POLITECNICO DI TORINO

Corso di Laurea Magistrale in Ingegneria Aerospaziale

Master's Thesis

Development of a SimScape model for prognostic of electro-hydraulic actuator mechanical components

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April 2022

ABSTRACT

The aeronautical industry is constantly evolving in terms of efficiency, consumption, accessibility, environmental impact and, in particular, safety. Each product, whether it is intended for civil or military use, is subject to continuous study and validation in order to meet compliance requirements. Nowadays, thankfully, the use of computers and calculators has made this process easier and faster, as well as more accurate.
One of the main steps to validate and study a product is to simulate its behaviour digitally. Not only this process allows a more complex study, but it also saves time and costs.

This thesis will examine this study and the validation process for a servovalve model in an electro-hydraulic control system.
The study will focus not only on the simulation of the actuator behaviour, but also on the phenomena that affect the behaviour (e.g. pressure drops, mechanical backlash and friction).

In the initial part of this thesis, there will be an introductory description of the different actuation systems and the different servovalve models that make up their architecture, of which, afterwards, the different operating characteristics and relative non-linearities will be analysed.
A Simulink model of the servovalve will first be studied and analysed. Then, this model, whose lements and results will be analyzed, will be used as a reference model.
The SimScape model is then presented, describing the choices and solutions that characterized the modeling and made it possible to achieve similarity with the reference model.

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Chapter 1

Flight Commands

1.1 Control Surfaces

One of the main aspect of an aircraft is the ability to maintain its stability during flight, complying with the requirement of the controllability and maneuverability. In order to make this happen, it is necessary to have the presence of movable control surfaces.
These surfaces, changing the shape of the wings and the tail, have the purpose to modify the overall aerodynamics forces on the aircraft in order to obtain changes in the attitude of the aircraft.

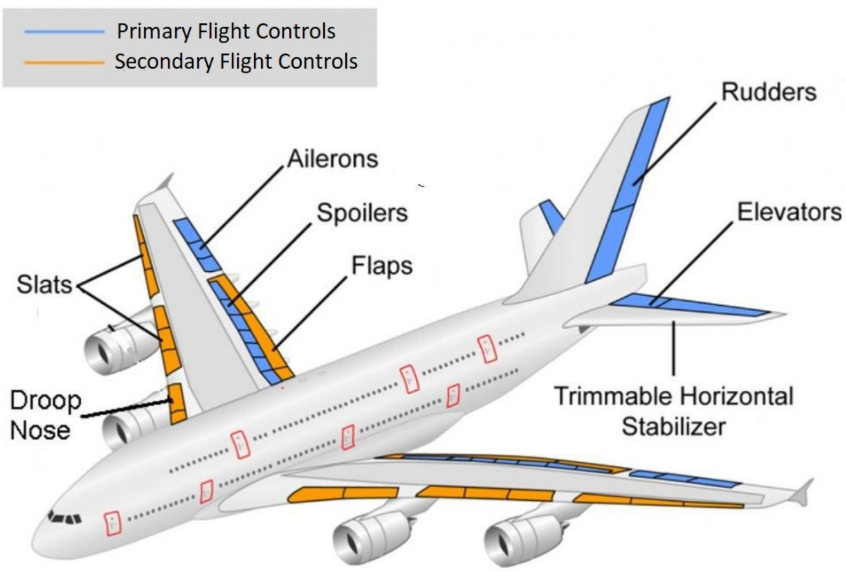


Figure 1.1: All of the control surfaces on a fixed-wing aircraft

The movable control surfaces on the aircraft are divided into two categories of flight commands:

- ***Primary Flight surfaces***

The surfaces that belong to this category are those surfaces that are used to modify the forces and momentum around the three aircraft principal axes of motion.
Modifying the motion and deflection of these surfaces, the pilot is able to control the aircraft in its attitude ad path. Due to this fundamental role, these surfaces are constantly moved by pilot and are constantly subject to forces and momentum, therefore they have to be *Stable, Instinctive and Sensitive.*

The requirement of *Stability* is referred to the ability of a flight control to maintain, even with external perturbation, its condition of equilibrium.
The requirements of *Instinctive* is referred to the relationship between the input given by the pilot and the movement of the aircraft (e.g. to a positive deflection of the cloche, the pilot wants the aircraft to pitch).
Lastly the flight control has to be *Sensitive*, which means that pilot needs to feel the force caused by the deflection of the surface during the flight, and this feedback has to be also proportional to the aircraft speed.

- ***Secondary Flight surfaces:***

These commands differently from the first one, are less important for the control of the aircraft in fact are used to compensate, during the flight and in certain maneuvers, those variations in weight, velocity and attitude and also to reduce the effort of the pilot during the flight.

All of these surfaces has to be moved or deflected from their nominal position to the position requested by the pilot, and in order to make this happen it is used a dedicated system, namely the *Flight Control System.*

1.2 Flight Control System

In a conventional fixed-wing aircraft the flight control system consists in all of the subsystems that ensure the control of the aircraft path and the attitude during the flight.
It is usually made of all the control surfaces over the wings and tail, the cockpit controls and all the other connections between these last two.
In addition, another important element that is implemented in the modern aircraft is the FCC (*Flight Control Computer*), which takes as input the command given by the pilot and all the data from the sensor installed on the aircraft, processes them and gives, as output, the command to the actuators of control surfaces.

There are three types of command transmission:

1.2.1 Mechanical control system

This kind of control system uses the mechanical transmission on command and it is usually transmitted manually to the system by the pilot.
This system uses a collection of mechanical parts as rods, gears, tension cables and also chains to transmit the commands directly, from the pilot, to the control surfaces.
This kind of transmission is now an obsolete system because the force to deflect the control surfaces is directly proportionated to the surface area and the flight speed, that means that all the force to move the control surfaces is given by the pilot.
Moreover, this kind of transmission is not able to give an high level of accuracy and stability.
Nowadays it is primarly used in small aircrafts (like in the Cessna Skyhawk).

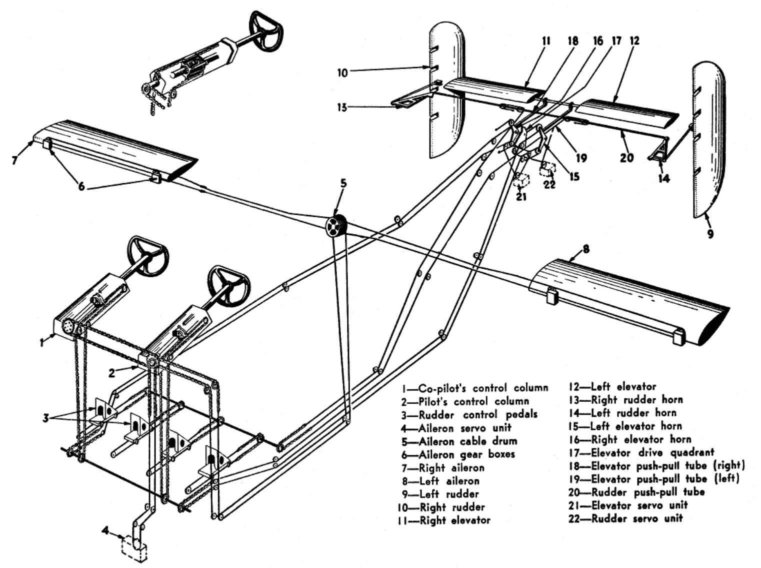


Figure 1.2: A scheme of mechanical control system

1.2.2 Hydro-mechanical control system

Due to the increase of the complexity of the most recent aircrafts, the mechanical system even increased in complexity and size, which means an increase of the weight of the aircraft and also the rise of the effort of the pilot during the flight.
The Hydro-mechanical system is less complex and lighter than the mechanical.
This system is made of two different parts. The hydraulic part, which uses

pumps, valves and tanks to converts an hydraulic pressure into a movement of the control surface. The mechanical part that connects the pilot and the cockpit system with the hydraulic part using rods, gears and cables.

1.2.3 Electrical control system

This kind of control system is the newest system implemented in modern aircrafts.
The idea behind this system is to remove those mechanical parts that makes the control system complex and heavy, using only digital signals. Not only does this new system increase the performances of the entire vehicle due to the reduction in weight, but it also makes the entire system easier to maintain.

The last element of a control system, before the control surfaces described earlier, is the actuator. This element main purpose is to move the control surfaces proportionally to the command given by the pilot and the FCC.

1.3 Actuators

All of the surfaces described earlier need to be moved from their nominal position, once the pilot gives the command to the control system. The actuator is the element between the flight control system and the control surfaces that makes this happen.
Depending on what kind of flight control system is implemented in the aircraft and on the performances needed by the system, it is possible to find different types of actuators.

1.3.1 Hydromechanical Actuators (HMA)

This type of actuator is made of a three-centered lever from which the pilot commands arrive. The displacement of this lever generates a movement of a spool, which has the role of opening or closing the connection with the hydraulic part.
According to the movement of the spool, a pressure difference is generated between the two chambers of an hydraulic cylinder, which causes the movement of the jack.
The latter element, the jack, is directly connected to the “user”, which in this case is the control surface.
This kind of actuator, although it is considered to be a safe and reliable actuator, is nowadays used increasingly less, due to its weight and complexity.

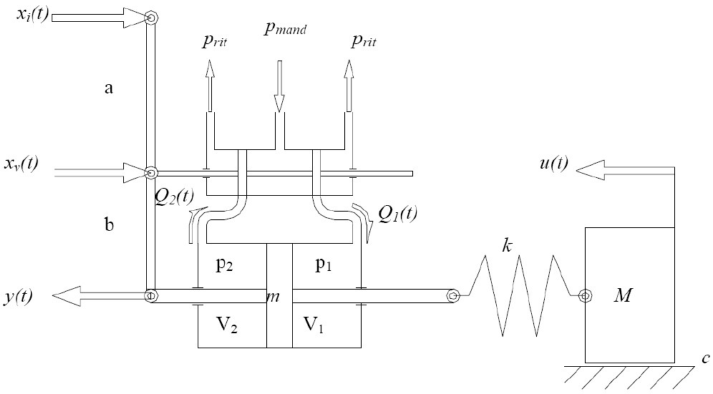


Figure 1.3: Scheme of an Hydromecanical actuator (HMA)

1.3.2 Electromechanical actuator (EMA)

Currently in aircrafts, whether military or civil, the actuation system is controlled hydraulically. However, in recent decades the aviation industry is moving more and more towards a "more-electric" concept, both to increase the precision and performance of the system, but especially because the development of increasingly innovative and reliable electric motors, allows to achieve high performances.
Electromechanical actuators overcome the problems of using an hydraulic system and directly transform the electrical signal into a mechanical signal.

In addition, all pressure losses due to the hydraulic system are avoided, leaving only those problems related to the mechanical part (e.g. friction).

One of the most used models of electromechanical actuator is the one formed by an electric motor, which converts the command of the pilot, in a mechanical signal to a worm screw. The worm screw is connected to a gearbox and a ballscrew, which has the task both to transfer the command to the screw and to transform the rotative signal in a linear signal.

However, even if the system is safer and more precise, it does not allow to obtain high output power, which is necessary to move the primary control surfaces.
The development of this technology is obviously in continuous growth.

Figure 1.4: A model of an Electromechanical actuator(EMA)

1.3.3 Electrohydraulic actuator (EHA)

In order to overcome the problems of hydro-mechanical actuators and the current shortcomings of electro-mechanical ones, electro-hydraulic actuators are increasingly used in modern aircraft.
In these systems, the mechanical part that used to characterize HMAs has been removed. The command given by the pilot is no longer transmitted mechanically but it is converted, through a sensor, into a digital signal which is then converted into a mechanical movement in the spool.
From the spool to the control surface, the operation of the actuator is similar to that of the model described above.
The advantage of this type of actuator over the previous one is that, since there is no mechanical connection, the system requires less maintenance and it is faster, the overall weight of the system is reduced and better accuracy is obtained.

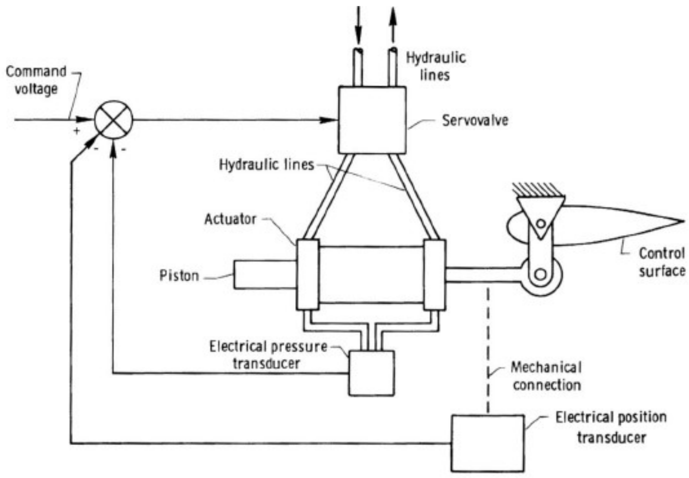


Figure 1.5: A scheme of an Electrohydraulic actuator

This type of actuator, the electrohydraulic one (EHA), is the one that will be considered and studied during this thesis work.

Chapter 2

Servovalves

The actuator is the element within the control system that allows the transmission of the command, provided by the pilot, to the control surface. Inside the actuator, the main element is the servovalve, which has the task of connecting the hydraulic circuit of the actuator, varying the flow rates and pressures, with the mechanical circuit that will move the surface.
Usually this type of servovalve, by means of a closed feedback loop and a PID controller, also allows to manage possible errors due to external loads or to the failure to reach the required position upstream.

In the following paragraph, three types of servovalves will be described.

2.1 Servovalve Types

2.1.1 Direct Drive Valve (DDV)

In this type of servovalve, the movement of the spool, i.e. the element that connects the chambers of the actuator with the hydraulic circuit, is driven by a magnetic force. The system is formed by a solenoid, which will be traversed by currents generated by an electronic card. This very electronic card has the task of transforming the input of the pilot and the feedback signal, into the two voltages in the solenoid. The error evaluation is managed by a linear sensor (LVDT), which makes the servovalve precise and stable.

The advantage in this servovalve is that it reduces the stages of operation, which allows for reduced leakage losses and non-linearities generated by spool control. Moreover, in this type of servovalve it is possible to obtain high powers with extreme precision, it is in fact mainly used in military aircrafts.
On the other hand, being its operation defined by a magnetic force, it is very sensitive to the high-frequency electromagnetic interference.

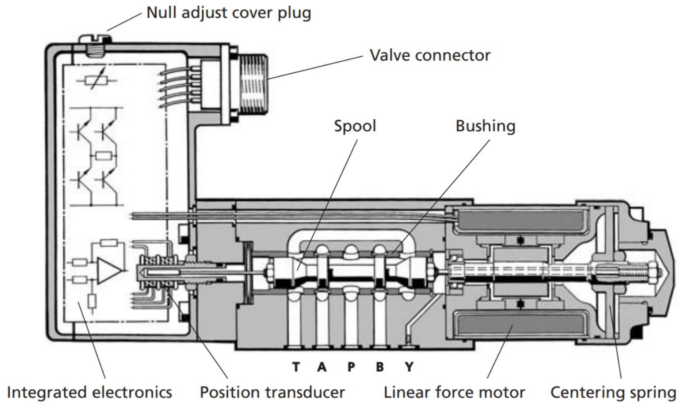


Figure 2.1: Scheme of Direct Drive Valve

2.1.2 Jet Pipe Valve

In this servovalve model, the displacement of the spool is managed by a pressure difference at its extremes.
Upstream of the servovalve a torque motor moves the position of a jet pipe which, moving from its neutral position, changes the flow rate in the two chambers extreme to the spool. This change in flow rate generates a change in pressure and, consequently the spool moves under hydraulic force, opening or closing the chambers of the actuator.

This type of servovalve allows high precision and lower weight than other models, however it is subject to several problems related to microscopic particles inside the fluid, which could lead to an obstruction of the jet or the nozzles, thus causing a wrong operation.

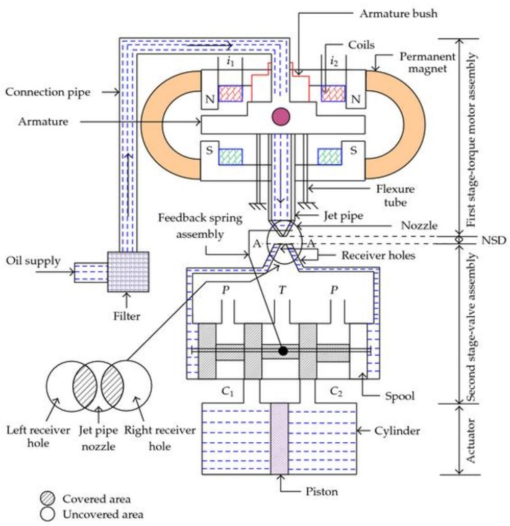


Figure 2.2: Scheme of Jet Pipe Valve

2.1.3 Flapper-nozzle valve

The Flapper-nozzle valve, that will be analyzed in this thesis, is also the most common in control systems.

This servovalve is also called Two-Stage Flapper-Nozzle Valve, because it is formed by two stages.

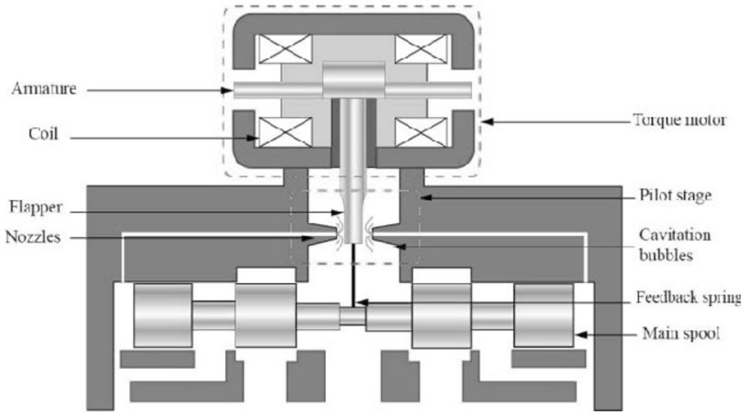


Figure 2.3: Scheme of Flapper-nozzle valve

The first stage, which takes the input from the pilot and the error signal from the control logic, is the stage related to the electric torque motor which, based on the command, will create a potential difference in the windings inside the motor.
The current produced will generate an induced magnetic field that will interact with the permanent magnets attached to the anchor, causing it to rotate.

The flapper, being rigidly connected to the anchor, will also be subjected to a rotation, which will vary its relative position between two hydraulic nozzles. The nozzle towards which the flapper is approaching will be characterized by an increase in hydraulic resistance, compared to the one from which the flapper has moved away. In these valves, the hole size of the nozzles is in the range of $0.25 - 0.5mm$, while their distance from the flapper is about $0.06 - 0.075mm$.

This variance will generate a pressure difference that will act on the second stage of the servovalve, which is the hydraulic one.

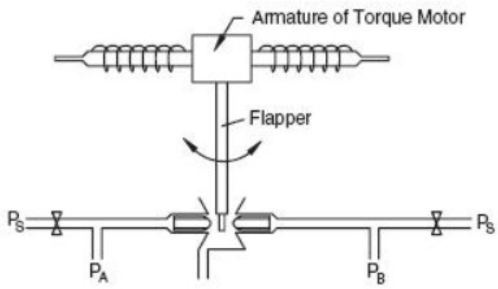


Figure 2.4: Detail of the first stage of Flapper-nozzle valve

In this stage, the differential pressure generated by the nozzles will act on the two ends of the spool.
Depending on the pressure and on the surface of the spool, a force will be generated and it will move the spool from its neutral position.
The spool, moving inside the sleeve, will connect the two chambers of the hydraulic cylinder with the hydraulic pressure of supply and with the hydraulic pressure of the tank. This additional pressure difference, in the chambers of the actuator, will generate a force and thus a displacement of the jack, to which, in this case, the control surface is connected.

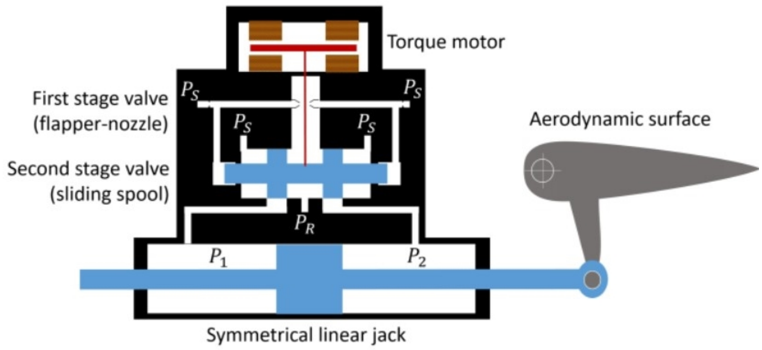


Figure 2.5: Flapper-nozzle valve with the Jack and the surface control

2.2 Servovalve characteristics

Knowing just how servo valves work is not enough to know how to model and analyze them.
A fundamental aspect in describing any type of servovalve, is to know its characteristics, that is to know the laws, the parameters and the non-linearities of which the servovalve is affected.
In the flapper-nozzle valve, in particular in the second stage, several aspects come into play that together allow to obtain a displacement of the actuator and therefore the generation of a mechanical force.

However, proceeding upstream, this result is obtained thanks to the generation of a pressure difference, and therefore an hydraulic force, between the two chambers of the cylinder. This pressure difference, however, is instead caused by a variation of the inlet and outlet flow rates in the two chambers. In the initial part of the second stage, it easy to see that the variation of the flow rate in the chambers is generated because of the spool movement, which moving from his *Null Position*, puts in communication the two chambers of the cylinder with different pressures of delivery and supply.
The correlation between all these aspects is described by the characteristic of the valve, through which it is possible to study and predict its behavior.

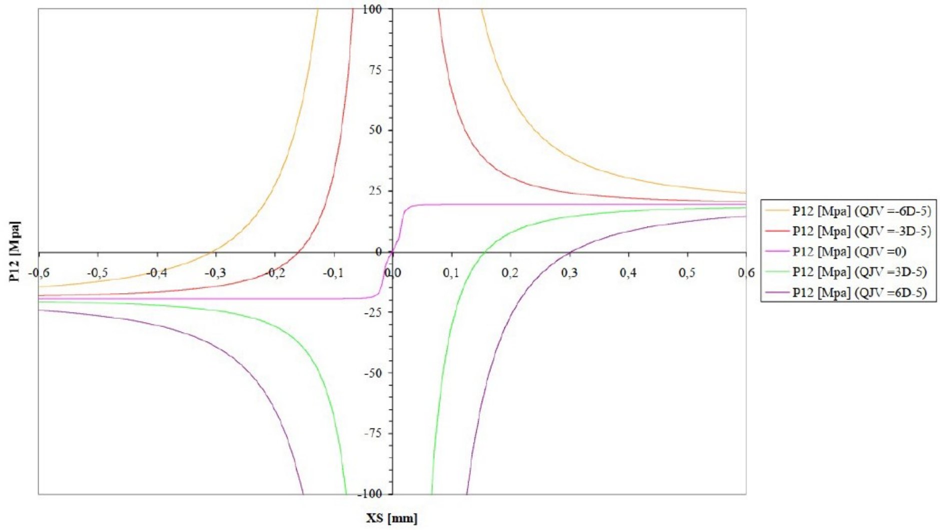


Figure 2.6: Characteristic of a Servovalve

Using this feature and making appropriate simplifications, it is possible to link the pressure and the flow rate inside the servovalve, with the displacement of the spool inside the sleeve.
Through the use of two gains, one in flow and one in pressure, it is possible to linearize the operation of the servovalve and then its characteristic.

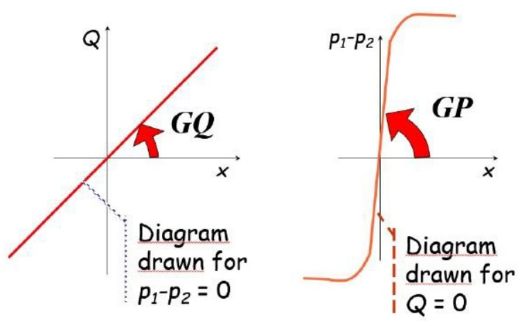


Figure 2.7: Gains of the servovalve

$$GQ = \left(\frac{QJ}{XS}\right)_{\text{P12=0}} \qquad GP = \left(\frac{P12}{XS}\right)_{\text{QJ=0}}$$

Where GQ and GP are the gains in flow and pressure, respectively, both evaluated under zero differential pressure (P12 is the pressure difference between the two chambers in the cylinder) and zero flow condition.
Both gains are related to spool displacement (XS).
This first description of the servovalve allows a first linearization of its operation.

2.3 Servovalve Non-Linearities

In this description of all the non-linearities that a servovalve could have, and electro-hydraulic valve as the Flapper-nozzle will be analyzed, starting from the first stage, the electric one, to the idraulic one.

2.3.1 Hysteresis

In valves characterised by an electrical stage, as in EHAs, it is important to take into account the possible non-linearities that can affect this stage. First of all is the possible occurrence of *Hysteresis*.
Hysteresis is a phenomenon that causes a system to have a quantity dependent not only on the instantaneous values of that quantity, but also on the values previously assumed.
In the case of magnetic hysteresis, ferromagnetic substances tend to retain their magnetization even when the induced magnetic field is zero.
In servo-valves, even under zero current conditions, this phenomenon leads to a magnetic field other than zero and therefore a non-zero flow rate downstream.
In order to obtain a non-null work, a greater flow rate is needed, therefore a greater opening of the lights and consequently a greater input current to the electric motor.

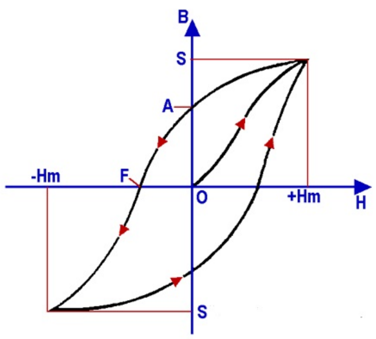


Figure 2.8: Tipical magnetization curve

2.3.2 Threshold

Similar phenomenon, that affects the current required by the motor is the *Threshold*. In this phenomenon, the variation and delay in the operation is due to the mechanical play of the servovalve.
In order to open or close the valve, there will be a certain range in which an increase or decrease in current does not correspond to any change in the flow rate and therefore to any kind of work.

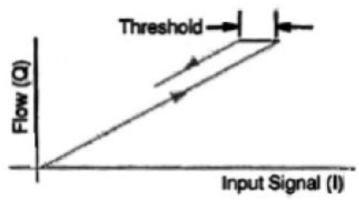


Figure 2.9: Effect of a threshold

2.3.3 Spool-lap condition

One of the first problems that occur in the proper functioning of a servovalve, particularly in an electrohydraulic valve, is related to the geometry and construction of what is the spool and its condition of covering the sleeve ports.
The conditions in which the spool can be found are of three types and each condition has a different effect on the operating characteristic.

- ***Over-lap condition***

In this condition, the spool has a geometry that is bigger than the light of the sleeve (usually not more than 20% of the light). In this case the valve is characterized by few leakage losses in the spool but on the other hand the valve will have a delayed response.

- ***Zero-lap condition***

In this condition the size of the spool is equal to that of the light, so in the case of null-position, the spool manages to totally cover the light.
Valves with this characteristic will have a lower time constant, which makes the system faster in response, however, leakage losses may occur.

- ***Under-lap condition***

In this condition the spool, unlike the previous ones, has a smaller dimension than the light.
This type of geometric condition leads to have a continuous flow of hydraulic oil that does not work, but on the other hand, it allows to have a time constant smaller and therefore an even faster system.

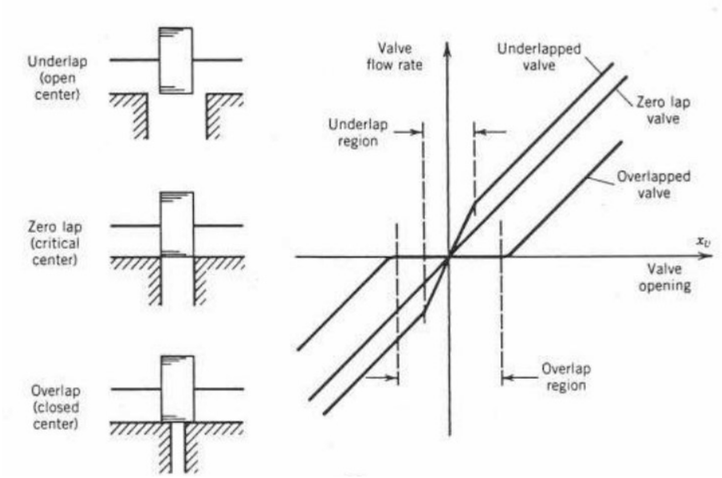


Figure 2.10: Lap conditions

2.3.4 Backlash

Due to the presence of interconnected mechanical elements, and due to the impossibility of obtaining a perfect fit, the *Backlash* phenomenon arises.
This usually means that dead band of displacement within which the movement of one body does not generate forces or movement in the other. This non-linearity unfortunately tends to increase over time due to wear, which tends to increase this gap.

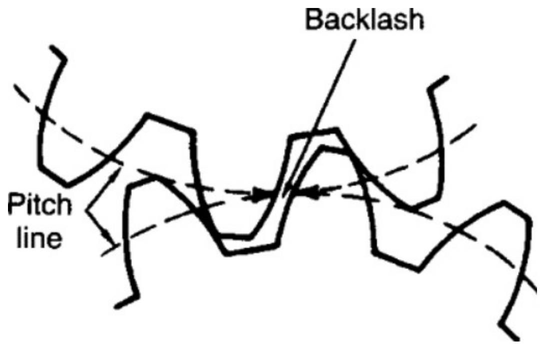


Figure 2.11: Backlash in a gear connection

2.3.5 Friction

One of the most important aspects to take into account when modelling and studying a mechanical system in motion is friction.
This non-linearity drastically modifies the performance of the system if it is not properly considered .
To be able to study it, a mathematical model that allows to evaluate every aspect of it under different conditions, is required, and in order to do this, it is necessary to make considerations and simplifications.

Friction is a miscroscopic and dissipative phenomenon that is generated between two bodies in contact and can be divided into two first categories, depending on whether or not the two bodies are in motion.
In the first case it is possible to talk about dynamic friction and this tends to oppose this movement by generating an opposite and contrary force.
The second case referes to static friction and it tends to keep the two bodies in contact with each other.

Furthermore, friction can be divided into three categories, depending on the type of contact between the two surfaces.

- *Sliding Friction:* occurs when two bodies crawl together.
- *Rolling Friction:* occurs when two bodies roll between them.
- *Viscous Friction:* occurs when a body moves in a fluid and comes into contact with the molecules of the fluid.

In this thesis, different mathematical models will be presented to describe sliding friction and therefore the relative static and dynamic friction, since it is the type of friction that most affects the operation of a flapper-nozzle servovalve.

It has been shown that sliding friction is strongly linked to the contact surface between the two bodies, and this phenomenon has been attributed both to the miscroscopic ridges and valleys of the surfaces that come into contact, but also to a chemical interaction between the materials that make up the bodies.
As a first approximation, it is possible to define static and dynamic sliding friction respectively as follows:

$$F_s = \mu_s F_N$$

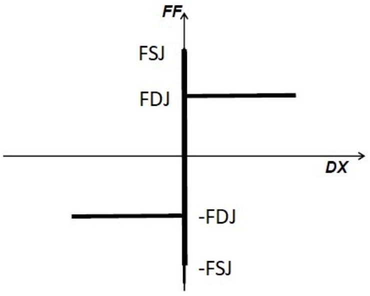
$$F_d = \mu_d F_N$$

Where μ it the adimensional friction coefficient, and it is not the same for static and dynamics friction and depends on property of the surfaces, and also where F_n represents the normal force between the surfaces.

However, the static friction force, within a certain speed range, is only dependent on the normal force, but once a "limit " has been exceeded, other viscous effects are also added to the friction force, which vary as a function of speed.

