2kg)
J_d	total load-side reflected desired/virtual/impedance inertia (m^2kg)
r_d	total load-side reflected inertia ratio
J_g	load-side reflected gearbox inertia (m^2kg)
J_M	inertia sum of components at motor-side (m^2kg)
J_L	inertia sum of components at load-side + J_g (m^2kg)
A_s	surface are of contact (m^2)
μ	reduced mass between human and robot/cobot (kg)
m_H	effective human body part mass (kg)
m_R	effective mass of robotic link (kg)
M	total mass of moving parts (kg)
m_L	effective payload mass (kg)
m_l	robotic link mass (kg)

Symbol	Description
Number of Teeth	
Z_S	number of teeth of sun S
Z_{Ps}	number of teeth of planet Ps
Z_{Pa}	number of teeth of planet Pa
Z_{Pb}	number of teeth of planet Pb
Z_{Ra}	number of teeth of ring Ra
Z_{Rb}	number of teeth of ring Rb
Torques, Forces and Pressures	
$\tau_M, (\bar{\tau}_M, \hat{\tau}_M)$	motor delivered/measured torque, (reference, estimated) (Nm)
τ_s	reaction torque (Nm)
$\tau_d, (\hat{\tau}_d)$	disturbance torque, (estimated) (Nm)
$\tau_g, (\hat{\tau}_g)$	gravity torque, (estimated) (Nm)
$\tau_f, (\hat{\tau}_f)$	friction torque, (estimated) (Nm)
$\tau_{bl}, (\hat{\tau}_{bl})$	backlash torque, (estimated) (Nm)
τ_{stic}	stiction compensation torque signal (Nm)
G	gravity torque constant (Nm)
F_T	transient force (N)
p_T	transient pressure (Pa)
F_S	quasi-static force (N)
p_S	quasi-static pressure (Pa)
F_{im}	impact force (N)
Stiffness and Damping Coefficients	
K_s	gearbox stiffness (Nm)
K	human body part stiffness (N/m)
K_d	virtual/desired/impedance stiffness (Nm)
ξ	relative damping ratio (%)
C	human body part damping (Ns/m)
C_d	virtual/desired/impedance damping (Nm/s)
ξ_d	virtual/desired/impedance relative damping ratio (%)
System Theory Parameters and Related	
t	time (s)
s	Laplace-transform variable (rad/s)
z	Z-transform variable (rad/s)
T_s	sampling period (s)
T	LPF time-constant (s)
T_{nat}	natural period (s)
t_f	final time (s)
g	generic frequency symbol (rad/s)
$G(s)$	generic LTI system
$G_n(s)$	estimate of $G(s)$
$q(s)$	Laplace transform of generic $q(t)$
$\dot{q}$	derivative of generic q
Σ	LTI state-space system
A	state matrix
B	input matrix
D	disturbance matrix
C_{out}	output matrix
$L = [l_1, l_2]^T$	Luenberger observer matrix gain
$F, (F^{-1})$	observability matrix (and inverse)
I	identity matrix
$x, (\hat{x}, z)$	state vector (estimated, Luenberger estimation)
u	input signal
y	output signal
$w, (\hat{w})$	disturbance signal (estimated)
v	sliding mode vector
$\alpha_1, \alpha_2, \alpha_3$	HOSM parameters
Other Symbols	
K_t	B BLDC torque constant (Nm/A)
N	number of points in time
α	average value over N samples
i_p	phase current (A)
E	maximum permissible energy (J)
ω_d	virtual/desired/impedance eigen-pulsation (rad/s)
A_d	virtual/desired/impedance amplitude ($1/(Nm)$)
$c_{coulomb} = \tau_c$	rotational coulomb friction coefficient (Nm)
$c_{visc} = \nu$	rotational viscous friction coefficient (Nms)
c_{mesh}	rotational meshing friction coefficient
l	robotic link length (m)

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Chapter 1

Introduction

1.1 Motivation

The field of human-centered robots and Human-Robot-Interaction (HRI) is increasingly gaining traction, given its potential to revolutionize interactions between humans and machines in everyday environments [1]. Central to the realization of this vision is the need for robots to co-exist safely and synergistically with humans. Traditional approaches to ensure the safety of these interactions have predominantly relied on the integration of costly and heavy-torque sensors within robotic actuators or on reduced torque resolution. These sensors are crucial for detecting the presence of humans and modulating robotic behavior accordingly [2]. In safe Human-Robot-Interaction (HRI), the robot must detect human interaction, requiring the need of a force sensor, a contact sensor or a joint torque sensor. These sensors will increase the cost, size and mass of the robot [3]. At the same time they reduce the bandwidth and stiffness of the actuator, and induce extra noise [4]. One possible solution is the use of highly backdrivable actuators, which are generally safer as they can yield under certain external forces and could be, in some cases, even backdriven by hand [3].

The R&MM group of the VUB has developed a novel Wolfrom-based gearbox design that promises to mitigate these limitations. This innovative layout not only potentially reduces the weight of robots by up to 50%, thereby enhancing their utility, environmental sustainability, and safety, but also introduces a transparent gearbox mechanism. This transparency facilitates the extraction of internal gearbox properties through indirect measurement techniques. For example, as discussed in this thesis, it enables the estimation of output torque based solely on input-side variables, offering a promising alternative to conventional torque sensing, which typically relies on direct measurements at the gearbox output [2].

1.2 Goals

The primary objective of this project is to explore and validate novel torque sensing methods that leverage the unique properties of the newly developed gearbox layout. By eliminating the need for traditional torque sensors, this research aims to significantly reduce the cost and complexity of human-centered robots, making them more accessible and safer for everyday use. Specifically, the project will focus on investigating the three points presented below.

- To what extent can motor current serve as an accurate indicator of gearbox input torque, and how effectively can it be used to estimate output torque, especially for the estimation of impacts?
- How can the safety of human interaction and collision be ensured for a link connected to a motor-gearbox actuator operating in position and/or speed control?
- Can we develop a new algorithm for high ratio gearboxes that accurately predicts the impact torque?

The ISO 15066 norm will provide the necessary requirements for the safety guarantee in HRI applications. The metrics and their respective values of the given research questions are given in table 1.1, where the last two requirements are related to this ISO norm. It is important to highlight that these research questions are formulated for experiments with a clock frequency of $1kHz$.

Accuracy metrics	Value
Duration of impact estimation	50%
Position control	1%
Impact torque estimation	10%
Constant torque estimation	5%

Table 1.1: Values linked to the research questions. The duration of impact accuracy is the relative error between the actual duration of an impact and the estimated one. Position accuracy is related to the difference between the wanted load-side angle and the actual angle.

1.3 Outline

chapter 2 reviews the state-of-the-art in gearbox technologies, torque sensing, and control strategies relevant to human-robot interaction. chapter 3 presents the modeling and control architecture of the novel Wolfrom-based gearbox, along with simulation results in MATLAB/Simulink. chapter 4 describes the experimental validation using two test setups to assess safety and performance under realistic HRI scenarios. chapter 5 summarizes the key findings, addresses the initial research questions, and outlines directions for future work. Additional details, such as a model for backlash and ISO standards, are provided in the appendices.

Chapter 2

State of the Art

2.1 Gearboxes

2.1.1 Introduction

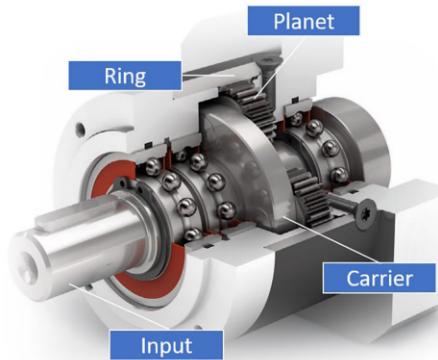
High torques are typically needed for conventional industrial applications. There exist several ways to achieve high torques. Typical drivetrains include a high speed, low torque electric motor in series with a high reduction gearing element [1]. Available technologies comprise planetary gear trains (PGT's), harmonic drives and cycloid drives. These all rely on the high speed, low torque electric motors [5]. Harmonic and cycloid drives are more frequently used in industrial robot applications and HRI, but all rely on the integration of a torque sensor at the output [1]. The direct-drive uses a high torque motor, disregarding the use of a transmission element. Advantages include simple dynamic control, good backdrivability [5] and a smaller impact of nonlinearity's such as backlash [1]. Quasi direct-drives have a compromise solution between direct-drive and traditional gear element actuation. It combines a high-torque motor with a low gear ratio gearbox to achieve high torques [5].

2.1.2 Gearbox types

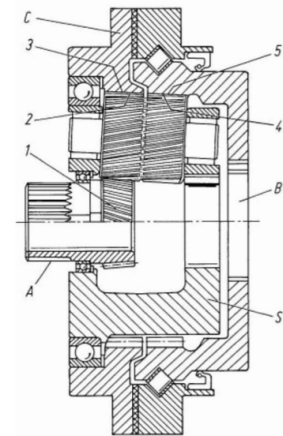
The most common gearbox types are presented in this section to establish an idea of the (dis)advantages across these types, justifying the choice for the Wolfrom-based gearbox [1] to be controlled in this master thesis. The common gearbox types include planetary gear trains (PGT's) (fig. 2.1a), Wolfrom PGT's (fig. 2.1b), harmonic drives (fig. 2.1c) and cycloid drives (fig. 2.1d).

Planetary gear trains

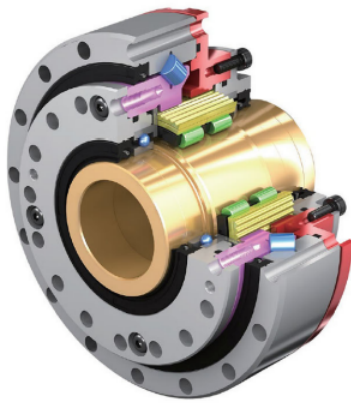
The planetary gear train is compact, versatile, and broadly used in power trains where power density matters. High ratios can be achieved by choosing a certain amount of teeth for the sun, planet and ring gears, however, mostly compromising efficiency. The Wolfrom PGT type is especially compact and seems to have gained interest in the robotics community [1], [3]. In Wolfrom-based gearboxes, high reduction ratios can be achieved but are characterized by large meshing losses [5]. The working principle is explained later in this chapter. Large ratios mean that a smaller motor can be used, reducing the overall weight and lowering the battery energy consumption [4]. As [6] proposes a torque scaling law to be proportional to the squared diameter



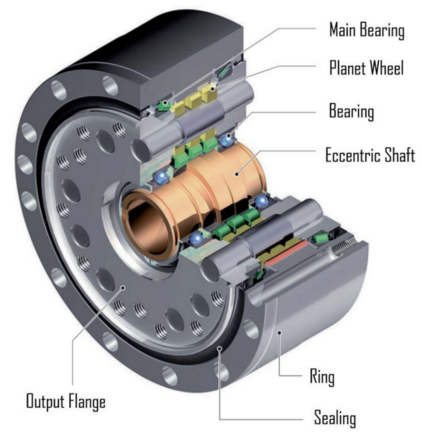
(a) Planetary gear train (PGT) [1].



(b) Wolf from PGT [1].



(c) Harmonic drive [1].



(d) Cycloid drive [1].

Figure 2.1: The four most common gearbox types. An review of these gearbox types can be found in [1].

of the gearbox, it makes sense that using a smaller motor and a larger reduction ratio element will result in an overall reduced actuator weight. This is depicted in fig. 2.2. PGT's also have the advantage of being less sensitive to changes in operating conditions. This makes them suitable for HRI applications. PGT's are most prone to backlash losses however.

Harmonic and cycloid drives

Harmonic drives have the best torque-to-weight ratios, low lost in motion due to backlash, but the worst peak efficiencies, which also depend largely on the operation conditions. Cycloid drives have similar size of the harmonic drive, but weights closer to those of PGT's. The torque-to-weight ratio is in the range between those of a PGT and a harmonic drive. They can withstand large loads and impacts with little maintenance required. Mechanical compliance is lower than

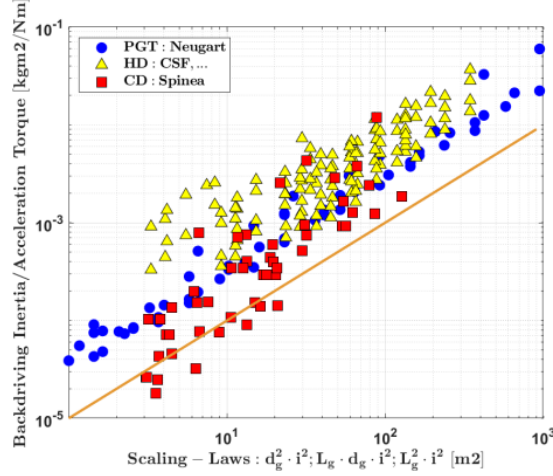


Figure 2.2: Correlation between backdriving inertia, acceleration (maximum repeatable) torque and the gearbox outer radius d_g [6]. It can be seen that the correlation between the backdriving inertia over the acceleration torque is proportional to the square of the outer diameter of the gearbox d_g .

PGT's and harmonic drives. These gearboxes have smaller and more condition-dependent peak efficiencies making them less suitable for HRI, especially in combination with relatively large torque-ripple [1]. However, the combination of cycloid drives with a conventional PGT's pre-gearing stage enables their broad acceptance in robotics [1].

2.1.3 Bilateral Drive

The bilateral drive is a two-stage Wolfrom-based planetary gear train designed by Hiroshi Matsuiki, Kenta Nagano and Yasutaka Fujimoto. A comparison between the gearbox used in this paper in terms of configuration is depicted in fig. 2.3. A Wolfrom PGT has at least two stages of PGT's where the planets rotate around the same shaft. Most of the time, the ring at the stage of the sun is fixed and the ring at the second stage serves as an attachment point for the output shaft. The creators of the bilateral drive use an optimization based approach to determine the number of teeth (NOT) and the profile shift coefficients of all the gears to maximize the total forward driving efficiency [3].

2.1.4 Novel Wolfrom-based gearbox

The novel Wolfrom-based gearbox is developed by Ailos and the R&MM group of the VUB. This three-stage Wolfrom-based gearbox uses the sun s as input and ring Rb as output [7]. The graphical representation of the gearbox is depicted in fig. 2.4a.

From fig. 2.4b and fig. 2.4c, the gearbox can be seen as a concatenation of two separate ones, where each gearbox has a different transmission ratio. The total transmission ratio is the product of the separate ratio's:

$$i = i_0 i_w = \frac{\omega_S}{\omega_C} \frac{\omega_C}{\omega_{Rb}} = \left(1 + \frac{Z_{Ra} Z_{Ps}}{Z_{Pa} Z_S}\right) \left(\frac{1}{1 - \frac{Z_{Pb} Z_{Ra}}{Z_{Pa} Z_{Rb}}}\right). \quad (2.1)$$



Figure 2.3: Comparison between the Bilateral Drive and the novel Wolfrom-based gearbox used in this paper in terms of configuration. See also fig. 2.4 that dives deeper in the different stages of the configuration of this latter. Key differences for the novel gearbox is its extra stage and higher gear ratio of ~ 222 compared to ~ 100 for the bilateral drive.

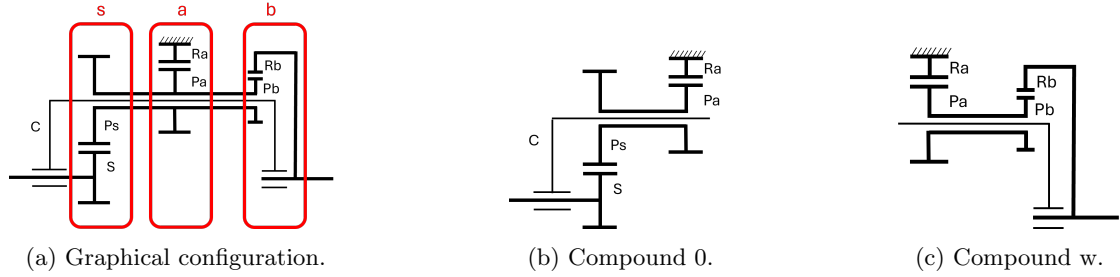


Figure 2.4: Configuration of the three-stage Wolfrom gearbox with only one meshing in each stage. S represents the sun, Ra the fixed ring, and Rb the ring gear of the respective stages s , a and b . C is the common carrier supporting the planetary gears Ps , Pa and Pb . The gearset is the superposition of compounds 0 and w [7].

In this formula, i stands for the gear ratio of a certain compound: i_0 for compound 0 (the pre-Wolfrom stage) and i_w for compound w (the basic-Wolfrom stage). Z_j stands for the number of teeth of part j : Ra for the ring of stage a , Rb for the ring of stage b , Ps for the planet of stage s , Pa for the planet of stage a , Pb for the planet of stage b and S for the sun. The addition of the extra s stage has the advantages of increasing the gear ratio and limiting the geometric dependency of i_0 to the outer diameter, resulting in the capability of separately optimizing the sun-planet meshing. Another advantage includes a more balanced Hertzian and bending load profile. The first compound 0 has a small sun S and two larger, thin planets Ps that are used for high speed and low torques. That means that the ratio i_0 is large. The second compound w has eight smaller and thicker planets (four planets Pa and four planets Pb) to cope with the larger torques. That implies that the ratio i_w is small.

2.2 Gearbox metrics and concepts

2.2.1 Introduction

In this section, four metrics for gearboxes are highlighted. They include backdrivability, mechanical compliance, backlash and efficiency.

2.2.2 Backdrivability

Across the literature, the concept of backdrivability is not entirely consistent. In [4], the authors provide a definition for backdrivability as: "the ability of an actuator that receives power from the external environment". The authors of [8] propose: "backdrivability is the performance of torque transmission from the load terminal input to the motor angular acceleration output". Whereas the authors of [5] describe it as: "backdrivability characterizes the ability of an actuator to operate in reverse direction, driven from its output". In these three references, the unit to describe backdrivability is either power, torque or speed respectively.

For robots in close contact with humans, safety must be guaranteed when they collide with each other. Advantages of backdrivability include increased safety on impact, wearability and the capability of manually programming the robot by demonstration, even when non-powered. In backdrivable actuators, transparency increases, possibly allowing for a (cheaper) torque-senseless control [5], which is the aim of this thesis. Another advantage is the bi-directional power flow possibility between human and actuator, especially interesting in rehabilitation devices or exoskeletons. It achieves higher overall efficiency and easier motion flow for such rehabilitation purposes [1]. Disadvantages include the incapability of holding a (static) position without the consumption of input power [6].

In general, most backdrivable actuators have a brake serving as fail-safe. In case of power shortage during operation, these brakes close instead of opening. This also implies that brakes continuously waste energy during operation. An overview of locking devices and brakes can be found in [9].

2.2.3 Mechanical/torsional compliance

There also seems to be confusion about the difference compared to mechanical compliance. In [4], compliance is stated as: "An actuator with high mechanical compliance refers to the fact that the output can be moved by a small external force, but it is possible that its backdrivability is still low because the input is hardly to receive the power of the external environments". [10] explains it as: "Mechanical compliance refers to the inherent flexibility of a mechanical system, specifically how much the output side of a drivetrain can be displaced when the input (e.g. the motor) is held fixed". This property reflects the elastic deformation of the components (e.g., shafts, gear teeth, housing) under load and is independent of programmed control or software-defined behavior. A compliant system will allow some output motion under external force even if the motor is not actively moving, provided backlash has been eliminated or compensated for [10].

2.2.4 Backlash

Gear backlash is the play between teeth. It is equal to the angle at which the output shaft of a gearbox can rotate without moving the input shaft and is characterized with zero torque transmission [11]. An example for a simple pinion-gear is provided in fig. 2.5a. In practice, there is always (a small) torque transfer, due to the friction in bearings for example [11]. Backlash induces a certain clearance between mating gears, which is needed to prevent binding between them at the contact point as well as double flank contact. Binding can lead to additional heat losses, noise, contact deformation, and thus to a potential preliminary failure of the gearbox [11].

Backlash is particularly important when it comes down to torque control; as the angle over which backlash occurs results in zero torque, and thus zero power transfer. This effect is prominent when the input speed reverses its direction. As a result, the output might see a zero torque

transfer over a certain time interval. It is very difficult to model and thus backlash should be reduced in the design phase of a gearbox.

Backlash can be measured using a single encoder placed at the input of the gearbox. The backlash property is then measured by maximally rotating the input and fixing the output. The backlash can be expressed as a relative angle between the two sides. In theory, no torque should be applied to either the input or output side. Otherwise, the stiffness contributes to the measurement.

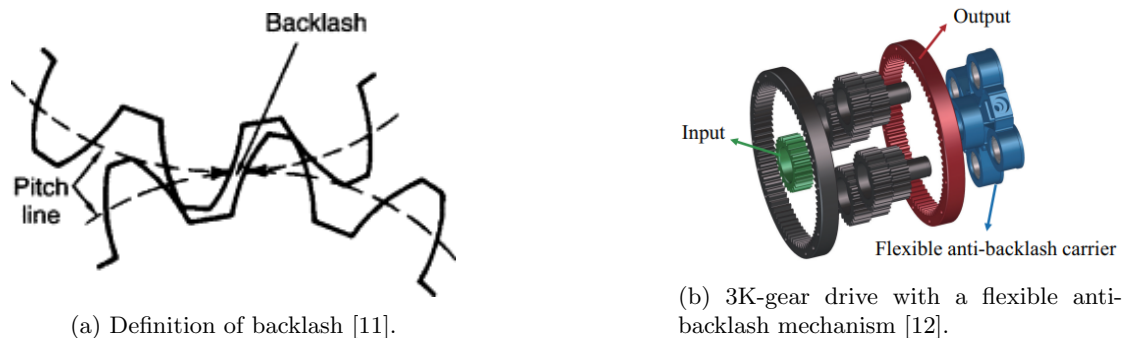


Figure 2.5: Backlash inside a gearbox is here defined as the angle at which the output shaft can rotate without moving the input shaft [11] [12]. Flexible elements such as an anti-backlash mechanism, depicted in the right, can be used to reduce or mitigate backlash effects. A disadvantage is that the gearbox compliance increases, i.e. the gearbox becomes less stiff, as will be explained later.

2.2.5 Gearbox efficiency

Another important distinction that must be made is the difference between forward and backward driving efficiency. The driving efficiency of a gearbox in general refers to the generated output power divided by the input power. In case of forward drive, the input refers to the motor-side and in case of backward drive, the load-side reflects the input. [3] proposes a simple mathematical framework based on a friction coefficient to prove that the backdriving efficiency is always lower than that of forward driving. The result is depicted in fig. 2.6.

In order to estimate the forward and backdriving efficiency, one must know the contributions that decrease these efficiencies in gearboxes. fig. 2.7 proposes a classification of torque losses in an electric motor and a gearbox drive [6]. Most manufacturers only provide maximum gearbox efficiency for a nominal operating point in their datasheets. The efficiency changes a lot over the entire operating range, making this single value impractical for real-time control.

2.3 Mathematical modeling of PGT's

2.3.1 Introduction

This section introduces the mathematical framework used to model the dynamic behavior of actuators incorporating planetary gear trains (PGT's), or any transmission element in general. The objective is to accurately represent the phenomena that affect torque transmission within these systems. Key aspects include the double inertia model, used to describe motor-load interactions

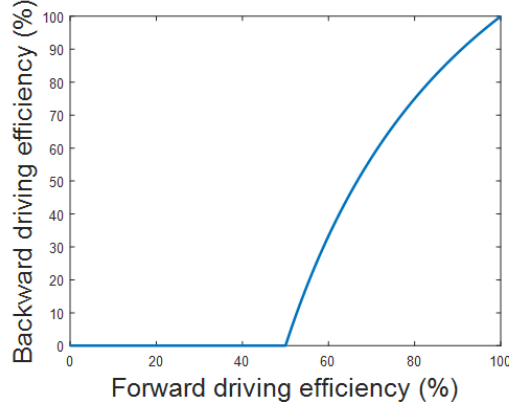


Figure 2.6: Forward versus backward drive efficiency of a gearbox [3].

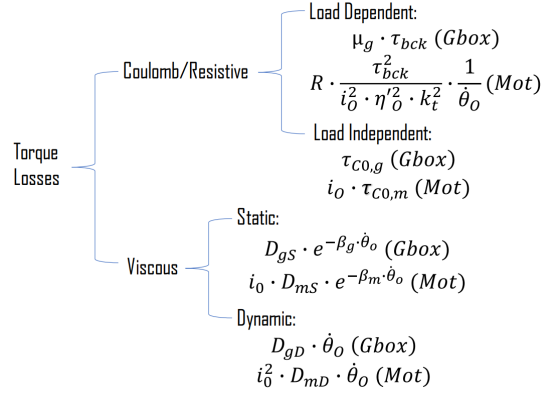


Figure 2.7: Classification of torque losses. Coulomb/resistive losses are losses due to dry friction, whereas viscous friction losses are losses due to flow. Each of these losses can be further categorized into a loss at the motor and gearbox side [6].

and backlash, as well as the modeling of gear efficiency losses, which introduces nonlinearity and discontinuity into the torque dynamics.

2.3.2 Dynamic actuator model

Double inertia model

To model the dynamics of joint actuators composing of a motor and gearbox, a double inertia system is frequently used in the literature. It is a second order, linear, coupled ordinary differential equation system and has multiple implementations. Based on [13], a general equation is proposed:

$$J_M \ddot{\theta}_M = \tau_M - \frac{\tau_s}{i} - \tau_f \quad (2.2)$$

$$J_L \ddot{\theta}_L = \tau_s - \tau_d - \tau_g \quad (2.3)$$

$$\tau_s = K_s \left(\frac{\theta_M}{i} - \theta_L \right). \quad (2.4)$$

Eq. (2.2) represents a dynamic equation of the motor-side, eq. (2.3) of the load-side and eq. (2.4) models the reaction torque between motor- and load-side to the flexible nature of materials used inside gearboxes [13]. The time signals θ_M and θ_L are the motor- and load-side angles respectively. The torque signals τ_M , τ_s , τ_f , τ_g and τ_d represent the motor, reaction, friction, gravity and disturbance torque respectively. Most of the time, the motor torque is proportionally modeled with the drawn current and gravity solely depends on the configuration. In the case of a single-degree-of-freedom robotic link attached to the load-side, the gravity torque depends on θ_L . Notice that no Coriolis and/or centrifugal forces are modeled as this is the dynamic equation for a single degree-of-freedom actuator. A block scheme of the model is shown in fig. 2.8. Using this representation, backlash can be modeled via the saturation block *backlash*. When the load- and motor-side angles are larger than the backlash of the gearing element, the output of the block is the backlash angle itself. It is important to mention that this block introduces a nonlinearity.

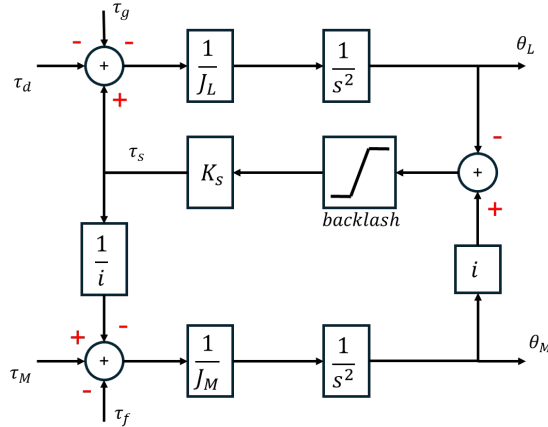


Figure 2.8: Block diagram of the double inertia model typically used for rotary actuators.

The disturbance τ_d incorporates the torques due to impacts and reaction effects. Examples of the latter are normal torques acting on the joint. Most of the time, gravity and disturbance are represented as a single load-dependent torque. Since the goal of this thesis comes down to estimating the disturbance torque without the influence of gravity, the two are separately modeled. This disturbance could represent a human wielding on the load-side of the gearbox and is thus time-dependent.

J_M and J_L are the inertia's of all elements at motor-side and load-side respectively. K_s is a stiffness hyperparameter, because it is generally not known for stiff elements such as gearboxes. However, some exceptions such as described in fig. 2.5b exist, where an anti-backlash mechanism is added to the load-side of the gearbox. Due to the flexible nature of such elements, the total gearbox stiffness K_s can be estimated as the stiffness of the anti-backlash mechanism, because small stiffness's have a large influence in a serial mechanical system [12].

2.3.3 Gearbox efficiency models

Drivetrain losses

Efficiency models describe torque losses as a function of speed and load acting on the motor and gearbox. Most of the time, such models are implemented in a real-time environment, where they can serve as a feedforward compensation [4]. Generally, losses inside a gearbox can be divided into load-dependent and load-independent losses, as in fig. 2.9. Load-dependent losses come from friction due to moving parts, whereas load-independent losses originate from effects of wind and/or oil [14].

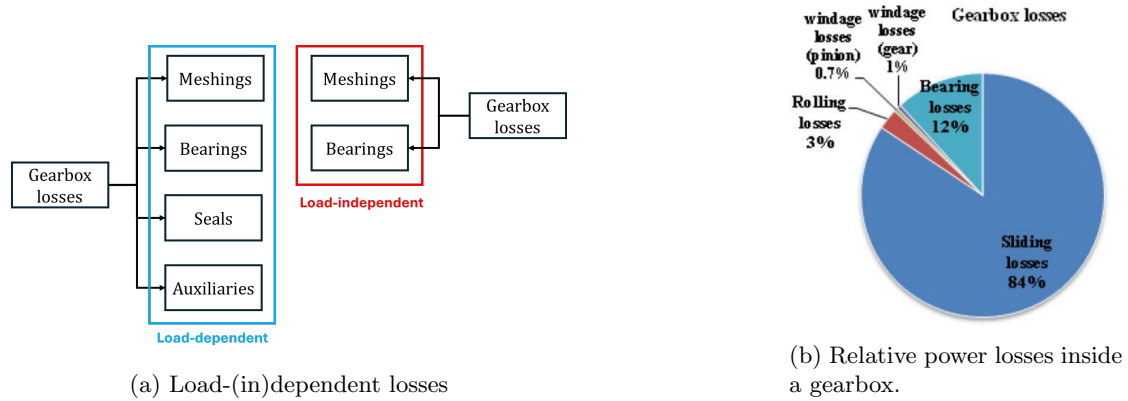


Figure 2.9: Load-dependent and load-independent losses inside a gearbox on the left and the relative power losses in a gearing system on the right [14].

The meshing losses in fig. 2.9a are rolling and sliding losses. The power losses due to meshings are proportional to the rotational speed and the transferred torque, in case of a single meshing [14]. Churning losses arise due to the interaction of gears with lubricant (oil). They are further divided into drag losses and the interaction of the gear pair with the surrounding medium at the mesh interface. A last class of losses is the windage losses introduced by drag on the rotational movement of gears. A pie chart estimating the contributions of these losses to the overall efficiency is depicted in fig. 2.9b [14].

Del Castillo efficiency model

In a gearbox, meshing efficiency depends on internal power flows and is not just the product of all individual meshing losses. This is what the mathematical models of Del Castillo and Pennestri concern in an analytical way. Both methods start by building a table representing the structure of its circuits. A circuit always composes of two meshing gearwheels and a carrier. The number of circuits j in a PGT equals the number of links n minus 2:

$$j = n - 2. \quad (2.5)$$

Building these tables allows the programmer to implement formulas (possibly speed-dependent) of the simple meshing efficiencies. Several simple meshing efficiency models exist, called 'ordinary efficiencies'. Mostly, these are determined experimentally [15] or an analytical expression is used. Examples of the latter include the formulas of Buckingham or Niemann and Winter, as described in [16]. Another example includes the Anderson-Loewenthal ordinary efficiency

model [14]. A detailed step-by-step plan for calculating the total efficiency for a PGT where the ordinary efficiencies are individually known is provided in [17].

Pennestri efficiency model

Calculating the PGT efficiency with the method of Pennestri also implements the use of the circuit table but differs from the method of Del Castillo by also building a power flow circuit. In such charts, each block represents a circuit and the connection lines represent meshing pairs. More information on how to build these charts can be found in [15]. An implementation of the tables and power flow charts of the method of Del Castillo and Pennestri can be found in table 2.1 and fig. 2.10.

circuits	Links		
	i	j	r
a	1	5	4
b	2	5	4
c	3	5	4

Table 2.1: Circuits and links for the gearbox depicted in fig. 2.10. Circuits are composed of three elements: a meshing gear pair and a carrier that keep the gears center distance fixed. Links are either central links (like carriers) or planets [17].

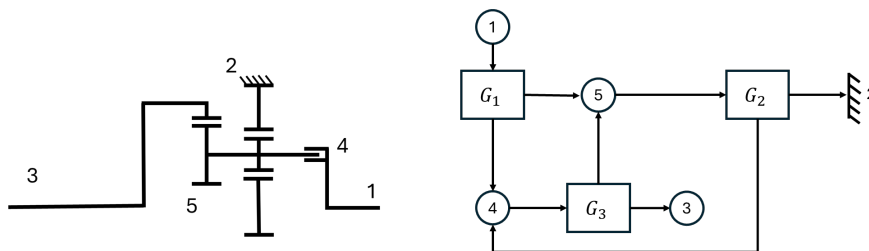


Figure 2.10: The method of Pennestri for calculating the total gearbox efficiency using the individually known meshing efficiencies. The right figure is a power flow representation of the gearbox depicted on the left. table 2.1 explains the circuits labeled a , b , c .

2.4 Torque estimation strategies

2.4.1 Introduction

Several control algorithms for an single-degree-of-freedom rotary actuator exist in the literature, mostly based on the double inertia model, or variants resulting from it. The double inertia model and the variant used in this paper was previously explained in section 2.3.2. The type of controller must be carefully considered, as the use of force sensors or encoders increases the response time of backdrivable actuators, thus reducing HRI compliance [4]. Given the goal of this thesis, this section is dedicated to the existing literature concerning methods of estimating the disturbance (or environmental) torque that a robotic actuator can experience.

2.4.2 Bandwidth

Before diving into the methods of torque sensing, the bandwidth of the double inertia system is investigated to showcase the difficulty of torque sensing that high ratio gearboxes experience. Bandwidth of a mechanical rotational system decreases with the increase of transmission ratio and/or total reflected inertia due of the increase in natural frequency. It can also be shown mathematically by considering the double inertia system without gravity τ_g and friction torque τ_f :

$$J_M \ddot{\theta}_M = -\frac{\tau_s}{i}, \quad (2.6)$$

$$J_L \ddot{\theta}_L = \tau_s - \tau_d, \quad (2.7)$$

$$\tau_s = K_S \left(\frac{\theta_M}{i} - \theta_L \right). \quad (2.8)$$

Transforming eq. (2.6) till eq. (2.8) to the Laplace domain and solving for $\frac{\ddot{\theta}_M(s)}{-\tau_d(s)}$ results in

$$\frac{\ddot{\theta}_M(s)}{-\tau_d(s)} = \frac{1}{\frac{J_M J_L i}{K_S} s^2 + J_M i + \frac{J_L}{i}}. \quad (2.9)$$

The system governed by this transfer function can be interpreted as the frequency response of load-side disturbance onto motor-side acceleration. The bandwidth of this system is defined as the frequency interval over which the attenuation is less than 70.7% in the logarithmic scale. Using this definition, a graph showing the influence of changing the load-side inertia J_L and gear ratio i is depicted in fig. 2.11. This also proves the statement in [4] that low ratios gearboxes have large bandwidth. Considering that the novel gearbox has a relatively large gear ratio of ~ 222 , this could pose a difficulty when designing the control scheme. The greatest influence, however not shown in the graph, is the stiffness parameter K_s . A high stiffness results in a higher bandwidth.

2.4.3 Disturbance observers

In this section, the 'typical' disturbance observer (addressed as the 'LPF DOB', where LPF stands for low pass filter) and a multi-encoder disturbance observer (MEDOB) are explained. Also, the mathematics of the higher-order-sliding-mode (HOSM) observer are highlighted for a specific class of systems. Concerning the latter, [18] explains an implementation for a brushless DC motor (BLDC) and [19] highlights the general theory behind it. The most relevant sections of those papers are reproduced here, but for a more complete view, one can refer to them.