Coulomb's model

One of the first mathematical models to describe friction, describing it as a function of speed, is the Coulomb's model.



(a)

$$FF = \begin{cases} F_{att} & se\ v = 0 \wedge |F_{att}| \leq FSJ \\ FSJ \cdot sgn(F_{att}) & se\ v = 0 \wedge |F_{att}| > FSJ \\ FDJ \cdot sgn(F_{att}) & se\ v \neq 0 \end{cases}$$

(b)

Figure 2.12: Model and equation of Coulomb's friction model

In this model, the friction is modeled considering both the relative velocity and the Force applied between the two bodies.
If the relative velocity between the two bodies is equal to zero, than the friction force could be between FSJ (static friction force) and F_{att}, depending on the value of the latter. On the other hand, if the velocity becomes different from zero, the friction force is equal to FDJ (dynamic friction force).
The only problem with this model is that it does not describe what happens to the friction force when the relative velocity is very small, tending almost towards zero.
This situation was instead evaluated and studied by Striebeck, who showed that, for velocity ranges of the order of $10^-6[m/s]$, friction does not change from one value to another as instantaneously as described in Coulomb's model.

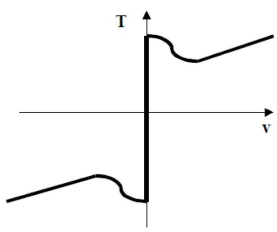
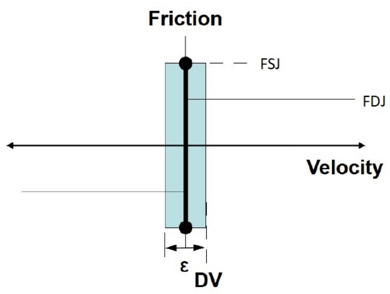


Figure 2.13: Striebeck's curve

In the modelling work, it is therefore essential to be able to describe all these friction-related aspects in the best possible way, also taking into account the risk that overly complex numerical problems may arise.

Karnopp friction model



(a)

$$f = \begin{cases} MIN(MAX(-FSj, h), FSJ) & se\ |v| \le \varepsilon \\ v = 0 & se\ |v| \le \varepsilon \\ sgn(v) * FDJ & se\ |v| > \varepsilon \end{cases}$$

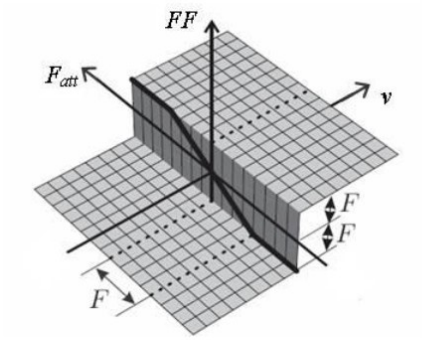
(b)

Figure 2.14: Model and equation of Karnopp's friction model

This model finds a solution to the problem of the Coulomb model with regard to velocities close to zero.
Karnopp adds a region of velocity around the null value ϵ. If the value of the relative velocity lies within this band, then the friction force will be equal to the minimum value between the FSJ and the value of the F_{att}, otherwise the value will be equal to the FDJ.
Using it solves a priori the problem of choosing the value of the friction force in the null speed range, on the other hand it needs to have an accurate definition of the null speed range.
If the size of this band is too small, there is a risk of not considering the phenomenon of adhesion in its entirety; if it is too large, there is a risk of overestimating the phenomenon. The choice of ϵ also depends on the integration step, which should be slightly smaller than the null speed band so that any reduction in speed can be more easily detected.

Borello fricion model

This model, named after the professor at the Polytechnic of Turin who developed it, removes the need to add a region around zero speed without neglecting the phenomenon of adhesion.
This model at each integration step evaluates the sign of the velocity.



(a)

$$FF = \begin{cases} -F_{att} & se \quad v = 0 \ \wedge \ |F_{att}| \leq FSJ \\ -\operatorname{sgn}(F_{att}) \cdot FSJ & se \quad v = 0 \ \wedge \ |F_{att}| > FSJ \\ -FDJ & se \quad v \neq 0 \end{cases}$$

(b)

Figure 2.15: Model and equation of Borello's friction model

If between one integration step and the next, the velocity has the same sign it imposes the condition of dynamic friction and therefore the friction force is equal to FDJ.
If between one integration step and the next, the velocity changes sign, then the model will reset it to a null value, and the friction condition will be the static one.
This model is the one that will be used in the following modelling and comparison work of the servovalve.

Chapter 3

Computational Simulink Model

The advent of computers, and therefore of computational modelling, has given a strong boost to the industrial production process. In the aeronautical sector in particular, not only has this improved the overall quality of the products and systems used, but has also made everything much faster and significantly less expensive.

Testing and validating most products or systems no longer means creating a physical model to be inserted into a series of test benches or experiments. Moreover, these kind of tests do not allow to obtain particular situations and operating environments, which means that it is not possible to obtain a complete and definite analysis of what the real behavior of the system could be during its entire life.
Simulation and analysis models are not only used in the design phase, but are also used during the whole operative life.
Knowing in advance how a certain system should behave under a given condition, makes it possible to anticipate any faults caused by malfunctions, so that it can be repaired at the next opportunity.

Modelling a system means, first of all, knowing everything that makes it up, each subsystem and element, knowing the relationships between them and knowing the mathematical and physical laws that allow it to function. Some systems, however, are so complex that they necessarily require a series of approximations and simplifications.
Obviously, this process must be done with full knowledge of the facts, since too much simplification could erroneously lead the model to not following reality.

3.1 Reference Model

3.1.1 General Overview

During the development of the thesis, a Simulink reference model has been used.
This model, made in the package Simulink of MatLab software, has been created by the professor Dalla Vedova Matteo.

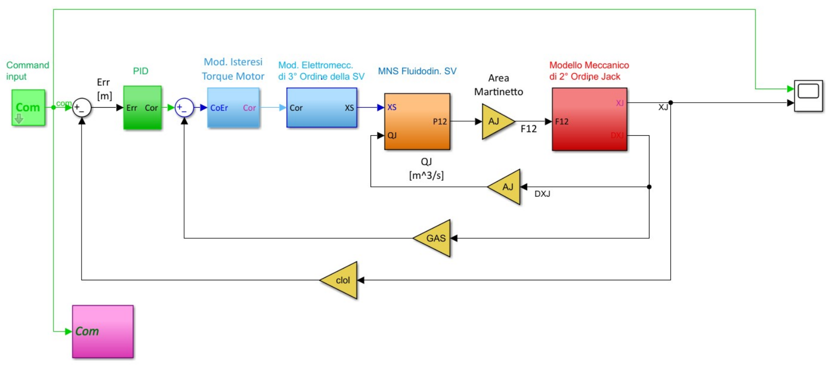


Figure 3.1: Reference Simulink model of electro-hydraulic servovalve

As already mentioned, the type of valve to be analysed in this thesis is a typical electro-hydraulic flapper-nozzle valve.
Since it is the type valve mentioned earlier, it is necessary to take into consideration not only the two stages that characterise it (the first electric-magnetic stage and the second hydraulic stage) but also the stage prior to the servo-valve and the actuator.

3.1.2 Command Subsystem

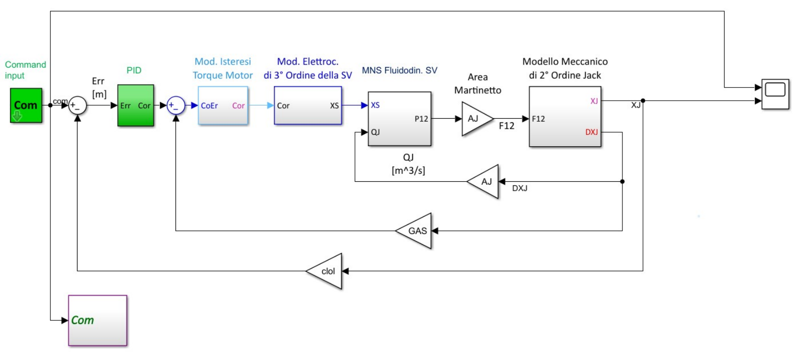


Figure 3.2: Light up of Command Subsystem in the Simulink model

The first block to be found in the logic flow of the model is the command block.
This element is used by the user in order to decide what type of input the user wants to provide to the system.
Opening this block, an interface will be found where the user can choose the type of input and the parameters that characterise it.

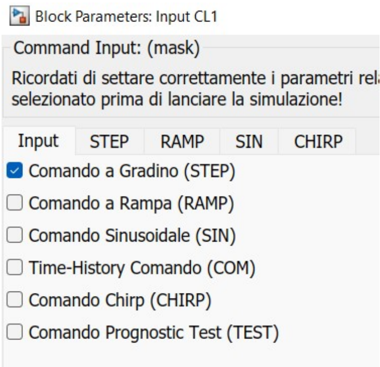


Figure 3.3: User interface for the command input

The user can enter one of the provided input types (Step, Ramp, Sine, Chirp, Time hystory and a Prognostic test command) as input or even several together.
The choice of command type obviously depends on the type of situation and response to be obtained or studied.
Being a model used for a servovalve for the actuation of the mobile surfaces of an aircraft, the type of input is of the order of a displacement and therefore of the deflection that the pilot would like to give to the control surface.

The input provided will be linked to the feedback signal of the system.
The value of the instantaneous end position reached downstream by the actuator is subtracted from the input.
This parameter represents the system error, i.e. how far the system is from the commanded position.
The fundamental element of this section, the one that receives the error in input, is the PID controller.

The PID controller consists of three analysis branches, each of which takes the error as input.

- ***Proportional branch***

This branch multiplies the error by a GAP gain, so that the output is proportional to the error value.

- ***Integrative branch***

This branch tries to reduce the stationary error, by taking into account the hystory of it, integrating the error in the time domain. Here the error is multiplied by the gain GAI.

- ***Derivative branch***

This branch tries to predict what the future trend of the error might be, so that it is possibile to "brake" it if it is tending to zero too quickly.
The error here is multiplied by a GAD gain.
Usually a low-pass filter is associated with this gain in order to reduce the error.

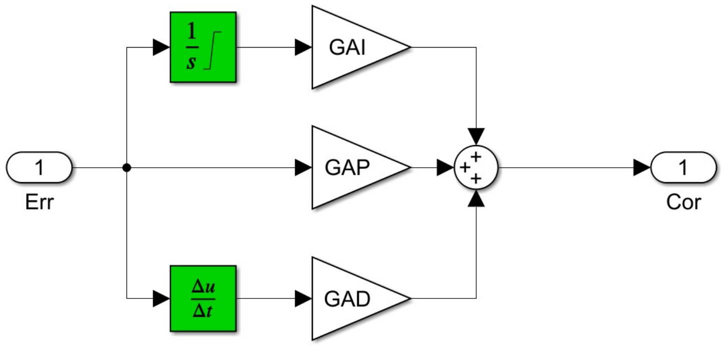


Figure 3.4: Controller PID

The error analysed as described is supplied to the first block characterizing the servovalve.

3.1.3 First Stage

The second subsystem that will be analyzed, is the one related to the second stage of the servovalve.
This subsystem is dived into two blocks: the Torque motor Hysteresis and the Eectromechanical model.

Torque motor hysteresis

The error given previously by the PID block, is now given into another block which has the role to simulate the first non-linearities of the servovalve, the hysteresis.

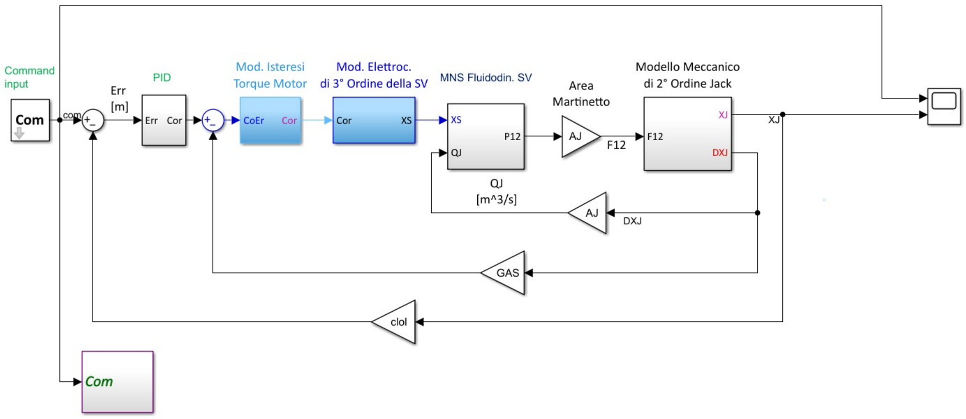


Figure 3.5: Light up of the First Stage in the Simulink model

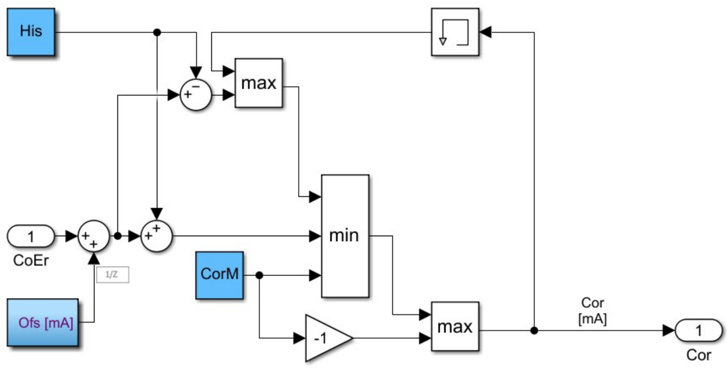


Figure 3.6: Torque motor Hysteresis block

In this block it is possible to notice that the effect on the current, depends on how it is varying.
If the input current is increasing, the hysteresis value will be subtracted, but if the current is decreasing, the hysteresis value will be added.
Either way, the output current to this block cannot exceed a value given by two extremes (CorM and -CorM).

In this block, not only is the hysteresis model inserted, but also two other non-linearities are simulated.
Thanks to the offset block (Ofs), to the input current signal, this value is added relative to any residual currents in the system.
Moreover, in the Ofs block, is added to the current signal via the Band-limited White Noise block, also another signal related to what may be the electromagnetic interference that the system can hear.

Electromechanical model

The current signal now enters what is the block related to the electro-mechanical model.

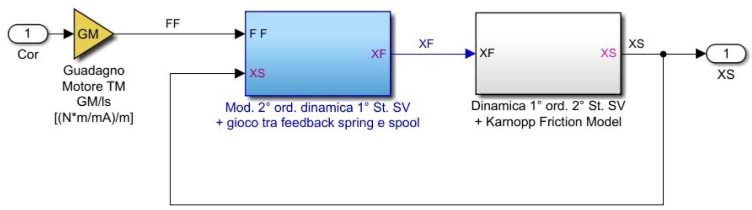


Figure 3.7: Electromechanical model blocks with light up ot the first stage block

The current signal is first of all transformed, multiplying it by the gain of the motor GM, in the value of the torque provided by the electric motor. This torque moment is relative to the center of gravity of the flapper, to which the torque moment is applied, therefore it means that at the output from the GM gain is just the torque force (FF) applied to the flapper.

To evaluate what the dynamics of the flapper subject to the FF force is, it is necessary to analyze what are all the forces at play.

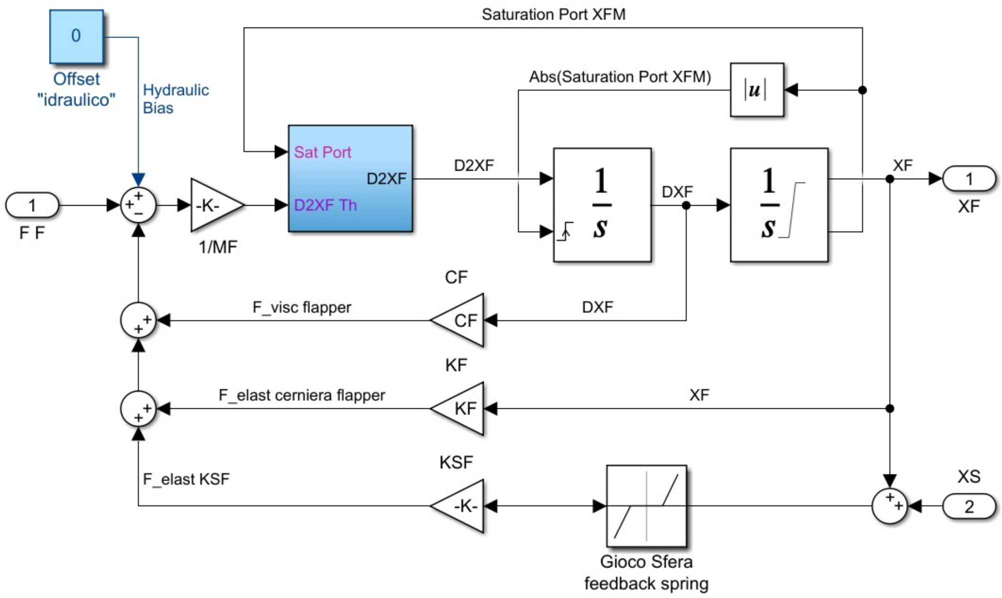


Figure 3.8: Model of flapper dynamics

It is already clear from the model that, in addition to the input torque force provided by the torque motor,it is necessary to consider the forces opposing this torsion.
These forces can be of three types:

- ***Vicous force***

This force is related to the environment in which the flapper is immersed. The flapper, being immersed in a fluid-dynamic oil, will necessarily be subject to viscous forces which oppose to its movement.
This force, in fact, is obtained multiplying the velocity of the flapper by a coefficient CF.

- ***Elastic force***

This eleastic force is linked to the fact that the flapper is tied to the anchor by a spring which allows it to rotate.
This force is obtained by multiplying the displacement of the flapper by a coefficient KF.

- ***Bendig force***

This force is related to the type of connection between the flapper and the spool. The flapper has a small ball at the end which fits into a small cavity in the spool.
This connection therefore generates a torque force.

Considering therefore all those forces which have an effect on the rotation and, as a consequence therefore on the translation of the flapper, it is possible to evaluate, by means of the second law of dynamics, what the acceleration is subjected to.
By integrating twice the signal of the acceleration it is possible to obtain both the rotation speed of the flapper, which also serves to evaluate the viscous forces mentioned above, and the translation, which serves both for the evaluation of the elastic recall and as input for the next block.

The other block in the Electromechanical model is the first block of the second stage of the valve.

3.1.4 Second Stage

The position of the flapper (x_F) is supplied as input to the second block inside the electromechanical model, i.e. the block relating to the second stage of the servo-valve

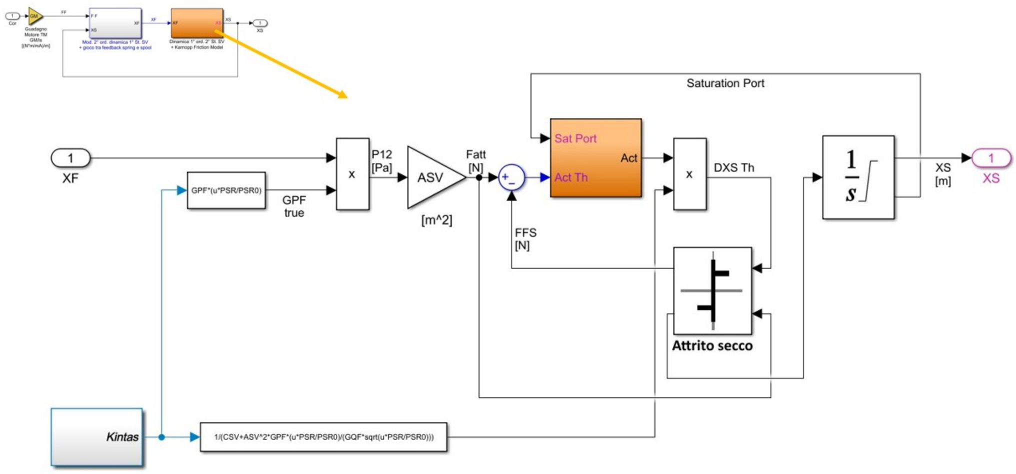


Figure 3.9: Model of the second stage of the servovalve

As described in the previous chapters, the displacement of the flapper is important in relation to the two nozzles that connect to the hydraulic chambers of the spool.
When approaching one of the two nozzles, the flapper will change the hydrodynamic resistance in the system, which will result in the generation of a pressure difference.
This pressure difference will act on the area at the ends of the spool.
Since the output of this block must be the displacement of the spool, it is necessary to evaluate the dynamics of the spool and therefore to evaluate the forces involved.
The balance of forces on the spool will then be:

$$F_{idr} = F_{spool}$$

F_{idr} is the hydraulic force generated by the hydraulic resistance caused by the flapper, and it depends by the spool area (ASV) and pressure difference generated (P_{12f})

$$F_{idr} = ASV \cdot F_{spool}$$

With regard to the forces acting on the spool, it is necessary to evaluate the dynamics of the spool.
There are a number of forces at work, however only two fo them have an important influence on the dynamics:

- ***Inertial Force***

$$F_{inertial} = M_{sv} \cdot \ddot{x}_s$$

- ***Viscous Force***

$$F_{viscous} = C_{sv} \cdot \dot{x}_s$$

As described in the previous chapters, the characteristic of the servo-valve allows to define the pressure difference by means of a gain (GPF).
As can be seen in the simulink model, the input is related to another fundamental parameter, relating to the hydraulics of the system.
As the system is driven by hydraulic oil, there is a need to consider any impurities in the oil itself.
These impurities could vary the nozzle areas and therefore change the pressure acting on the spool.
For this reason, it is necessary to modify the pressure gain (GPF).
This modification is taken into account by the K_{intas} block.
The pressure gain, which becomes (GPF_{true}) is now linked to another.

$$GPF_{true} = GPF_{design} \cdot u$$

where:

$$u = 1 - K_{intas}$$

The variable K_{intas} is user-defined, and can be between 0 and 1.

If a null value is defined ($K = 0$) then the pressure gain does not need to be modified as there is no contamination and therefore no clogging of the nozzles.
If, on the other hand, a unit value is defined ($K = 1$), the system then does not function due to complete blockage.

It is now possible to define the forces balance on the spool:

$$\dot{x}_S \left(C_{SV} + ASV^2 \frac{GPF}{GQF} \right) = ASV \cdot GPF \cdot XF$$

However, the force acting on the spool is linked to another acting force, namely the frictional force between the spool and the jacket.
The friction force is defined by the **Attrito secco** block, where the model used to define it is the Borello model, which follows the rules described in the previous chapters.

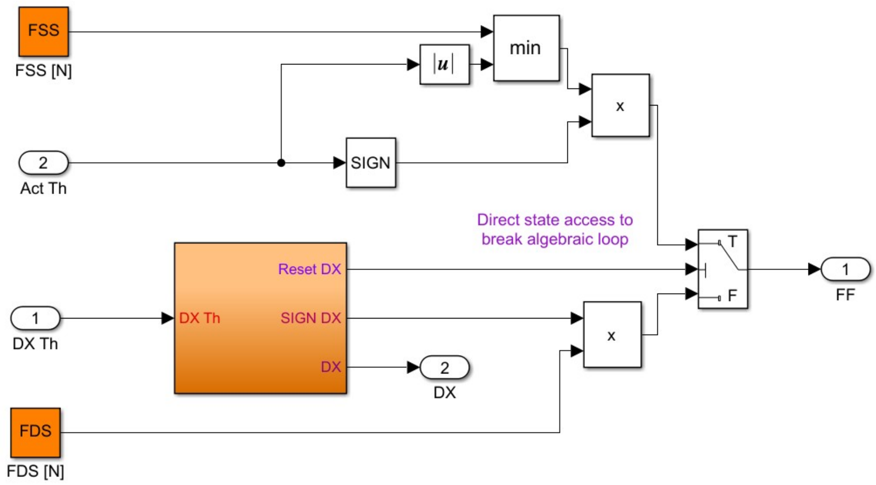


Figure 3.10: Borello's friction model

This block will evaluate the speed of the spool, and will then give as an output, the friction force (static o dynamic) (FF) that will oppose to the dynamic force described before.

$$\dot{x}_S \left(C_{SV} + ASV^2 \frac{GPF}{GQF} \right) = ASV \cdot GPF \cdot XF - FF$$

Fluid dynamic model

The output of the second stage block described earlier, is the spool displacement.
The displacement of the spool is related to the opening of the liner ports, which generates the connection of the valve with the chambers of the actuator and, therefore, the generation of the pressure difference that will move the jack.
The simulink model block, which is going to do this evaluation, is the valve fluid dynamics model block.
Two different fluid dynamics models can be selected in this block.
The first is a model based on linearization from the valve characteristic, exploiting what are the two gains.

$$GP = \left(\frac{P12}{XS} \right)_{QJ=0} \qquad GQ = \left(\frac{QJ}{XS} \right)_{P12=0}$$

Exploiting them it is then possible to link the pressure difference in the two chambers of the actuator, with the displacement of the spool.

$$P12 = GP \cdot XS - \frac{GP}{GQ} QJ$$

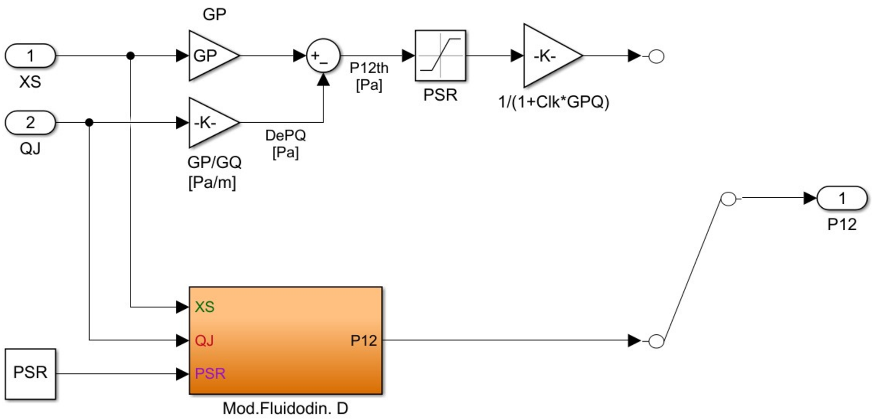


Figure 3.11: Fluid dynamic model choice

In this first choice it is clear how the pressure is limited, through a saturation block, by a minimum value (-PSR) and a maximum value (PSR).
It is also evaluated, through the final gain and a coefficient (CL_k) what can be the pressure drops caused by leakages

The second model instead does not take into account the linearization made previously, but instead considers in the pressure difference, two different losses.

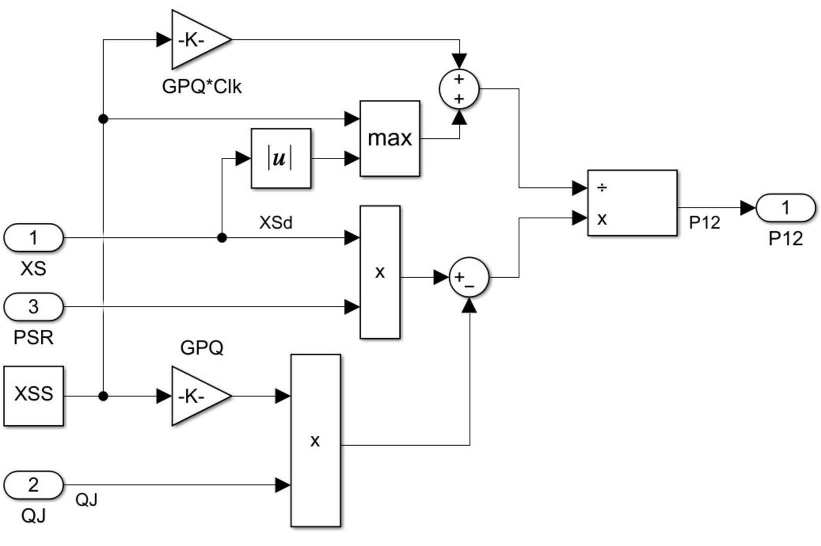


Figure 3.12: Fluid dynamic model in the second block

$$P_{12} = P_{12P} - P_{12Q} - P_{12LK}$$

Where P_{12p} is the differential pressure provided by the system, due to the opening of the lights:

$$P_{12p} = GP \cdot XS$$

To this pressure is then subtracted a first pressure loss, which considers the loss due to the drained (QJ):

$$P_{12Q} = \frac{GPS}{GQ} \cdot QJ$$

The second one takes in account the pressure loss due to the leakage:

$$P_{12LK} = P_{12} \cdot CL\_K \frac{GPS}{GQ}$$

Before obtaining the pressure difference, it is necessary to define two other parameters.

$$XSS = \frac{PSR}{GP} \qquad GPQ = \frac{GP}{GQ}$$

Where PSR is the differential pressure that takes into account the pressure changes.
Where instead GPQ, as shown, is the ratio between the two gain of the valve, which drive to definition of:

$$GP = \frac{PSR}{XSS}$$

and so:

$$GPS = \frac{PSR}{\max\left(|XS|,\ XSS\right)}$$

It is now possible to write the final equation that describes the pressure difference in this fluid dynamic model:

$$P_{12} = \frac{PSR \cdot XS - GPQ \cdot XSS \cdot QJ}{\max\left(|XS|, XSS\right) + GPQ \cdot XSS \cdot CL_k}$$

In this thesis, the last model described will be the one used as the reference model.

3.1.5 Actuator

The last element of the model is the actuator.
The output of the fluid dynamic model is the differential pressure P12, which needs to be given to the actuator sysetm.

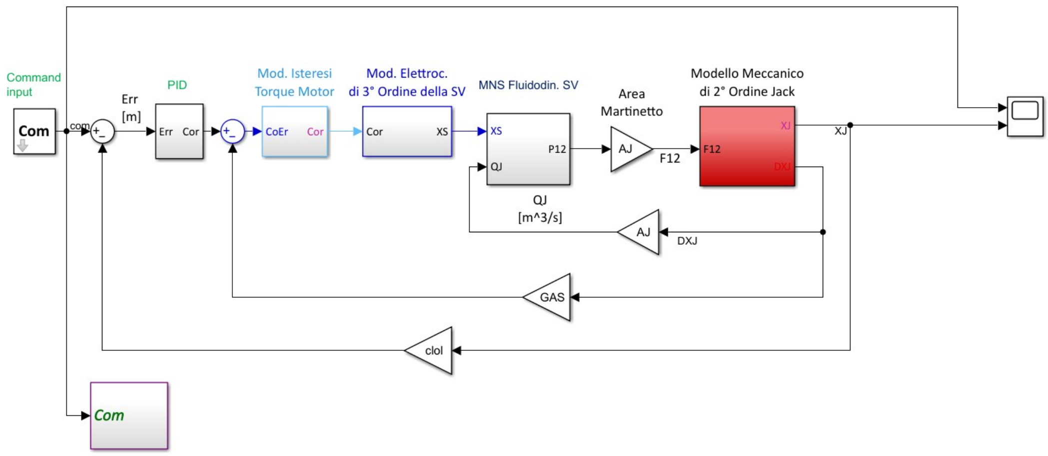


Figure 3.13: Light up of the actuator block

Having the pressure difference, it is necessary to provide as input to the jack block, what is the hydraulic force exerted on it.
The cylinder model used is that of a classic Double-acting Cylinder, that is a hydraulic cylinder characterized by two chambers separated by a piston, and both connected to the hydraulic system of the valve.