Typical disturbance observers

A disturbance observer is a mathematical tool that can identify the disturbance in a time-invariant system. Consider for this a simple system governed by

$$J \ddot{\theta} = \tau_M - \tau_d, \quad (2.10)$$

with J , θ , τ_M and τ_d equal to an inertia of the motor + load, angle of the motor, the motor torque and disturbance torque respectively. The transfer function relating the motor torque to the output position would yield

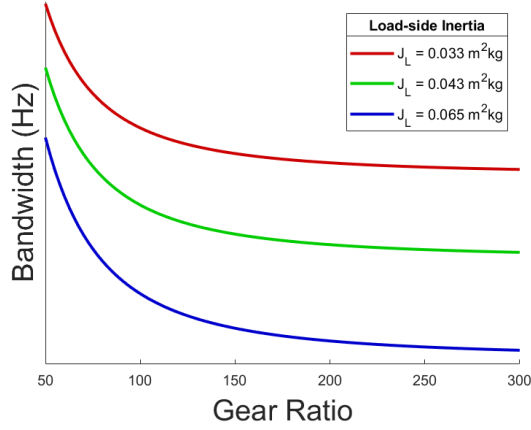


Figure 2.11: Influence of total reflected inertia and gear ratio onto the bandwidth of the double inertia model. The following values are used: $K_s = 1\text{e}4 \text{ Nm/rad}$ and $J_M = 2.09\text{e-}04 \text{ kgm}^2$. Although not shown on the graph, the greatest influence onto the bandwidth is the stiffness hyperparameter K_s . In general, this parameter is unknown and thus the ordinate has no units on its axis. This graph is to solely showcase that large ratio decreases bandwidth.

$$G(s) = \frac{1}{Js^2}. \quad (2.11)$$

The purpose of the DOB is to estimate (in this paper, generally denoted with a hat) the disturbance signal τ_d with the available outputs and inputs of the system: θ and τ_M . This estimated signal can then be compensated for, so that the resulting system becomes linear. In order to do this, the transfer function (or more generally matrix) of the actual plant $G(s)$ must be known. What is commonly done to design such a DOB is to use a reduced-order model $G_n(s)$. For instance, the reduced order-transfer function for eq. (2.11) would be

$$G_n(s) = \frac{1}{Jgs}. \quad (2.12)$$

Frequently, a low pass filter (LPF) is used onto the output of the DOB because noise might be introduced when taking the derivative of a signal. The frequency parameter g is frequently set equal to the cut-off frequency of this low pass filter (LPF). Examples have been described in [12], [20], [13], [21], [22] and [8]. A general scheme for these references is provided in fig. 2.12a. Paper [20] uses a somewhat different approach in that sense that filtering the signal containing the output is performed as well, as can be seen in fig. 2.12b.

MATLAB has implemented an active disturbance rejection control (ADRC) block in Simulink that estimates the real plant dynamics with a first or second order black box approach. An advantage of the ADRC is that the plant dynamics do not have to be known in order to estimate the disturbance.

Multi-encoder-based disturbance observer (MEDOB)

A multi-encoder-based disturbance observer uses two encoders at both the input and output of the gearbox, allowing to estimate the external disturbance τ_d in eq. (2.4) without knowing the

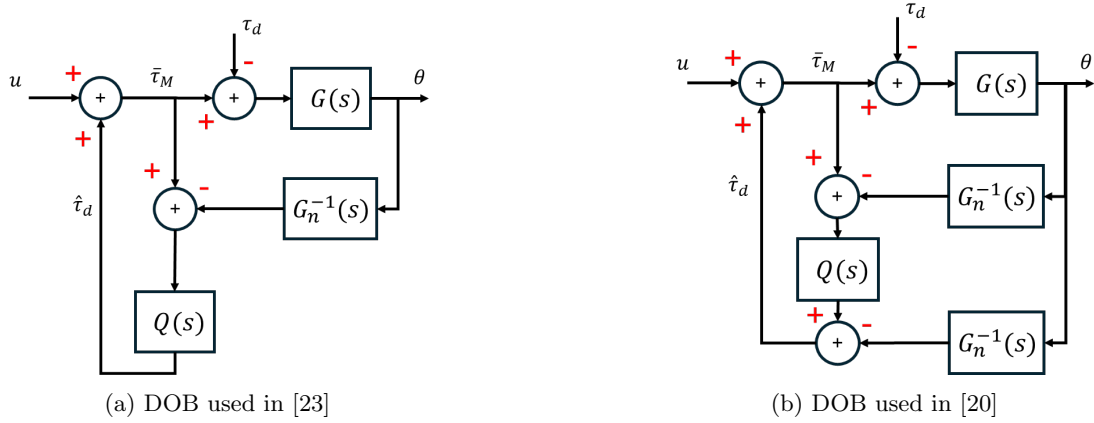


Figure 2.12: Typical disturbance observers (DOB's) used in several papers. The transfer function $G(s)$ represents the actual plant, $G_n(s)$ the lower-order estimated plant and $Q(s)$ a low pass filter (LPF). The signal u is the input with the (estimated) disturbance compensated for. In the rest of this thesis, the DOB in fig. 2.12a will be addressed as the 'LPF DOB'.

stiffness parameter K_s . Consider the double inertia system eq. (2.2) and eq. (2.3) without gravity τ_g . Taking the Laplace transform yields:

$$J_M s^2 \theta_M(s) = \tau_M(s) - \frac{\tau_s(s)}{i} - \tau_f(s), \quad (2.13)$$

$$J_L s^2 \theta_L(s) = \tau_s(s) - \tau_d(s). \quad (2.14)$$

Solving for eq. (2.13) and eq. (2.14) to the disturbance torque $\tau_d(s)$ yields:

$$\tau_d(s) = i\tau_M(s) - iJ_M s^2 \theta_M(s) - J_L s^2 \theta_L(s) - \tau_f(s). \quad (2.15)$$

As one can see, the stiffness K_s is not needed inside this signal. In accordance with the DOB scheme in fig. 2.12b, fig. 2.13 can be build to estimate the disturbance torque τ_d using the MEDOB scheme. The authors of [13] use the speeds ω_M and ω_L as inputs for this MEDOB, whereas the authors of [20] and [22] use the angles θ_M and θ_L as the inputs. The latter also uses a pseudo-differentiator to compute the time-derivative of the angles. If an encoder can measure the speed, it should be exploited.

Variant 1: anti-backlash mechanism

In the next smaller sections, some variants of the typical disturbance observer are highlighted. The gearbox presented in [12] uses an anti-backlash mechanism. As a result, the stiffness K_s has its value equal to the stiffness of the anti-backlash mechanism. It is assumed that the stiffness of the gearbox is much higher than the stiffness of the anti-backlash mechanism, therefore one can neglect the flexing of the gearbox. This is an exception of where the stiffness parameter K_s can be determined.

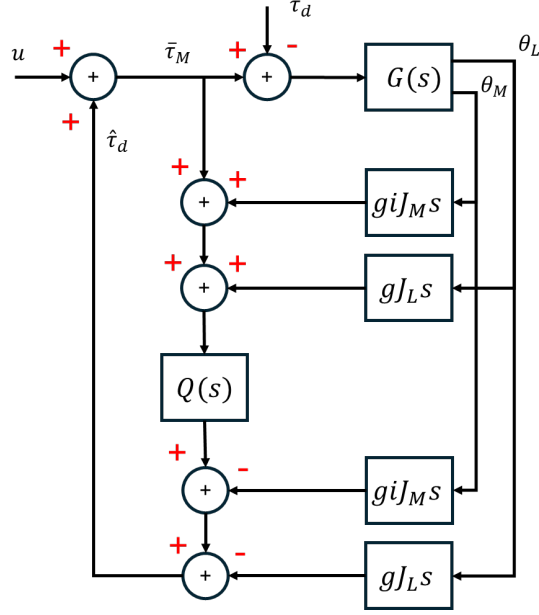


Figure 2.13: Multi-encoder disturbance observer (MEDOB) used in the papers [20], [13] and [22]. The difference with the 'LPF DOB' is that there must be two encoder signals (multi-encoder) present for an actual application. $G(s)$ represents the double inertia system as in fig. 2.8.

Variant 2: exploiting backlash

In [20] backlash is used to detect impact and contact of a link at load-side with the environment faster compared to the disturbance observer. The extra torque due to backlash τ_{bl} can be estimated as

$$\hat{\tau}_{bl}(s) = J_L s^2 \left(\frac{\theta_M(s)}{i} - \theta_L(s) \right). \quad (2.16)$$

This extra torque is then added to the signal of $\hat{\tau}_d$ so that this torque can be fed into the motor reference torque $\bar{\tau}_M$. The authors also expand the double inertia system by adding a disturbance torque to the right side of eq. (2.2).

Variant 3: nonlinear backdrivability improvement

The authors of [20] use an impedance controller with a nonlinear gained estimated external disturbance to ease backdrivability. Due to the inherent backdrivability of the gearbox used in this thesis, diving in such complexity will not be done.

Variant 4: Bilateral drive control

Paper [13] implements a controller to improve backdrivability of the gearbox and to estimate & control the motion of the user without the use of a torque sensor for the Bilateral Drive. Similar to [20], a MEDOB is used rather than estimating the stiffness parameter K_s . The bilateral drive has a large gear ratio, thus when 'in operating' mode, the motor torque reference $\bar{\tau}_M$ is set to the estimated disturbance torque $\hat{\tau}_d$ divided by the gear ratio i :

$$\bar{\tau}_M = -\frac{1}{i} \hat{\tau}_d \quad (2.17)$$

The mode identification (either 'in operation' or 'stopped') is done using a machine learning algorithm that determines the current mode based on the measurements of a nonlinear support vector machine at the motor-side.

Variant 5: resonance ratio control

Paper [21] models a general actuator system as a two-mass-spring-system between motor and load. Due to the elasticity of the system, resonance might occur at certain frequencies. The authors in this paper then propose a resonance ratio control, instead of the conventional PID controller, to guarantee robustness and to suppress oscillations of the system.

Variant 6: back-drive assist control

[22] uses a back-drive assist control based on feedforwarding eq. (2.16) into the motor torque $\bar{\tau}_M$. It is called an 'assisted control based on angular acceleration', due to the second derivative s^2 . Experimentally, it is shown that this type of control can detect smaller reaction torques by contact with the environment, compared to the measurement of a MEDOB. This method also achieves a smaller load-side equivalent inertia and assists in the start of driving.

Variant 7: equating forward- and backward drive inertia

[8] builds transfer functions for both forward-drive and backward-drive inertia. A controller is then developed in such a way that the motion characteristics of forward- and backward-drive are the same to improve overall backdrivability. Additionally, a reaction force observer (RFOB) is implemented to account for all disturbance torques. Thus, this method does not require a friction torque model as in fig. 2.14.

Conclusion, similarities and differences between the model variants

All papers listed above ([12], [20], [13], [21], [22] and [8]) have in common the absence of need for a torque sensor. Papers [20], [13] and [22] all use a friction feedforward controller based on experimental data. Such an experimental data graph is shown in fig. 2.14. A dead zone is implemented around 0rad/s to make sure the actuators do not feedforward a friction torque due to viscous friction which is not present. It is indeed possible that encoders can give a non-zero speed indication when in reality the gearbox is not turning.

Higher order sliding mode observer

Another approach for the estimation of unknown input signals (disturbances) is the use of a HOSM controller in combination with a Luenberger observer. The authors of [18] have successfully implemented this to estimate the disturbance onto the dynamics of a BLDC motor. The general theory behind this HOSM DOB is explained in [19], but the most relevant information is repeated here for a single-input-single-output, linear system with 2 states in state space form:

$$\begin{cases} \dot{x} = Ax + Bu + Dw \\ y = C_{out}x \end{cases} \quad (2.18)$$

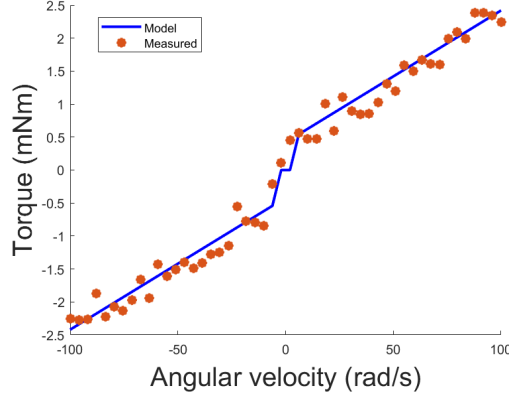


Figure 2.14: A theoretical friction model used in [20], [13] and [22]. The friction model is measured using experimental data. Notice that only viscous friction can be modeled with this method. A dead zone around 0rad/s is implemented to make sure the controller does not actuate when speed could be non-zero. This could be due to the noise introduced by the encoder for example.

Where A , B , D and C_{out} are system matrices, x is the state vector, y is the output and w is the disturbance that one wants to estimate in real-time. Notice that the output is a direct function of the states and not of the input u . First, all states are estimated with the use of a Luenberger observer:

$$\dot{z} = Az + Bu + L(y - C_{out}z), \quad (2.19)$$

where z and L are the state estimates and the matrix of the observer respectively. Due to the non-zero disturbance signal w , this state estimation z will not converge to the actual states of the system x . The high-order sliding mode (HOSM) differentiator ensures the finite time convergence of the resulting estimation error $z - x$ to zero. The improved estimation of the states

$$\hat{x} = z + Fv \quad (2.20)$$

will drive this error to zero. The signal v is updated by means of a higher-order-sliding-mode differentiator:

$$\begin{cases} \dot{v}_1 = -\alpha_3 M^{1/3} |v_1 - (y - C_{out}z)|^{2/3} \text{sign}(v_1 - (y - C_{out}z)) + v_2 \\ \dot{v}_2 = -\alpha_2 M^{1/2} |v_2 - \dot{v}_1|^{1/2} \text{sign}(v_2 - \dot{v}_1) + v_3 \\ \dot{v}_3 = -\alpha_1 M \text{sign}(v_3 - \dot{v}_2). \end{cases} \quad (2.21)$$

[19] explains that setting the parameters $\alpha_1 = 1.1$, $\alpha_2 = 1.5$ and $\alpha_3 = 2$ works well for systems with less than or equal to six states. The matrix gain F in eq. (2.20) can be calculated by inverting the observability matrix of the observable system:

$$F^{-1} = \begin{bmatrix} C_{out} \\ C_{out}(A - LC_{out}) \end{bmatrix}. \quad (2.22)$$

The signal $v = [v_1, v_2, v_3]^T$ can be used to reconstruct/estimate the disturbance signal w as

$$\hat{w} = (C_{out}AD)^{-1}(v_3 - (a_1v_1 + a_2v_2)), \quad (2.23)$$

where the parameters a_1 and a_2 are the coefficients of the expression

$$(-1)^2 \det(A - LC_{out} - sI) = s^2 - a_2s - a_1. \quad (2.24)$$

This HOSM DOB is a continuous time controller. [19] also explains that for sufficiently small sampling period T_s , the HOSM part can be discretized using the method of Euler. The disturbance estimator scheme for the general linear system of eq. (2.18) is depicted in fig. 2.15.

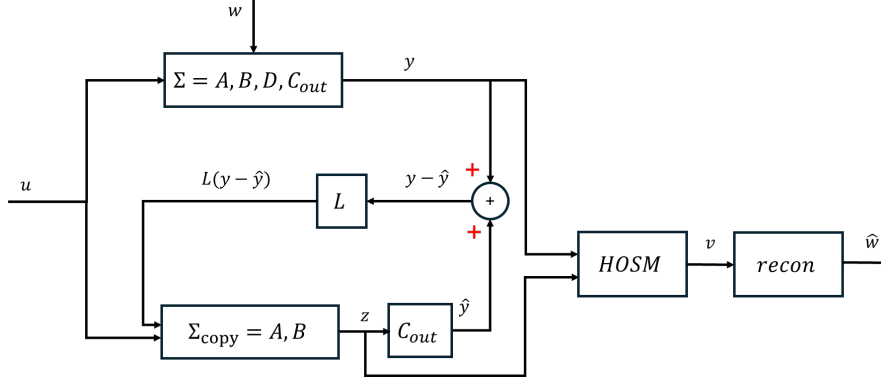


Figure 2.15: Disturbance observer using a Luenberger observer and a higher-order-sliding-mode (HOSM) differentiator to estimate the disturbance for a general linear system in continuous states space representation. The system Σ is governed by eq. (2.18). The HOSM-block implements the solution of the nonlinear higher-order-sliding-mode differentiator eq. (2.21) and the reconstructor block *recon* implements the estimation of the disturbance by applying eq. (2.23).

2.4.4 Measuring torque with a torque sensor

Several types of torque sensors are described in table 2.2 and listed hereunder, as well as some advantages and disadvantages. The table does not provide a comparison between the devices, but rather a list of (dis)advantages of each technology. As it is rather difficult to find state-of-art information about an actual quantitative analysis across these methods.

- In the rotary torque sensors, a lower stiffness part of the shaft twists when torque is applied. The displacement is measured with an (optical) encoder, which is proportional to that torque [24], [25].
- In a surface-bonded strain gauge sensor a strain gauge is bonded to the rotating shaft that slightly deforms when torque is applied. This gauge is usually in the form of a balanced Wheatstone bridge. When torque is applied, the bridge becomes unbalanced and a voltage can be measured which is proportional to that torque [25], [26].
- Capacitive force sensors work on the principle that the capacitance of two conductive elements changes when the distance between varies. This is measured using oscillators, switched-capacitor circuits or capacitive bridges [25], [27], [28].

- Piezoelectric torque sensors have piezoelectric materials inside. When exposed to physical stress, the material generates a measurable voltage [25], [29].
- Magnetoelastic sensors are based on the Villari effect. It is the phenomenon of a change in magnetic state under the influence of the mechanical stress. Sensor coils located in close proximity to the magnetically encoded region sense this change of magnetic field [25], [30].
- In the rotary transformer torque sensors, there is one stationary transformer coil and another is fixed on the rotating shaft. The rotation of the shaft induces a voltage in the stationary transformer, which is proportional to the torque applied [31].
- The transense SAW technology has made a innovative torque sensor which can be found in [32].

2.4.5 Measuring torque with current

Typically, the current inside any electric motor is measured. This signal then provides useful information to estimate the torque generated by the motor and is measured anyway for the torque closed-loop control of a commercial motor. [33] proposes a real-time torque measurement based on measuring only one out of three phase currents of a BLDC motor. Even for constant torque and power, this signal will be a periodic time signal. The formula for the estimated torque is

$$\hat{\tau}_M = ||2K_t i_p| - 0.75\alpha| + 0.75\alpha. \quad (2.25)$$

Where K_t and i_p are the torque constant and an instantaneous single phase current measurement respectively. α is the average value of electromagnetic torque over a number N of previously recorded sampling points:

$$\alpha = \frac{1}{N} \sum_N \hat{\tau}_M. \quad (2.26)$$

N must be defined with care. Small values for N result in non-smooth measurements, whereas large values for N result in slower real-time estimation.