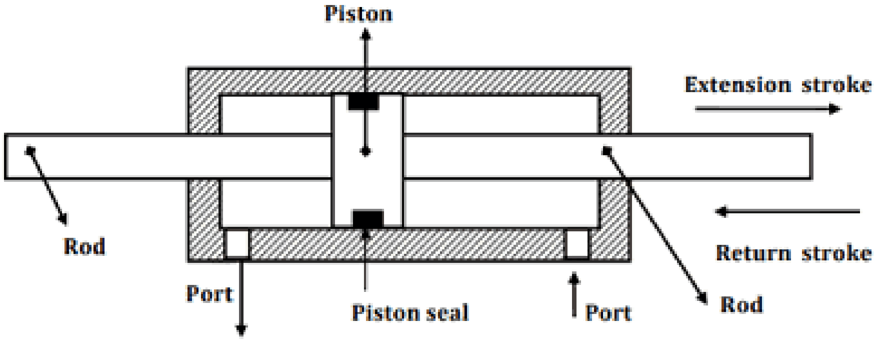


Figure 3.14: Double-acting cylinder with the piston-rod

The pressure difference at the outlet of the block of the fluid dynamic model already represents the pressure difference between the two chambers of the cylinder, therefore, the force exerted on the piston results to be:

$$F_{12} = AJ \cdot P_{12}$$

However the piston will not be subjected only to the force provided by the difference. For this reason, in order to describe the dynamics of Jack, it is necessary to evaluate all the forces that come into play in the system.
The jack, being formed by a cylinder and a rod, to which is then connected the mobile surface, is characterized by its own inertia that will affect the dynamics. The piston is also immersed and moves in the hydraulic oil, this leads to have a viscous force that opposes the translation motion.
It is also necessary to take into account any external forces that may act on the jack.

In this section, which is related to the study for aeronautical applications, this external force is linked to the aerodynamic forces acting on the mobile surface.

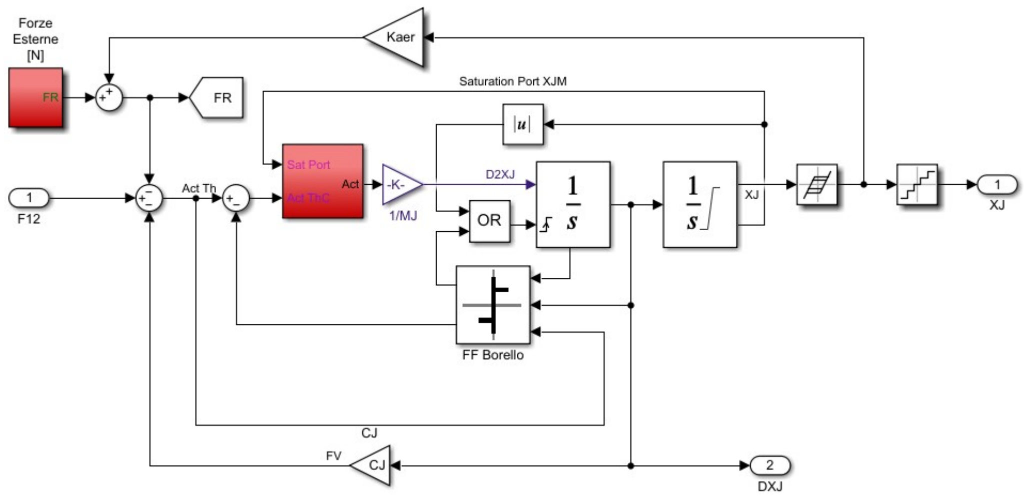


Figure 3.15: Model of the dynamics of the actuator

In the model it is also analyzed and evaluated the friction that is generated by the movement of the jack inside the cylinder.
The Borello friction model, in the friction block, evaluates this force, which opposes the motion.
The output of the second integrator, which is nothing else than the position of the jack and therefore its final position downstream of the command, is, before being sent to the feedback branch, modified through the Backlash block, which as described in the previous paragraphs, has the task of taking into account both those mechanical clearances that there may be in a mechanical system, necessary to the construction, but also those generated by any wear.

3.2 Simplified Model

The following model is a Simulink computational modeling of a flapper-nozzle valve, however unlike the model seen before, this one turns out to be simpler computationally speaking, but less accurate.

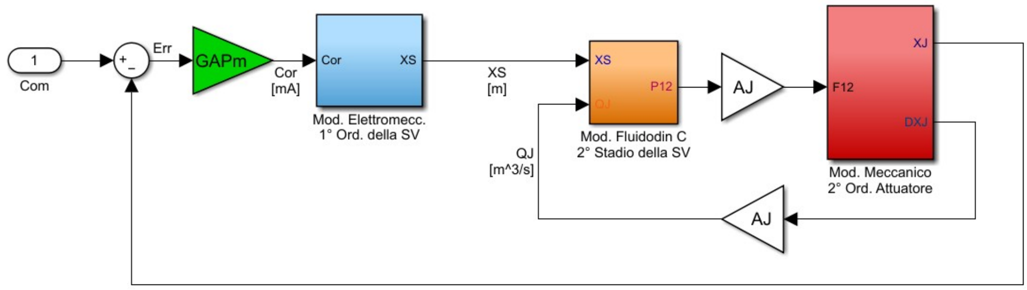


Figure 3.16: Simplfied Simulink model

The main difference between the two models is in the first stage of the servo valve.

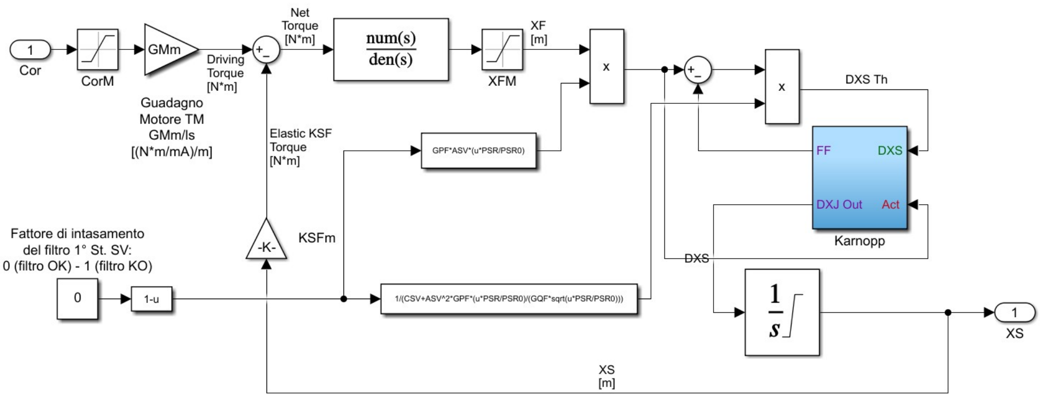


Figure 3.17: Model of the first stage of the valve in the simplified model

First of all, the signal and the error are not fed to a PID controller, but only a proportional gain is used.
Another difference is that the system is not evaluated by taking into account what are the real effects, but it is considered as a second order system.
The input current is related to the flapper displacement by a simple transfer function:

$$\frac{x_F}{Cor} = K \cdot \frac{1}{\frac{1}{\omega_n^2} + \frac{\zeta}{\omega_n} + 1}$$

Where the coefficient K takes into account the relation between the stiffness characterizing the flapper.

Another difference between the two models lies in the fluid dynamic model of the valve.
In the one used in the simplified model, the pressure difference is evaluated, downstream of the opening of the lights, by linearizing the valve with two gains.

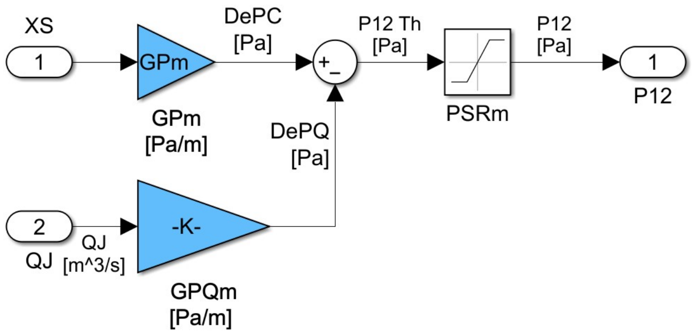


Figure 3.18: Model of the first stage of the valve in the simplified model

Finally, friction is evaluated using Karnopp friction model, which is less precise but simpler than Borello's, as seen in the description above.

Chapter 4

Computational SimScape model

4.1 General Overview

SimScape is an extension of the Simulink package, which enables the modelling of complex multidomain physical systems.
The main difference between Simulink and Simscape is that Simulink is mainly used for mathematical models where the data flow is linked to purely numerical modelling, whereas SimScape is linked to a data flow of physical quantities.

Like the Simulink model described above, the SimScape model of an EHA (with the flapper-nozzle valve), will be divided into four subsystems:

- ***Controller***
- ***First stage***
- ***Second stage***
- ***Actuator***

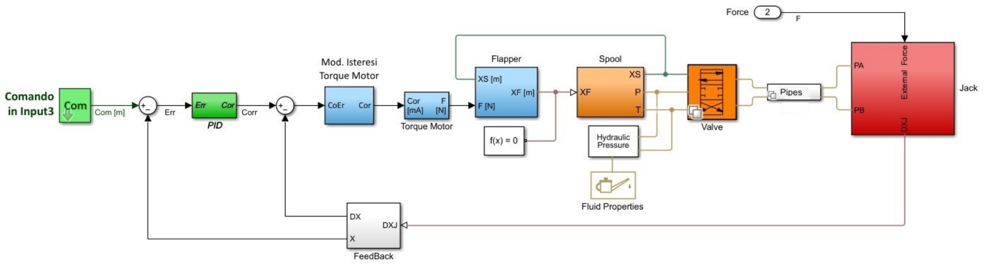


Figure 4.1: General overview of the SimScape model

4.2 Command Subsystem

The first subsystem in the model is obviously the command input block. Since this system is linked to a digital signal of a data given by the pilot, and since the modelling of this part of the system is external to the planned thesis, the same command subsystem used in the Simulink model is implemented.

The first block is where the user can enter different commands with different patterns and characteristics.
Being the command related to the desired deflection at the moving surface, it is expressed in meters and is then compared with the feedback branch, where the final position reached by the actuator is measured.
The error is then provided as input to a PID controller.

4.3 First stage

Like the simulink model, the error signal taken into analysis by the PID controller is then given as input to the second subsystem of the model, the first stage of the servo valve.

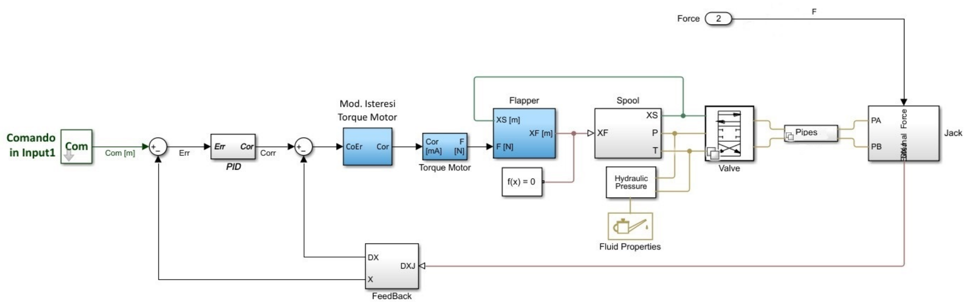


Figure 4.2: Light up of the first stage in the SimScape model

The blocks relating to what is upstream in the first stage are similar to those in the Simulink model.
To begin with, there is the implementation of the hysteresis of the electromagnetic system and the torque motor model, in which the output is the torque force applied to the flapper.

The first block modeled with SimScape is the block related to the flapper.

Flapper

The torque produced by the torque motor is used to rotate the anchor to which the flapper is attached.
As in the simulink block where the flapper's dynamics are studied, the forces acting on the flapper must be taken into account in this simscape model.

As mentioned above, the idea behind SimScape is to model a system by considering its various physical connections.
In each system modelled on simscape, it is necessary to insert what is called a ***Reference***, and the type of the latter varies according to the type of physical signal that characterizes the studied model.
In this model, as well as those that will be characterised by dynamics linked mainly to mechanical translation, the reference used is the *Mechanical Translational Reference*. This element represents a point of reference or "anchorage" to which all elements, that are rigidly bound together or to the ground, must be connected.

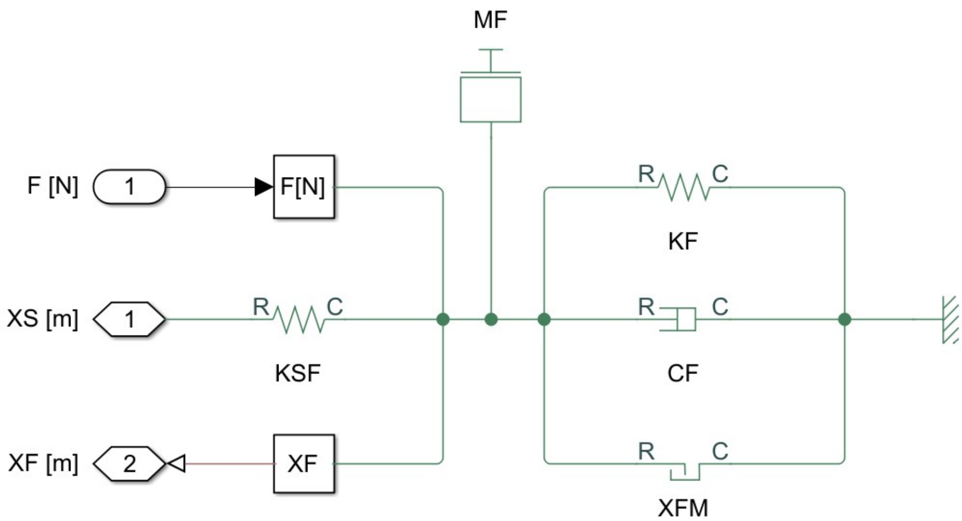


Figure 4.3: SimScape model of the flapper

As can be seen in the flapper model, the flapper is simulated as a mass with its own inertia (MF).
The mass, and therefore the flapper, is physically linked to a reference, which ideally represents the stiffness of the whole servo-valve.
Between the mass and the reference, however, there are physical elements that represent certain aspects of how the first stage of the valve is built.

- ***Translational spring KF***

This element represents the torsional rigidity of the hinge, which connects the flapper to the torque motor anchor.
It is modelled with the physical SimScape element of a translational spring, where the only parameter to be entered is the elastic stiffness.

- ***Translational damper CF***

This element, the translational damper, is used to model and consider the viscous forces the flapper is subjected to when immersed in hydraulic oil.
The parameter required by this block is precisely the oil damping coefficient.

- ***Translational hard-stop XFM***

The flapper, subjected to the torsion provided by the torque motor, will approach one of the two nozzles required to vary the fluid-dynamic resistance.
However, the translation of the flapper will be limited by what was said before, so it will be necessary to limit its translation in the model, and this is done by using the translational hard stop block.
This block takes into account not only the upper and lower limits within which the flapper can move (xfm and -xfm), but also the type of impact the flapper may face.
In order to do this, this block models the two limits as a spring and a damper, and by varying these parameters, it is possible to define the type of impact.

  – *Stiffness and damping applied smoothly through transition region, damped rebound*

  – *Full stiffness and damping applied at bounds, undamped rebound*

  – *Full stiffness and damping applied at bounds, damped rebound*

In the present work, the second model is used.

With regard to the input provided to this subsystem, there is a Simulink signal that comes from the block relating to the torque motor, which, through an *Ideal Force Source* (the **F[N]** block), generates the force that translates the flapper.
The mass inserted in the simscape model and therefore subject to an external force, will undergo its translation, which is measured by an *Ideal Translational Motion Sensor* (block XF).
This sensor allows not only the position of the mass to be measured, but also its translation speed.
Of the two measured data, only the position, and therefore the translation of the flapper, will be provided as input to the next subsystem block, the *SPOOL*.

4.4 Second stage

The study regarding the modelling of the second stage of the servovalve is addressed in the thesis by the colleague G. Caruso.
In this sub-chapter, it will only be presented in a general overview, just for the sake of completeness.

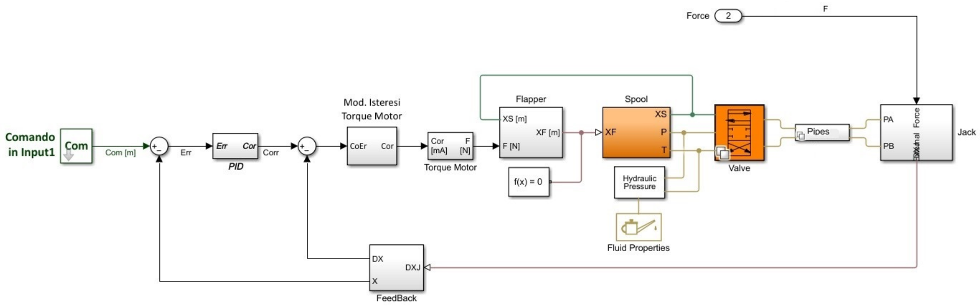


Figure 4.4: Light up of the second stage in the SimScape model

Spool

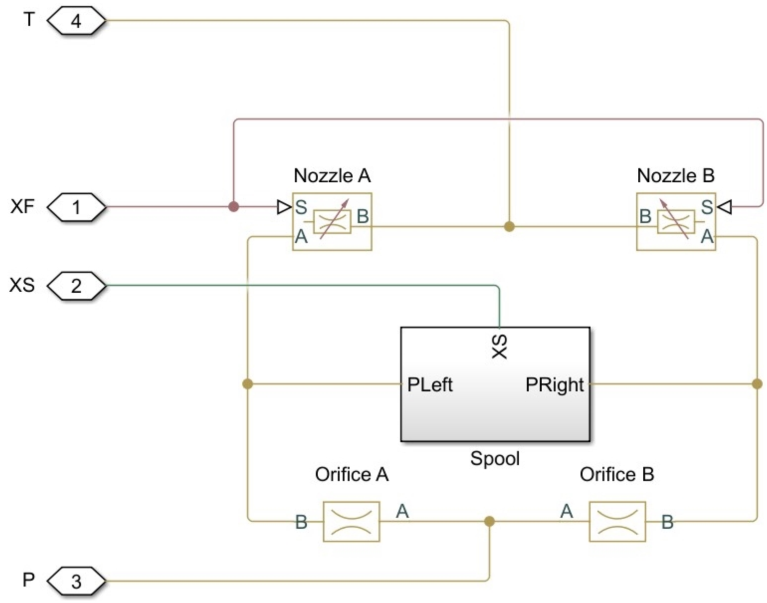


Figure 4.5: SimScape model of the spool

As mentioned above, the output of the flapper model will be the input to the spool dynamics model.

As can be seen from the model, the flapper position (XS) will be provided to two blocks, which represent the nozzles towards which the flapper approaches, and that allow the generation of the fluid dynamic resistance at the ends of the spool.
The position therefore (XS) will go to vary what is the area of two nozzles, *Nozzle A* and *Nozzle B*, and consequently go to vary the pressure in entrance to the block of the dynamics of the spool.
The spool is instead modelled in the *Spool dynamic block*, where its dynamics is modelled. The main aspect of this block is related to the compressibility of the oil, which after several analyses is excluded from the study in the modelling of the servovalve.
The output of the block relating to the dynamics of the spool, is that linked to the displacement of the spool and therefore to the opening of the liner lights.
As already widely described, the valve, with the opening of the ports through the translation of the spool, connects the chambers of the actuator with the different delivery and return pressures.

Valve

This information, the translation of the spool, is then used as input in the block related to the valve where a comparison is made between different valve models available in the simscape library, a comparison which was extensively addressed by the colleague Caruso, mentioned above.

4.5 Actuator

As already described, the study and modelling is done for a servo-valve operating upstream of a fixed-wing Aircraft control surface. The latter must therefore be set in motion both when the aircraft is stationary and when cruising, where it is necessary to counteract the aerodynamic forces acting on it. The surface must therefore be able to move in both directions, allowing both negative and positive deflection, therefore the actuator must be able to generate forces in both directions.

In the Simulink model used as a reference, the actuator has been modelled by considering its dynamics.
Therefore, following the second law of dynamics, it is possible obtain that:

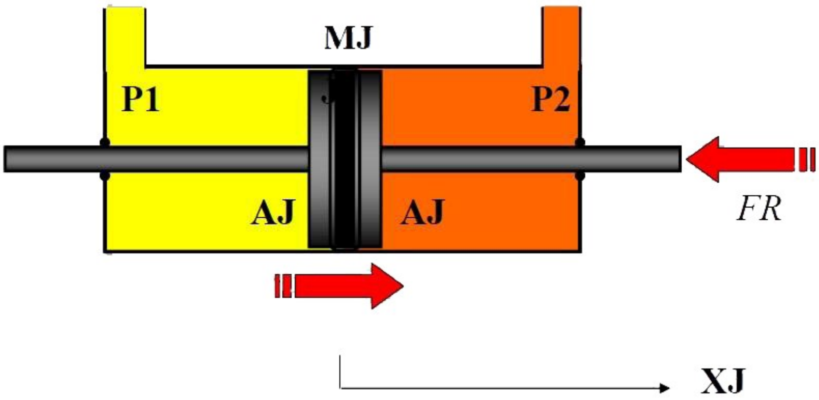


Figure 4.6: Scheme of a double-acting cylinder

$$MJ \cdot \frac{d^2 XJ}{dt^2} + CJ \cdot \frac{dXJ}{dt} = AJ \cdot P12 - FR$$

A different approach must be taken in simscape because of the different modelling approach.
The type of actuator used in this modelling is therefore a double-acting cylinder.
This type of actuator consists of two chambers in which the fluid, or, in this case, the hydraulic oil, acts alternately in both chambers due to a pressure difference.
Connected to the cylinder, which moves between the two chambers, there is a rod which is then connected to the user.
In the modelling carried out in this thesis, both the modelling of the actuator and friction, and therefore the choice and analysis of the blocks provided by the SimScape library, were carried out.

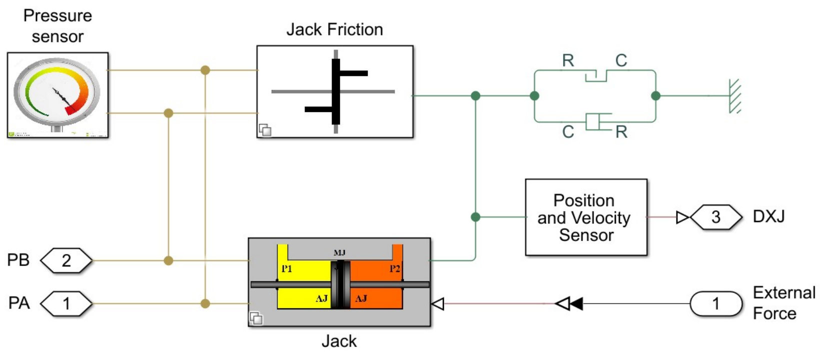
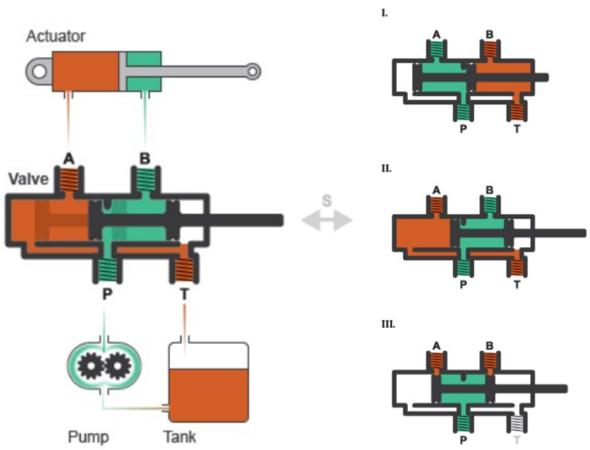


Figure 4.7: Overview of the Simscape model of the actuator

4.5.1 Hydraulic Cylinder

As can be seen from the image above, in which the simscape model of the actuator is shown, there are several blocks that group together the various subsystems used in the modelling.
As can be seen, there are two inputs provided to the actuator model, and these are the pressures (PA and PB) that will go into the two chambers of the actuator.
The magnitude of the two pressures depends on the displacement of the spool and therefore depends on the hydraulic connection that is created.
The chambers of the actuator, which are characterised by having an equal initial pressure (Pi), can be connected to either the Supply pressure (P_{supply}) or the Tank pressure (P_{tank}).



(a) (b)

Figure 4.8: General setup of a valve and different position of the spool

It is assumed that the tank pressure is greater than the supply pressure, and the displacement of the cylinder rod is positive if to the right.

Following the image above, three cases can be defined.

– *Case I*

If the spool displacement connects chamber A of the cylinder with the supply pressure ($PA = P_{supply}$) and chamber B with the tank pressure ($PB = P_{tank}$) then there will be a negative rod displacement.

– *Case II*

If the spool displacement connects chamber A of the cylinder with tank pressure ($PA = P_{tank}$) and chamber B with supply pressure ($PB = P_{supply}$) then there will be a positive rod displacement.

– *Case III*

If the spool displacement creates no connection with the supply or tank pressure, then there will be no pressure difference between the two chambers and, therefore, no displacement.

In the SimScape, it is possible to see that the pressure signal is also supplied to two other blocks.
One block relates to a simple pressure sensor system, used to evaluate the variation of pressures in the chambers and thus the pressure difference that moves the piston.
The other block relates to the friction block in the actuator, which will be analysed later.
It is therefore necessary to model the block relating to the double-acting cylinder, taking into account everything that has already been mentioned above.
In order do that, it is necessary to use those blocks of the simscape library that allow to transform hydraulic energy into mechanical energy.
In particular two different model of the Double-Acting Cylinder will be presented.

Translational Hydro-Mechanical Converter (Two Spool)

It is possible to provide a first example of modelling of the actuator using the "Translational Hydro-Mechanical Converter" block.

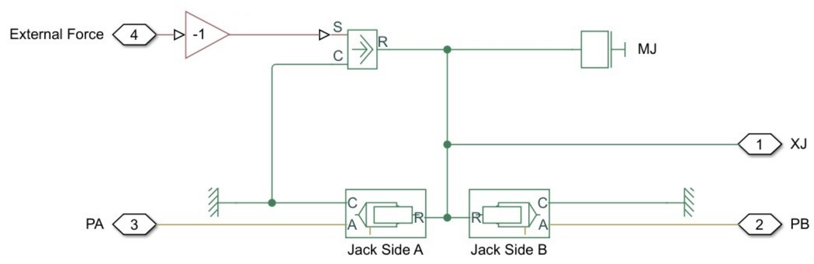


Figure 4.9: SimScape model of the actuator with two Translational Hydro-Mechanical converter

The Translational Hydro-Mechanical element, if used on its own, could be considered as a single-action cylinder.
The pressurized fluid is supplied to a chamber that transforms the pressure change, caused by a constant oil flow, into a mechanical translation of a rod.
However this block, simulating the conversion from hydraulic to mechanical energy in a single chamber, cannot be used alone in this thesis.
As can be seen in the image above, two blocks of this type were in fact used.
The two "mechanical" ports C of the two blocks are both linked to a translational mechanical reference, to indicate the anchorage of the piston to a possible fixing structure on the aircraft.
The other two mechanical ports R of the two blocks instead are linked together, to simulate the fact that they belong to the same element, i.e. the rod of the double-acting cylinder.
The two pressure inputs are supplied separately to the hydraulic ports of the two blocks.
This model, therefore takes into account the displacement of the cylinder in the "Jack side A", against a pressure variation (PA), and binds them to the displacement of the cylinder in the "Jack side B", against a pressure variation (PB).
The physical signal is then linked to a mass MJ, which represents the mass of the jack, and in particular, the mass of the piston head inside the cylinder.

This translation (XJ) is then supplied to the friction block of the jack and to the position and velocity sensor.

The only information required by this block is the area of the piston, needed to calculate the force acting on it, and the positive direction of translation. There is another information that the block asks for, and it is related to the compressibility of the oil.

As mentioned previously, this block also takes into account the variation of a flow rate supplied to the piston chamber, in fact the law used to model this SimScape block, takes into account only this variation to convert the flow into a pressure and then into a force.

$$q = \frac{d\left(\frac{\rho}{\rho_l^0} V\right)}{dt}$$

$$F = A \cdot p$$

The density value entered in the flow calculation can take into consideration both the pressure difference and the compressibility of the oil.
In the block it is possible to choose whether or not to take this into account. If the compressibility is "activated", other aspects such as the *Bulk modulus* of the oil and the amount of air trapped in the fluid must be taken into account. It will also be necessary to consider the dead volume of the actuator and the specific heat ratio.
As will be showed shortly, in this model the compressibility will be set to OFF.

Another element inserted and connected to the signal of translation of the mass, is an *Ideal Force Source*.
This block is used to generate a force that simulates those external forces that can act on the user. In fact the signal before entering the source is multiplied by a gain equal to -1, only to indicate that the force will be opposite to the translation

The subsystem of the external load, will be the same in the following second model of the jack.

Double-Acting Cylinder block (simple)

The simscape library provides a block that simulates a double-acting cylinder. In particular, two blocks are available: a "simple" one and a more complete one. The difference between the two will be shown later, but now the model of the actuator using the "simple" block will be shown.

As can be seen from the graphical representation, this block alone simulates the behaviour of a cylinder, in which the difference in pressure between two chambers generates a force that sets a piston in motion to which a rod is connected. The latter will be connected to the user.
The external structure of the cylinder, whose port is C, is connected to the mechanical reference, while the actuator rod, R, is connected to the MJ mass.
The output signal here is also the cylinder rod translation signal.

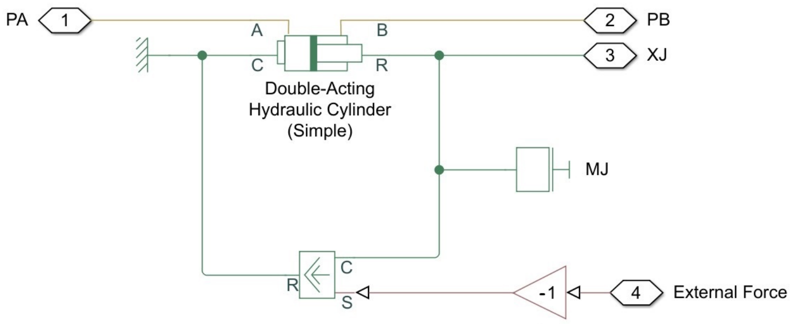


Figure 4.10: SimScape model of the actuator with the Double-Acting cylinder (simple) block

The difference between this block and the one used in the previous model (the *Translational Hydro-Mechanical Converter*) is that this one, in addition to asking for the area of the cylinder and the positive direction of translation, asks for information regarding the structure of the actuator.
A required information is for example the Piston maximum travel between caps (piston stroke) but also coefficients describing the type of penetration between the bodies (in this case the piston head and the two ends of the cylinder).

$$F = A_A \cdot p_A - A_B \cdot p_B - F_c$$

$$q_A = A_A \cdot v$$

$$q_B = A_B \cdot v$$

Where F_c is the binding reaction of the cylinder head to the cylinder end, and v is the cylinder rod speed.

The assumptions and limitations of this block are therefore:

- Friction between moving parts is not considered.
- Inertia effects are not considered.
- Fluid compressibility is not considered.
- Leakage flows are assumed to be negligible.
- The hard stops are assumed to be fully inelastic, as explained above.

Double-Acting Cylinder block

As mentioned above, the library provides two models for the simulation of a double-acting cylinder, a simple model and a more complete one.