	Rotary	Strain gauge	Capacitive	Piezoelectric
Price	cheap	cheap	expensive	expensive
Lifetime	till breakage	till breakage		
Use				
Bandwidth	reduced	reduced	no impact	high freq only
Stiffness	reduced	no impact	no impact	
Static Load	difficult	impossible	possible	possible
Size	large	small	small	small
T-dependent	yes	yes		compensation
Noise			resistant	
Disturbance				compensation
Resolution	encoder dep.	voltage dep.		
Contact?	needed	needed	not needed	needed
Accuracy				precise
	Magnetoelastic	Transformer	SAW	MEM's
Price	expensive		expensive	
Lifetime				
Use		difficult	easy	for testing
Bandwidth	no impact		no impact	
Stiffness			no impact	
Static Load	difficult			
Size	small	large	small	small
T-dependent	yes		T-measurement	
Noise	linear		immune	
Disturbance	robust			
Resolution	high			
Contact?	not needed	not needed	not needed	
Accuracy	low hysteresis	precise	precise	

Table 2.2: Comparison between 8 types of torque measurements. The table provides an enumeration of (dis)advantaged. The words 'till breakage' means that the device should keep its performance until the shaft brakes. 'dep.' means dependent, so 'encoder dep.' means that the resolution of the device depends on the encoder that is integrated with the device. 'no impact' means that installing the device does not result in a change in bandwidth of the entire driveline. 'linear noise' means that the noise can be filtered out. This is because the transfer function of the device is linear.

Chapter 3

Model & Control of the Novel Wolfrom-based Gearbox

3.1 Introduction

Firstly, a model for the drivetrain/actuator system will be formulated, based on the previously explained double inertia model. Secondly, a control algorithm for a link connected to this actuator is developed. It is then controlled in an HRI environment. This controller + actuator system will be implemented in MATLAB/Simulink to perform simulations and test the performance of the control in terms of position tracking and safety against collisions with humans. The parameters of the scheme will be chosen to make sure that the link, which in this case represents the robot, is in accordance with the ISO 15066 norm. The real-time implementation and experimental validation are presented in chapter 4.

3.2 ISO regulation for robotic devices and cobots

The information needed to perform simulations and tests from the ISO 15066 regulation regarding safety in robots and cobots is reproduced here. Regarding this standard, there are two types of contact: a transient type, where the human is impacted by a moving part of the robot/cobot, and a quasi-static type, which includes clamping or crushing situations. For each of these cases, the ISO standard provides maximum values for the force and/or pressure of robot-human collision, as depicted in fig. 3.1. The values for F_T , p_T , F_S and p_S depend on the specific body area of impact and are provided in appendix B, fig. B.1. The maximum relative speed $v_{rel,max}$ between a human and the robot/cobot is also provided by the standard:

$$v_{rel,max} = \frac{F_T}{\sqrt{\mu K}} = \frac{p_T A_s}{\sqrt{\mu K}}. \quad (3.1)$$

With K equal to the effective stiffness of the human body model and A_s the surface area of contact. This value can be found in tables such as fig. B.1 [34], provided in appendix B. μ is the reduced mass between human and robot/cobot:

$$\mu = \frac{1}{\frac{1}{m_H} + \frac{1}{m_R}}, \quad (3.2)$$

where m_H is the effective mass of the human body part, determined from fig. B.2, in appendix B. m_R is the effective mass of the robotic link:

$$m_R = \frac{M}{2} + m_L, \quad (3.3)$$

with M and m_L equal to the total mass of moving parts of the robot/cobot and the effective mass of payload respectively. The standard also provides an expression for the maximal permissible energy transfer between robot/cobot and human:

$$E = \frac{1}{2} \mu v_{rel,max}^2. \quad (3.4)$$

These maximum energy values E are provided in appendix B, in fig. B.3. These equations lead to three requirements that must be fulfilled in simulation, depending on what body part is impacted.

- The speed on impact must be below the value determined from eq. (3.1), when assumed that the body part is stationary.
- The force on impact cannot exceed the values in fig. B.1, depending on the part that is hit by the robot/cobot (transient region). Also, the forces after collision must be below the quasi-static limit.
- Detection time of human collision must be smaller than 0.5s, as provided by [34]. This is an important requirement, but is only highlighted in the validation section.

The norm does not provide a value for the damping factor of the human body part, but a good estimate will improve the simulation model. For instance, the relative damping factor of $\xi = 40\%$ is assumed based on the findings of viscoelastic properties of human facial skin [35]. A 'human' damping factor C can be determined based on this relative damping factor:

$$C = 2\xi\sqrt{Km_H}. \quad (3.5)$$

3.3 Impedance control for safe HRI

To ensure safe physical interaction between humans and robots, impedance control is often used to shape the dynamic response of the system to external forces [36].

Impedance control uses a combination of feedforward and PD control to achieve desired system behavior depending on external disturbances. On the other hand, the computed torque method (CTM) uses a combination of feedback linearization, feedforward and a PID controller to achieve position, speed and/or acceleration control. Impedance control is used more frequently where interaction with the environment could possibly occur, whereas CTM focuses more on robust position control, making it more dangerous in human environments [37].

The desired stiffness K_d , damping C_d and inertia J_d of the actuator should be chosen according to the environmental disturbances. The subscript 'd' stands for 'desired' and these parameters will be addressed as the 'virtual parameters' of the impedance control. Impedance control forces an actuator to follow a desired position and/or velocity when not constrained by the environment. An example where the environment changes from unconstrained to constrained in HRI is when a robotic link hits a human (part).

Two types of impedance control can be implemented. In what follows, the mathematics of this control strategy for a single-degree-of-freedom rotational joint is explained, for both types.

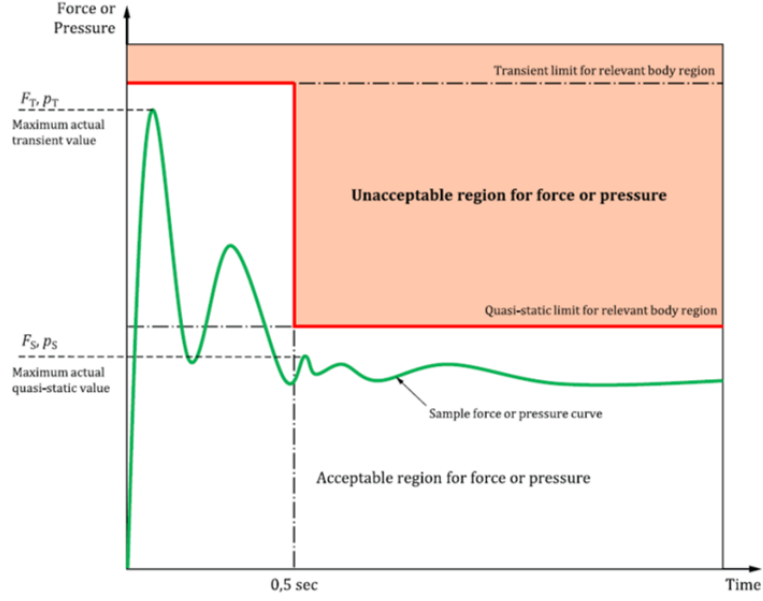


Figure 3.1: Graphical representation of the acceptable and unacceptable forces or pressures on impact (transient region) and constant forces/pressures (quasi-static region).

Consider the simplified double inertia system, without the modeling of backlash, and reflected to the load-side L :

$$J\ddot{\theta}_L + i\tau_f + \tau_d + \tau_g = i\tau_M. \quad (3.6)$$

The parameters are the same as before and $J = i^2 J_M + J_L$ is the total load-side reflected inertia of the entire drivetrain. J_M is the sum of inertia's of all components at the motor-side (sun-side) and J_G is the sum of all components at load-side of the gearbox + the gearbox reflected inertia itself. In eq. (3.6), τ_d represents the environmental torque (disturbance torque). It could represent a human trying to rotate the link or a reaction torque due to collision. When τ_d is zero, the desired position and/or speed should be reached and when τ_d is non-zero, the kinematics ($\dot{\theta}_L$, θ_L , θ_L) should be commissioned by that disturbance. Type one impedance control sets the motor reference torque $\bar{\tau}_M$ equal to [36]:

$$i\bar{\tau}_M = \frac{J}{J_d}(J_d\ddot{\theta}_L + C_d\dot{\theta} + K_d\tilde{\theta} - \hat{\tau}_d) + i\hat{\tau}_f + \hat{\tau}_g + \hat{\tau}_d. \quad (3.7)$$

Type two sets the motor reference torque equal to:

$$i\bar{\tau}_M = \frac{J}{J_d}(J_d\ddot{\theta}_L + C_d\dot{\theta} + K_d\tilde{\theta}) + i\hat{\tau}_f + \hat{\tau}_g. \quad (3.8)$$

$\tilde{\theta} = \bar{\theta}_L - \theta_L$ is the virtual position (error signal) and $\bar{\theta}_L$ is the reference signal. When the motor time constant is considered small, the actual motor torque can be assumed equal to the reference torque, $\tau_M = \bar{\tau}_M$ and assuming perfect modeling, cancellation can be achieved, resulting in: $\tau_f = \hat{\tau}_f$, $\tau_g = \hat{\tau}_g$, $\tau_d = \hat{\tau}_d$. Then, inserting eq. (3.7) and eq. (3.8) into eq. (3.6) yields

$$J_d\ddot{\theta} + C_d\dot{\theta} + K_d\tilde{\theta} = \tau_d \quad (3.9)$$

for type one, and

$$J_d \ddot{\tilde{\theta}} + C_d \dot{\tilde{\theta}} + K_d \tilde{\theta} = \frac{J_d}{J} \tau_d \quad (3.10)$$

for type two. These expressions explain how the resulting dynamics act when the drivetrain is constrained by the environment. Both types prove that when τ_d is zero, the virtual position $\tilde{\theta}$ goes to zero and when non-zero, the virtual position is imposed by that torque τ_d [38]. The difference, however, lies in the coefficient in front of the disturbance torque τ_d . For type one, the disturbance torque must be measured (for instance, using a torque sensor) or estimated (for instance, using a disturbance observer) or a combination of both (redundant measurement), whereas type two does not require a model for the estimated disturbance. Therefore, type two has an easier implementation. For type one however, one can choose each of the three virtual parameters independently. This can be explained by comparing the Laplace transform of the resulting dynamics eq. (3.9) to the general expression for a second-order low pass filter transfer function:

$$\frac{\tilde{\theta}(s)}{\tau_d(s)} = \frac{1}{J_d s^2 + C_d s + K_d} = \frac{A_d}{\frac{s^2}{\omega_d^2} + \frac{2\xi_d}{\omega_d} s + 1}, \quad (3.11)$$

where A_d , ω_d and ξ_d are the desired (virtual) amplitude, natural pulsation, and damping ratio respectively. When introducing the desired inertia ratio $r_d = J/J_d$, one can rewrite the natural pulsation as

$$\omega_d = \sqrt{\frac{r_d}{A_d J}}. \quad (3.12)$$

When rearranging the Laplace transform of eq. (3.9) and eq. (3.10) to eq. (3.11), one can see that only two out of three virtual parameters can be chosen independently from each other for type two; whereas all three can be chosen independently for type one:

$$J_d = \frac{J}{r_d}, \quad (3.13)$$

$$C_d = 2\xi_d \sqrt{\frac{J}{r_d A_d}} \quad (3.14)$$

and

$$K_d = \frac{1}{A_d}. \quad (3.15)$$

Thus, eq. (3.13), eq. (3.14) and eq. (3.15) provide a way to transform the more intuitive parameters A_d , r_d and ξ_d to the virtual parameters J_d , C_d and K_d in an automatic way. These formulas can be used for a better way of fine-tuning the virtual parameters for a drivetrain when performing experiments as explained in the next bullet points.

- In order to have 'smooth' control, the desired damping coefficient ξ_d is, for instance, set to 99%. This will result in a critically dampened system when the robotic link is backdriven.
- The amplitude parameter A_d will impose the angle over which the link drives back when a disturbance of $\tau_d = 1Nm$ is applied. Choosing this parameter is especially difficult when the gear ratio i is large because it depends on the bandwidth of the disturbance observer. When the DOB gives a (small) false non-zero result, the link will not reach the desired

position in case of large virtual amplitude, because of the high reduction ratio of the gear element. In addition, in real-life applications, stiction plays an important role for high A_d values, as will be highlighted in chapter 4. Choosing A_d too low results in a stiff and thus less safe control in the vicinity of humans.

- The inertia ratio r_d has a large influence on the impact force. The higher this ratio, the lower the impact force. Thus, this value is best chosen as high as possible. However, it is limited by the real-time implementation as will be explained in chapter 4.

3.4 Friction model

3.4.1 Introduction

A friction model is build to characterize the gearbox performance. This model can be used to estimate the influence of the friction term τ_f onto the dynamics of eq. (3.6). In this paper, it was first assumed a scalar function $\hat{\tau}_f(\omega_M, \tau_d)$. The speed of the motor ω_M is available from the Hall sensors inside the BLDC motor, whereas τ_d is not. The most used type of friction model consists in the sum of coulomb friction and viscous friction. It was initially though that this model would not be sufficient, so an extra load-dependent loss that depends on the backdriving torque (in this case the disturbance torque τ_d) was added. When performing tests, a simple viscous and coulomb model seemed to suffice, so the load-dependent loss was discarded, as will be explained later in this section. The friction contributions that have been characterized for the novel Wolfrom-based gearbox are listed here.

- Stiction is the torque needed to initiate motion at zero speed. For backdrivable gearboxes, it is a quantity that should be measured from both the input side as well as the output side. This quantity is simply measured by linearly increasing the sun/ring torque and measuring the ring/sun speed. The stiction torque is then estimated as the torque at which the speed starts to rise parabolically.
- The coulomb friction is the constant torque that the motor must deliver when turning. This friction torque typically solely depends on the sign of the speed and is therefor called a speed-independent loss.
- Viscous friction is the speed-dependent torque and is most of the time linear with speed.
- Rolling and sliding torque losses are meshing losses proportional to the torque acting on the mating gears [14] and would be load (τ_d)-dependent.

At first, the idea was to use the data collected from measurements to build an accurate model that could predict stiction, viscous and coulomb friction as well as load-dependent losses. This model could then be used as a feedback linearization model in the real-time control. However, once tests were performed with the impedance control (see chapter 4), a simple viscous and coulomb model seemed to be sufficient. The advantage of this simple model is the fact that fewer parameters must be fine-tuned, once implemented on a test bench. Additionally, in an early stage of this master thesis, a model was proposed to incorporate the losses due to backlash in planetary gears. An explanation on how this could have been done is explained in appendix A. It is further not used as it drastically reduces the computation performance.

3.4.2 Empirical way of designing the friction model

In this section, an empirical way of designing the friction model is presented. This model is not used in the experimental validation because of its high complexity, as explained earlier. Since the analytical expressions for coulomb friction, viscous friction and meshing losses are known, the coefficients can be determined from experimental data with a fitting algorithm based on the least-squares method. This allows to rapidly and generically update the coefficients in case new data would become available. The model of the friction torque for this gearbox is the scalar function

$$\hat{\tau}_f(\omega_M, \hat{\tau}_d) = c_{coulomb} + c_{visc}\omega_M + c_{mesh} \frac{\hat{\tau}_d}{i}. \quad (3.16)$$

Where $c_{coulomb}$, c_{visc} and c_{mesh} are the coulomb, viscous and meshing torque friction coefficients respectively. Since the disturbance torque cannot be measured, the estimate must be used instead.

The stiction torque is not modeled in this expression because it is nonlinear. However it can be measured and added to the model as a subsystem. This subsystem would take in a torque and have its output equal to the same value, except for when the input is smaller than the stiction torque. Some models incorporate this via a Stribeck contribution, an example is explained in [39].

The coefficients are determined with a single, long running experiment on a test bench. The experimental setup comprises a BLDC motor connected to the sun-side of the gearbox and is controlled in speed. A larger BLDC motor is connected to the ring-side and is controlled in torque, thus serving as a load. Two torque sensors are connected to the sun- and the ring-side, as depicted in fig. 3.2.

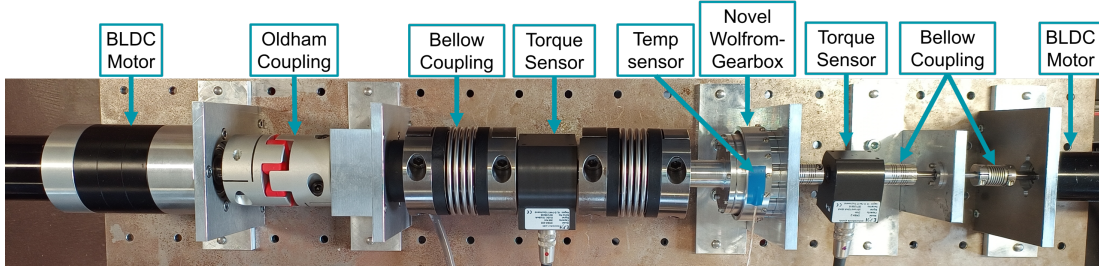


Figure 3.2: Experimental setup for the determination of the friction model $\hat{\tau}_f(\omega_M, \hat{\tau}_d)$. The smaller BLDC motor is connected to the sun-side of the gearbox via a bellow coupling and the larger BLDC motor to the ring-side.

To collect data for the determination of the coefficients in eq. (3.16), some couples of (ω_M, τ_d) should be measured. An automatic approach consists in sending speed steps as a reference signal to the small BLDC drive and at the same time, sending smaller torque steps as reference signal to the drive of the larger BLDC motor. An example of such reference signals is explained in fig. 3.3. At the same time, the torque signals from the torque sensors are captured.

Experiments pointed out that the friction model coefficients depend on the sign of ω_M and τ_d . This means that a distinction between the clockwise and counterclockwise motion must be made (sign of ω_M), as well as a distinction between power flow (sign of ω_M times τ_d). This could be explained by the fact that helical gears are used inside the gearbox. For right-handed helical gears, when the driver rotates clockwise, it pushes the driven gear in one axial direction. When

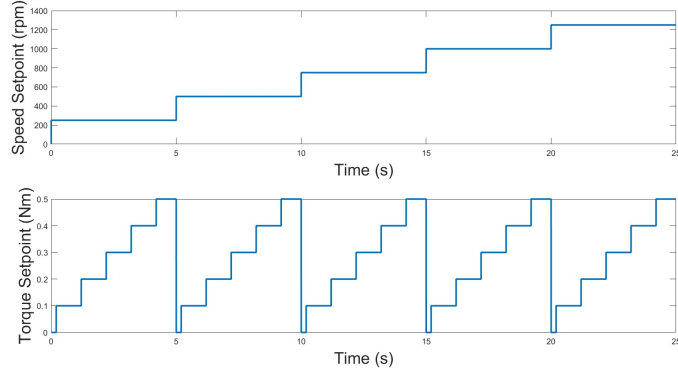


Figure 3.3: Reference signals for the speed controlled BLDC at the sun-side and the torque controlled BLDC at the ring-side of the experiment for the determination of the friction model. For each speed step, the torque step must be long enough to minimize the influence of inertia. In reality however, these steps should be smaller, and in a higher amount, to have a sufficient amount of data points.

the rotation direction reverses, the axial direction flips as well. This change in axial force might be responsible for the efficiency difference in rotation direction [40].