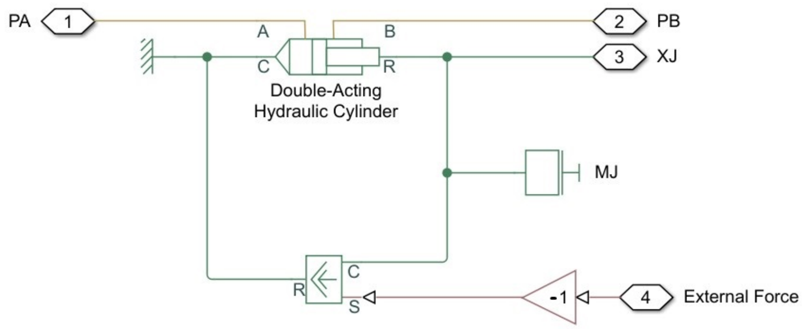


Figure 4.11: SimScape model of the actuator with the Double-Acting cylinder block

In the type of operation, both blocks are similar, that is in the transformation of hydraulic energy into mechanical energy.
The difference between the two blocks lies in the fact that the "simple" model does not take into account the compressibility of the fluid, while the more complex one also asks for information about the initial conditions of the fluid in the two chambers.
Another difference between the two blocks lies in the internal modelling of the hard stop.
The double-acting cylinder models the hard stop using an elastic model, where there is basically a combination of a spring and a damper.
The spring and therefore its stiffness, which is very high, represents the contact stiffness between the two bodies. This high stiffness can unfortunately also lead to oscillations which make the numerical calculation complex and inefficient.
The double-acting cylinder (simple) block, on the other hand, models the hard stop with an inelastic model, i.e. as a simple dynamic damper whose coefficient depends on the amount of penetration between the two blocks. This means that there are no oscillations in the event of a collision.

The output signal of the hydraulic cylinder is therefore linked to the cylinder translation. However, this signal is also linked to other blocks.
As can be seen in the general picture, a hydraulic damper and a hard stop have been incorporated into the model.
The damper is used to simulate the fact that the MJ mass, i.e. the piston head, is moving through the hydraulic oil with non-linear velocities, so it

will be subject to a damping force.
As far as the hard stop is concerned, since the last model used for the cylinder model is the double-acting cylinder (simple), which is characterized by a very mild hard stop, this block was inserted in parallel to the damper, in case the hard stop properties were to be modified outside the cylinder.

In modelling the servo-valve in this thesis, the use of the *Two Spool* block was preferred following the results obtained.

4.5.2 Friction Blocks

Another fundamental aspect in the modelling of the actuator is its friction. For a piston moving in a cylinder, and particularly for a double-acting cylinder, friction can have a major effect on the dynamics.
In this type of cylinder, the parts that are characterised by contact and then relative displacement are both the piston and the rod, which slide against the internal walls of the cylinder.
In addition, these two elements, which separate both the two chambers from each other and the chambers from the outside, require elements (*O-ring*) to prevent any fluid leakage, which usually are made with a mild steel or stainless steel. Those needs to isolate the inside of cylinder but also must allow the head and the rod to move.

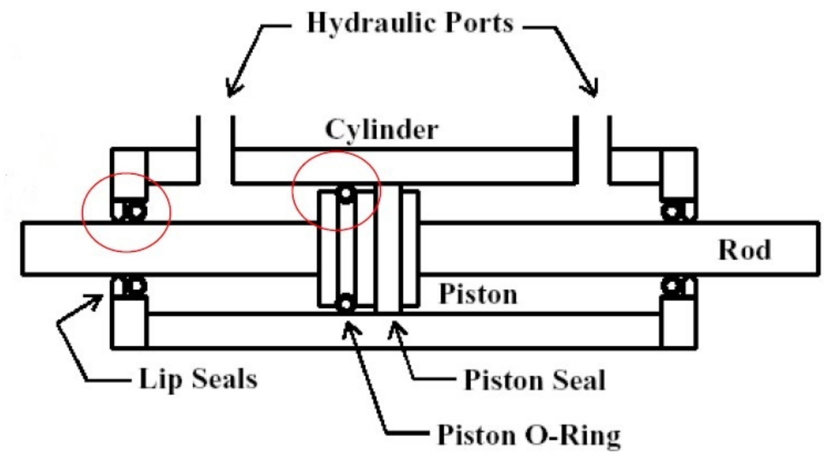


Figure 4.12: Details of where the friction could accour in a Double-Acting Cylinder

Furthermore, depending on the control provided, the actuator will have very low actuation speeds. These aspects further influence friction.
In the Simulink model, the friction model used belongs to the Borello's model, which has been implemented using the equations governing its dy-

namics.
This modelling must be carried out differently in SimScape.
The SimScape library provides several elements for modelling friction.
In the following modelling, two blocks will be compared and the results will be showed.

Translational Friction Block

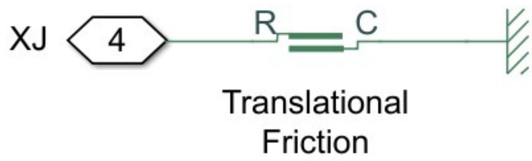


Figure 4.13: Block of the Translational Friction

This element simulates the sliding between two bodies moving towards each other, so the friction generated is assumed as a sum of Striebeck, Coulomb and viscous components.

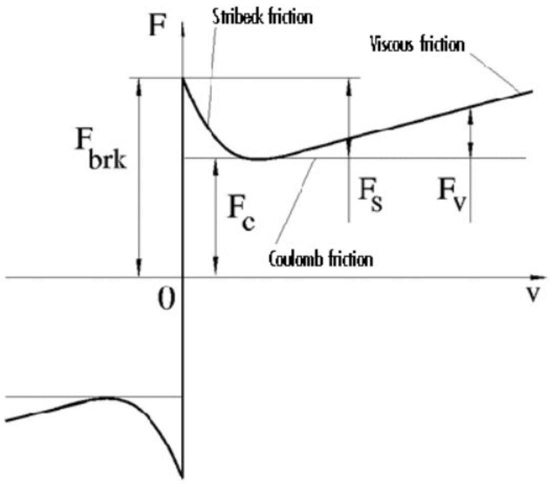


Figure 4.14: Friction Model in SimScape friction blocks

This block therefore takes into account the effects and how friction varies according to the relative speed between the bodies.

As can be seen in the graph above, the curve is the sum of three components:

- *Striebeck friction*

In this area, at very low speeds, usually of the order of $10^-6[m/s]$, the curve has a negative slope. It begins from to highest value of friction (**Breakaway friction**) till certain value of speed where the curve changes the course.

- *Coulomb friction*

After a specific value of velocity (**Breakway friction velocity**), the curve becomes a curve with zero derivative, and the friction force becomes constant with velocity (**Coulomb friction force**).

- *Viscous friction*

The curve now takes an increasing course with velocity, where the friction force is proportional to the velocity by a coefficient (**Viscous friction coefficient**).

In this model it is therefore necessary to indicate these parameters that best describe these three regions.

It should be remembered that the reference Simulink model uses the Borello model to describe the friction.
Considering this, the Breakway friction and Coulomb friction values are easily set, and equal to $400[N]$ and $200[N]$ respectively.
Regarding the speed regime within which friction can be approximated to only static friction (Breakway friction velocity) a value of $2e-5[m/s]$ has been imposed.
This implies that if the relative velocity is less than this value the friction will be at Breakaway friction value, if it is higher it will change value and will be equal to the Coulomb friction force.
This range has been obtained iteratively, until a convergence of values between the SimScape model and the Simulink model has been achieved.

Cylinder Friction block

Another element that the Simscape library provides for friction modelling is the Cylinder friction block.

In the cylinder block described before, one of the assumptions was that the friction between the piston and the inner surface of the cylinder was not taken into account.
This block allows more complete modelling of friction in the cylinder.

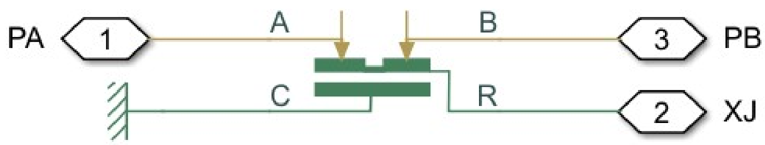


Figure 4.15: Block of the Cylinder Friction

$$F_C = F_{pr} + f_{cfr}(p_A + p_B)$$

In this block, friction is modelled not only as a function of speed, but also in relation to the pressures in the cylinder chambers.

Where the F_{pr} is th Preload Force, caused by the seal squeeze during assembly and the f_{cfr} is the coulombic friction coefficient required to bind the friction proportional to the pressure in the chambers.

This proportionality between pressure and friction, however, risks leading to a limit cycle in the numerical calculation, in the case where the relative speed between the two blocks is very small, almost close to zero.
This consideration leads the block to ask for a parameter (*linear region velocity threshold*) which defines a region in which friction is considered to be only linearly proportional to velocity.
SimScape suggests a value between 10^6 and 10^4.

$$F = K \cdot v$$

Another parameter that this block requires, unlike the previous one, is the definition of a coefficient (*Transition approximation coefficient*).
Through this coefficient and through the relation:

$$c_v \simeq \frac{4}{v_{min}}$$

It is possible to define the minimum relative velocity for which the friction is equal to the minimum value in the Striebeck friction region.
There are other parameters that depends on how the cylinder is made, for which they were assumed and evaluated iteratively due to lack of data.

In the following modelling, the two blocks will be compared and the results will be showed.
The cylinder friction block is the one that better follows the reference model.

4.6 Pipes

SimScape, as has been said many times, is an environment in which it is possible to model systems following a physical approach, where the various connections are made not on the basis of the numerical relationship, but on the basis of the physical one.
As can be seen in the model described above, the flow of information follows a physical logic.
However, one aspect that could be followed in the modelling of the servovalve, in particular a valve characterised by a hydraulic system, is that of the connection itself, i.e. the modelling of the hydraulic pipes.
A brief analysis was conducted on the types of pipes and hydraulic connections that SimScape provides.

In particular, this analysis was carried out on the connection downstream of the valve and before the actuator, since in the actual construction of the actuator system there cannot be direct connection between the two systems. This feature, could lead to the addition of other aspects, such as pressure losses, which could influence the behaviour of the complete system.

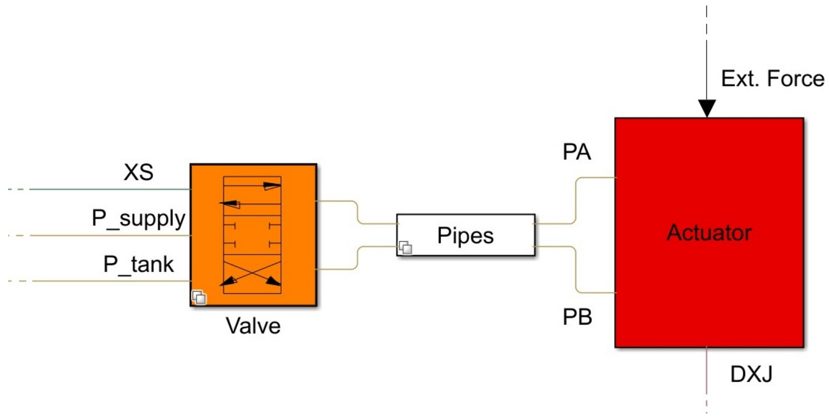


Figure 4.16: Connection between Valve and Actuator with the Pipe block

In the model as well as in this thesis, three different conditions of hydraulic connection will be shown.

- *The case where there is no connection at all, and therefore no modelling of the pipes*
- *The case in which the pipes are modelled and, in particular, are considered as rigid pipes.*
- *The case where the pipes are modelled as flexible pipes.*

Simscape provides a hydraulic block that simulates pipelines with circular and non-circular cross-sections.
In this analysis, however, pipes with a circular cross-section are considered. Parameters that are also required are related to geometry and physical characteristics, such as diameter, length, or internal roughness.
Another characteristic that this model requires is the type of pipe, i.e. whether to set flexible wall or rigid.
This choice is only possible if a pipe with a circular cross-section is imposed. The main difference between the two types of walls is that in the case of flexible pipes, which are taken into account when the pipe plays a fundamental role in the operation of the entire system and the wall compliance needs to be considered, thus making the simulation even more complex.
Considering rigid pipes, on the other hand, makes the simulation numerically simpler because this aspect is not taken into consideration.

The limits and assumptions that are applied in this block are:

- *Flow is assumed to be fully developed along with the length of the pipe.*
- *Fluid inertia is not considered.*
- *The block takes into consideration friction losses along with the length of the pipe and the compressibility of the fluid.*

The comparison between the three different cases is shown below.
From the results obtained, it was decided not to consider pipes in the modelling.

4.7 Hydraulic System

With regards to the hydraulic system, and thus the definition of supply and tank pressures, a simple model was used.

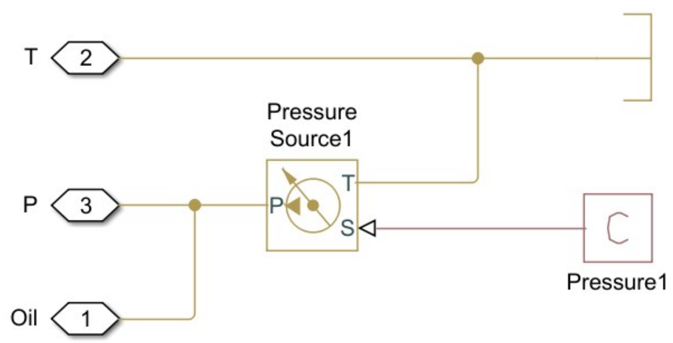


Figure 4.17: Model of the Hydraulic System

For both tank and supply pressures, it is necessary to use a hydraulic pressure source.
By providing a dimensionless numerical value of the pressure via a constant block, the desired pressure can then be generated.
However, it is also possible to simulate the tank pressure using only the hydraulic reference making the system simpler and makes no difference.
This system is linked to the fluid properties block which allows to choose between different types of hydraulic fluids/oils.
The oil used in this model is *Skydrol LD-4*, which has the following characteristics.

- **Density** $= 954.06[kg/m^3]$
- **Viscosity Cynematic** $= 5.96942cSt$
- **Bulk Modulus** $= 1.17304 \cdot 10^9[Pa]$ (where the value of trapped air in the fluid is 0.005)
- **Fluid Temperature** $= 70[°C]$

Chapter 5

Model Analysis

5.1 Cylinder and friction blocks analysis

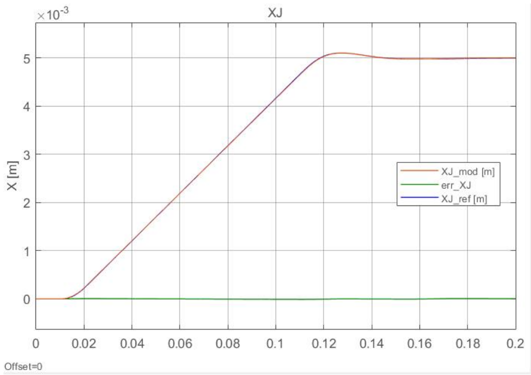
In this chapter, the analyses and comparisons carried out in order to choose those blocks that make it possible to obtain a SimScape model that best follows the Simulink reference model will be shown. This chapter, in particular, will compare the blocks relating to the modelling of the actuator cylinder and the friction. The actuator system is located downstream of the servovalve, it is therefore necessary that in the servovalve, the blocks and elements used in its modelling are already those that best follow the reference model. As previously mentioned, this study was carried out by the colleague Giuseppe Caruso, which led to the conclusion that for the valve system, the *4-Way-Valve* block was used as the *Cylinder-friction* for the friction in the spool. This analysis will also evaluate the error and thus how far the SimScape model differs from the reference model.
Six different combinations will be analysed, depending on the three blocks relating to modelling the actuating cylinder (Two-spool, Double-acting cylinder (simple) and Double acting cylinder) and the two blocks relating to modelling friction in the actuating cylinder (translational friction and cylinder friction).To each of combination, a step command and a ramp command will be provided. The intensity of such command is intended to stress the model to show how it works when far from the nominal condition.

Step Inputs	**Ramp Inputs**
m	m/s
0.005	0.5
0.001	0.1
0.0005	0.05

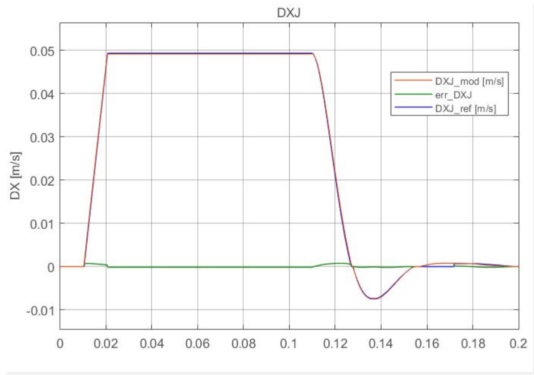
Considering that the actuator system is the main system analyzed in this chapter, a graph of the position and the velocity of the jack will be showed. The actuator affects the entire valve system as well. For this very reason, a graph showing pressure differences, provided by the valve up to the cylinder, will be showed.

5.1.1 Two spool - Translational friction

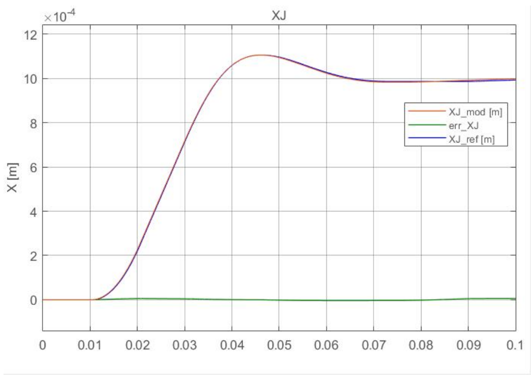
Step Commands



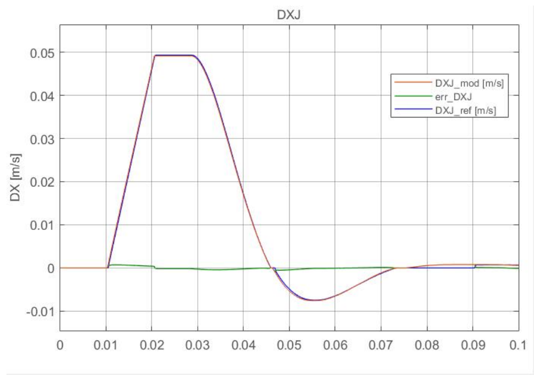
(a) Jack position with a step command=0.005 m



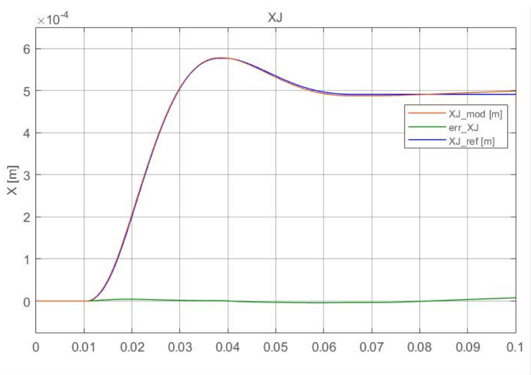
(b) Jack velocity with a step command=0.005 m



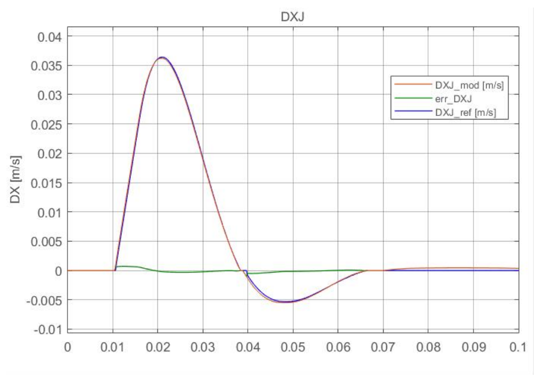
(c) Jack position with a step command=0.001 m



(d) Jack velocity with a step command=0.001 m

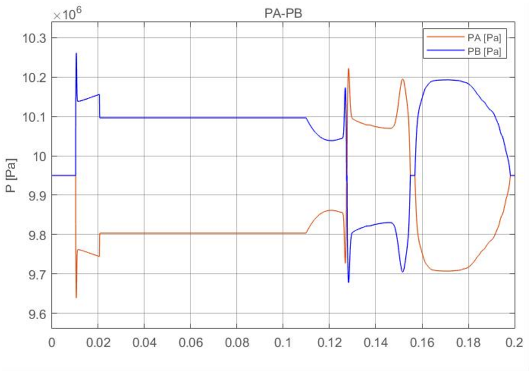


(e) Jack position with a step command=0.0005 m

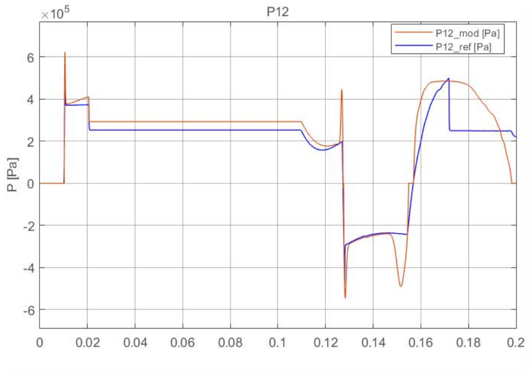


(f) Jack velocity with a step command=0.0005 m

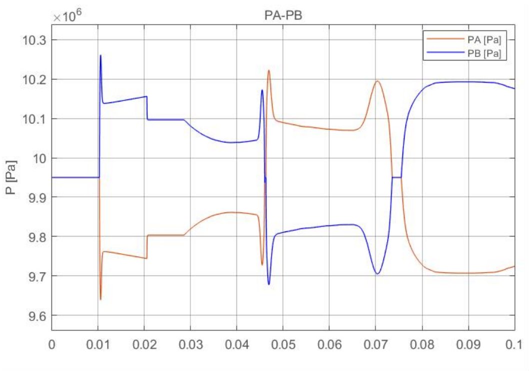
Figure 5.1: Position and Velocity of the actuator with three different step commands and with the **Two spool - Translational friction** combination



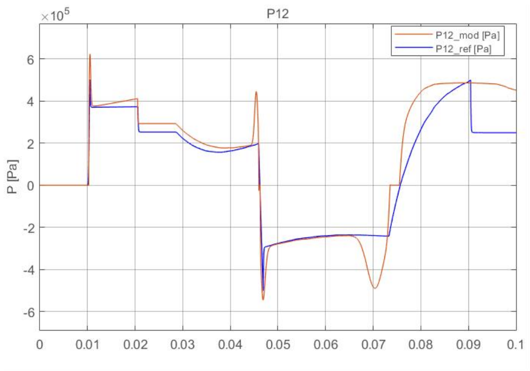
(a) Pressure in the chambers of the cylinder with a step command=0.005m



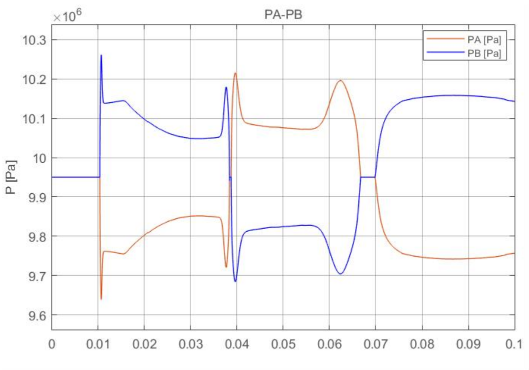
(b) Pressure Difference in the actuator with a step command=0.005m



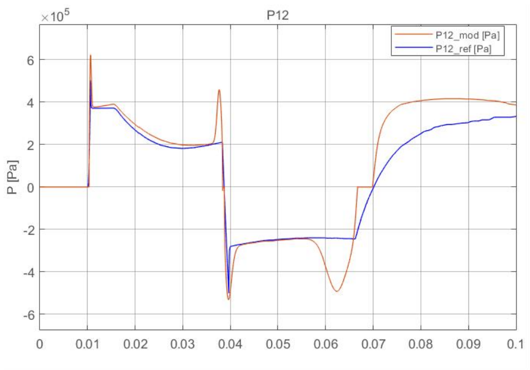
(c) Pressure in the chambers of the cylinder with a step command=0.001m



(d) [Pressure Difference in the actuator with a step command=0.001m



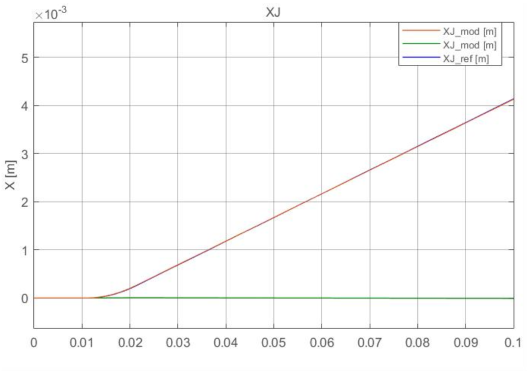
(e) Pressure in the chambers of the cylinder with a step command=0.0005m



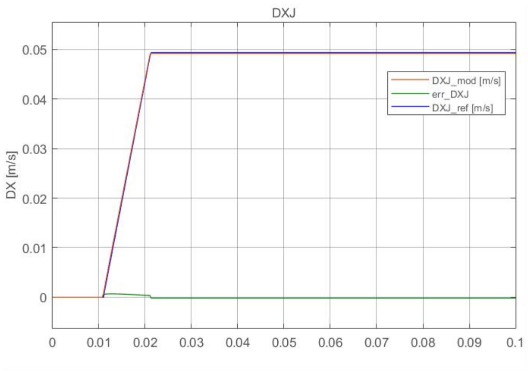
(f) [Pressure Difference in the actuator with a step command=0.0005m

Figure 5.2: Pressure and difference pressure in the chambers of the actuator with three different step commands and with the **Two spool - Translational friction** combination

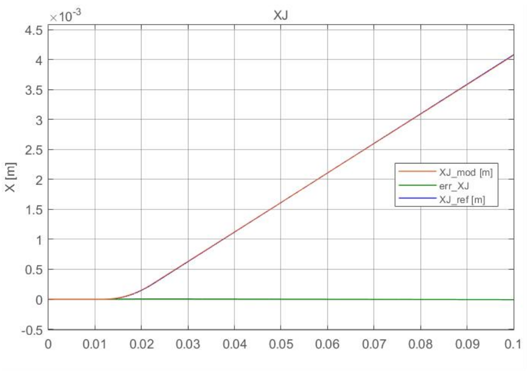
Ramp Commands



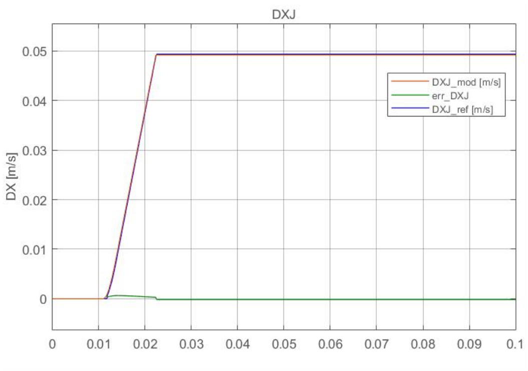
(a) Jack position with a ramp command=0.5 m/s



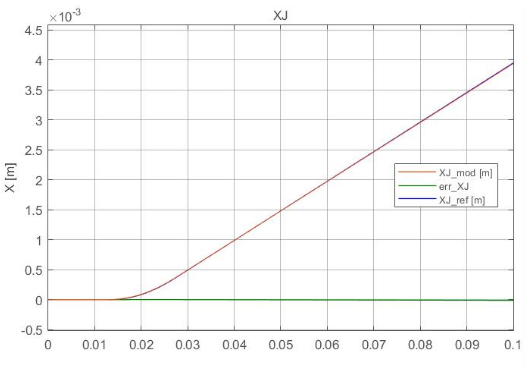
(b) Jack velocity with a ramp command=0.5 m/s



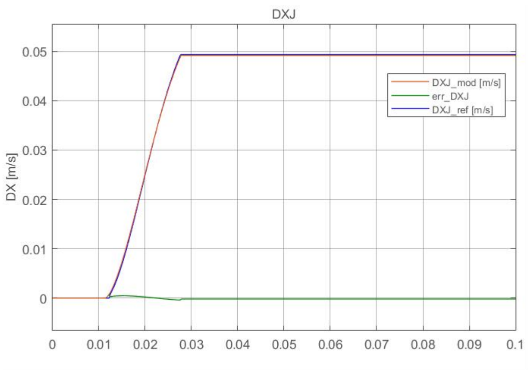
(c) Jack position with a ramp command=0.1 m/s



(d) Jack velocity with a ramp command=0.1 m/s

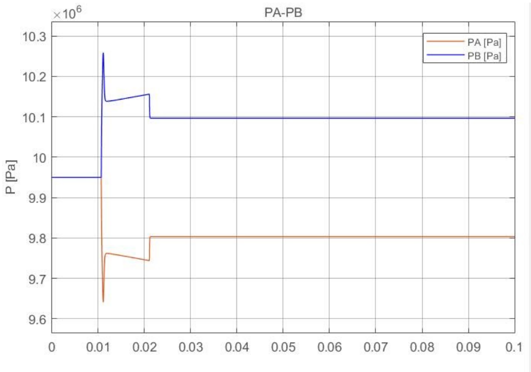


(e) Jack position with a ramp command=0.05 m/s

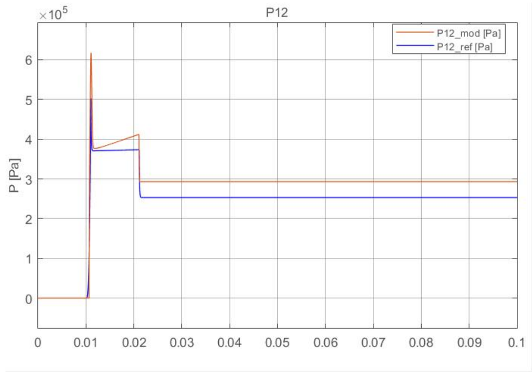


(f) Jack velocity with a ramp command=0.05 m/s

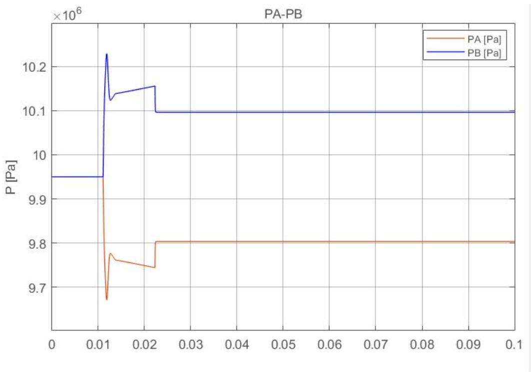
Figure 5.3: Position and Velocity of the actuator with three different ramp commands and with the **Two spool - Translational friction** combination



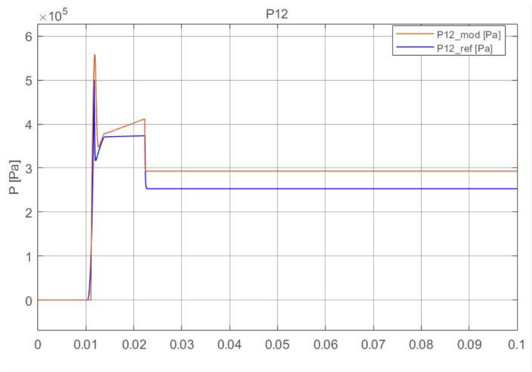
(a) Pressure in the chambers of the cylinder with a ramp command=0.5 m/s



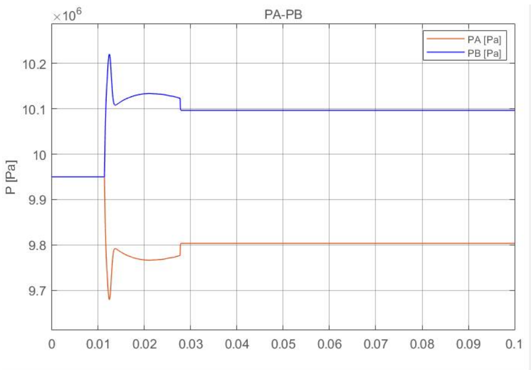
(b) Pressure Difference in the actuator with a ramp command=0.5 m/s



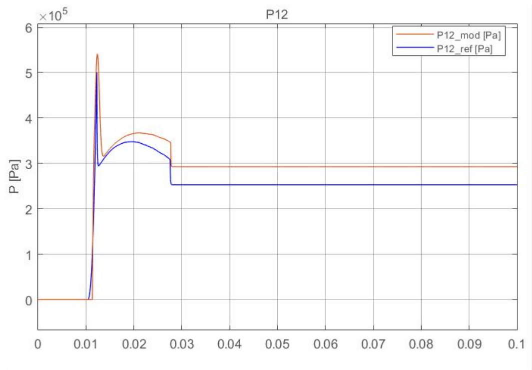
(c) Pressure in the chambers of the cylinder with a ramp command=0.1 m/s



(d) [Pressure Difference in the actuator with a ramp command=0.1 m/s



(e) Pressure in the chambers of the cylinder with a ramp command=0.05 m/s

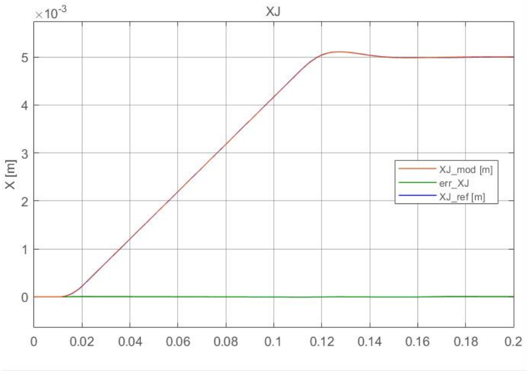


(f) [Pressure Difference in the actuator with a ramp command=0.05 m/s

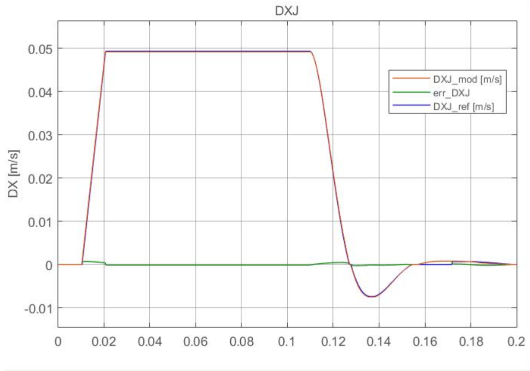
Figure 5.4: Pressure and difference pressure in the chambers of the actuator with three different ramp commands and with the **Two spool - Translational friction** combination

5.1.2 Two spool - Cylinder friction

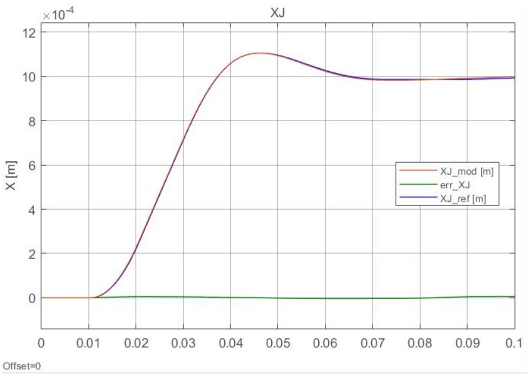
Step Commands



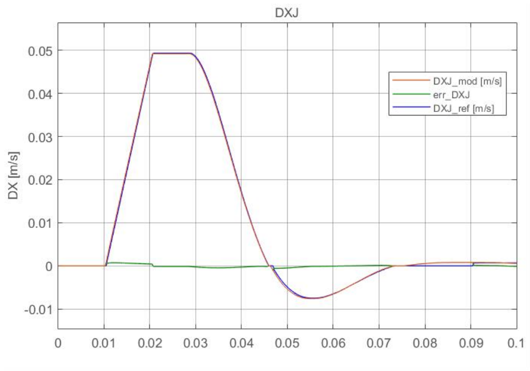
(a) Jack position with a step command=0.005 m



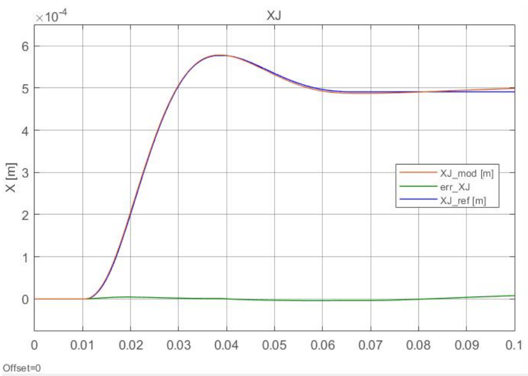
(b) Jack velocity with a step command=0.005 m



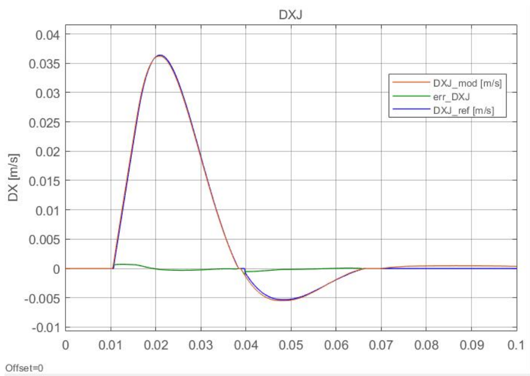
(c) Jack position with a step command=0.001 m



(d) Jack velocity with a step command=0.001 m

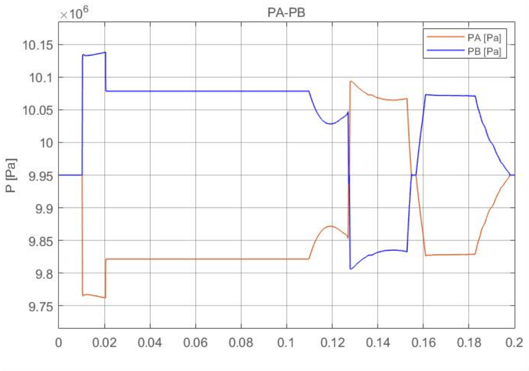


(e) Jack position with a step command=0.0005 m

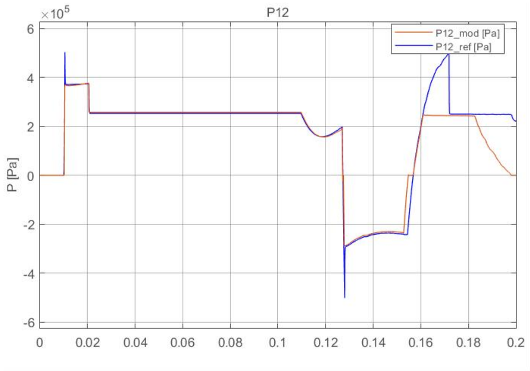


(f) Jack velocity with a step command=0.0005 m

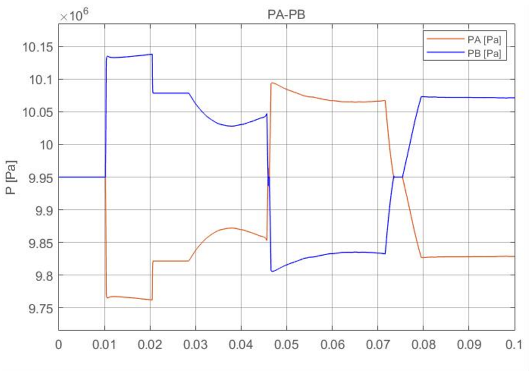
Figure 5.5: Position and Velocity of the actuator with three different step commands and with the **Two spool - Cylinder friction** combination



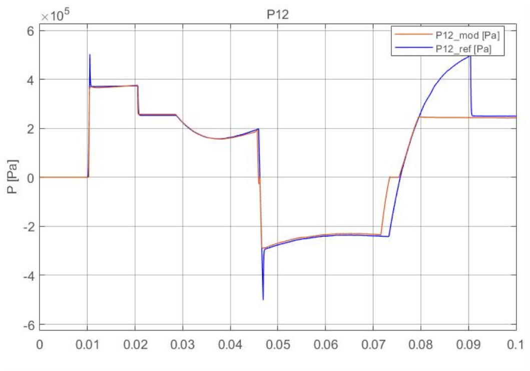
(a) Pressure in the chambers of the cylinder with a step command=0.005m



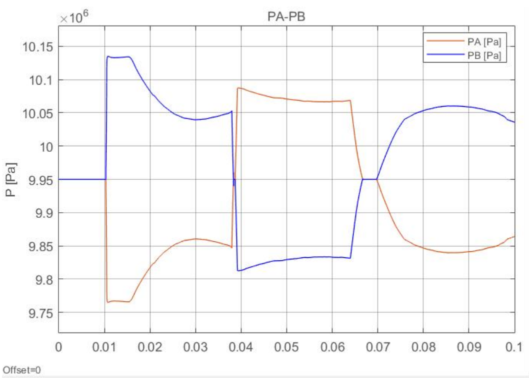
(b) Pressure Difference in the actuator with a step command=0.005m



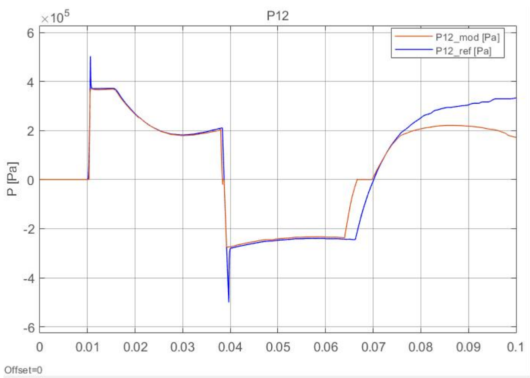
(c) Pressure in the chambers of the cylinder with a step command=0.001m



(d) [Pressure Difference in the actuator with a step command=0.001m



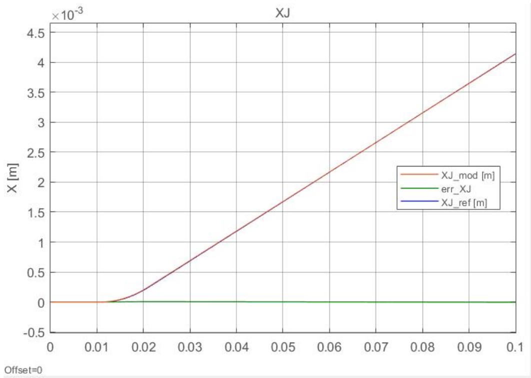
(e) Pressure in the chambers of the cylinder with a step command=0.0005m



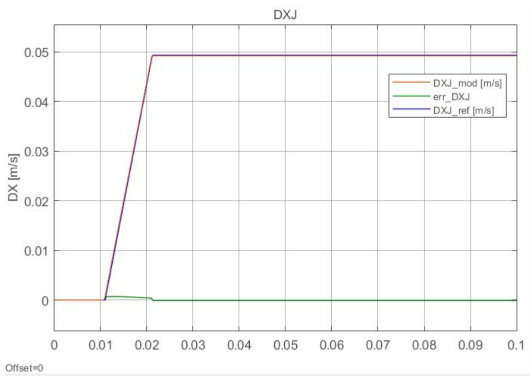
(f) [Pressure Difference in the actuator with a step command=0.0005m

Figure 5.6: Pressure and difference pressure in the chambers of the actuator with three different step commands and with the **Two spool - Cylinder friction** combination

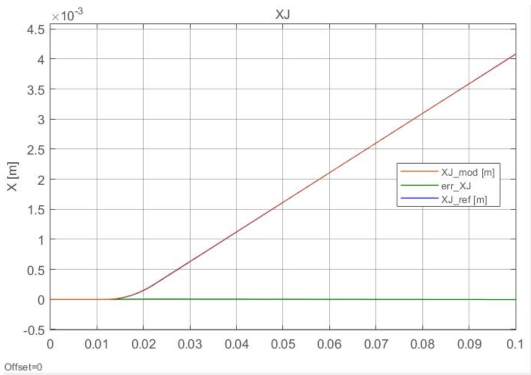
Ramp Commands



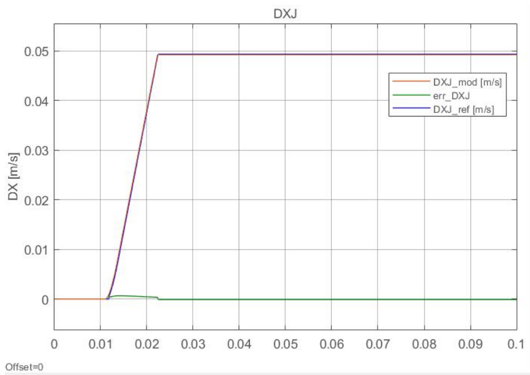
(a) Jack position with a ramp command=0.5 m/s



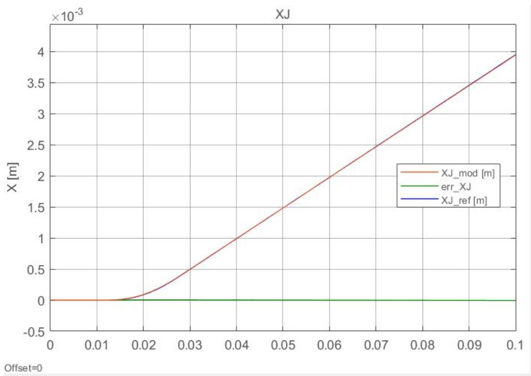
(b) Jack velocity with a ramp command=0.5 m/s



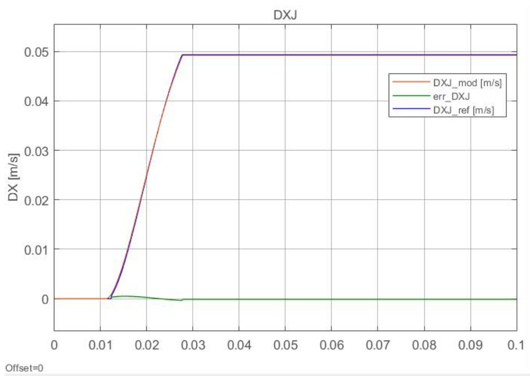
(c) Jack position with a ramp command=0.1 m/s



(d) Jack velocity with a ramp command=0.1 m/s

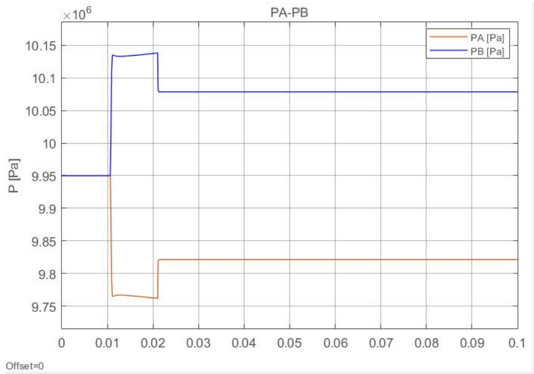


(e) Jack position with a ramp command=0.05 m/s

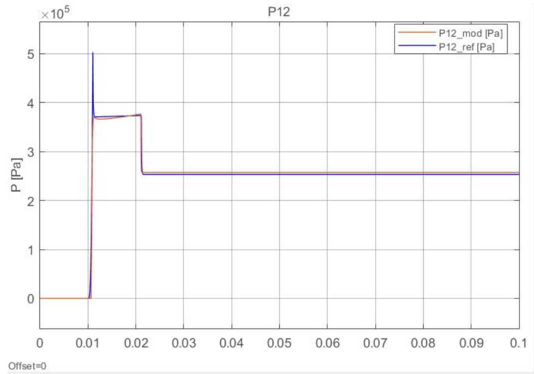


(f) Jack velocity with a ramp command=0.05 m/s

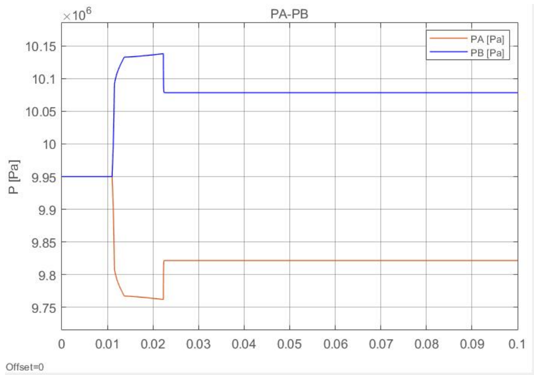
Figure 5.7: Position and Velocity of the actuator with three different ramp commands and with the **Two spool - Cylinder friction** combination



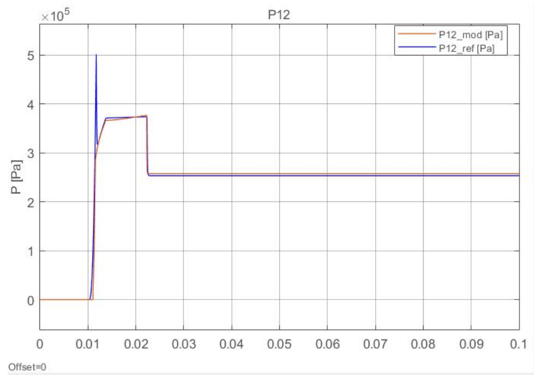
(a) Pressure in the chambers of the cylinder with a ramp command=0.5 m/s



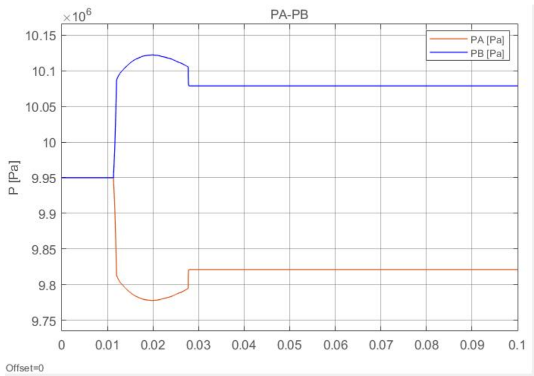
(b) Pressure Difference in the actuator with a ramp command=0.5 m/s



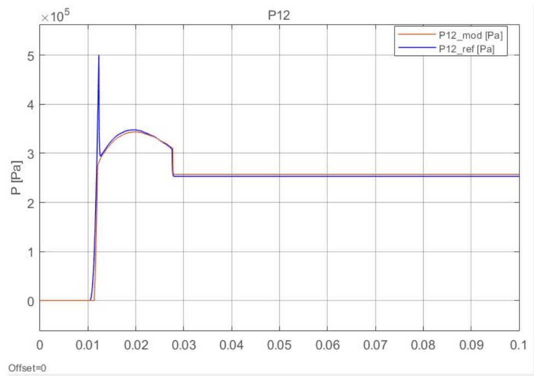
(c) Pressure in the chambers of the cylinder with a ramp command=0.1 m/s



(d) [Pressure Difference in the actuator with a ramp command=0.1 m/s



(e) Pressure in the chambers of the cylinder with a ramp command=0.05 m/s

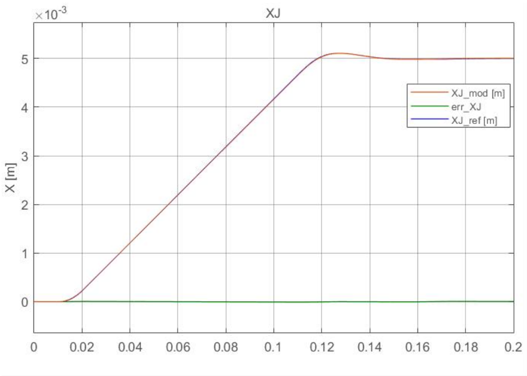


(f) [Pressure Difference in the actuator with a ramp command=0.05 m/s

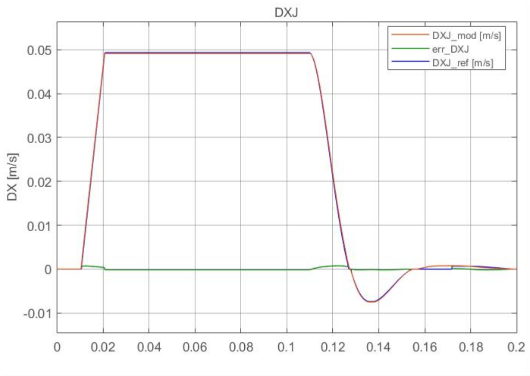
Figure 5.8: Pressure and difference pressure in the chambers of the actuator and with three different ramp commands with the **Two spool - Cylinder friction** combination

5.1.3 Double-acting Cylinder (simple) - Translational friction

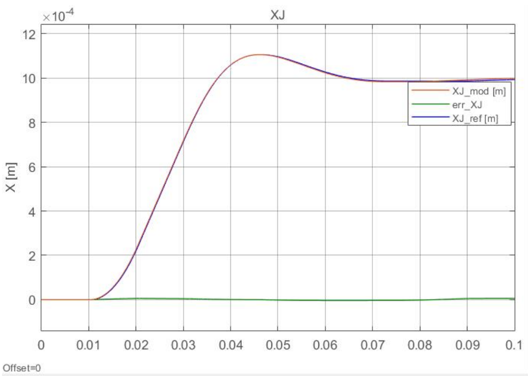
Step Commands



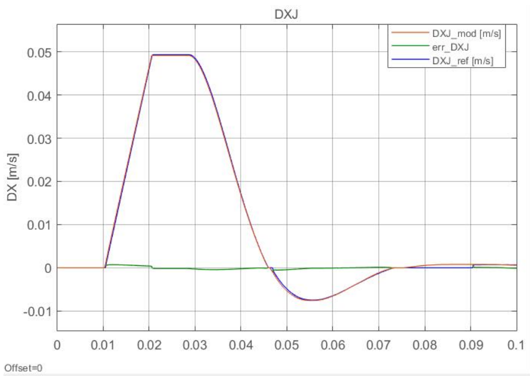
(a) Jack position with a step command=0.005 m



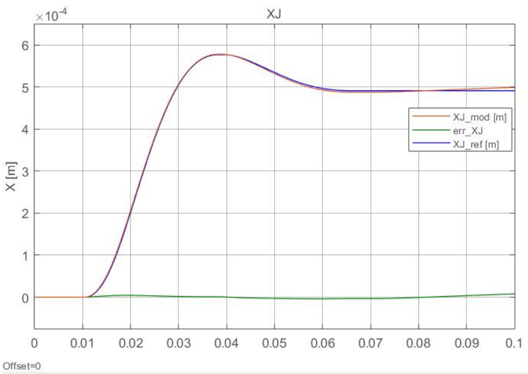
(b) Jack velocity with a step command=0.005 m



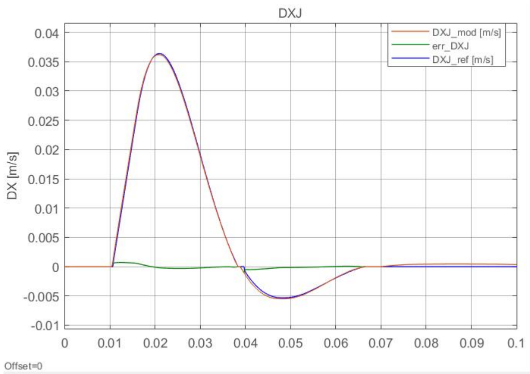
(c) Jack position with a step command=0.001 m



(d) Jack velocity with a step command=0.001 m

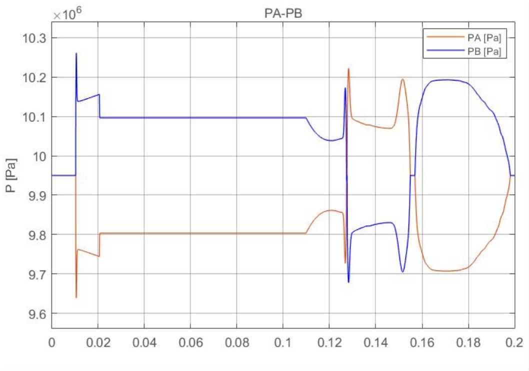


(e) Jack position with a step command=0.0005 m

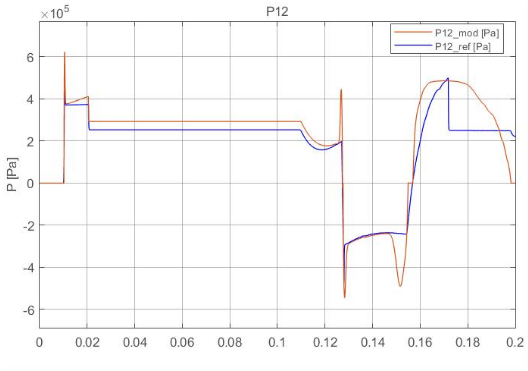


(f) Jack velocity with a step command=0.0005 m

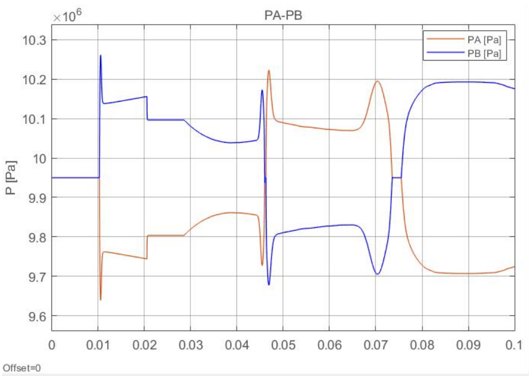
Figure 5.9: Position and Velocity of the actuator with three different step command and with the **Double-acting Cylinder (simple) - Translational friction** combination



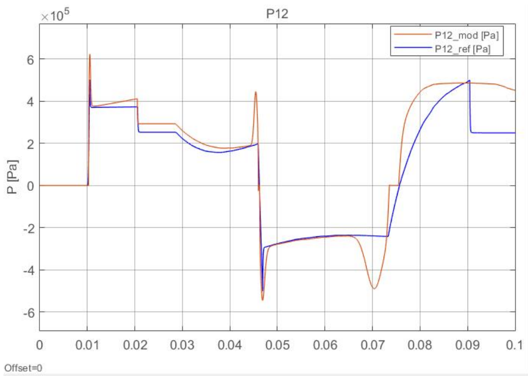
(a) Pressure in the chambers of the cylinder with a step command=0.005m



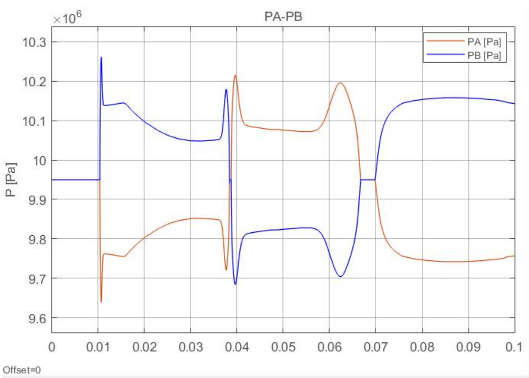
(b) Pressure Difference in the actuator with a step command=0.005m



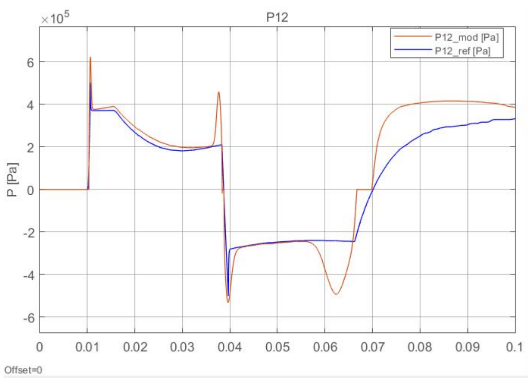
(c) Pressure in the chambers of the cylinder with a step command=0.001m



(d) [Pressure Difference in the actuator with a step command=0.001m



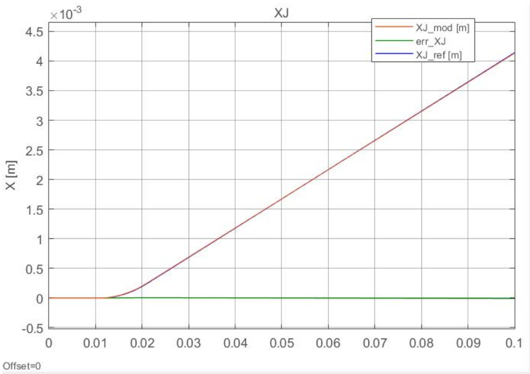
(e) Pressure in the chambers of the cylinder with a step command=0.0005m



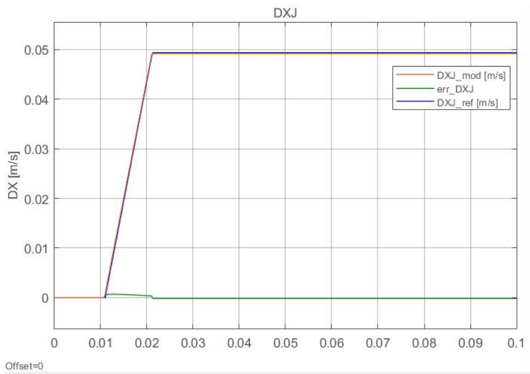
(f) [Pressure Difference in the actuator with a step command=0.0005m

Figure 5.10: Pressure and difference pressure in the chambers of the actuator with three different step commans and with the **Double-acting Cylinder (simple) - Translational friction** combination

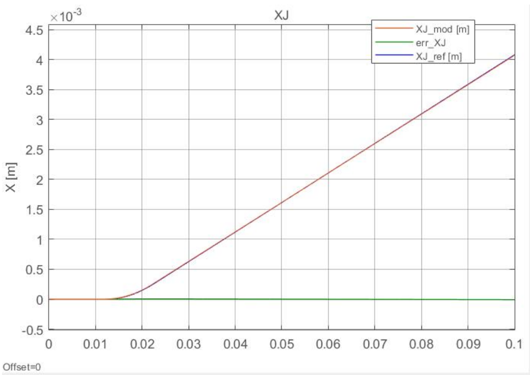
Ramp Commands



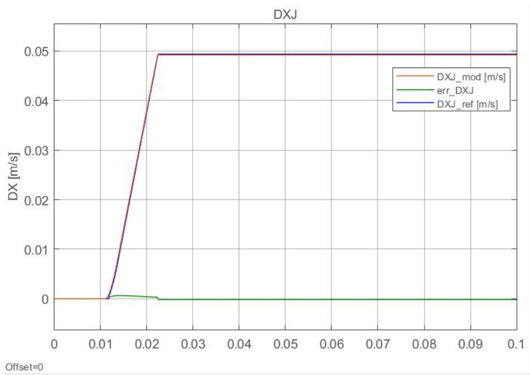
(a) Jack position with a ramp command=0.5 m/s



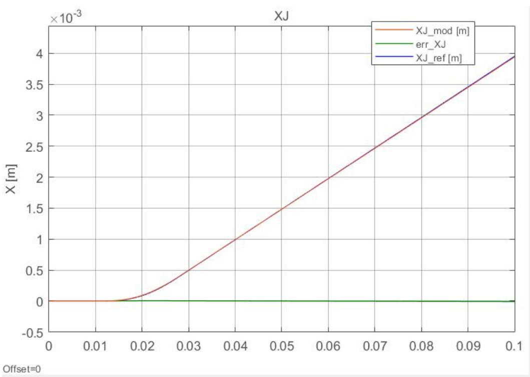
(b) Jack velocity with a ramp command=0.5 m/s



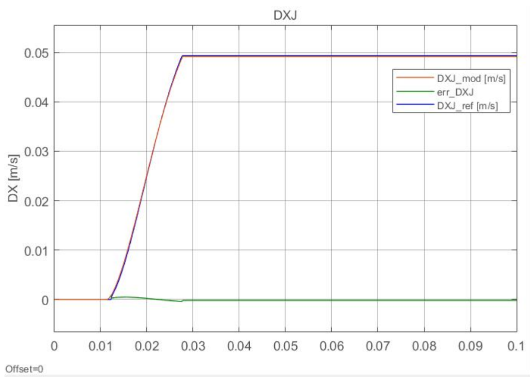
(c) Jack position with a ramp command=0.1 m/s