An explanation of how the coefficients for the friction model eq. (3.16) are determined in an automatic way can be reviewed in fig. 3.4. The signals $\bar{\omega}_M(t)$ and $\bar{\tau}_d(t)$ are the reference signals from fig. 3.3 that are send to the small and larger BLDC motor of the experimental setup fig. 3.2 respectively. From the small torque sensor and the larger torque sensor, the torques $\tau_M^{meas}(t)$ and $\tau_d^{meas}(t)$ can be extracted respectively. The time-dependent friction $\tau_f^{meas}(t)$ can be computed as $\tau_M^{meas}(t) - \tau_d^{meas}(t)$. Some post-processing with a non-causal Butterworth filter is needed to extract meaningful information. The non-causal filter will make sure that no delay is introduced into the measurements. Taking the pseudo-inverse of the transpose of the filtered time signals will yield the coefficients of the friction model eq. (3.16). Since the friction model depends on the sign of speed and environmental torque, the data gathered in the block 'Friction torque time signal' must be split into four parts: quadrant Q_1 where $\bar{\omega}_M > 0$, $\bar{\tau}_d > 0$; quadrant Q_2 where $\bar{\omega}_M < 0$, $\bar{\tau}_d > 0$; quadrant Q_3 where $\bar{\omega}_M < 0$, $\bar{\tau}_d < 0$ and quadrant Q_4 where $\bar{\omega}_M > 0$, $\bar{\tau}_d < 0$.

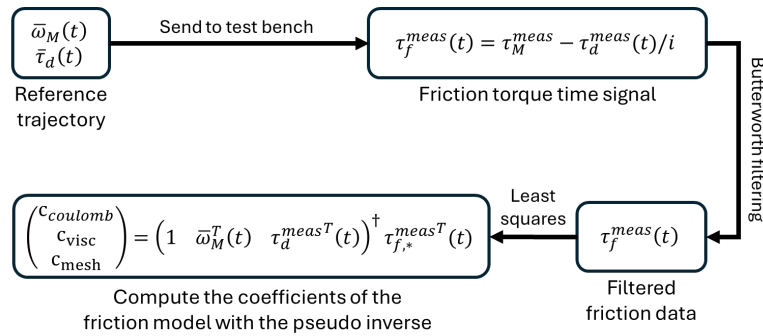


Figure 3.4: A summary of the method used to determine the coefficients of the friction model.

As already explained, this model seemed too complicated for real-time implementations because of its many parameters to be fine tuned on the actual drivetrain. Also, because of the four-quadrant approach, a discontinuity arises around the lines $\{\omega_M \in \mathbb{R} : \tau_d = 0\}$ and $\{\tau_d \in \mathbb{R} : \omega_M = 0\}$, when crossing quadrant boundaries.

3.5 Inertia & dynamics

For the impedance control eq. (3.7) to work properly, the total load-side reflected inertia J must be known accurately. In this section, the load-side reflected gearbox inertia J_g is calculated, taking into account the inertia's of the components of the gearbox. This expression can then be used to determine J as

$$J = i^2 J_M + (J_g + J_l) = i^2 J_M + J_G, \quad (3.17)$$

where J_l is the inertia of the robotic link at the load-side, and any other element that is connected to it. The configuration of the gearbox is depicted in fig. 3.5 and will be used to derive an analytical formula for the load-side reflected inertia J_g .

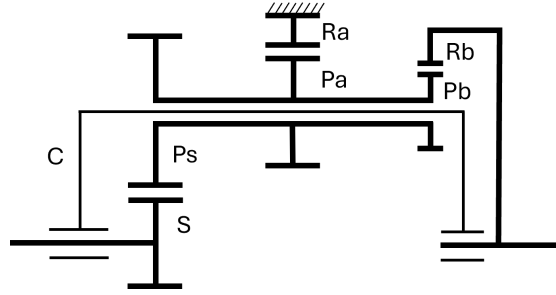


Figure 3.5: Configuration of the novel Wolfrom-based gearbox. S is the driven sun (motor-side) and Rb is the driving ring (load-side).

The inertia's of the separate components of the gearbox can be determined from CAD files. However, these values must be weighted with coefficients that depend on the gearing ratio between them. To this end, the Willis-equations can be written as

$$\frac{\omega_S - \omega_C}{\omega_{Ps} - \omega_C} = -\frac{Z_{Ps}}{Z_S}, \quad (3.18)$$

$$\frac{\omega_{Pa} - \omega_C}{\omega_{Ra} - \omega_C} = \frac{Z_{Ra}}{Z_{Pa}} \quad (3.19)$$

and

$$\frac{\omega_{Pb} - \omega_C}{\omega_{Rb} - \omega_C} = \frac{Z_{Rb}}{Z_{Pb}}. \quad (3.20)$$

The ring Ra is stationary and the three planetary stages are rigidly connected to each other. This implies that $\omega_{Ps} = \omega_{Pa} = \omega_{Pb}$, thus eq. (3.18) till eq. (3.20) can be solved for $\frac{\omega_C}{\omega_{Rb}}$, $\frac{\omega_{Ps}}{\omega_{Rb}}$ and $\frac{\omega_S}{\omega_{Rb}}$. Substituting these expressions into the formula for the reflected inertia

$$J_g = \left(\frac{\omega_C}{\omega_{Rb}}\right)^2 J_C + \left(\frac{\omega_{Ps}}{\omega_{Rb}}\right)^2 (2J_{Ps} + 4(J_{Pa} + J_{Pb})) + \left(\frac{\omega_S}{\omega_{Rb}}\right)^2 J_S + J_{Rb}, \quad (3.21)$$

results in an expression for the gearbox load-side reflected inertia:

$$J_g = i_0^2 J_C + (i_a i_w)^2 (2J_{Ps} + 4(J_{Pa} + J_{Pb})) + i^2 J_S + J_{Rb}. \quad (3.22)$$

Where J_j represents the mass moment of inertia (determined from CAD files) around the rotational axis in a fixed frame of component j . This can be either S , Ps , Pa , Pb , Ra or Rb . i_w and i_0 are separate gear ratio's as in [7]:

$$i_w = \frac{1}{1 - \frac{Z_{Pb} Z_{Ra}}{Z_{Pa} Z_{Rb}}}, \quad (3.23)$$

$$i_0 = 1 + \frac{Z_{Ra} Z_{Ps}}{Z_{Pa} Z_S} \quad (3.24)$$

and i_a is a newly defined gear ratio:

$$i_a = 1 - \frac{Z_{Ra} Z_{Rb}}{Z_{Pa} Z_{Rb}}. \quad (3.25)$$

Where Z_j is the number of teeth of component j .

3.6 Disturbance observer

3.6.1 Overview of the observers

In order to use the second type of impedance control eq. (3.7), an estimate $\hat{\tau}_d$ for the disturbance torque τ_d must be available as a real-time signal. Based on the different types of disturbance observer listed in section 2.4.3, four different types of DOB's are build and listed below. A comparative study in terms of impact estimation is performed in the experimental validation section, chapter 4.

- The first disturbance torque observer is based on fig. 2.12a, where the speed and acceleration are estimated with a Tustin derivative with low pass filter from the measurement of an encoder placed at the load-side of the gearbox. In what follows, it will be called the 'load LPF DOB'.
- A second DOB only uses the Hall sensors inside the BLDC to measure position and hence has the advantage that no extra encoder must be placed at the load-side of the gearbox, lowering cost, volume and mass even more. This is interesting for a commercial integrated actuator. In this document, it is called the 'motor LPF DOB'.
- The third DOB implements a Luenberger observer to estimate the speed and a HOSM differentiator to estimate the influence of disturbance onto that measurement. Rather than linearizing the dynamics, a feedback linearization is employed. Based on the general HOSM observer theory derived in [19] and [18], a disturbance observer for the novel Wolfrom-based gearbox is derived. This scheme uses the encoder at the load-side to estimate the speed, but the speed measurement from the BLDC motor can be used as well. This will be called the 'HOSM DOB'.
- The last DOB uses the multi-encoder scheme of fig. 2.13, called the MEDOB, requiring both the load-side encoder as the position measurement from the BLDC.

Some of these disturbance observers require the acceleration of the motor. Since only the position is known (at load-side), a double derivative must be implemented. A simple Euler derivative to estimate the acceleration from position has a useless signal-to-noise-ratio [41]. That is why a double Tustin derivative with LPF is used to estimate the speed and acceleration from a position measurement. It comes down to discretizing the plant-expression for a derivative in Laplace transform s multiplied with the expression of a first-order LPF ($\frac{1}{Ts+1}$) using the causal Tustin discretization technique:

$$s = \frac{2}{T_s} \frac{z-1}{z+1}. \quad (3.26)$$

This yields an expression for the transfer function $D(z)$ that can be used to estimate speed from position and acceleration from speed:

$$D(z) = \frac{2z-2}{(2T+T_s)z+(T_s-2T)}. \quad (3.27)$$

T is the time-constant of the first-order LPF and T_s is the sample time. It was experimentally validated that $D(z)$ yields faster speed and acceleration measurements from a real-time position signal compared to the (simpler) Euler derivative with LPF.

3.6.2 Analytical expression for the HOSM DOB

Feedback linearizing the friction model $\hat{\tau}_f$ and gravity term $\hat{\tau}_g$ to the measurement of the torque signal τ_M yields a new input

$$u = \hat{\tau}_M - \hat{\tau}_f - \frac{1}{i} \hat{\tau}_g \quad (3.28)$$

that makes the system dynamics of eq. (3.6) linear. It allows the latter to be written as a continuous, state space representation:

$$\frac{d}{dt} \begin{bmatrix} \theta_L \\ \dot{\theta}_L \end{bmatrix} = \begin{bmatrix} 0 & 1 \\ 0 & 0 \end{bmatrix} \begin{bmatrix} \theta_L \\ \dot{\theta}_L \end{bmatrix} + \begin{bmatrix} 0 \\ \frac{i}{J} \end{bmatrix} u + \begin{bmatrix} 0 \\ -\frac{1}{J} \end{bmatrix} \tau_d. \quad (3.29)$$

Since the encoder at the load-side only measures position, the output of the system is

$$\theta_L = \begin{bmatrix} 1 & 0 \end{bmatrix} \begin{bmatrix} \theta_L \\ \dot{\theta}_L \end{bmatrix}. \quad (3.30)$$

This means, following the formula of eq. (2.24), that $a_1 = -l_2$ and $a_2 = -l_1$ where l_i are the individual elements of the Luenberger observer matrix gain L . By substituting the system matrices of eq. (3.29) into eq. (2.23), an expression for the estimated disturbance torque $\hat{\tau}_d$ can be computed for:

$$\hat{\tau}_d = -J(v_3 + l_2 v_1 + l_1 v_2). \quad (3.31)$$

A general block scheme of the HOSM DOB for this plant was previously given in fig. 2.15, and can be directly applied to the Wolfrom-based gearbox when one considers that the blocks *recon* correspond to eq. (3.31) and *HOSM* to eq. (2.21). Since the BLDC motor is equipped with Hall sensors, speed can be measured with sufficient accuracy. However, as will be explained in section 3.7, the HOSM DOB should estimate a disturbance faster compared to the simpler DOB scheme of fig. 2.12a in case the load-side encoder is used to calculate the speed. It shows the advantage of the HOSM DOB. The disadvantage, however, is its larger scheme and more parameters to finetune.

3.7 Simulations of the drivetrain

3.7.1 Drivetrain model in MATLAB/Simulink

In order to implement impedance control, the motor must be controlled in torque. A Maxon BLDC motor is used in simulation and for the tests performed in chapter 4 [42]. A simple motor model is implemented that acts as a first-order low pass filter with cut-off frequency equal to the phase inductance divided by the phase resistance of this motor. This is done to improve the accuracy of the model in simulation.

The dynamic equation eq. (3.6) is transformed to the Simulink model of fig. 3.6 where friction, gravity and collision are simulated. The inputs of this system (in red) are the reference torque $\bar{\tau}_M$ and the disturbance torque τ_d . The outputs of this system (in blue) are the load-side position θ_L , the motor-side speed ω_M and the reaction force at impact F_{im} . This impact force is measured between the contact pad 'Spherical Solid: pad' and the human body part 'Brick Solid: mH', where 'mH' stands for m_H , the impacted body part as prescribed by the ISO 15066 norm. The block 'Spatial Contact Force' is responsible to model this interaction. 'Rotational Multibody Interface' makes the connection between Simulink and Simscape Multibody. This is helpful since the controller is programmed in Simulink whereas the dynamic simulation is done in the multibody environment of Simscape. Gravity is modeled via the multibody environment blocks, friction as a decrease in motor delivered torque and stiction torque via the blocks 'Stiction'. As explained in section 3.4, these are added at both the load- as motor-side: if the input is smaller than the stiction torque, the output will be zero.

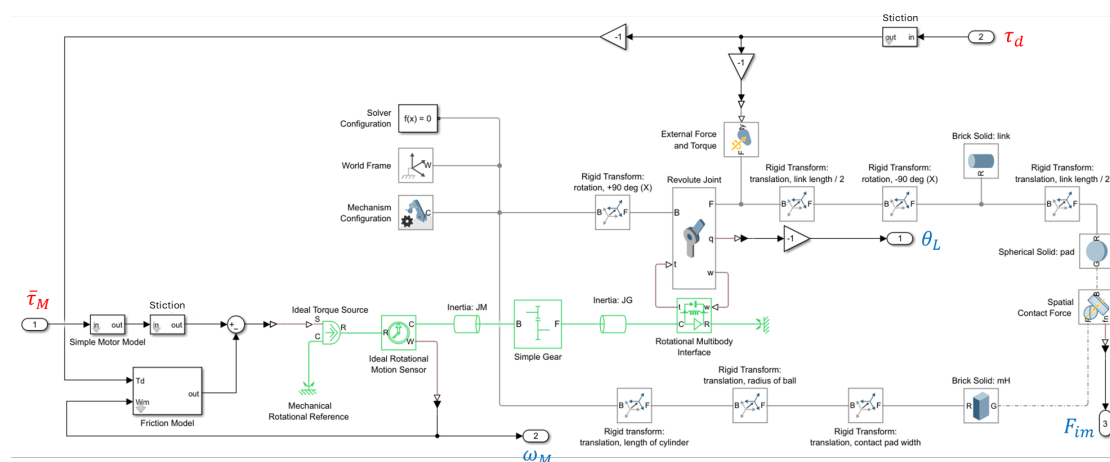


Figure 3.6: eq. (3.6) transformed to a continuous Simulink model.

A discretized version of this model will be helpful, as will be explained later. Consider for this the continuous state space representation previously given by eq. (3.29). This model can be discretized with the `c2d()` MATLAB command that yield discrete system matrices. A disadvantage of this model is that it does not allow the connection to the multibody environment and thus impacts must be modeled in another way.

3.7.2 Control architecture in MATLAB/Simulink

The complete control architecture is presented in fig. 3.7. This scheme can be used for the control of the system in fig. 3.6 (and its discretized version), as well as for the experimental validation in chapter 4. In the latter, the torque reference signal $\bar{\tau}_M$ is send to the motor drive of the BLDC motor. In what follows, each block is explained and the connection between other chapters is highlighted.

The 'Feedback linearization' contains the feedforward of gravity due to the mass of the link and the gearbox + motor combined friction. Due to the high complexity of the friction model based on empirical data, a simpler model was adopted: ν and τ_c are the viscous and coulomb friction parameters respectively. In the gravity compensation part, G contains the weight-torque of the robotic link. The disturbance observer 'DOB' and stiction compensator ' τ_{stic} ' are explained in section 3.6 and section 4.3.4 respectively. The latter proves to be helpful for low virtual stiffness, as explained in chapter 4. The Tustin derivative (section 3.6) is implemented on the left side of the scheme to compute the load-side rotational speed form the encoder measurement.

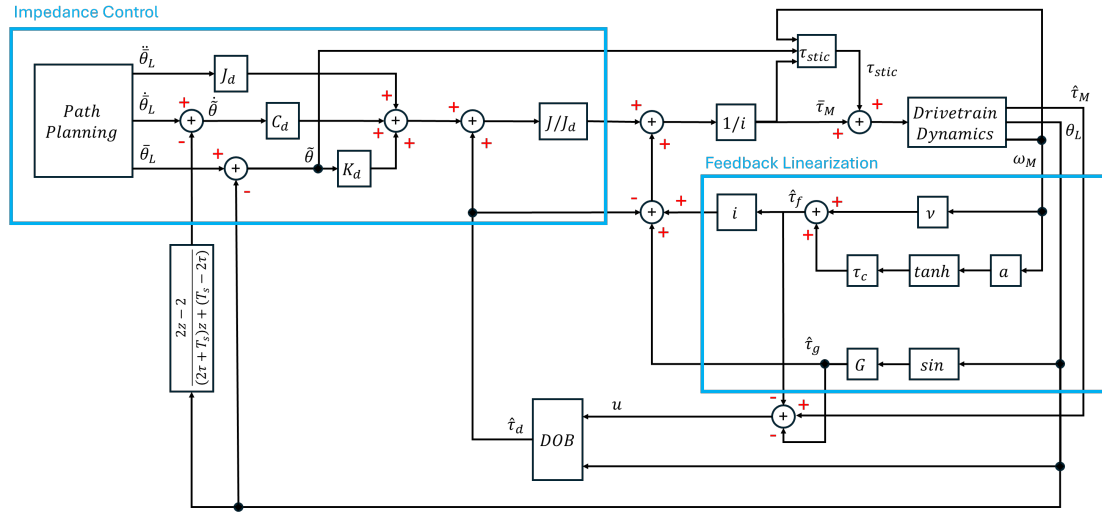


Figure 3.7: Block diagram of the control scheme used for simulations, as well as the real-time implementation in chapter 4. For the real-time implementation, the reference signal for the motor torque $\bar{\tau}_M$ is send to the drive of the BLDC, which is connected to the sun-side of the gearbox. In case of simulations, the 'Drivetrain Dynamics' block represents the model of fig. 3.6, in case of a continuous simulation.

The reference load-side position signal $\bar{\theta}_L$ is generated in the block 'Path Planning'. This block generates a linear-segment-with-parabolic-blends signal as in fig. 3.8. Using this implementation over a constant position reference has two distinct advantages. The speed $\bar{\Omega}_L$ can be set by the user, to a maximum (and constant!) value corresponding to the linear segment; and the reference acceleration signal can be used as a feedforward term in the impedance control. In case of using a constant reference, the part $J_d \ddot{\theta}_L$ of eq. (3.7) is always zero and thus of no use. Also, as explained earlier, the speed cannot increase above a certain threshold determined by the ISO 15066 regulation. Based on [43], the trajectory is generated as:

$$\bar{\theta}_L = \bar{\theta}_L(t) = \begin{cases} \frac{\bar{\Omega}_L}{2t_b} t^2 & t \in [0, t_b) \\ \frac{\bar{\Theta}_L - \bar{\Omega}_L t_f}{2} + \bar{\Omega}_L t & t \in [t_b, t_f - t_b) \\ \bar{\Theta}_L - \frac{\bar{\Omega}_L}{2t_b} (t_f - t)^2 & t \in [t_f - t_b, t_f] \end{cases} \quad (3.32)$$

when assumed to start at $\bar{\theta}_L(t=0) = 0$. The final time t_f is the time at which $\bar{\theta}_L$ reaches the reference $\bar{\Theta}_L$. t_b is the blend time, the duration of one parabolic blend, which is computed as

$$t_b = \frac{\bar{\Omega}_L t_f - \bar{\Theta}_L}{\bar{\Omega}_L}. \quad (3.33)$$

The blend time is fixed once the parameters $\bar{\Theta}_L$, $\bar{\Omega}_L$ and t_f are chosen. This also means that the condition

$$\frac{|\bar{\Theta}_L|}{t_f} \leq |\bar{\Omega}_L| \leq 2 \frac{|\bar{\Theta}_L|}{t_f} \quad (3.34)$$

must be fulfilled in order for the LSPB trajectory to be possible.

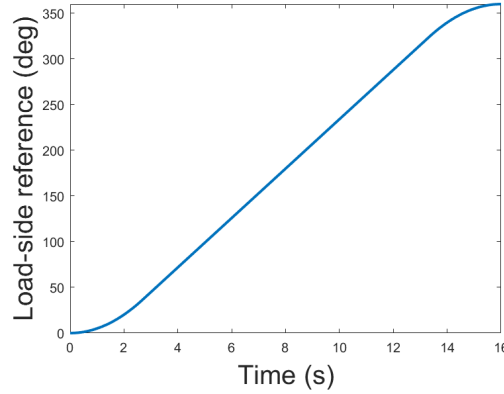


Figure 3.8: The reference signal in case a load-side position $\bar{\Theta}_L = 360deg$ degrees is to be reached. For the first $2.68s$, corresponding to the blend time t_b , the position reference increases parabolically; followed by a linear increase in position and another parabolic trajectory afterwards. This allows the system to be controlled in position with a large region of constant speed. The linear section is featured by the constant speed $\bar{\Omega}_L$.

3.7.3 Simulations in continuous and discrete time

Four simulation scenarios will be shown to demonstrate the performance: three simulation with the continuous and a fourth with the discrete model.

Continuous simulations

The first simulation is to showcase the position tracking performance of the impedance control eq. (3.7). In a second simulation, the link tries to reach a desired position when a constant disturbance torque $\tau_d = 10Nm$ is applied. In a third simulation, the robotic link will hit a simulated human contact and the impact force F_{im} will be measured in both the transient and quasi-static case. For each of the simulations, the actual and desired position of the robotic link is plotted. The result of the three continuous simulations is shown in fig. 3.9.

A first simulation is performed without any disturbances. The black load-side position trajectory is underneath the red curve showcasing that in the absence of disturbances and human interaction, the impedance controlled link can reach the position.

In the second simulation, a constant disturbance of $10Nm$ is applied for the entire time. The link can still be controlled in position, where the difference between reference trajectory and actual load angle is a constant after around 2 seconds. Type one of impedance control is used, thus the steady state error corresponds to: $10Nm = K_d \tilde{\theta}$, which is desired for safety applications.

In the third simulation, the link reaches a constant speed of $1000rpm$ (at motor-side) and then hits the 'skull and forehead' at $\theta_L = 270deg$, in accordance with the ISO 15066. The damping is chosen in accordance with eq. (3.5). When this disturbance torque is divided by the length of the link ($l = 145mm$), its maximum value of impact reaches $320N$, which is the limit of impact with 'skull and forehead'. The quasi-static force reaches a value of around $172N$, which is $42N$ too high in accordance with the norm. This could be easily solved by changing the values of the impedance control after an impact has been detected, but is not further investigated in this thesis.

These simulations were carried out with total inertia $J = 10m^2kg$, which is approximately 10 times the total inertia of the drivetrain used in the experimental validation of chapter 4. This is to showcase what would happen in case of higher inertia, for instance, when the drivetrain were to be implemented in a cobot with multiple-degrees-of-freedom.

Discrete simulation

From these simulations, it is concluded that the desired performances can be reached in case of a continuous simulation with perfect sensors. However, in reality, the controller works at a clock cycle of $1kHz$ and the measurements for the position and torque from the motor are noisy. As explained in section 3.6.1, the noise will be amplified and the estimation $\hat{\tau}_d$ might become unstable. To this end, a fourth simulation is performed on the discretized system. Unfortunately, the discrete model cannot incorporate the Simscape impact model of fig. 3.6 since the latter only works in a continuous simulation environment. Instead, an impact is simulated with τ_d equal to a step of amplitude $40Nm$ for a duration of $0.5s$. Gaussian noise is added to the measurement of motor torque τ_M , the measurement of load-side position θ_L and motor speed ω_M . Simulation results that compare the HOSM DOB and LPF DOB of fig. 2.12a are presented in fig. 3.10.

Again, the controller tries to follow the LSPB path when two impacts occur in time: one when the motor is turning at a constant speed of $1000rpm$ and another when the link reaches the desired end-position. This first impact occurs at $t = 10s$ and the second at $t = 16s$, each for a duration of $0.5s$ and with $40Nm$ of amplitude. In what follows, a discussion for this discretized simulation result is provided.

Since the LPF DOB is a dynamic system on its own, it needs some time to be calibrated as can be seen in the beginning of the estimated disturbance torque: the actual disturbance is zero, whereas the estimation takes some time before reaching that value. This is also the case for the HOSM DOB, when the starting condition of the Luenberger observer does not coincide

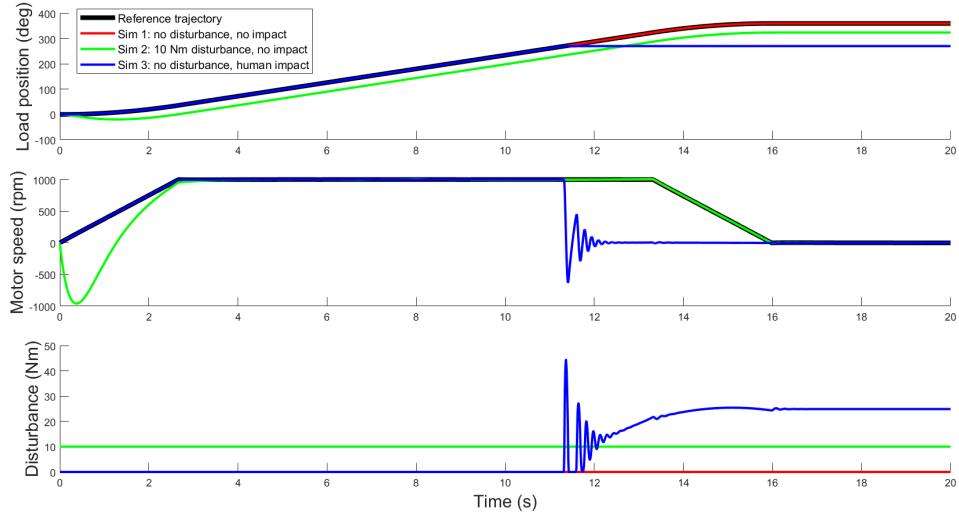


Figure 3.9: Three simulations are performed to validate the impedance controller as well as to validate the choice of the impedance parameters for the continuous model. In all three simulations, the link is controlled to follow the LSPB, as in fig. 3.8, corresponding to the black curve. The first figure shows the load-side position in time, the second shows the motor-side speed of the actuator and the third figure shows the actual disturbance applied to the link connected to the actuator. Across all three figures, the color code is the same: the black curve correspond to the reference trajectory, the red curves are the results from the first simulation, green from the second and blue from the third simulation. No noise is simulated, so the estimated equals the actual disturbance.

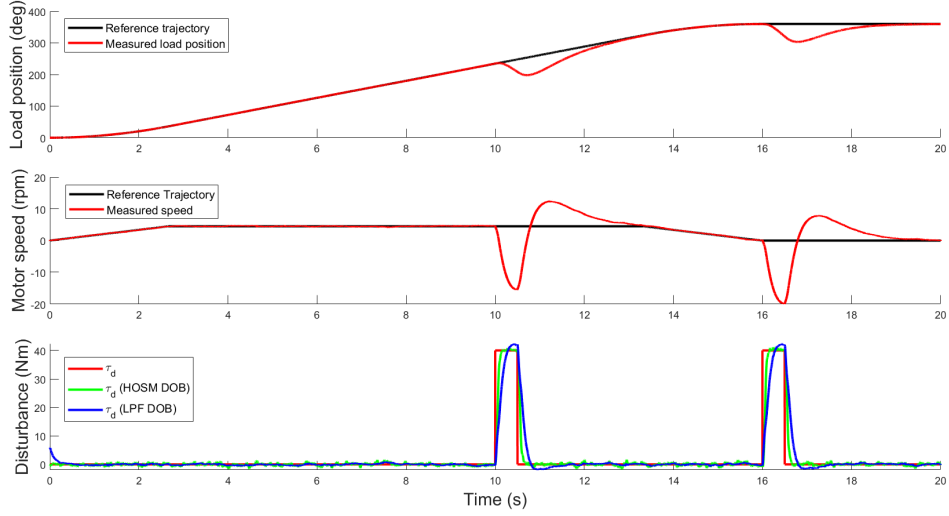


Figure 3.10: A fourth simulation is performed to validate the impedance controller in case of a discretized drivetrain and the addition of sensor noise to the measurements of torque, position and speed. The first graph shows the load-side reference position in black and the actual position in red. As impacts occur, the link drives back a bit, as desired for safe control. At time $t = 16.5s$, the position error goes to zero, as no disturbance torque is applied anymore. The second graph shows the speed with the same color convention. In the third graph, the red curve showcases the actual disturbance torque, the green is the estimated with the HOSM DOB and the blue curve with the LPF DOB.

with the actual load-side position $\theta_L(t = 0)$. Since most of the time the absolute position would be known, this would not impose problems for this type of DOB.

Both DOB's are capable of estimating the impact at constant speed as well as at constant position. The rather accurate estimation $\hat{\tau}_d$ is presumably due to the backdrivability of the gearbox. For backdrivable gearboxes, the influence of a non-zero disturbance τ_d results in an increase in current and consequently in motor torque. This signal can be used inside the DOB scheme, which is not the case for a non-backdrivable gearbox.

As hypothesized in section 3.6, the HOSM DOB can indeed detect the impact faster as it reaches the peak value of $40Nm$ faster compared to the LPF DOB. Also, when the actual applied disturbance drops to zeros again, this DOB reaches that value faster as well.

Due to the sliding nature of the HOSM part, there is 'virtual' induced noise onto the estimation of the disturbance torque. This can be seen in the region where the actual disturbance is zero. However, due to the high gear ratio, this noise is suppressed at the motor-side.

These simulations show that, even in the presence of noise, the HOSM DOB can make around a 2 times faster estimation of the disturbance torque. This can be explained by the fact that the derivative is estimated via an observer, rather than differentiating, filtering and thus slowing down the position signal [41]. Disadvantages include the computationally more expensive algorithm and the fact that the performance heavily depends on the sampling frequency [19].

Chapter 4

Experimental Validation

4.1 Introduction

This chapter presents the experimental validation of the proposed control framework developed in chapter 3. Two experimental setups are used to evaluate the performance of the system in real-time. The first part of this chapter focuses on fine-tuning and parameter identification of the drivetrain model, impedance control strategy, and disturbance observers. These experiments ensure that the virtual parameters (J_d , C_d , K_d) and observer settings match the physical behavior of the test bench. The second part evaluates the system's behavior under conditions relevant to Human-Robot Interaction (HRI). Key performance metrics, such as position tracking accuracy and impact torque estimation, are assessed using the standardized Pilz PRMS device.

4.2 Fine-tuning the parameters of the control scheme

4.2.1 Introduction

In a first setup, the parameters of the proposed control scheme fig. 3.7 are fine-tuned experimentally with six tests. Using the second type of impedance control, the feedback linearization and some virtual parameters can be tested simultaneously, as no disturbance estimation must be provided for this type. Secondly, the virtual parameters K_d and C_d are validated. Lastly, the disturbance observer is added: first in open-loop to make sure that no unexpectedly high disturbance estimation torques are detected, then in closed-loop using the first type of impedance control. Additionally, some bounds and parameter choice considerations are provided.

The experimental setup for this section is depicted in fig. 4.1. A 250Watt Maxon EC45 brushless DC motor (part nr. 136210) ¹ incorporating a 2-channel, 10,000 CPR US Digital encoder ² was driven by a Maxon EPOS4 70/15. The control is done via the MATLAB/Simulink environment. For validation, a Lorenz DRBK-2 torque sensor with a $\leq 0.5\%$ accuracy (serial nr. 307125816) is connected in between the EC45 motor and the gearbox. A $1kHz$ network in a Beckhoff Twin CAT master environment was used in which the control scheme of fig. 3.7 was implemented.

¹<https://www.maxongroup.com/maxon/view/product/136210>

²<https://www.usdigital.com/products/encoders/incremental/kit/e6/?q=E6-10000-984-IE-D-H-D-3>

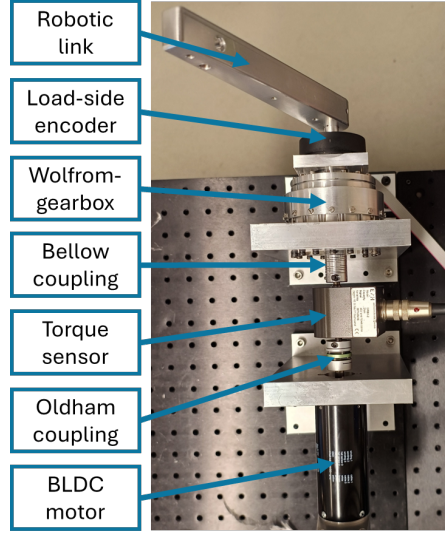


Figure 4.1: Experimental setup used to fine-tune the feedforward parameters, the impedance control and the disturbance observer of the control scheme.

4.2.2 Test 1: fine-tuning the gravity model

When choosing all virtual parameters and desired kinematics zero, one would, based on eq. (3.6), expect the system to behave as

$$i\bar{\tau}_M \approx i\tau_M = \hat{\tau}_g = G\sin\theta_L, \quad (4.1)$$

$\forall\theta_L$. The contribution of the (simple) friction model is zero because it solely depends on speed. When the link is manipulated from one position to another, it should remain stationary and stable at that new position. Based on this principle, the parameter G can be estimated. This parameters can be more accurately determined from CAD files, given one knows exactly what material the link is made of.

4.2.3 Test 2: fine-tuning the friction model and influence of virtual stiffness

Consider for this experiment the second type of impedance control with virtual parameters $J_d = J = 1.25m^2kg$, $C_d = 0Nms$ and $K_d = 3.2Nm$. The inertia is determined form eq. (3.17). The resulting dynamics should, according to eq. (3.10), behave as

$$J_d\ddot{\theta} + K_d\tilde{\theta} = 0. \quad (4.2)$$

The link should behave as an undamped system for a non-zero initial position with a natural period T_{nat} equal to

$$T_{nat} = 2\pi\sqrt{\frac{J_d}{K_d}}. \quad (4.3)$$

The friction model

$$\hat{\tau}_f(\omega_M) = \tau_c + \nu \text{sign}(\omega_M) \quad (4.4)$$

and its parameters τ_c and ν can be fine-tuned by investigating their influence onto the stability and periodicity of the swing. In other words, once the link is released at a certain angular position, it should return at that exact position after T_{nat} seconds and this for an infinite amount of time. When this is the case, one can assume that the friction model parameters are correctly set. When the virtual stiffness is increased, the natural period should decrease, determined by eq. (4.3). Like so, the influence of K_d can be investigated and validated.

Consider for this experiment the same friction model as explained in fig. 3.7, where the $\tanh()$ function is replaced by a $\text{sign}()$ function. It allows for the slope parameter a to be fine-tuned independently from the coulomb and viscous friction parameters τ_c , ν . The advantage of the $\tanh()$ function is to eliminate the chatter each time rotational speed would change direction, allowing the torque reference signal $i\bar{\tau}_M$ to be smoother.

4.2.4 Test 3: measuring drivetrain inertia

The previous experiment can be conducted only when the inertia J is in fact equal to the actual load-side reflected inertia of the drivetrain. Now that the gravity and friction models are determined, the inertia J can be measured. When applying the reference signal

$$i\bar{\tau}_M = \hat{\tau}_g + i\hat{\tau}_f + c^{te} \quad (4.5)$$

to the motor drive, the inertia can then be calculated as a time varying signal:

$$\hat{J}(t) = \frac{c^{te}}{\ddot{\theta}_L(t)}. \quad (4.6)$$

Taking the mean of this signal over a certain time range leads to an estimate of the inertia which should be compared to the value computed with eq. (3.17). When $\hat{J}$ and J have the same order of magnitude, it can be concluded that the inertia is correct. Since the determination of the friction model depends on this inertia, it can also be concluded to be correct.

4.2.5 Test 4: influence of virtual damping

Using type two impedance control with virtual impedance parameters $J_d = J$, $K_d = 3.2Nm$ and

$$C_d = 2(99\%) \sqrt{J_d K_d}, \quad (4.7)$$

the link should critically dampen when released at a certain non-zero position, in the absence of any environmental torque. No overshoot should be observed when the link reaches the position $\theta_L = 0deg$. When this is the case, the impedance parameters and control scheme can be assumed to be correctly programmed.

4.2.6 Test 5: fine-tuning the disturbance observer(s)

In this experiment, the second type of impedance control is used with all virtual parameters set to zero except for the inertia ratio $r_d = J/J_d$. Consider the LPF DOB that measures the encoder position at the load-side:

$$\hat{\tau}_d = i\tau_M - i\hat{\tau}_f - \hat{\tau}_g - J\ddot{\theta}_L. \quad (4.8)$$

For the actual implementation, the signals $\ddot{\theta}_L$ and τ_M should be filtered. Since the second type of impedance control is employed, the estimation of eq. (4.8) is not actually used in closed-loop. That means that the signal is real-time available, but it has no contribution to the reference torque $i\bar{\tau}_M$ signal. The idea of this experiment is that when the link is backdriven, one should see an increase in the signal $\hat{\tau}_d$. When the link is released, the estimation should become zero. Since no torque sensor is used at the load-side of the setup, it is impossible to know the accuracy of this estimation, thus this is a pure qualitative experiment.