(d) Jack velocity with a ramp command=0.1 m/s

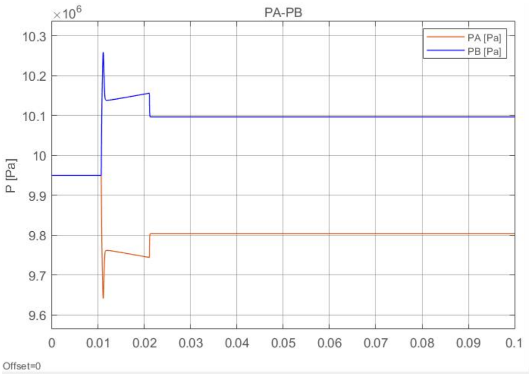


(e) Jack position with a ramp command=0.05 m/s

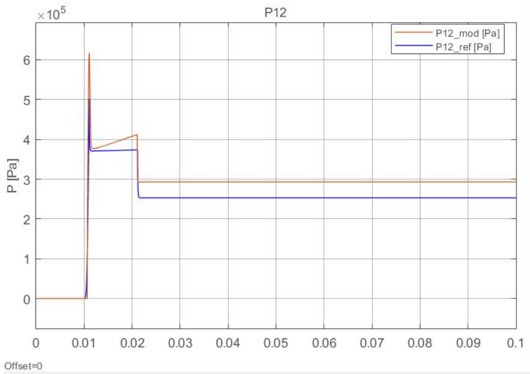


(f) Jack velocity with a ramp command=0.05 m/s

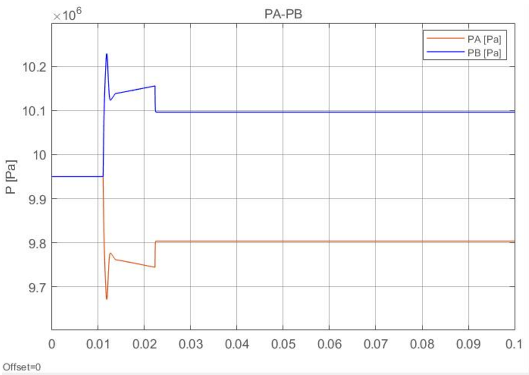
Figure 5.11: Position and Velocity of the actuator with three different ramp commands and with the **Double-acting Cylinder (simple) - Translational friction** combination



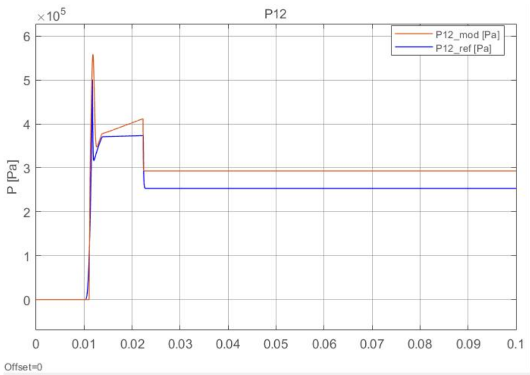
(a) Pressure in the chambers of the cylinder with a ramp command=0.5 m/s



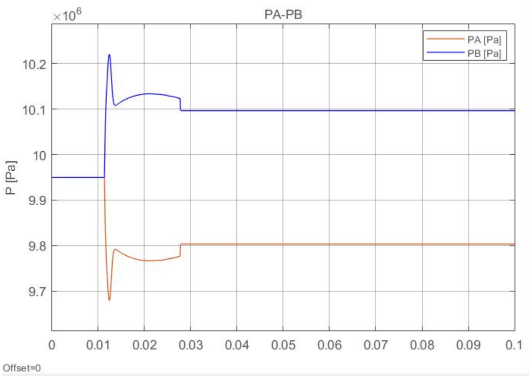
(b) Pressure Difference in the actuator with a ramp command=0.5 m/s



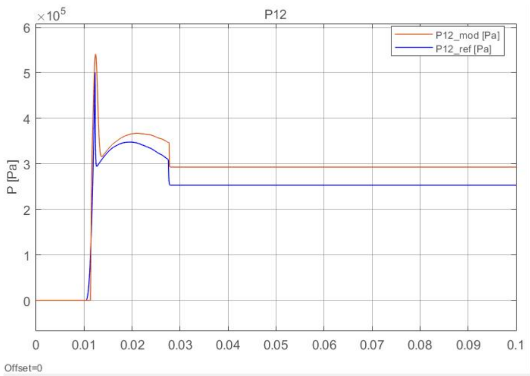
(c) Pressure in the chambers of the cylinder with a ramp command=0.1 m/s



(d) [Pressure Difference in the actuator with a ramp command=0.1 m/s



(e) Pressure in the chambers of the cylinder with a ramp command=0.05 m/s

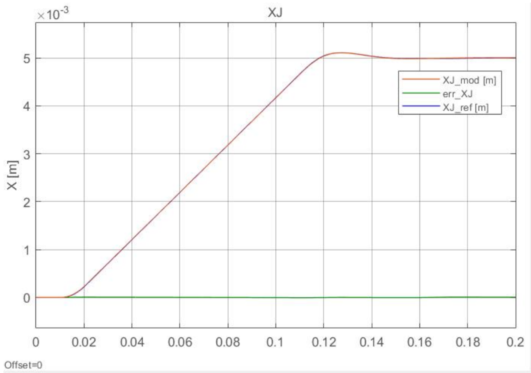


(f) [Pressure Difference in the actuator with a ramp command=0.05 m/s

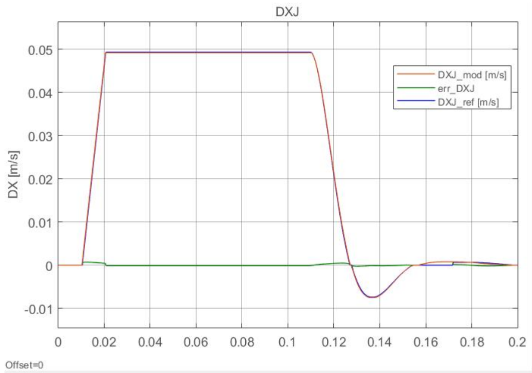
Figure 5.12: Pressure and difference pressure in the chambers of the actuator with three different ramp commands and with the **Double-acting Cylinder (simple) - Translational friction** combination

5.1.4 Double-acting Cylinder (simple) - Cylinder friction

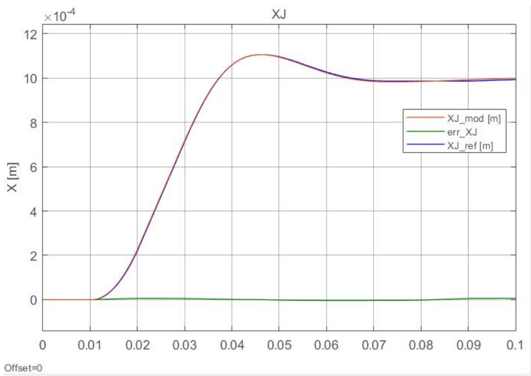
Step Commands



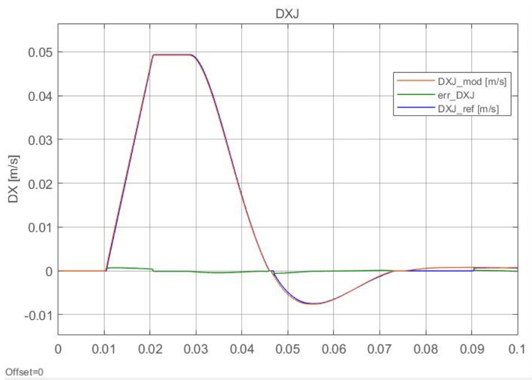
(a) Jack position with a step command=0.005 m



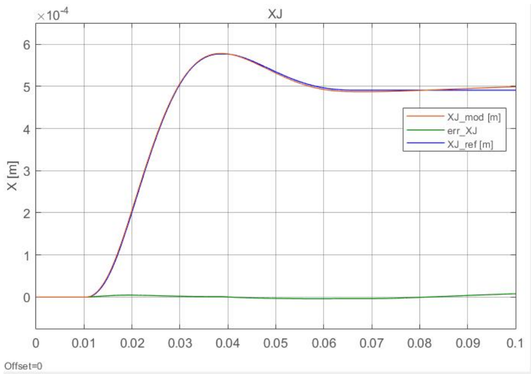
(b) Jack velocity with a step command=0.005 m



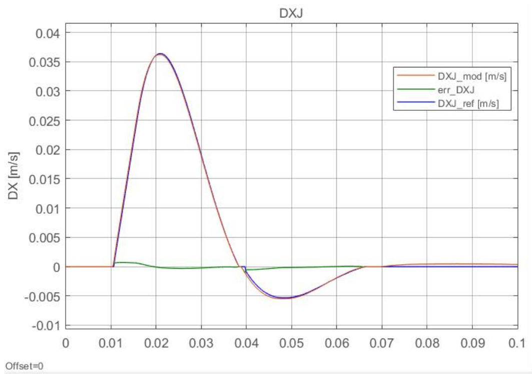
(c) Jack position with a step command=0.001 m



(d) Jack velocity with a step command=0.001 m

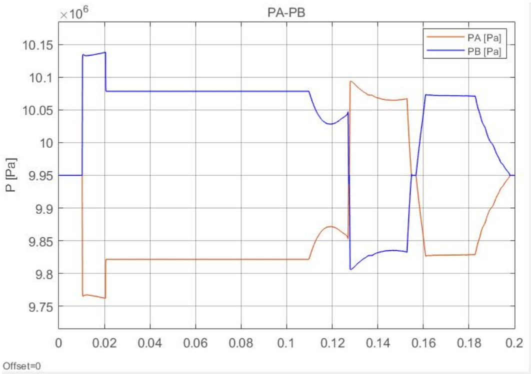


(e) Jack position with a step command=0.0005 m

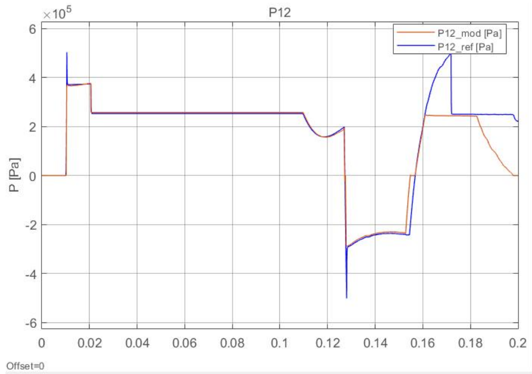


(f) Jack velocity with a step command=0.0005 m

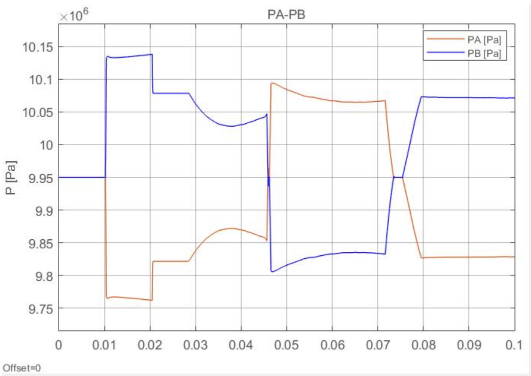
Figure 5.13: Position and Velocity of the actuator with three different step command and with the **Double-acting Cylinder (simple) - Cylinder friction** combination



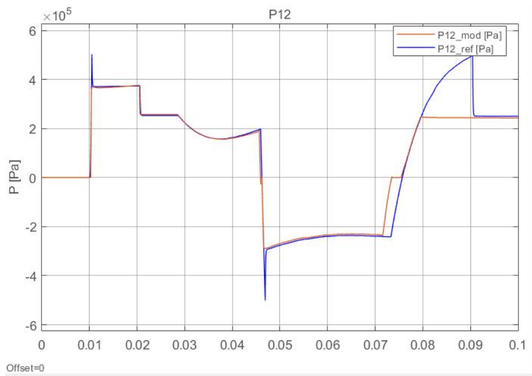
(a) Pressure in the chambers of the cylinder with a step command=0.005m



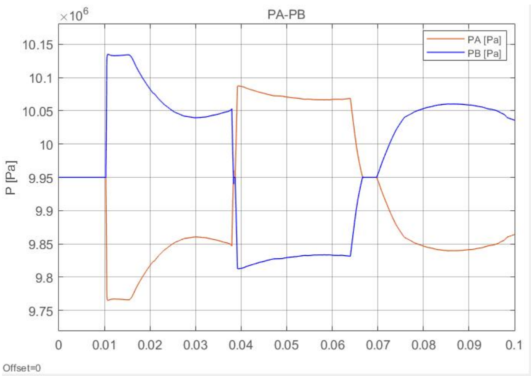
(b) Pressure Difference in the actuator with a step command=0.005m



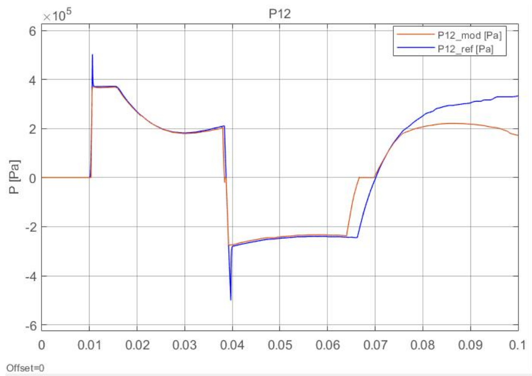
(c) Pressure in the chambers of the cylinder with a step command=0.001m



(d) [Pressure Difference in the actuator with a step command=0.001m



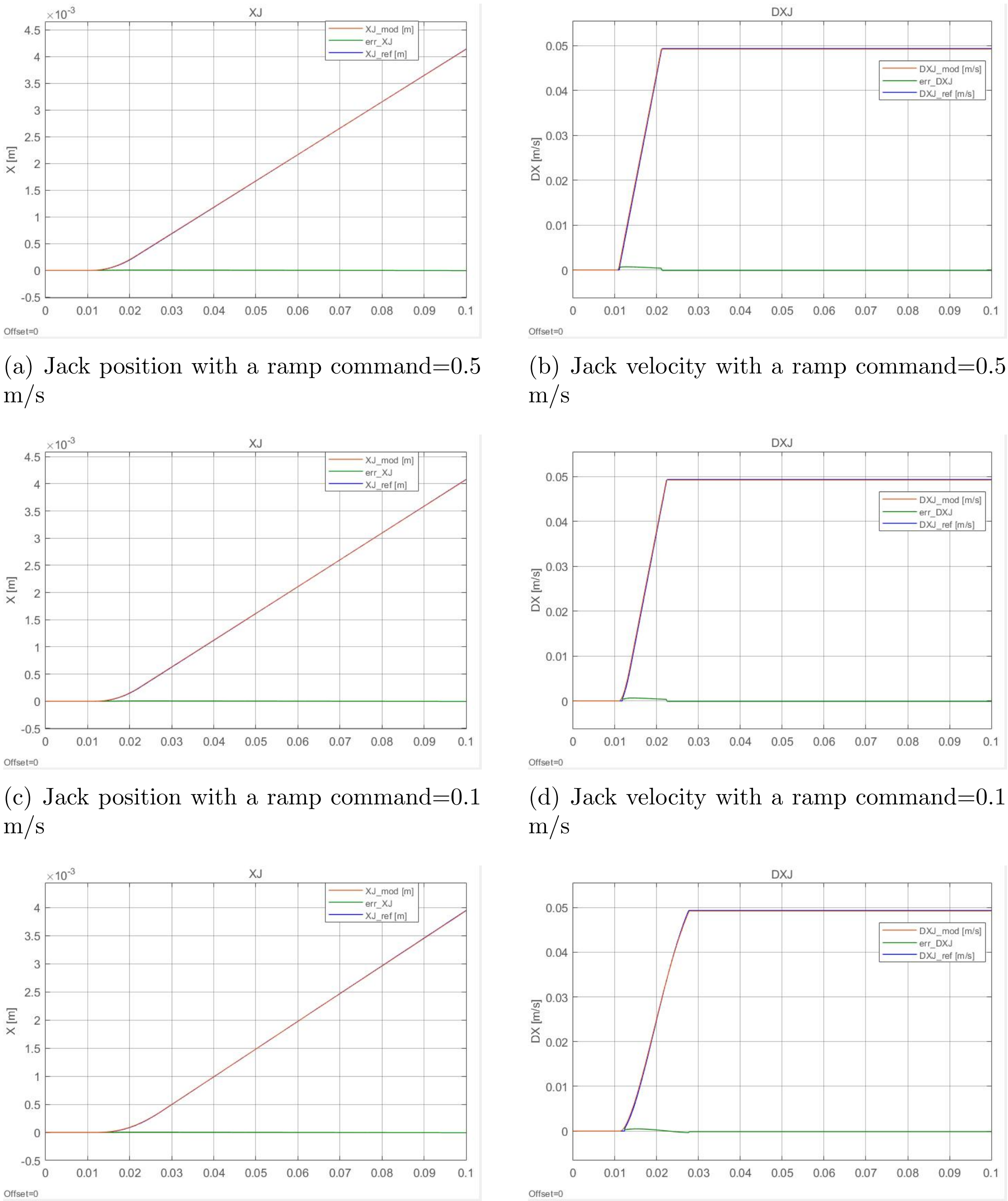
(e) Pressure in the chambers of the cylinder with a step command=0.0005m



(f) [Pressure Difference in the actuator with a step command=0.0005m

Figure 5.14: Pressure and difference pressure in the chambers of the actuator with three different step commands and with the **Double-acting Cylinder (simple) - Cylinder friction** combination

Ramp Commands



(a) Jack position with a ramp command=0.5 m/s

(b) Jack velocity with a ramp command=0.5 m/s

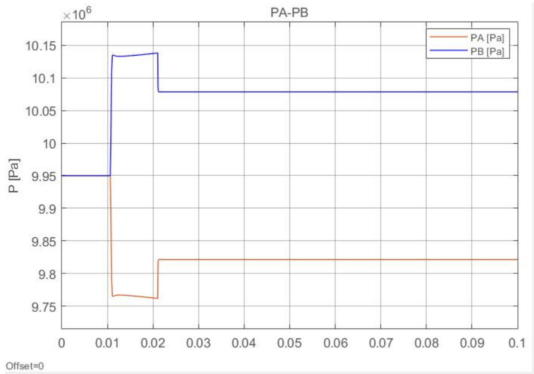
(c) Jack position with a ramp command=0.1 m/s

(d) Jack velocity with a ramp command=0.1 m/s

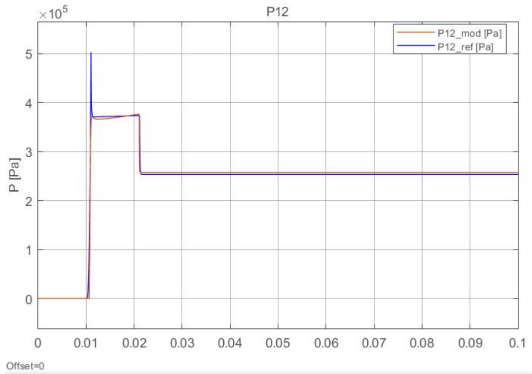
(e) Jack position with a ramp command=0.05 m/s

(f) Jack velocity with a ramp command=0.05 m/s

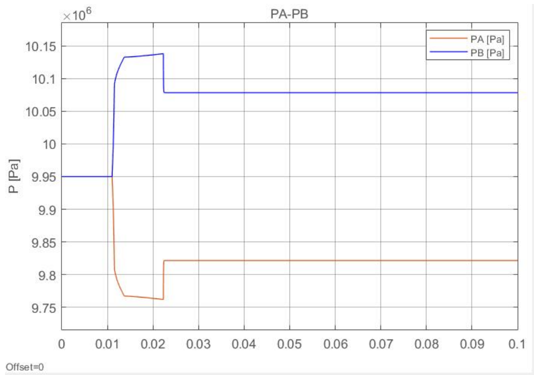
Figure 5.15: Position and Velocity of the actuator with three different ramp commands and with the **Double-acting Cylinder (simple) - Cylinder friction** combination



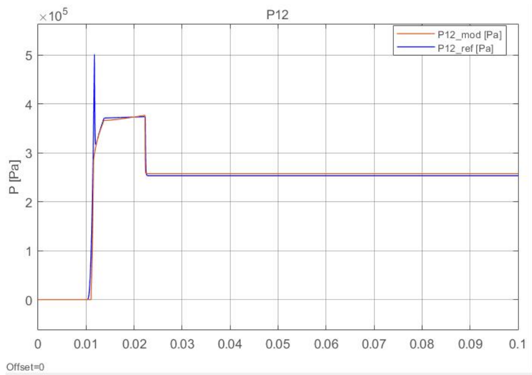
(a) Pressure in the chambers of the cylinder with a ramp command=0.5 m/s



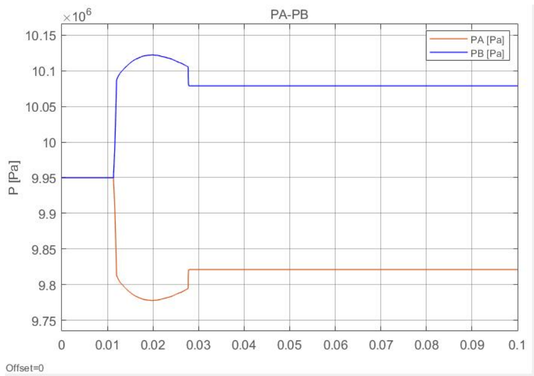
(b) Pressure Difference in the actuator with a ramp command=0.5 m/s



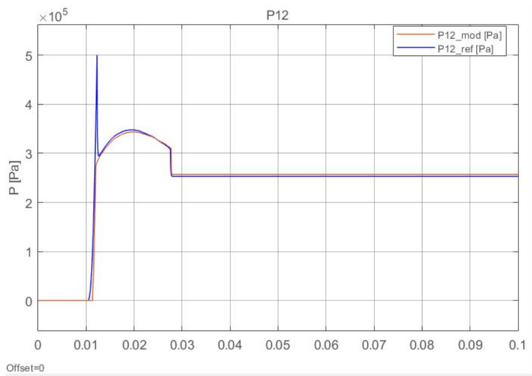
(c) Pressure in the chambers of the cylinder with a ramp command=0.1 m/s



(d) [Pressure Difference in the actuator with a ramp command=0.1 m/s



(e) Pressure in the chambers of the cylinder with a ramp command=0.05 m/s

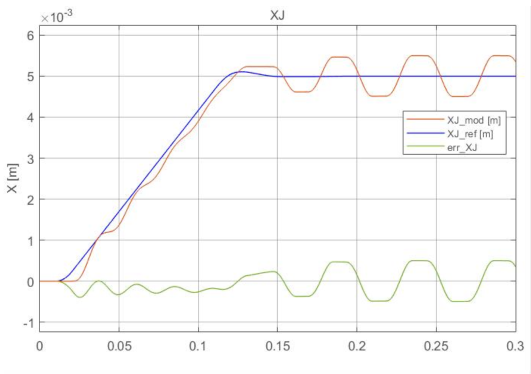


(f) [Pressure Difference in the actuator with a ramp command=0.05 m/s

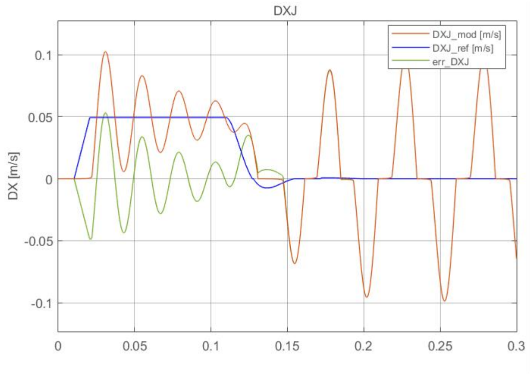
Figure 5.16: Pressure and difference pressure in the chambers of the actuator with three different ramp commands and with the **Double-acting Cylinder (simple) - Cylinder friction** combination

5.1.5 Double-acting Cylinder - Translational friction

Step command

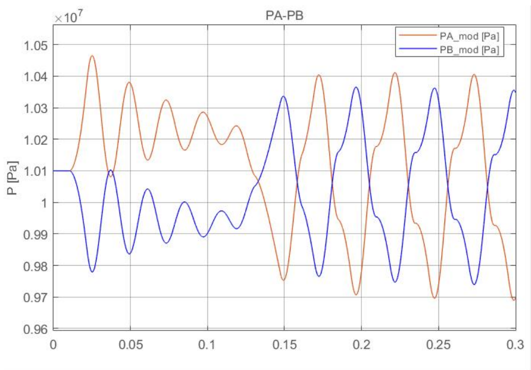


(a) Jack position with a step command=0.005 m

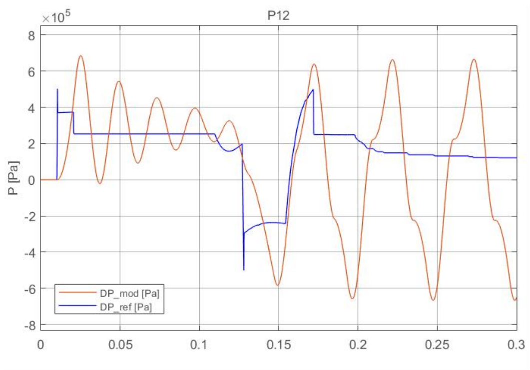


(b) Jack velocity with a step command=0.005 m

Figure 5.17: Position and Velocity of the actuator with a step commands



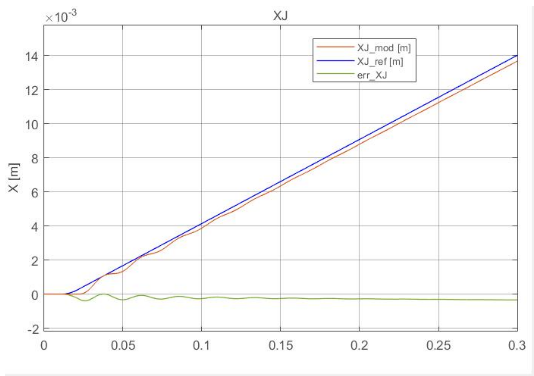
(a) Pressure in the chambers of the cylinder with a step command=0.005 m



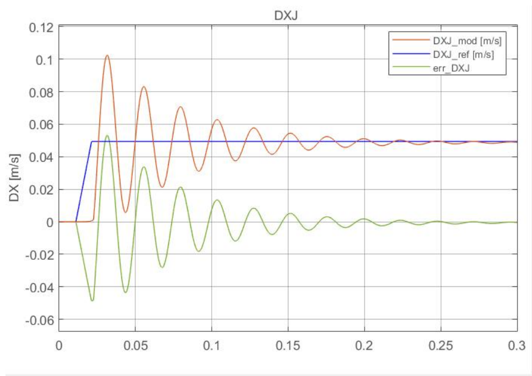
(b) Pressure Difference in the actuator with a step command=0.005 m

Figure 5.18: Pressure and difference pressure in the chambers of the actuator with a step commands and with the **Double-acting Cylinder - Translational friction** combination

Ramp command

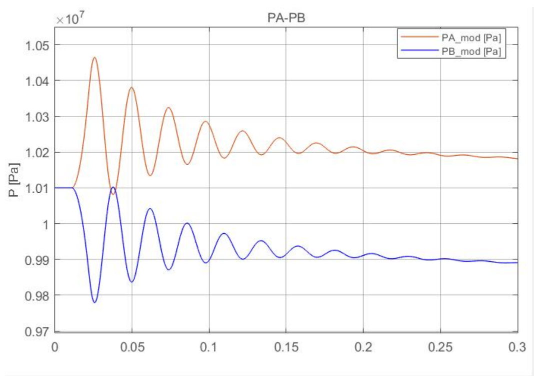


(a) Jack position with a ramp command=0.5 m/s

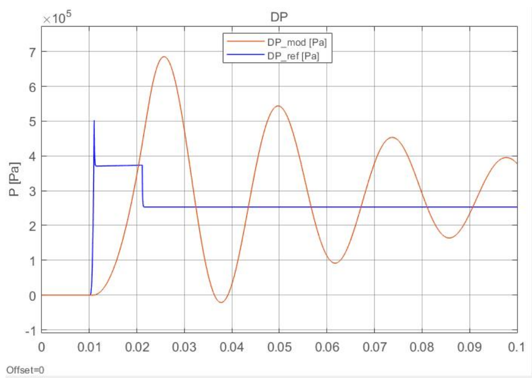


(b) Jack velocity with a ramp command=0.5 m/s

Figure 5.19: Position and Velocity of the actuator with a ramp commands



(a) Pressure in the chambers of the cylinder with a ramp command=0.5 m/s

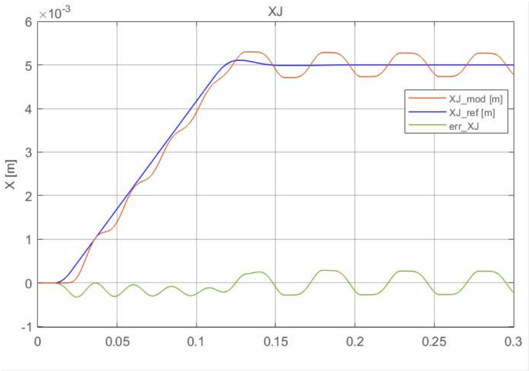


(b) Pressure Difference in the actuator with a ramp command=0.5 m/s

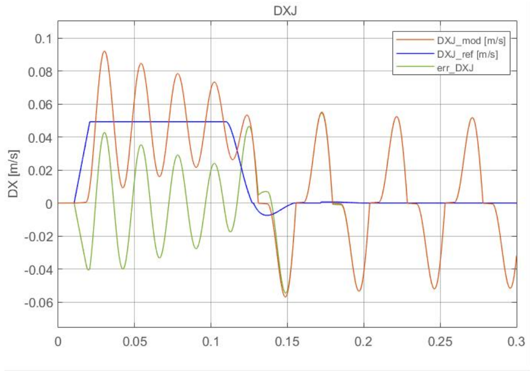
Figure 5.20: Pressure and difference pressure in the chambers of the actuator with a ramp commands and with the **Double-acting Cylinder - Translational friction** combination

5.1.6 Double-acting Cylinder - Cylinder friction

Step command

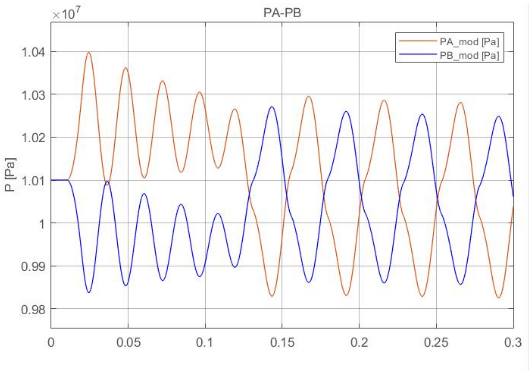


(a) Jack position with a step command=0.005 m

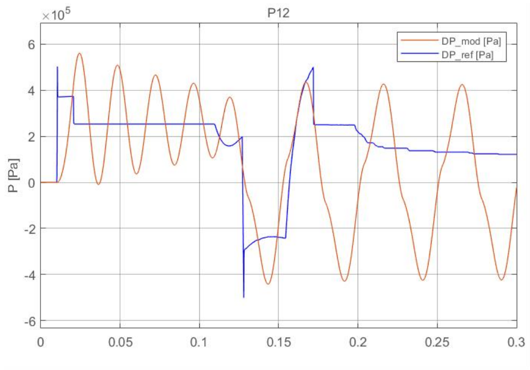


(b) Jack velocity with a step command=0.005 m

Figure 5.21: Position and Velocity of the actuator with a step commands



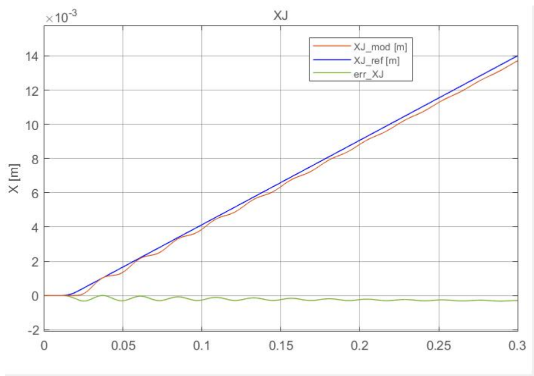
(a) Pressure in the chambers of the cylinder with a step command=0.005 m