The use of the experiment is to see if no unrealistically high torques are detected in the estimation, because they could yield unsafely high accelerations when switched to the first type of impedance control (closed-loop). It is however very important to fine-tune the time constants of the Tustin derivatives and the DOB filters to have a meaningful measurement of acceleration and disturbance estimation. The latter should also be done during this test.

4.2.7 Test 6: influence of inertia ratio

In this experiment, the same parameters as in the previous one are used and the increase of inertia ratio r_d is investigated. To this end, switch to the first impedance type and set the inertia ratio to a value that is a little bit higher than one. The link should now come into a state in which it can be manipulated easier compared to the unactuated state. That means that a certain observed acceleration should require less torque. Notice that the uncontrolled link is manipulatable, because of the inherent backdrivability of the gearbox by hand.

The r_d ratio cannot be increased indefinitely however, as instability behaviors will arise. It is hypothesized that this is due to the fact that low pass filters introduce a delay in measurements. This behavior is pronounced when one tries to move the link from a certain position to another. Due to the filters, the estimated disturbance lags behind the actual one, resulting in a non-zero reference torque that is applied to the motor at the instance the user tries to stop the link. As a result, the link moves a bit further than expected. The user intuitively tries to compensate for it by applying a torque in the opposite direction. That extra disturbance is estimated later in time and the link starts to accelerate in the opposite direction, consequently overshooting the position a second time. The user again tries to compensate for this and the cycle repeats, resulting in unstable behavior. For the novel gearbox, a maximum ratio of $r_d = 2.5$ can be applied before this instability phenomenon appears.

4.2.8 Some struggles and wins regarding the fine-tuning process

These six experiments are a systematic way of fine-tuning and investigating the influence of model parameters from the control scheme fig. 3.7. When trying out the experiments in section 4.2.6, it is easy to make mistakes. That is why this particular order of fine-tuning and mistake troubleshooting the parameters is proposed.

The discretized model and the way of converting intuitive parameters to the virtual parameters (explained in section 3.3) really helped troubleshooting and finding suitable parameters to be tested onto the test bench. For example, as explained in section 3.7.2, not all choices of LSPB parameters lead to meaningful reference trajectories. When condition eq. (3.34) is not satisfied for instance, it is possible for the reference acceleration to reach values that would lead the torque reference signal to be unsafely high with respect to the ISO norm. Preventing this by preliminary running a simulation resulted in a safe testing environment and a faster experimental validation.

4.3 Experiments in an HRI environment

4.3.1 Introduction

To assess the system under realistic HRI conditions, a second experimental setup was created using the Pilz PRMS (Pilz Robot Measurement System) device. It is a standard-compliant collision measurement in accordance with the ISO 15066 norm and will be used to test the performance of disturbance observers measuring impact forces/torques. The setup is depicted in fig. 4.3 and the same hardware was used as described in section 4.2.1. The Pilz instrument is used to detect the contact force between the link and the yellow surface (the pad), which imitates a certain human body part, in accordance with the ISO norm. For instance, when one wants to investigate a collision between cobot/robot and the skull of a human, the black contact pad and spring with stiffness $K = 150kN/m$ should be installed. An overview of which springs and contact pads correspond to the human body parts is presented in table 4.1 [34].

	K (N/mm)	C (color)
Skull and forehead	150	Black
Face	75	Black
Neck	50	Black
Back and shoulders	35	Blue
Chest	25	Black
Abdomen	10	Yellow
Pelvis	25	Black
Upper arms and elbow joints	30	Blue
Lower arms and wrist joints	40	Black
Hands and fingers	75	Black
Thighs and knees	50	Blue
Lower legs	60	Blue

Table 4.1: Depending on which body part is impacted by a robot/cobot, the spring with stiffness K and contact pad with the corresponding color should be installed inside the Pilz PRMS instrument.

The link will be controlled in position when an impact occurs. In case the end-position reference $\bar{\Theta}_L$ is larger than the position of the Pilz instrument, an impact is inevitable, as explained in the visual representation of fig. 4.2. The (maximum) impact force should be less than the values prescribed by the ISO 15066 norm (fig. 3.1), depending on the body region the link impacted. Also, the duration of impact should be less than $0.5s$ [34]. Three experiments are performed to test safety and performance of the control scheme. After each impact, the link is impedance controlled to rotate back to an initial position. The difference between this angle and the actual encoder read-out will serve as a position accuracy metric.

4.3.2 Experiment 1: influence of impact speed

For safety reasons, the speed of impact in the next experiments is set to a low value for actual pick-and-place purposes. Consequently, a preliminary experiment is conducted where the influence of increasing the speed onto the impact force is investigated. To this end, the yellow contact pad and spring $K = 10kN/m$ are used. The impedance parameters are $K_d = 5Nm$, $C_d = 3.4Nms$

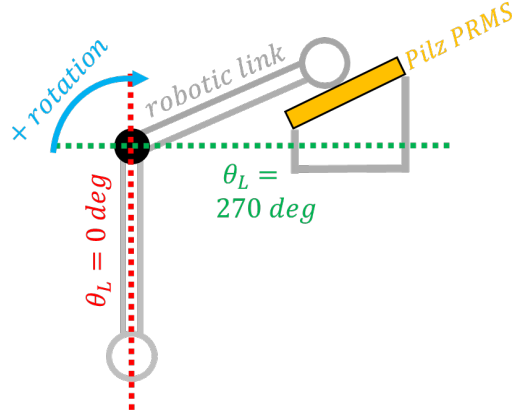


Figure 4.2: A visual explanation of how a position controlled link will impact with the Pilz instrument if the end-position reference is chosen at $\Theta_L = 270deg$.

(critical dampening) and $J_d = J/2$ ($r_d = 2$). The HOSM DOB is used in closed-loop (first type of impedance control) to estimate the disturbance and to act accordingly. The results that relate the rotational impact speed (at motor-side) to the impact force, measured with the Pilz instrument and estimated using the DOB, are shown in fig. 4.4. One can prove that the correlation between them is linear. To this end, equate the kinetic energy of the link to the impact energy onto the instrument [34]:

$$\frac{1}{2}\mu v_{rel,max}^2 = \frac{1}{2}\mu \frac{l^2 \omega_M^2}{i^2} = \frac{1}{2} \frac{1}{K} F_{im}^2. \quad (4.9)$$

All variables in this equation are explained previously. Solving this equation to the ratio

$$\frac{F_{im}}{\omega_M} = \frac{l}{i} \sqrt{JK} \quad (4.10)$$

proves the linear fit in fig. 4.4. The linear behavior of the HOSM DOB to measure impact force/torque from impact speed means that the measurements from the next experiment can be extrapolated to higher speeds (with confidence bounds increasing for higher speeds).

The stiffness and contact pad chosen in this experiment correspond to the 'abdomen' body part as prescribed by the ISO 15066 norm [34]. From fig. B.1, the transient impact force cannot exceed $110N$ for this application. The results in fig. 4.4 tell that the speed for this link of $145mm$ cannot exceed $3200rpm$ at motor-side. Considering the gear ratio of 222, this corresponds to a $0.2m/s$ speed at load-side, which is not a lot. However, these claims must be taken with care, as impact force depends on the application and body part of impact. For instance, when this actuator were to be implemented inside a robot/cobot, the reflected inertia would be higher.

4.3.3 Experiment 2: impact disturbances & duration estimation

Impact torque estimation

For several virtual (K_d) and real (K) stiffness couples, the estimation performance of four different types of DOB's to the actual impact disturbance torques are investigated and compared. The experiment was carried out by controlling the link in position with a constant speed of $2000rpm$ at motor-side, where the link is following a LSPB trajectory as described in fig. 3.8.

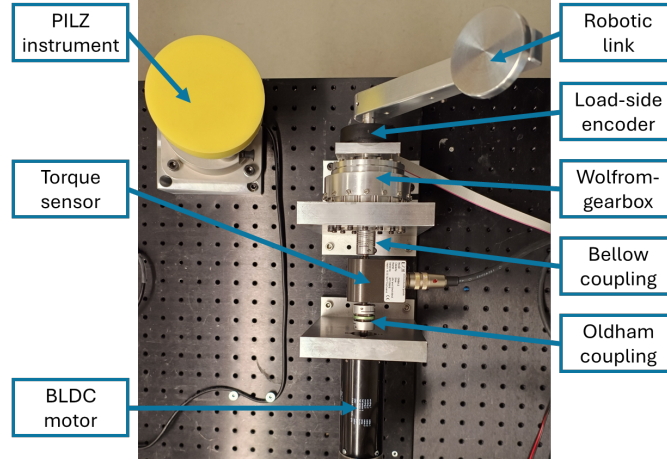


Figure 4.3: The setup used to measure impact torques, quasi-static torques and to investigate the position accuracy metric.

The end-reference angle $\bar{\Theta}_L$ is higher than what can be reached, so an impact will occur. Measurements are captured in table 4.2, where each value is computed as the average over three recorded impacts. In order to compare the performance across observers, the impact torque is divided by rotational speed. As explained by eq. (4.10), this ratio should be a constant and thus a quantitative analysis between the DOB types can be conducted.

For a comparative study, one must really look at the relative percentage errors. It can be concluded that all models experience a better impact detection for soft (low K values) impacts, with the MEDOB having the best performance. As explained earlier, it is also the most expensive and complex setup to build as an encoder at the load-side must be provided. It also has the advantage of being applicable to non-back drivable actuators [22] [8], which is irrelevant here because of the inherent backdrivability of the novel gearbox.

Overall, one can say that the errors of impact estimation could be around 40% and 20% for large and low stiffness (K) respectively, which means that the corresponding requirement in table 1.1 could not be reached. Anyway, these errors are rather large and are hypothesized to originate from the fact that the backdriving efficiency is generally lower than the forward driving one [3]. At impact, the power flow reverses sign from load to motor. Due to the inherent backdrivability of the gearbox, this is reflected in the current and thus in the measurement of the motor torque τ_M . As the simple friction model does not make a distinction between the power flow direction, the efficiency is overestimated at impact and consequently, the disturbance estimation underestimated. A way to increase the impact estimation would lie in the use of the four-quadrant friction model. This however drastically increases the complexity of experimentally fine-tuning the parameters. Additionally, a way to deal with the discontinuities would be needed. An easier way of dealing with the problem of underestimation would be to model the viscous and coulomb parameters depending on the power flow (τ_M multiplied with ω_M). This ultimately leads to a two-quadrant friction model and thus half the amount of parameters to fine-tune.

Impact duration estimation

In order to validate the duration of impact estimation for each of the observers, the relative error between the actual impact duration (measured from the Pilz instrument) and the estimated one

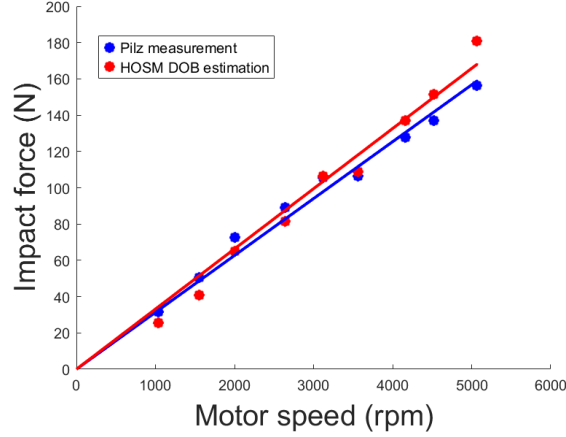


Figure 4.4: Influence of impact speed to the impact force. As expected by equating kinetic energy to impact energy, the correlation between them is linear.

(with the disturbance observers) was also computed and can be found in table 4.3. The impact time t_{im} is computed as the mean of three recorded impacts with the Pilz PRMS instrument, whereas $\hat{t}_{im}$ is the estimated time from the respective disturbance observer. From this data it is hard to identify which observer has the best performance in terms of estimating the impact duration, since they all have the best performance over at least one experiment. However, one can conclude that the estimation error is less than 50% over all experiments. That means that the first metric in table 1.1 can be reached for the novel gearbox.

4.3.4 Experiment 3: quasi-static disturbances & position accuracy

Quasi-static disturbance estimation

A last experiment is performed to test the accuracy of the HOSM DOB to detect quasi-static disturbances. These types of disturbances are prescribed by the norm as clamping or crushing forces/torques [34]. For different values of impedance parameters, the reference position was set beyond the position of the Pilz instrument so that a constant torque is applied to the contact pad after impact. The results of the experiment are presented in table 4.4.

The last experiment in that table was performed with a small virtual stiffness. As a consequence, the position was not reached and thus a small push had to be applied for the link and the Pilz instrument to start touching. It could explain the high relative error. The highest value of constant torque inside this table corresponds to a force of $24N$, which is far below the quasi-static limit as prescribed by the ISO norm. However, for a more complete study of how this gearbox would behave in an HRI environment, measurements should be conducted when the gearbox is integrated inside a multi-degree-of-freedom robotic arm.

The relative errors imply that the requirement for a constant disturbance estimation of at least 5% accuracy is not reached, as prescribed by the requirements of table 1.1. This could be explained by the fact that the link is not applying a torque perfectly in the middle of the contact pad, or by the Pilz incapability of measuring relatively small forces. For high virtual stiffness the relative error is low, and vice versa for low stiffness. Taking into consideration the fact that for a real pick-and-place application the virtual stiffness should be much higher, these experiments show that the novel Wolfrom-gearbox might be applicable for such applications without the need

		Experiment 1 $K_d = 1Nm$ $C_d = 1.5Nms$ $K = 10kN/m$ $C = \text{yellow pad}$	Experiment 2 $K_d = 5Nm$ $C_d = 3.4Nms$ $K = 10kN/m$ $C = \text{yellow pad}$	Experiment 3 $K_d = 1Nm$ $C_d = 1.5Nms$ $K = 150kN/m$ $C = \text{black pad}$	Experiment 4 $K_d = 5Nm$ $C_d = 3.4Nms$ $K = 150kN/m$ $C = \text{black pad}$
load LPF DOB	τ_d/ω_L (Nms)	1.11	1.00	2.17	2.33
	$\hat{\tau}_d/\omega_L$ (Nms)	0.63	0.78	0.66	0.76
	%error	43.3	21.8	69.5	67.2
motor LPF DOB	τ_d/ω_L (Nms)	1.11	2.18	2.09	2.32
	$\hat{\tau}_d/\omega_L$ (Nms)	0.96	0.99	1.39	1.36
	%error	13.9	5.7	38.5	41.2
HOSM DOB	τ_d/ω_L (Nms)	1.21	1.09	1.77	2.25
	$\hat{\tau}_d/\omega_L$ (Nms)	0.95	0.99	0.99	1.25
	%error	21.2	8.7	43.9	44.5
MEDOB	τ_d/ω_L (Nms)	1.11	1.00	1.90	2.35
	$\hat{\tau}_d/\omega_L$ (Nms)	0.99	1.02	1.30	1.51
	%error	10.6	-0.72	31.9	35.5

Table 4.2: Testing the performance of different observers to impact torques. The orange rows are computed as the relative error (%) between actual impact torques (τ_d/ω_L , from the Pilz instrument) and estimated impact torques ($\hat{\tau}_d/\omega_L$, from the corresponding disturbance observer). The 'motor LPF DOB' uses only the measurement of motor speed and the 'load LPF DOB' uses the measurement of the encoder placed at the load-side in fig. 4.3. The 'MEDOB' uses both the encoder at the load-side and the speed measurement from the motor. Each of the values inside this table are computed as the average over three recorded impacts.

for a torque sensor at the output of the gearbox. However, this really can only be said when it were tested when integrated in a multi-degree-of-freedom robotic arm. Also, the same hypothesis as for the impact experiments could be applicable: since the backdriving efficiency is lower than the forward one, the disturbance estimation is underestimated. This can be shown in simulation. If the two-quadrant model were be used inside the dynamic equations, but the simple friction model in the feedback linearization, one would see that the estimated impact and quasi-static torque are smaller than the actual ones.

Position accuracy

As explained in the introduction of these experiments, the link is position-controlled where start and end point coincide. The difference between the reference position and the position measured by the load-side encoder can serve as a metric of position accuracy. The measurements are captured in table 4.5. The research question requirement (table 1.1) of 1% error accuracy is easily reached. However, for low virtual stiffness, the relative error is too high for a pick-and-place application. This is further investigated with a stiction compensator.

		Experiment 1 $K_d = 1Nm$ $C_d = 1.5Nms$ $K = 10kN/m$ $C = \text{yellow pad}$	Experiment 2 $K_d = 5Nm$ $C_d = 3.4Nms$ $K = 10kN/m$ $C = \text{yellow pad}$	Experiment 3 $K_d = 1Nm$ $C_d = 1.5Nms$ $K = 150kN/m$ $C = \text{black pad}$	Experiment 4 $K_d = 5Nm$ $C_d = 3.4Nms$ $K = 150kN/m$ $C = \text{black pad}$
load	t_{im} (s)	0.54	0.55	0.32	0.33
	$\hat{t}_{im}$ (s)	0.65	0.55	0.54	0.51
	%error	-20.9	-0.30	-69.9	-56.1
motor	t_{im} (s)	0.60	0.66	0.37	0.39
	$\hat{t}_{im}$ (s)	0.48	0.46	0.42	0.35
	%error	21.0	30.3	-12.7	11.0
HOSM	t_{im} (s)	0.57	0.58	0.36	0.37
	$\hat{t}_{im}$ (s)	0.39	0.46	0.39	0.42
	%error	31.7	21.0	-9.8	-14.1
MEDOB	t_{im} (s)	0.53	0.59	0.32	0.35
	$\hat{t}_{im}$ (s)	0.37	0.41	0.32	0.30
	%error	30.0	30.6	-1.6	13.7

Table 4.3: Comparison across the different disturbance observers in terms of estimating the duration of the impact.

Stiction compensation for low virtual stiffness

When low virtual stiffness K_d is chosen (lower than $2Nm$ to $3Nm$), the position accuracy drops significantly. This is presumably due to stiction, the torque that is needed to initiate rotary movement of the drivetrain. For low stiffness, the reference torque $\bar{\tau}_M$ that is send to the BLDC drive is not high enough the overcome this stiction for small values of position error $\tilde{\theta}$. A so called 'stiction compensator' based on boolean logic is proposed to resolve this problem.