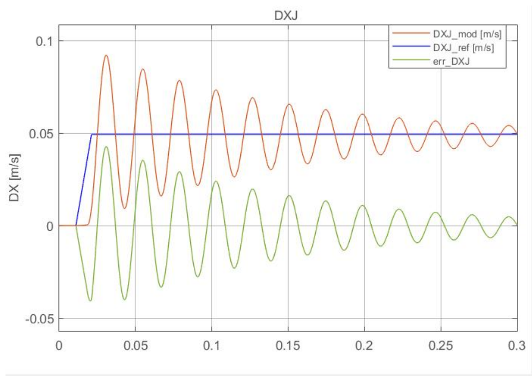
(b) Pressure Difference in the actuator with a step command=0.005 m

Figure 5.22: Pressure and difference pressure in the chambers of the actuator with a step commands and with the **Double-acting Cylinder - Cylinder friction** combination

Ramp command

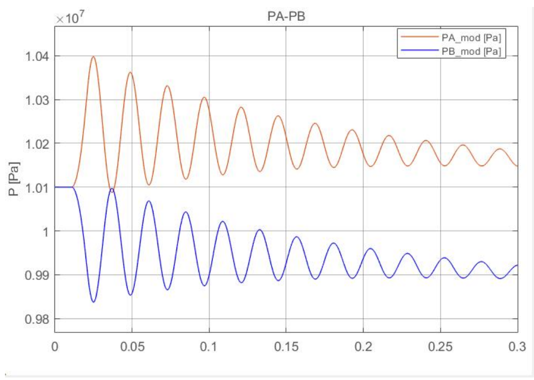


(a) Jack position with a ramp command=0.5 m/s

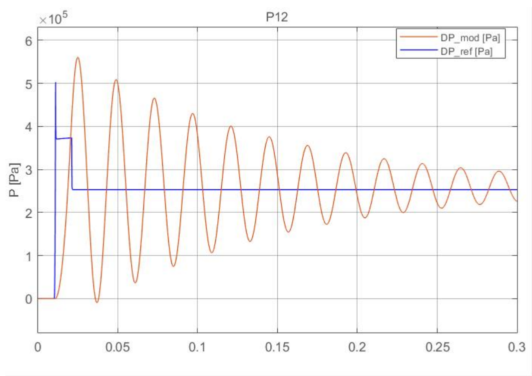


(b) Jack velocity with a ramp command=0.5 m/s

Figure 5.23: Position and Velocity of the actuator with a ramp commands



(a) Pressure in the chambers of the cylinder with a ramp command=0.5 m/s



(b) Pressure Difference in the actuator with a ramp command=0.5 m/s

Figure 5.24: Pressure and difference pressure in the chambers of the actuator with a ramp commands and with the **Double-acting Cylinder - Cylinder friction** combination

In order to have a complete view of what the results are, so as to make a more precise analysis, the error will also be measured, in particular the value of the Mean Absolute Error (MAE), of the position and speed of the jack between the reference model and the developed one.
The error will be measured for each type of command studied and for each combination.

$$MAE = \frac{1}{n}\sum_{i=1}^{n}|Y_{i-mod} - Y_{i-ref}|$$

Two spool - Translational friction

Step Command $[m]$	**XJ Mean Absolute Error** $[m]$	**DXJ Mean Absolute Error** $[m/s]$
0.005	$3.55e-6$	$2.23e-4$
0.001	$2.9e-6$	$2.55e-4$
0.0005	$2.56e-6$	$2.43e-4$

Ramp Command $[m/s]$	**XJ Mean Absolute Error** $[m]$	**DXJ Mean Absolute Error** $[m/s]$
0.5	$2.98e-6$	$1.91e-4$
0.1	$2.91e-6$	$1.86e-4$
0.05	$3.30e-6$	$1.70e-4$

Two spool - Cylinder Friction

Step Command $[m]$	**XJ Mean Absolute Error** $[m]$	**DXJ Mean Absolute Error** $[m/s]$
0.005	$2.94e-6$	$1.87e-4$
0.001	$3.06e-6$	$2.63e-4$
0.0005	$2.64e-6$	$2.48e-4$

Ramp Command $[m/s]$	**XJ Mean Absolute Error** $[m]$	**DXJ Mean Absolute Error** $[m/s]$
0.5	$2.37e-6$	$1.56e-4$
0.1	$2.28e-6$	$1.53e-4$
0.05	$2.23e-6$	$1.37e-4$

Double-acting cylinder (simple) - Translational friction

Step Command $[m]$	**XJ Mean Absolute Error** $[m]$	**DXJ Mean Absolute Error** $[m/s]$
0.005	$3.55e-6$	$2.23e-4$
0.001	$2.9e-6$	$2.55e-4$
0.0005	$2.56e-6$	$2.43e-4$

Ramp Command $[m/s]$	**XJ Mean Absolute Error** $[m]$	**DXJ Mean Absolute Error** $[m/s]$
0.5	$2.98e-6$	$1.91e-4$
0.1	$2.91e-6$	$1.86e-4$
0.05	$3.30e-6$	$1.70e-4$

Double-acting cylinder (simple) - Cylinder Friction

Step Command $[m]$	**XJ Mean Absolute Error** $[m]$	**DXJ Mean Absolute Error** $[m/s]$
0.005	$2.94e-6$	$1.87e-4$
0.001	$3.06e-6$	$2.63e-4$
0.0005	$2.64e-6$	$2.48e-4$

Ramp Command $[m/s]$	**XJ Mean Absolute Error** $[m]$	**DXJ Mean Absolute Error** $[m/s]$
0.5	$2.37e-6$	$1.56e-4$
0.1	$2.28e-6$	$1.53e-4$
0.05	$2.23e-6$	$1.37e-4$

Double-acting cylinder - Translational friction

Step Command $[m]$	**XJ Mean Absolute Error** $[m]$	**DXJ Mean Absolute Error** $[m/s]$
0.005	$2.07e-4$	0.0212

Ramp Command $[m/s]$	**XJ Mean Absolute Error** $[m]$	**DXJ Mean Absolute Error** $[m/s]$
0.5	$2.09e-4$	0.0118

Double-acting cylinder - Cylinder friction

Step Command $[m]$	**XJ Mean Absolute Error** $[m]$	**DXJ Mean Absolute Error** $[m/s]$
0.005	$1.69e-4$	0.0212

Ramp Command $[m/s]$	**XJ Mean Absolute Error** $[m]$	**DXJ Mean Absolute Error** $[m/s]$
0.5	$1.881e-4$	0.0154

5.1.7 Observations block analysis

The various results of the model were included, particularly those relating to the variation of the position of the actuator during the simulation at a given input, as well as its actuation speed.
As previously mentioned, the actuator also influences the fluid dynamics of the servovalve, which is the reason why the results for the pressures in the two chambers of the actuator were also included.
Since the developed SimScape must be compared with the Simulink reference model, the average error values are also presented, i.e. how much the model (Mod) deviates on average from the reference model (Ref).
This analysis was conducted for the different combinations for modelling the actuator.

At first sight, it can already be seen that for the combination in which the block used for the cylinder is the Double Acting cylinder (complete), there are fluctuations in the pressures of the two chambers. This trend then affects the actuator dynamics, as can be seen in the speed and position graph. The blockage then leads to a jack position that oscillates around the reference position with oscillations that decrease more and more. In fact, the average error on the position, although greater than the other combinations, is small. This is not the case for the speed of the actuator, which instead, is characterised by fluctuations of the same amplitude as the speed of the reference model.
This trend is linked to the fact that this block, unlike the other presented, takes into account those parameters which characterise the compressibility of the oil, an aspect that has been shown several times to be not considered in the Simulink reference model and consequently not implemented in the SimScape model.
This consideration leads therefore to the exclusion of the block even after a first simulation with only one step input and one ramp input, even with the Translational friction block and the Cylinder friction one.

Regarding the other combinations, it is possible to see that the results obtained between the two blocks related to the cylinder element, both *Two-spool* and *Double-acting Cylinder(simple)*, obtain results almost identical to those of the reference model.
The only difference that it is possible to evaluate, is related to the response of the system to the input, where the developed model presents very small oscillations. These oscillations are caused by a different fluid dynamics, given by the working environment (SimScape being an environment where there is a more real approach, takes into account more aspects related to the physics of the problem). This effect is more visible in the graphs related to the pressure curves, where the model formed by the *Two-spool*, for the same reason described earlier, better flows the trend of the pressure difference evaluated in the reference model.

Given the remarkable equality of the results obtained, it has been decided to implement in the final SimScape model, the model of the actuator formed by the two *Translational Hydro-Mechanical Converter* rather than the double acting cylinder.
The choice has been guided by the fact that it is a type of element that unlike the others, results to be simpler, since the only information it requires is the area of the cylinder.
The double acting cylinder is still a valid element, above all if in the SimScape and reference model, the fluid dynamic aspect of the system is better analyzed.

As far as the friction modeling block is concerned, the choice falls on Cylinder friction.
If only the system formed by the *Two-spool* is taken into consideration, and if the effects of the two friction blocks are analyzed, it is possible to notice very slight changes in the results related to dynamics, with a better trend of the error in the actuation speed, measured in the case of the Cylinder friction block implementation.
This aspect has been particularly considered, since in a ramp type input, and in particular in the case of a ramp with a very low slope, the risk of negative effects caused by friction are greater and more likely to modify what is the real trend of the system.
Moreover, since the servovalve is modeled considering it within an aeronautical actuation system, a higher accuracy in the actuation speed could be preferable.
Moreover, in the comparison related to the friction blocks, it is possible to see a considerable difference in the pressure trend.
If we take into consideration only the system formed by *Two-spool*, that is the block that better allows to follow the Simulink model, and if we analyze the friction effects, it is possible to see how the Cylinder friction block gives results that better match the reference model.

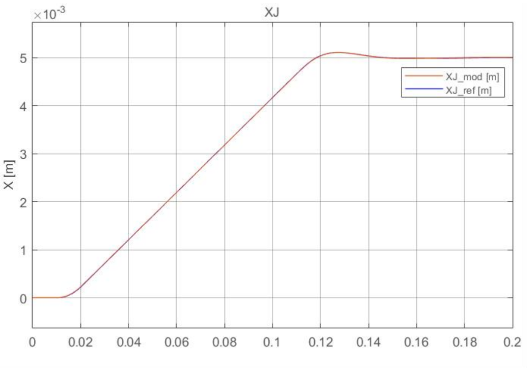
In conclusion, it can be said that the blocks that best follow the reference model, as far as the Actuator System si concerned, are the *Two-spool* combined with the *Cylinder friction* block.

5.2 Two-spool Compressibility analysis

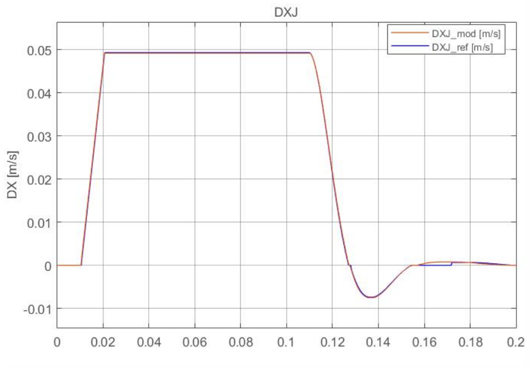
Once we have established which are the blocks that best allow to model the system in SimScape in order to be able to follow the reference system even more closely, it is possible to take a closer look at some of the blocks. The analysis carried out previously led to the decision that the system which best models the actuator cylinder is the combination of two Translational Hydro-Mechanical Converter.

As explained in the previous chapter, this block allows in the definition phase of its parameters to enable, or not, the compressibility of the hydraulic fluid. It has repeatedly been shown that compressibility is not taken into account in the Simulink reference model, thence, in order for the SimScape model to be able to follow it, it must be also excluded in the latter. The analysis carried out in this paragraph is presented both to show how compressibility affects the system as well as for the sake of completeness.
The SimScape model, in which only the blocks that best allow the simulation are implemented, is given an input of type Step with intensity equal to $0.005[m]$, and the dynamics of the jack is shown in the case in which the compressibility in the Cylinder block is deactivated and activated.

Compressibility OFF

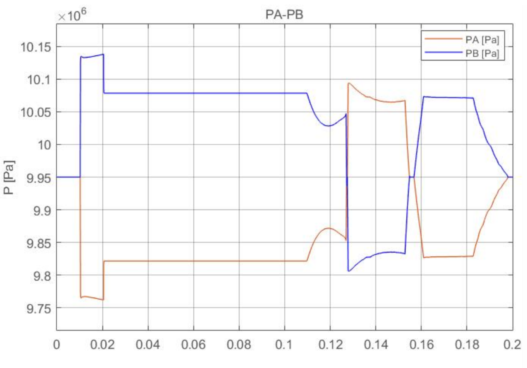


(a) Jack position with a step command=0.005 m

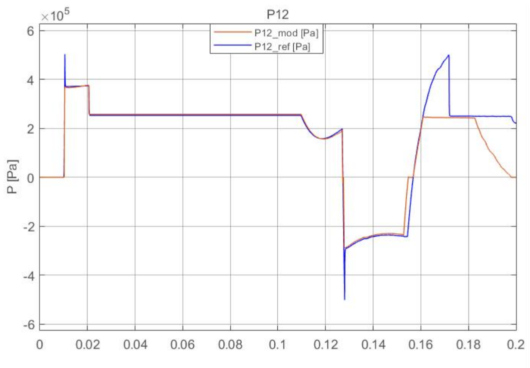


(b) Jack velocity with a step command=0.005 m

Figure 5.25: Position and velocity of the jack with the compressibility OFF



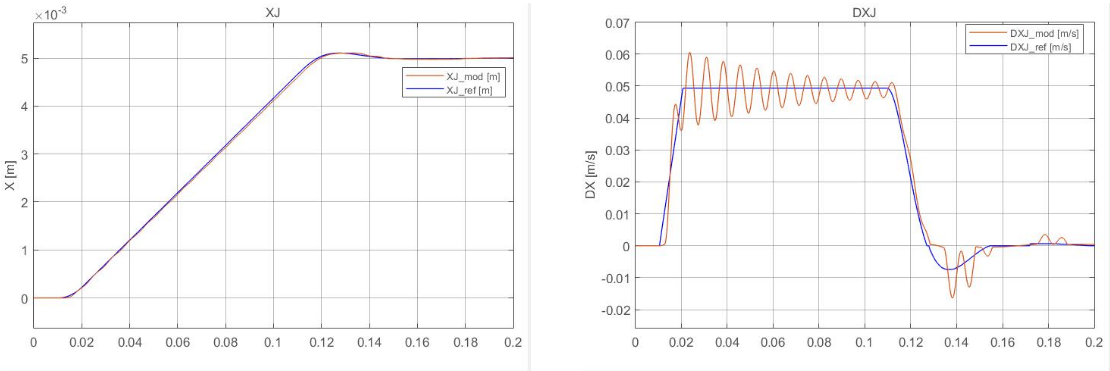
(a) Pressure in the chambers of the cylinder with a ramp command=0.5 m/s



(b) Pressure Difference in the actuator with a ramp command=0.5 m/s

Figure 5.26: Pressure and difference pressure in the chambers of the actuator with the compressibility OFF

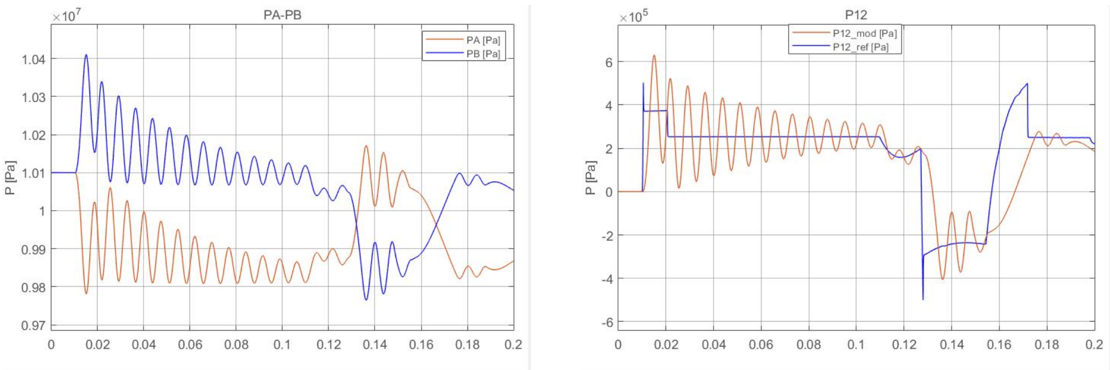
Compressibility ON



(a) Jack position with a step command=0.005 m

(b) Jack velocity with a step command=0.005 m

Figure 5.27: Position and velocity of the jack with the compressibility ON



(a) Pressure in the chambers of the cylinder with a ramp command=0.5 m/s

(b) Pressure Difference in the actuator with a ramp command=0.5 m/s

Figure 5.28: Pressure and difference pressure in the chambers of the actuator wit the compressibility ON

As it is possible to see from the results obtained, even if the Two-spool is the block that best models the cylinder of the actuation system, the compressibility must not be considered.
Considering the compressibility in the block means that, although the system tends to follow the reference model, it does so with a trend characterised by continuous fluctuations in the pressure, which translates into fluctuations in the parameters of the dynamics.
This is therefore also the proof that this SimScape model must necessarily ignore the compressibility of the fluid.

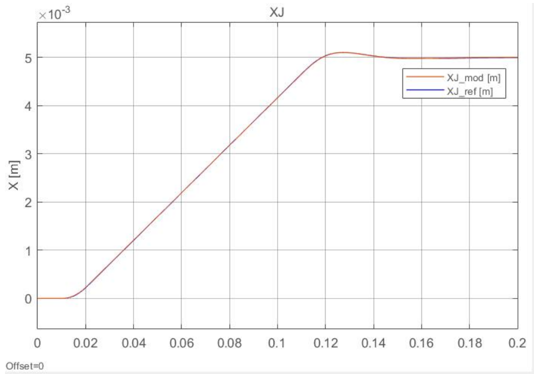
5.3 Pipes analysis

As described in the previous chapter, another aspect that can be modelled in the SimScape environment is that of the pipes that characterise the hydraulic connection between the servo-valve and the actuation system. Will be analyzed the effect on the dynamics and fluidynamics of the actuation system.
Three case studies will be shown:

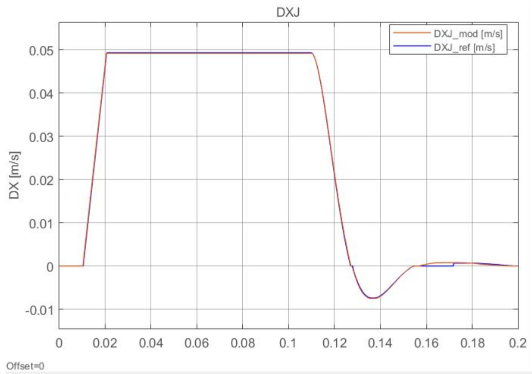
- The model without pipe implementation
- The model with the implementation of rigid pipes
- The model with the implementation of flexible pipes

All analyses were carried out with an input step of 0.005m.

NO pipes implemented

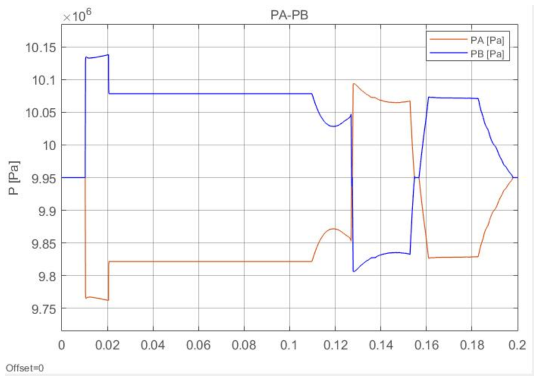


(a) Jack position with a step command=0.005 m

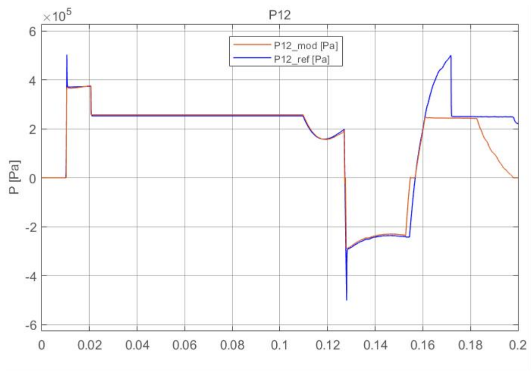


(b) Jack velocity with a step command=0.005 m

Figure 5.29: Position and velocity of the jack with no pipes implemented



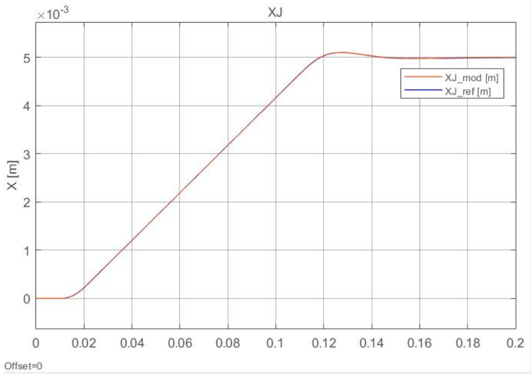
(a) Pressure in the chambers of the cylinder with a ramp command=0.5 m/s



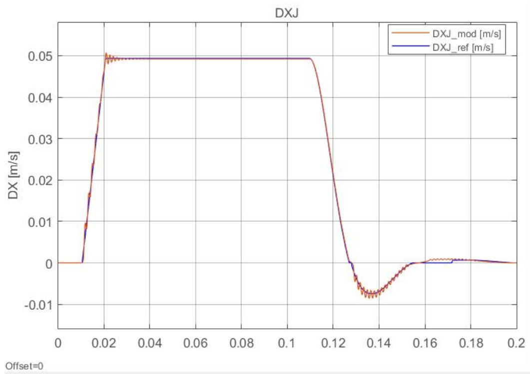
(b) Pressure Difference in the actuator with a step command=0.005 m

Figure 5.30: Pressure and difference pressure in the chambers of the actuator with no pipe implemented

Rigid pipes implemented

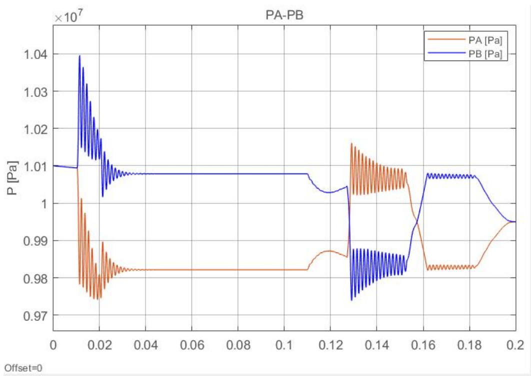


(a) Jack position with a step command=0.005 m

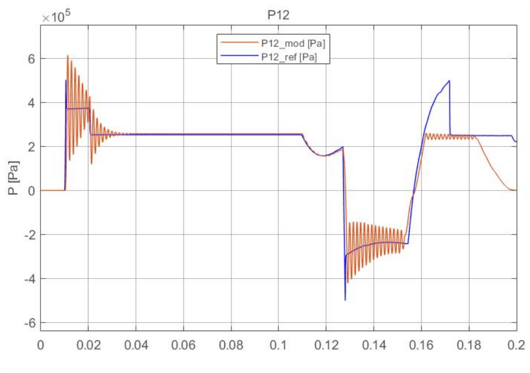


(b) Jack velocity with a step command=0.005 m

Figure 5.31: Position and velocity of the jack with rigid pipes implemented



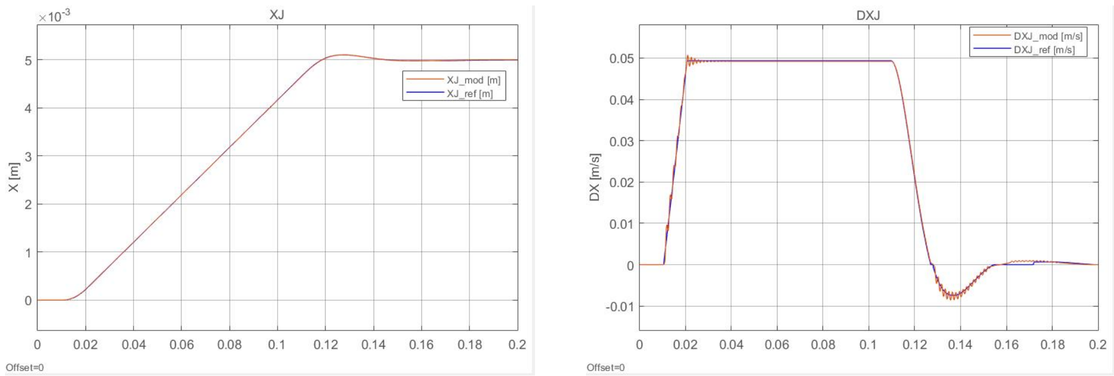
(a) Pressure in the chambers of the cylinder with a ramp command=0.5 m/s



(b) Pressure Difference in the actuator with a step command=0.005 m

Figure 5.32: Pressure and difference pressure in the chambers of the actuator with rigid pipe implemented

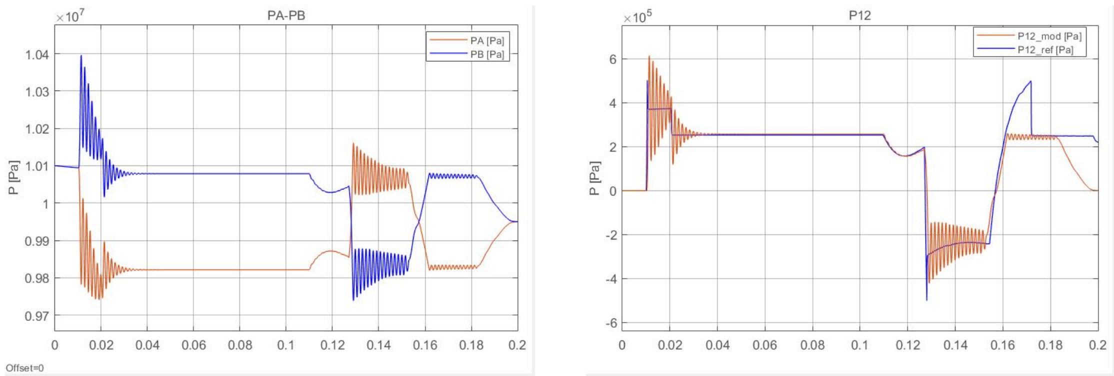
Flexible pipes implemented



(a) Jack position with a step command=0.005 m

(b) Jack velocity with a step command=0.005 m

Figure 5.33: Position and velocity of the jack with flexible pipes implemented



(a) Pressure in the chambers of the cylinder with a ramp command=0.5 m/s

(b) Pressure Difference in the actuator with a step command=0.005 m

Figure 5.34: Pressure and difference pressure in the chambers of the actuator with flexible pipe implemented

As can be seen, even if the system follows the trends of the reference model, slight oscillations accurs in the pressures, which have a slight effect on the actuation speed and a minimal effect on the position reached.
These oscillations only occur at points where the pressure changes drastically, whereas during the constant pressure trend, the SimScape model implemented with the pipes follows the Simulink model quite closely.
This behaviour occurs whether flexible or rigid pipes are implemented, precisely because this means that a different and more complete fluid dynamics model is being considered.
Therefore, it was decided not to implement pipes in the SimScape model.

5.4 External Load Analysis

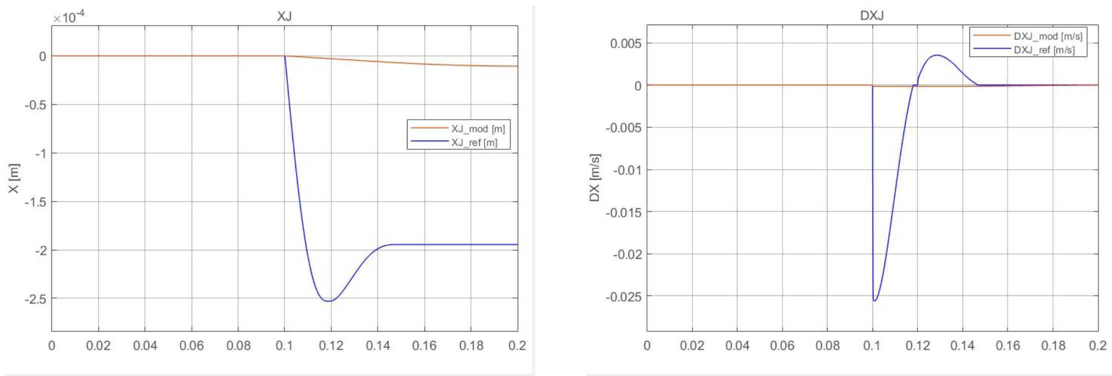
A final aspect to be evaluated in this chapter is how the system responds to an external load.
An external load in this discussion, where the servovalve is considered as if it were part of an aircraft control system, is associated with an aerodynamic force that is generated on the moving surface to which the system is connected.
These forces can be of different intensities and depend on various factors, such as whether the aircraft is on the ground or cruising, the speed of the aircraft itself and the size of the surface involved.
These forces then tend to displace the surface from its nominal position, and therefore the system has to compensate for this disturbance in order to return the error to zero.

In the Simulink model, a block similar to the command input block provides a force input type and its numerical value, which, when changed by a sign, is added to the hydraulic force generated in the actuator cylinder.
In a similar and simple way, the force is supplied to an *Ideal Force Source* via a numerical value from the same block of the simulink model. The physical force signal, after being changed sign, is tied to the mass branch that approximates the cylinder.

The system will be analyzed both in the case in which the system is in steady state condition and an external force acts on it, and in the case in which the system, responding to a command input is subject both to a force that opposes the command itself and to a force that instead favors its implementation.

In all three cases, the external force will be applied with a step input, with an intensity of 8000[N] and at a time instant of 0.1 [s].

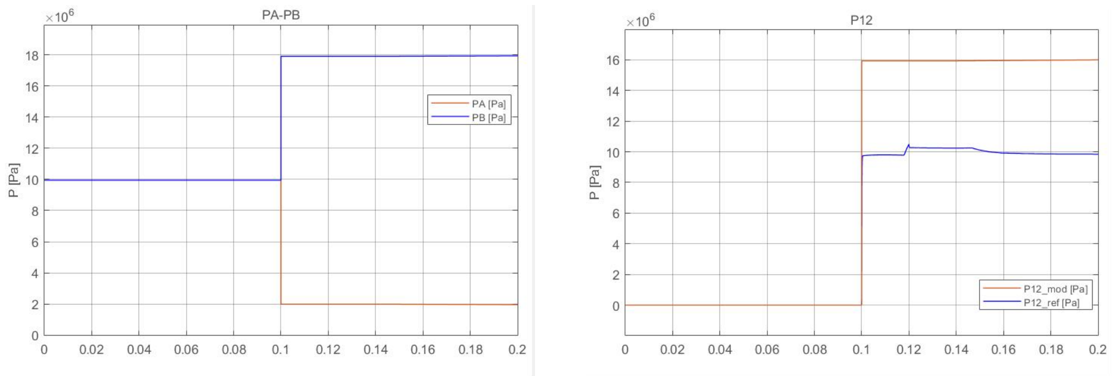
Only external load



(a) Jack position with a step command=0.005 m

(b) Jack velocity with a step command=0.005 m

Figure 5.35: Position and velocity of the jack when only external load are applied

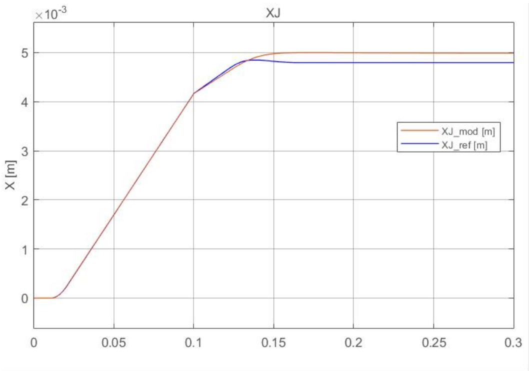


(a) Pressure in the chambers of the cylinder with a ramp command=0.5 m/s

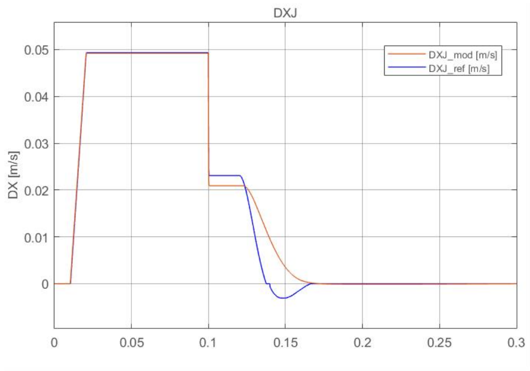
(b) Pressure Difference in the actuator with a step command=0.005 m

Figure 5.36: Pressure and difference pressure in the chambers of the actuator when only external load are applied

Load that opposes to the input

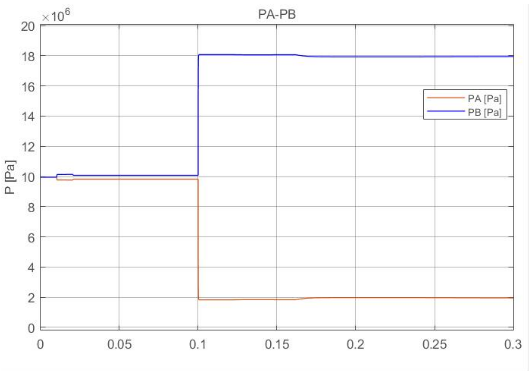


(a) Jack position with a step command=0.005 m

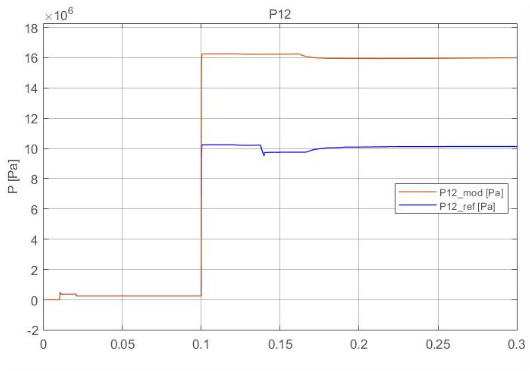


(b) Jack velocity with a step command=0.005 m

Figure 5.37: Position and velocity of the jack when the external oppose to the input



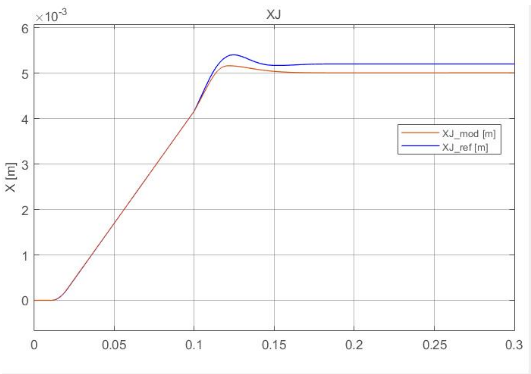
(a) Pressure in the chambers of the cylinder with a ramp command=0.5 m/s



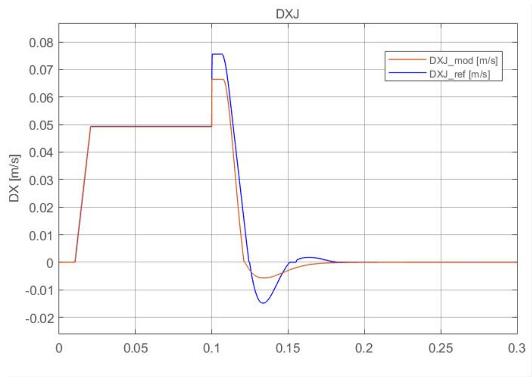
(b) Pressure Difference in the actuator with a step command=0.005 m

Figure 5.38: Pressure and difference pressure in the chambers of the actuator when the external oppose to the input

Load that favors to the input

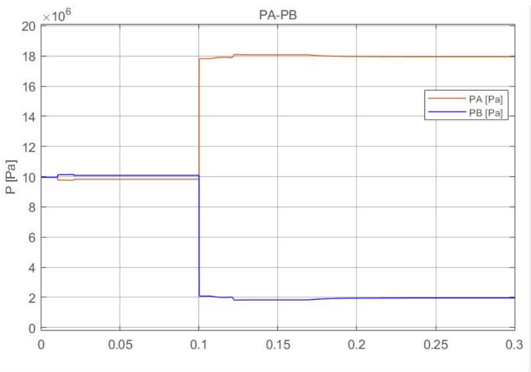


(a) Jack position with a step command=0.005 m

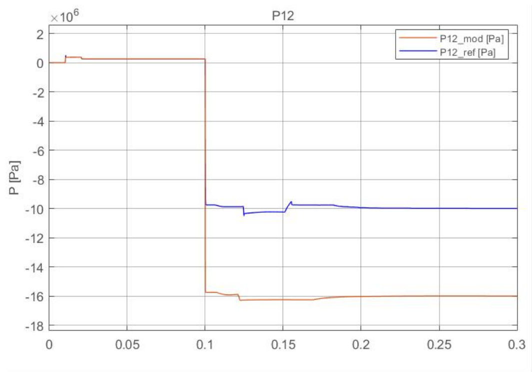


(b) Jack velocity with a step command=0.005 m

Figure 5.39: Position and velocity of the jack when the external oppose favors the input



(a) Pressure in the chambers of the cylinder with a ramp command=0.5 m/s



(b) Pressure Difference in the actuator with a step command=0.005 m

Figure 5.40: Pressure and difference pressure in the chambers of the actuator when the external favors the input

If an external force is applied during the implementation of a command, the system has a response in the dynamics that deviates slightly from the reference.
This is not the case if the system is subjected only to the external force and only has to compensate this perturbation.
In this case, as it is possible to see from the graphs, the model developed on SimScape, feels the external force but is unable to implement a compensation similar to that of the reference model.

In reality, this behaviour is not entirely incorrect, since the SimScape model tends to be more stable and rigid. This aspect can be seen in the fact that the reference system, after the perturbation, cannot bring itself to the initial or commanded condition, which instead the developed model does after a certain period of time. This difference in operation is due to the fact that the environment and the calculation logic of the two systems is different.
However, it is still not a correct answer, both because it does not follow the reference model and because it is far too rigid and less sensitive to perturbations.

Therefore, even if the implementation of the external load system is simple, the various simplifications made at the level of the fluid dynamics of the system, such as the repeated elimination of the compressibility of the oil, leads to inadequate operation.

Chapter 6

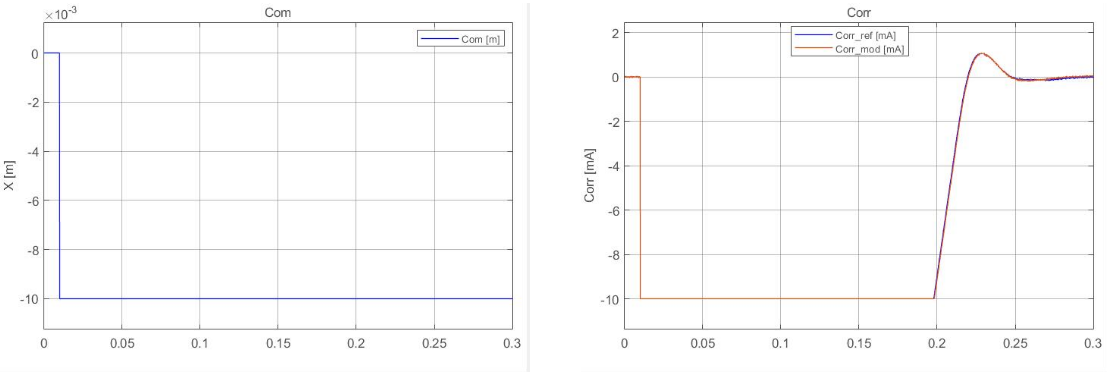
Model Results

Having studied the model and the various blocks that can be implemented in it, making all the various considerations and simplifications, a SimScape model was obtained which follows and respects the Simulink reference model quite faithfully.
In this chapter, the results will not be presented in order to evaluate them, but rather the final results of the final model will be shown.
In addition, not only the results of the actuation system will be shown as in the previous chapters, but the results of all systems that make up the SimScape model.

Three types of input will be provided to the system, and it will be shown how the system responds.

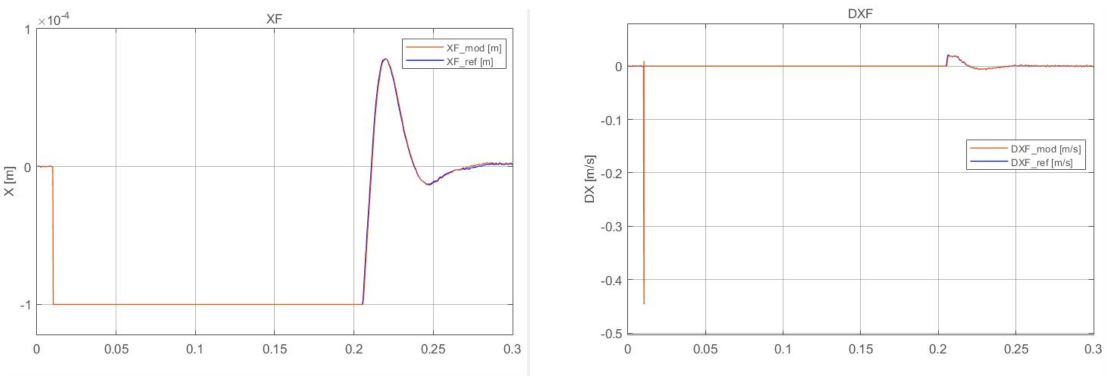
6.1 Step Command

Step Command	
Initial value	$0m$
Final value	$-0.01m$
Input instant application	$0.01s$
Step Time	$0.3s$



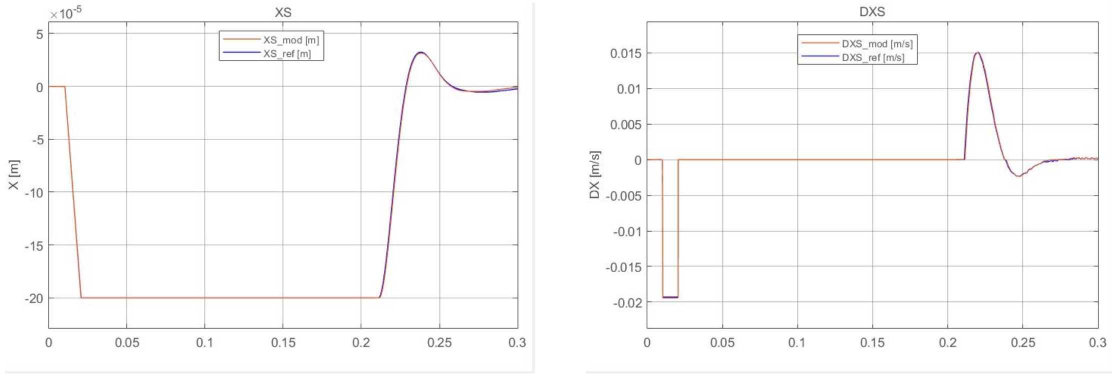
(a) (b)

Figure 6.1: Command and Current with a step command



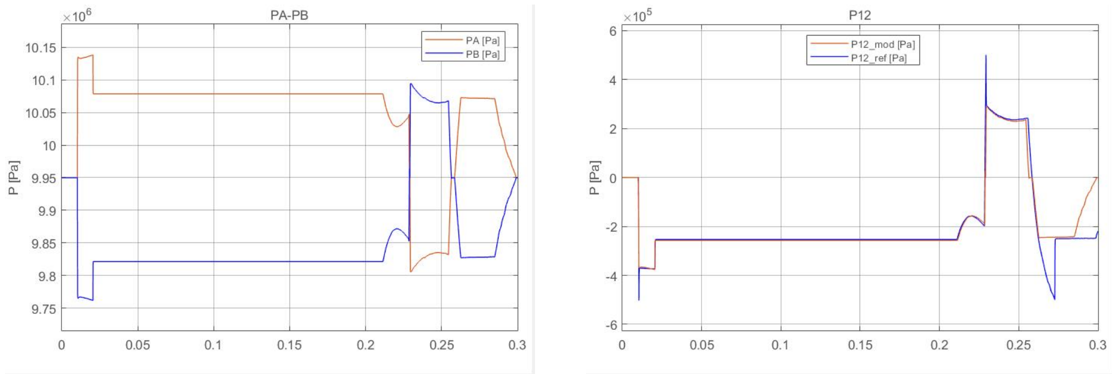
(a) (b)

Figure 6.2: Position and Velocity of the flapper with a step command



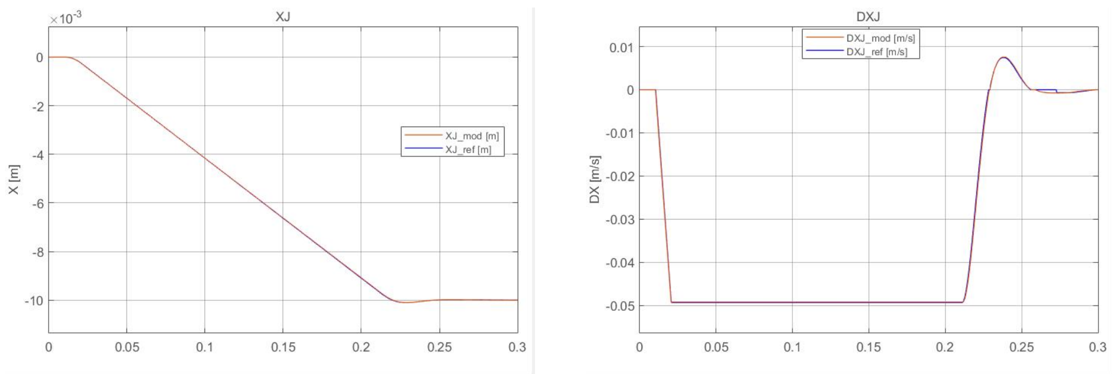
(a) (b)

Figure 6.3: Position and Velocity of the spool with a step command



(a) (b)

Figure 6.4: Pressures and Difference pressure with a step command

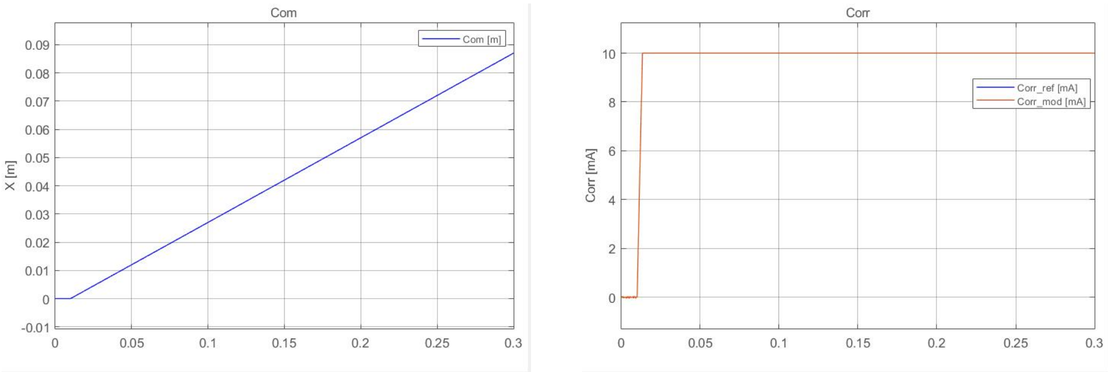


(a) (b)

Figure 6.5: Position and Velocity of the jack with a step command

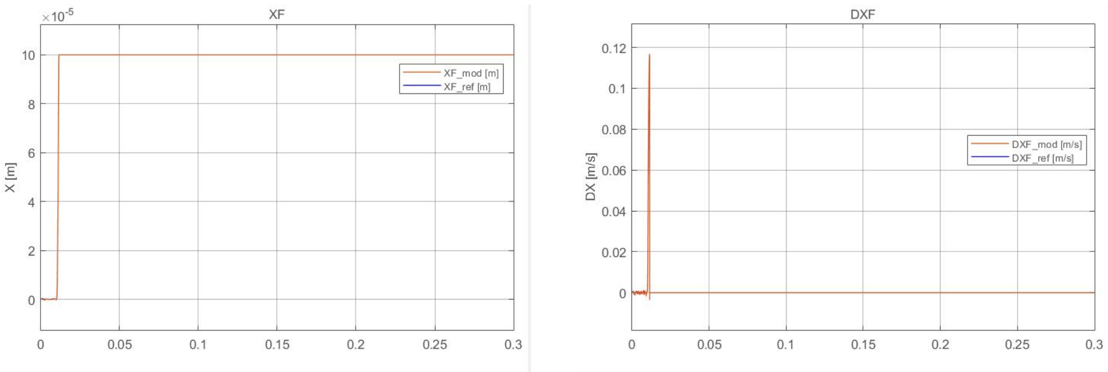
6.2 Ramp Command

Ramp Command	
Initial value	$0m$
Ramp slope	$0.3m/s$
Input instant application	$0.01s$
Step Time	$0.3s$



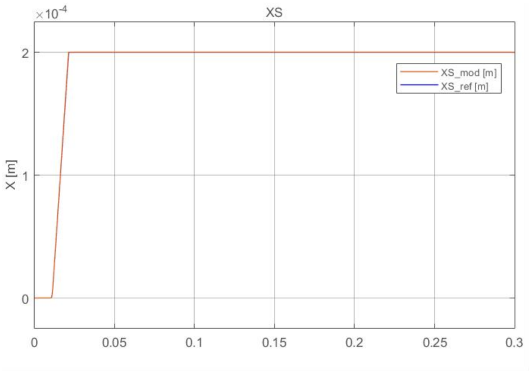
(a) (b)

Figure 6.6: Command and Current with a ramp command

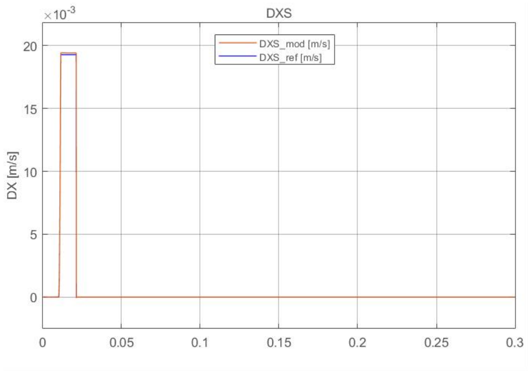


(a) (b)

Figure 6.7: Position and Velocity of the flapper with a ramp command

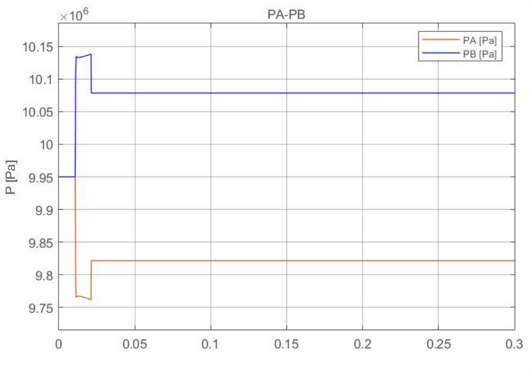


(a)

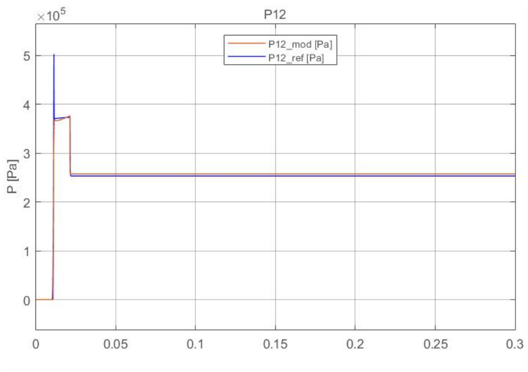


(b)

Figure 6.8: Position and Velocity of the spool with a ramp command

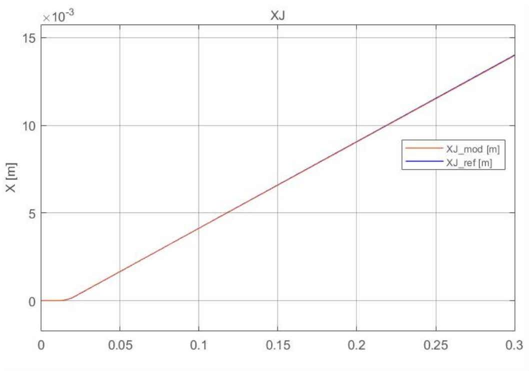


(a)

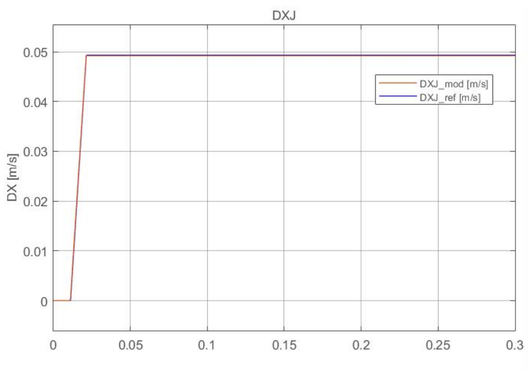


(b)

Figure 6.9: Pressures and Difference pressure witha ramp command



(a)

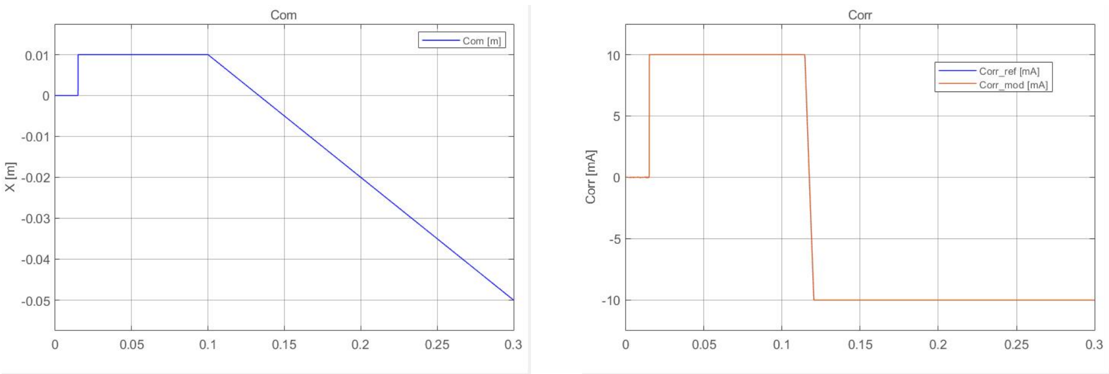


(b)

Figure 6.10: Position and Velocity of the jack with a ramp command

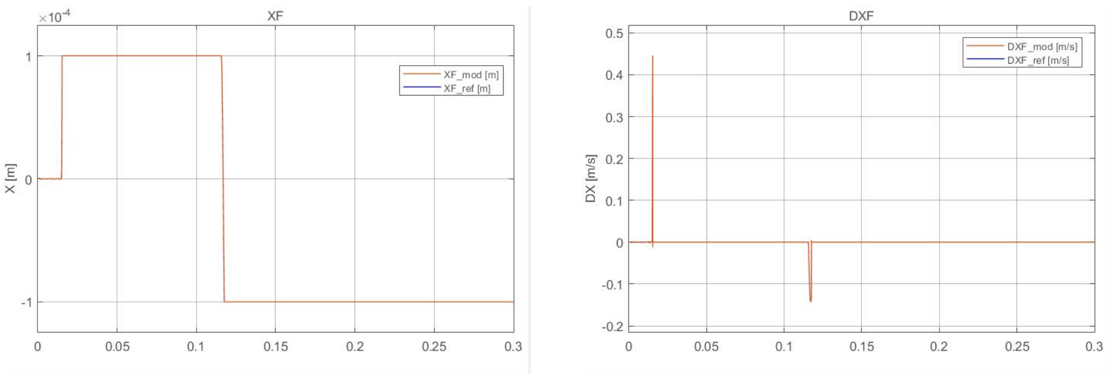
6.3 Simultaneous Commands

	Step Command	**Ramp Command**
Initial value	$0m$	$0m$
Final Value	$0.01m/s$	$-0.3m/s$
Input instant application	$0.01s$	$0.1s$
Step Time	$0.3s$	$0.3s$



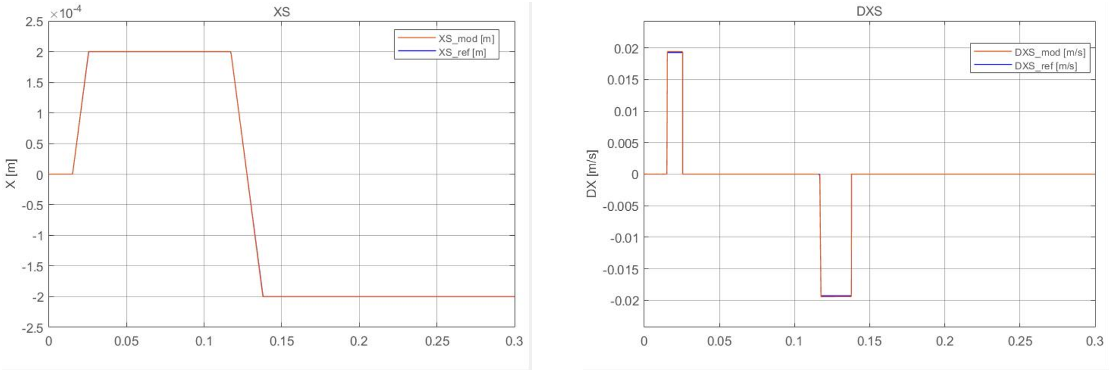
(a) (b)

Figure 6.11: Command and Current with simultaneous commands



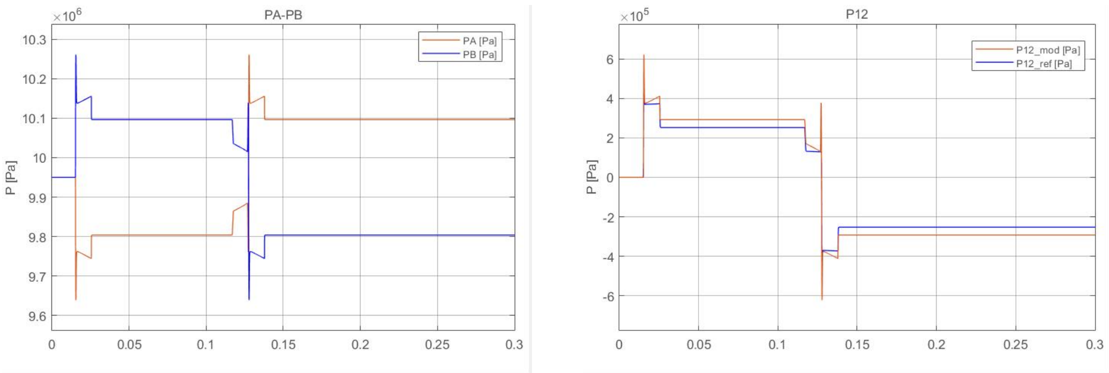
(a) (b)

Figure 6.12: Position and Velocity of the flapper with simultaneous commands



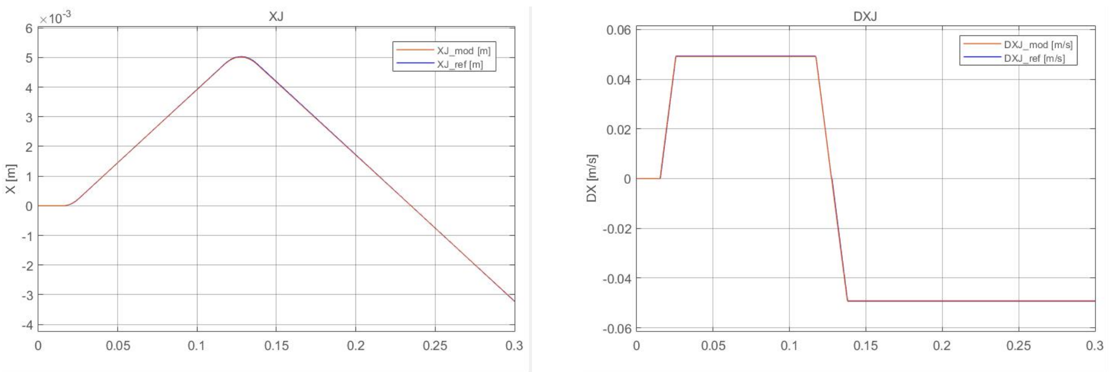
(a) (b)

Figure 6.13: Position and Velocity of the spool with simultaneous commands



(a) (b)

Figure 6.14: Pressures and Difference pressure with simultaneous commands



(a) (b)

Figure 6.15: Position and Velocity of the jack with simultaneous commands

Chapter 7

Conclusions and future works

The main purpose of this thesis is the development of a SimScape model of an electro-hydraulic actuator [EHA].
In particular, in the development, the aim was to shape this model in a way that it is close to a Simulink model of an EHA.
The need to develop this thesis, and therefore the related SimScape model, was to shape a new model that would use a more physical approach to simulation.

Simulink is a multi-domain program based on MATLAB, where dynamic systems can be modelled, analysed and simulated. Both models presented in this thesis have been developed in Simulink, the difference being that the reference model follows a purely numerical approach, whereas the relationships between the various blocks, and thus between the various subsystems of the EHA, have been modelled using mathematical relationships.
On the other hand, the model developed in this thesis uses Simulink as a working program, but it does so using one of its add-ons, SimScape.
SimScape makes it possible to model, analyse and simulate dynamic systems, even though through a more physical approach. The various elements and connections of the different blocks that characterise the EHA are modelled considering their physicality. Through SimScape, it is therefore possible to evaluate a system taking into account not only the mathematical laws that define its operation, but also those parameters linked to how it is physically made.
Furthermore, another substantial difference between the two working environments is that Simulink allows a unidirectional signal flow, while Simscape is bi-directional.
In the SimScape model developed in this thesis, the need to refer to a Simulink model allowed a fairly simple and straightforward comparison, but it also meant that several simplifications, and thus limitations to the modeling, had to be implemented.
The largest limitation that had to be imposed in this model is related to the fluid-dynamics of the system.

Although the Simulink model, it allows to obtain very valid data and quite likely with the real operation of an EHA, it takes little account of what is the real fluid dynamics, both because the software environment in which it was developed does not allow to do so, but also because this model requires a more detailed numerical modeling of fluid dynamics.
The operating environment of SimScape, as has been repeatedly shown during the analysis carried out in this thesis, largely allows to consider these aspects in the final model, which inevitably leads to results differences.
Therefore, the SimScape model developed and presented, allows to obtain data that are very close to what are the results of the Simulink model and, therefore, to that which is the real operation of an EHA. They can therefore be used for a possible series of prognostic tests, using characteristics mainly linked to the physicality of the component to be analysed.
However, higher simulation accuracy can be achieved if the fluodynamic aspect is modeled more thoroughly.
In addition to the fluid-dynamics of the system, it is also possible to implement thermodynamics in the simscape environment.
It is also possible to analyse and model the effects or factors influencing the thermal aspects of the actuation system.
Moreover, it is possible to improve the precision of the simulation, facing a different modeling of the system, studing and considering the great diversity of blocks that the SimScape library provides and that can be used in substitution or in addition to the blocks already considered.
Consequently, if the model needs to be implemented for more precise prognostics, the above considerations must be implemented as well in what will be the reference model.

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