The working principle can be summarized as: "When one wants to apply a torque ($i\bar{\tau}_M$ is non-zero), but the rotary speed is zero, and the reference position $\bar{\theta}_L$ is not reached, apply a small torque, higher than the Coulomb friction torque for a duration of one clock period T_s ". This expression can be transformed to boolean logic and is depicted in fig. 4.5. The parameters $\tilde{\epsilon}$, $\bar{\epsilon}$ and b can be seen as hyperparameters to be fine-tuned. A $\tanh()$ function is used, as opposed to a $\text{sign}()$ function to remove the discontinuity around $\tilde{\theta} = 0deg$. b can be seen as another hyperparameter that defines the slope of linear region around zero $\tilde{\theta}$.

This scheme is successfully applied to a real-life scenario for a stiffness $K_d = 0.5Nm$. Although this makes the drivetrain usable for low virtual stiffness position control, a question of relevance really arises. As explained above, the values of stiffness should actually be much higher in order to increase position accuracy. High stiffness also means that a robot/cobot returns faster to its reference position when a disturbance has moved the robot/cobot.

r_d (N.A.)	C_d (Nms)	K_d (Nm)	Disturbance (Nm)		%error
			Pilz: τ_d	HOSM: $\hat{\tau}_d$	
2	7	20	3.52	3.41	2.9
2	5	10	2.13	1.77	17
2	1.5	1	0.38	0.2	48
1.5	7	20	3.09	3.41	-10.4
1.5	5	10	1.93	1.77	8.5
1.5	1.5	1	0.38	0.22	41.8
0.5	7	20	3.09	3.47	-12
0.5	5	10	2.33	1.78	23.5
0.5	1.5	1	0.018	0.032	-78.2

Table 4.4: Testing the performance of the HOSM DOB to quasi-static environmental torques. The orange column is computed as the relative error (%) between the actual torque (τ_d) and the estimated torque ($\hat{\tau}_d$).

r_d	C_d (Nms)	K_d (Nm)	%position error
2	7	20	0.18
2	5	10	0.12
2	1.5	1	-6.9
1.5	7	20	0.045
1.5	5	10	-0.22
1.5	1.5	1	-5.2
0.5	7	20	0.25
0.5	5	10	-0.42
0.5	1.5	1	-2.2

Table 4.5: For different values of virtual parameters, the relative position accuracy is computed. It is the relative difference between the wanted (reference) position and the read-out of the load-side encoder.

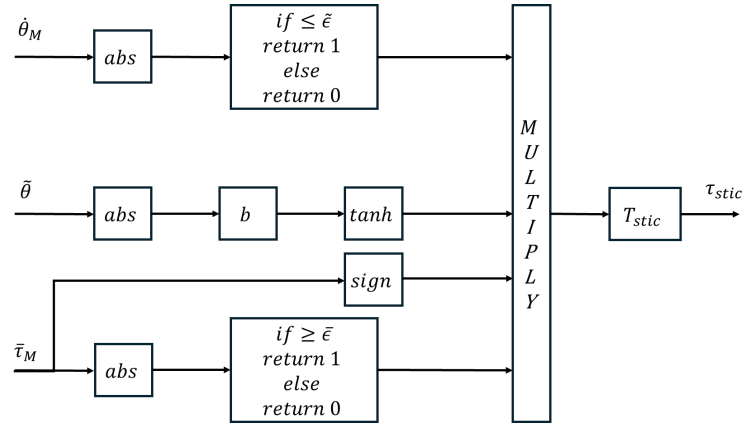


Figure 4.5: Block scheme for the stiction compensation in the control scheme of fig. 3.7. When the position is not reached and the impedance controller imposes a non-zero torque, but no rotary movement is detected, the reference torque is increased with T_{stic} for a duration equal to the sampling time T_s .

Chapter 5

Conclusion & Future Work

5.1 Conclusion

5.1.1 Summary overview

This thesis focused on torque-sensorless control of a newly developed high ratio, backdrivable Wolfrom-based gearbox.

To ensure safety inside an HRI environment, robotic joints have relied on the integration of heavy and expensive torque sensors at the output of the gearbox that monitor exactly the delivered torque. The goal of the master thesis was to investigate the feasibility of controlling a single-degree-of-freedom robotic link (driven by an actuator composing of a BLDC and the novel gearbox) without the integration of a torque sensor, using only the measurements available from the electric motor. To this end, an investigation was performed across existing literature concerning disturbance observers. These mathematical models allow a real-time estimation of disturbance(s) acting on a system, using only the available/already existing sensors. It was thus chosen to investigate the feasibility of such models onto the newly developed gearbox.

To this end, first a mathematic model for the drivetrain composing of a robotic link, the novel gearbox and electric motor was formulated, based on the double inertia system. This was transformed to a MATLAB/Simscape model in which impacts with the environment could be simulated. Since impedance control is commonly employed to ensure safe human-robot or human-cobot interactions, it was implemented here to enable position control of a robotic link in scenarios where human contact could occur at any moment in time. As mathematically shown, the chosen type of impedance control requires the real-time signal of the output delivered torque. As this type of control requires a torque signal and thus the integration of a physical torque sensor at the output of the gearbox, this signal was substituted by a higher-order-sliding-mode disturbance observed signal. Subsequently, it was tested in the simulation environment for the detection of impacts and constant disturbances. As different types of disturbance observers were used inside the impedance control scheme (and thus in closed-loop) a comparison study of performance could be made between them. It resulted that the newly developed higher-order-sliding-mode disturbance observer has around two times faster impact disturbance detection.

Two setups were build to test the performance and safety of the proposed control scheme for real-life scenarios. The first setup can be used for a step-by-step validation and fine-tuning of the parameters inside this control scheme. Experiments that imitate human-robot/cobot collisions together with the position accuracy were tested in the second setup. It was shown that the impacts as estimated by the observers are fast enough for the use in safe HRI applications and

that position error accuracy never exceeded 1% for relevant virtual stiffness's. Furthermore, the maximum ISO 15066 impact force of $110N$ was met for the link colliding with an 'abdomen' body part at $0.2m/s$.

5.1.2 Answers to research questions

Question 1

To what extent can motor current serve as an accurate indicator of gearbox input torque, and how effectively can it be used to estimate output torque, especially for the estimation of impacts?

Due to the inherent backdrivability of the gearbox, the current drawn from an electric motor at the driving side of the gearbox can serve as a measurement for the disturbance torque felt at the output of that gearbox. This was investigated by impact experiments. Overall, the estimation is around 20% for low stiffness (K) and 40% for larger stiffness's, with the dual position measurement observer yielding the lowest. Probably due the fact that two inertia torques (instead of one) are used for the estimation. These are acceptable error margins considering these impacts rise and decline in a time basis of only several times the sampling time. However, it was hypothesized that better results might be found when increasing the complexity of the friction model to one that would depend on power flow. More fine-tuning and/or focusing on only one of the disturbance observers might improve the prediction even more.

The relative errors of 40% for larger stiffness's and 20% for lower ones concerning the estimation of impact torques ultimately mean that the requirement of at least 10% accuracy is not reached (table 1.1). Also, the requirement of estimating constant torques with at least 5% of accuracy is not reached. This could have been because of the fact that the link is not perfectly impacting the Pilz instrument in the middle of the contact pad, or due to the Pilz incapability of accurately measuring low forces.

However, in the end, safe control was obtained with the simple friction model, making an increase in complexity possibly obsolete.

Question 2

How can the safety of human interaction and collision be ensured for a link connected to a motor-gearbox actuator operating in position and/or speed control?

In essence, experiments were conducted in which a single-degree-of-freedom robotic link was impedance controlled in position and speed. The biggest difference compared to type one impedance control (as in section 3.3) lies in the fact that the environmental torque was mathematically observed/estimated instead of measured.

The Pilz PRMS instrument served as a mechanical obstacle at which this link inevitably would impact with. This allowed to investigate the safety of the link when controlled in position. It was shown that the force of impact was in accordance with the ISO 15066 norm. However, in order to really investigate safety, it is concluded that experiments must be performed with the gearbox integrated in a multi-degree-of-freedom serial arm.

On the other hand, performance of the control scheme was investigated by means of position accuracy. It was concluded that the relative position error of 1% can be reached for larger virtual stiffness's ($K_d \geq 2$ to $3Nm$). The accuracy drops significantly in case of low virtual stiffness, presumably due to stiction: the reference torque corresponding to the product of K_d with position error (in radians) is lower than the torque needed to initiate movement. A stiction

compensator was introduced to deal with cases where low virtual stiffness could be relevant. It was experimentally proven that this allowed the position accuracy to drop below 1% again.

Question 3

Can we develop a new algorithm for high ratio gearboxes that accurately predicts the impact torque?

A mathematical framework based on the general theory of higher-order-sliding-mode disturbance observers was applied onto the drivetrain composing of the newly developed gearbox and an electric motor. As these models are generally applicable for linear systems, a feedback linearization strategy based on friction- and gravity compensation was performed. It was shown in simulation that this model would make two times faster predictions compared to the typical disturbance observers in the literature, probably due the fact that speed is estimated via an observer, rather than differentiating the position signal. In the experimental validation it was indeed proven that safe control can be achieved. However, it is hypothesized that impact estimation accuracy can be further improved. One could reflect on the feasibility of this control scheme for a commercial actuator. Given that the experiments were conducted with a clock frequency of $1kHz$, this is well-suited for commercial microcontrollers.

5.2 Future work

The main goal of answering the question of maintaining safety of a single-degree-of-freedom joint in an HRI application without the use of a torque sensor has been reached. However, it is really at the end of a project in which new questions and possible paths arise. In what follows, five possible improvements and new paths are highlighted for the future.

5.2.1 Combining the HOSM DOB and MEDOB

In the validation it was observed that the multi-encoder based disturbance observer yields the best estimation of impact torques onto the actuator. On the other hand, simulations pointed out that the HOSM DOB has a rather fast estimation of impacts. One could combine the accuracy of the MEDOB with the speed of the HOSM DOB to achieve a better approximation. Disadvantages include more parameters to be fine-tuned and more complex mathematics to derive the formula for the observer. Additionally, an encoder at the output of the gearbox must be provided.

5.2.2 Comparing the estimation with a torque sensor

As was proven in the experimental validation, the performance of disturbance estimation was relatively weak for low virtual stiffness. One way to possibly improve this problem is explained in the next point. Another reason could also be because of the Pilz PRMS instrument's incapability of measuring low impact forces. Instead of comparing the multiplication of the measured force with the length of the robotic arm with the estimated disturbance torque, the direct integration of a torque sensor at the output of the test setup could provide a better signal to compare the estimated torque with. In short, the hypothesis lies in the fact that it is assumed that measuring torque should have better accuracy compared to estimating this via a force measurement.

5.2.3 Improved power-flow dependent friction model

For simplicity reasons, the assumption was made that the forward and backdriving efficiencies are identical. However, based on the obtained data for the friction model, a more realistic model taking into account power flow directions to determine efficiency could potentially be worth the additional complexity. It would be interesting to see this combined with the improved HOSM + MEDOB disturbance observer.

5.2.4 Expansion to multi-degree-of-freedom robotic arm

Another relevant improvement could be the expansion of the model to an application of multiple-degree-of-freedom robotic arms, necessitating a further investigation onto increased complexity. For instance the addition of Coriolis and centrifugal forces/torques, which are not present in one-degree-of-freedom joints, making the amount of parameters to fine-tune and model increase a lot. It could be interesting and helpful to investigate this path further on, especially when the gearbox were to be implemented in a commercial robotic joint actuator.

5.2.5 Improved parameter choice using surrogate model

Although four different disturbance observers were build and tested for the application, focusing on one of them extensively might result in even better performance. For instance, the HOSM DOB has quite some parameters that have a lot of influence on speed and accuracy of the torque impact predictions. Using a surrogate model with these parameters as design variables and an objective function weighted between impact accuracy and detection duration might result in better performance. Parameters like cut-off frequencies for the Tustin derivatives might even be included inside these design variables.

Appendix A

Modeling backlash in simple meshings

Modeling backlash can be done by modeling the contact between gears as a mass spring damper system [44], or in Simscape using the 'simple gear' block. The user can choose between different types of models for backlash. One of them is via the use of the coefficient of restitution COR for rotating bodies [45]. This block drastically reduces the compile time and run-time of the simulation. That is why a mathematical framework has been build to identify the correlation between the efficiency loss and the coefficient of restitution COR in lashing gears. The coefficient of restitution COR for rotating bodies is:

$$COR = \frac{\omega_1^{after} + i\omega_2^{after}}{\omega_1^{before} + i\omega_2^{before}}, \quad (A.1)$$

where the subscript j stands for body j . The superscript *before* means the value of the quantity right before collision and *after* means right after collision. ω_1^{before} thus represents the speed of body 1 right before collision. Writing down the equation for conservation of momentum yields:

$$J_1\omega_1^{before} - J_2\omega_2^{before} = J_1\omega_1^{after} - J_2\omega_2^{after}. \quad (A.2)$$

and Conservation of kinetic energy:

$$\frac{1}{2}J_1(\omega_1^{before})^2 + \frac{1}{2}J_2(\omega_2^{before})^2 = \frac{1}{2}J_1(\omega_1^{after})^2 + \frac{1}{2}J_2(\omega_2^{after})^2 + E_{lost}, \quad (A.3)$$

where E_{lost} represents the lost energy in the collision. Solving eq. (A.1), eq. (A.2) and eq. (A.3) to ω_1^{after} , ω_2^{after} and E_{lost} allows to find an expression for the efficiency of the collision $\eta_{collision}$:

$$\eta_{collision} = \frac{\frac{1}{2}J_1(\omega_1^{before})^2 + \frac{1}{2}J_2(\omega_2^{before})^2 - E_{lost}}{\frac{1}{2}J_1(\omega_1^{before})^2 + \frac{1}{2}J_2(\omega_2^{before})^2}. \quad (A.4)$$

This formula (which is not written down because of its complexity) depends on the speeds of the two gears colliding ω_1^{before} and ω_2^{before} and the coefficient of restitution COR . The change in COR on the efficiency is shown in fig. A.1. The 1X synchronous line has been plotted additionally

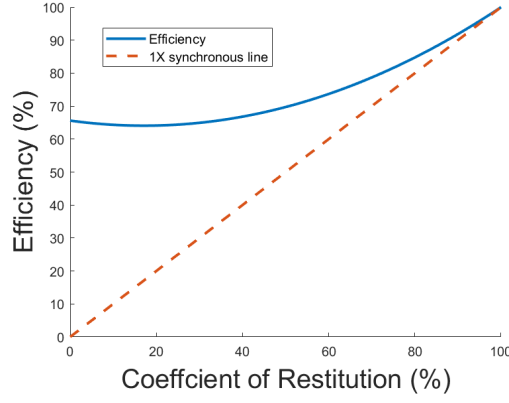


Figure A.1: Efficiency of a simple pinion (1) - gear (2) system when the pinion and gear have an angular velocity of resp. $\omega_1^{before} = 1$ rad/s and $\omega_2^{before} = 4$ rad/s for different values of the coefficient of restitution COR . The following parameters were used: $J_1 = 0.00075 \text{ kgm}^2$, $J_2 = 0.0025 \text{ kgm}^2$ and $i = 1.4$.

to showcase how nonlinear the relation $\eta_{collision} - COR$ is. It also shows that for low values of COR the efficiency is substantially higher than the COR .

For simulations, a COR of 0.84 should be used for steel gears [46]. For a simulation implementation, eq. (A.4) should be used in each meshing and reduce the output torque by $(1 - \eta_{collision})$ depending on the current speeds.

Appendix B

ISO 15066

In this section, some extra formation concerning the ISO 15066 norm is highlighted. The biomechanical limits for the quasi-static and transient contact cases are presented in fig. B.1. The forces and pressures F_T , p_T , F_S and p_S are also explained in that table. The effective mass and spring constants for a specific body part impacted by a robot/cobot are presented in fig. B.2.

Body region	Specific body area		Quasi-static contact		Transient contact	
			Maximum permissible pressure ^a p_s N/cm ²	Maximum permissible force ^b N	Maximum permissible pressure multiplier ^c P_T	Maximum permissible force multiplier ^c F_T
Skull and forehead ^d	1	Middle of forehead	130	130	not applicable	not applicable
	2	Temple	110		not applicable	
Face ^d	3	Masticatory muscle	110	65	not applicable	not applicable
Neck	4	Neck muscle	140	150	2	2
	5	Seventh neck muscle	210		2	
Back and shoulders	6	Shoulder joint	160	210	2	2
	7	Fifth lumbar vertebra	210		2	2
Chest	8	Sternum	120	140	2	2
	9	Pectoral muscle	170		2	
Abdomen	10	Abdominal muscle	140	110	2	2
Pelvis	11	Pelvic bone	210	180	2	2
Upper arms and elbow joints	12	Deltoid muscle	190	150	2	2
	13	Humerus	220		2	
Lower arms and wrist joints	14	Radial bone	190	160	2	2
	15	Forearm muscle	180		2	
	16	Arm nerve	180		2	
Hands and fingers	17	Forefinger pad D	300	140	2	2
	18	Forefinger pad ND	270		2	
	19	Forefinger end joint D	280		2	
	20	Forefinger end joint ND	220		2	
	21	Thenar eminence	200		2	
	22	Palm D	260		2	
	23	Palm ND	260		2	
	24	Back of the hand D	200		2	
	25	Back of the hand ND	190		2	
Thighs and knees	26	Thigh muscle	250	220	2	2
	27	Kneecap	220		2	
Lower legs	28	Middle of shin	220	130	2	2
	29	Calf muscle	210		2	

Figure B.1: Biomechanical limits [34].

Body region	Effective spring constant	Effective mass
	K	m_H
	N/mm	kg
Skull and forehead	150	4,4
Face	75	4,4
Neck	50	1,2
Back and shoulders	35	40
Chest	25	40
Abdomen	10	40
Pelvis	25	40
Upper arms and elbow joints	30	3
Lower arms and wrist joints	40	2
Hands and fingers	75	0,6
Thighs and knees	50	75
Lower legs	60	75

Figure B.2: Effective masses and spring constants for the body model.

Body region	Maximum transferred energy
	E
	J
Skull and forehead	0,23
Face	0,11
Neck	0,84
Back and shoulders	2,5
Chest	1,6
Abdomen	2,4
Pelvis	2,6
Upper arms and elbow joints	1,5
Lower arms and wrist joints	1,3
Hands and fingers	0,49
Thighs and knees	1,9
Lower legs	0,52

Figure B.3: Energy limit values based on the body region model.

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