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Research Article

Automatic Control System for an Oil-Hydraulic Actuator of a Scissor Lift

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Abstract

Lifting equipment's have the purpose to lift goods within their safe working load and design rules specified in standards. However, some applications require additional care regarding handling the load when this care is out of the standard's specifications.

This paper presents the control and supervision system development of a scissor lift table, which should ensure, during its motion, that the vertical speed is constant. This lift table is working alongside a system for handling boxes composed by 3 entry conveyors and 1 exit conveyor.

In this work, it was carried out the study of the direct and inverse kinematics to obtain the equations of position, speed and force for the oil-hydraulic actuator and the applicability of Denavit-Hartenberg's algorithm for these equations. It was proposed a hydraulic circuit, with the selection of the main hydraulic components and the design of the oil-hydraulic actuator (a cylinder) with the magnetostrictive transducer as well. A programmable logical controller was selected, as well as the sensors and actuators for the conveyor system and a program was developed which allows to automatically control the system.

The lift table's velocity control is done by equations which define an oil flow profile of the hydraulic cylinder, which ensures a constant vertical speed of the lift table. To improve the control of the system, initial and final flow ramps were added to the equations, with reduced oil flow at the target height.

It was observed that the use of Denavit-Hartenberg's algorithm revealed to be limited, on which it was necessary to use other calculation methods to obtain the equations of speed and force. However, the proposed oil-hydraulic circuit, as well as the developed speed control, allow the control of the lift table's position and speed.

Keywords: Scissor lift; Denavit-Hartenberg algorithm; Oil-hydraulic actuator; Magnetostrictive transducer; Proportional flow control valve; Programmable logic controller

Introduction

THIS paper presents the work developed for the project of an automatic control system for an oil-hydraulic actuator installed on a scissor lift.

To control the trajectory of a scissor lift, the knowledge about the mechanical system's kinematics as well as the available tools that allow analysis and characterization of the motion is important. Although there are tools, like Computer Aided Design/Computer Aided Engineering (CAD/CAE) software that can simplify the project of a mechanical engineer, it is fully important that the design engineer has background knowledge on the mechanical systems kinematics for his mind to be more design-oriented on the development of the mechanical systems.

On the lifting equipment sector there are mechanical systems such as clamps for lifting steel or marble plates and scissor lifts, for vertical lifting of persons or goods like a motorcycle in a repair workshop. The latter consists of a scissor mechanism assisted by a hydraulic cylinder connected to the scissor mechanism.

The scissor lift mechanism has been studied in different cases for the formulation of equations of the kinematics. Kirsanov [1] presented the generic equations for the calculation of the mechanical strength and deformation of the bars of a "n" scissor mechanism moving on a horizontal trajectory. Pervan et al. [2] presented a kinematic analysis of a car jack with a diamond shape mechanism, obtaining the equations for the vertical height and lift speed written in order to the angle of rotation of the power screw that actuates the lifting and lowering motion. They compared the results with a CAD simulation with difference observed on the initial values of the speed, since the kinematic analysis did not take into consideration the stationary state in the beginning of the lift.

The stability of the scissor lift equipment for lifting people is a safety concern for both users and manufacturers of this type of equipment. This subject was studied by Ronaghi et al. [3] on the risk of tipping over when this equipment is used in great heights. They concluded that even if all safety specifications are in effective use, there is the risk of tipping over, especially for heights above 5,49 m with a minimum side force of 623 N, which is a value that can be achieved by human force. Having this in mind, they recommend the usage of stabilizers to increase the contact area of the lift, increasing its static stability.

The tendency to tip over was also studied by Dong et al. [4] by using the same type of scissor lift platform in two series of tests: by impact against a curb, in a perpendicular and in an oblique direction and by impact on a hole. They concluded the dampening and elasticity of the scissor's structure only absorb part of the impact energy and the flexibility does not increase the vertical height threshold for the platform to tip over, recommending the increase of the structure rigidity.

To minimize the tipping over risk, a safety feature was developed by Shrivastava et al. [5] through a controller they named by Inertia Measurement Unit, which records the tilting angle of the scissor lift platform during its operation. Simultaneously it controls the hydraulic cylinders to stabilize the platform. The control system counts with a printed circuit with an accelerometer and a gyroscope. The measurement results improved from a ±1,5° angle measurement without the proposed system to a ±0,4° angle with the proposed system.

Some scissor lift mechanisms require the control of the vertical velocity. A control system was presented by Stawinski et al. [6] to control the vertical speed of a scissor lift, with two hydraulic cylinders and a fixed displacement pump. Position sensors were installed on the cylinders and a frequency inverter was installed on the pump's electric motor. They introduced a Quasi-Open Control Loop where the controller receives the signal from the position sensors and controls the frequency inverter. With this procedure, the oil flow that goes into the cylinder is controlled by the electric motor speed, which returns a specific oil flow. This controller is Quasi-Open because it receives geometrical data from an external source to calculate the "i" ratio which is a mathematical relation between de cylinder displacement and the scissor mechanism. They've done simulations in computer and in laboratory for three types of loads: without load, 50% of full load and full load, and they concluded that the platform is more stable with reduced amplitude and frequency of vibrations, through the reduction of the pressure pulsation.

Some scissor lifts were developed for applications other than industrial equipment. The project proposed by Báez et al. [7] is inspired in the difficulties that disabled people without lower limbs have when entering and exiting a wheelchair. The authors propose a scissor mechanism to raise or lower a secondary seat of a wheelchair. The disabled person uses this seat with mechanism to reach the primary seat of the wheelchair. The scissor mechanism is different from the standard scissor: it has two pairs of bars per side connected by the same end point. One pair of bars have the common point connected to the structure of an electric direct current motor and the other pair of bars have the common point connected to the nut of the power screw. The motor is connected to a power screw and a nut is also assembled to produce the horizontal actuating motion. The wheelchair is controlled remotely by a smartphone app, which sends signals to raise or lower the seat and signals for the movement control of the wheelchair. Also, a PID controller was added to control the response of the secondary seat and the wheels motion.

The energy used on scissor lift equipment is by standard the hydraulic oil, pressurized from a pump connected to an electric motor, which is fed with electric energy from the mains power. Some applications require that the scissor mechanism works without having to consume electrical power and Hariri [8] has presented a scissor mechanism to assist the cleaning of photovoltaic panels, without consuming any electrical power from the panels. These panels are installed on a dusty environment and a mobile rotary cleaning unit is installed on each panel. To ensure the motion of the cleaning unit, it has the scissor mechanism linked to the mobile cleaning unit by a wire and the mechanism is actuated by a Shape Memory Alloy (SMA) Wire Actuator. This SMA makes use of the thermal energy available at the production site to produce the desired motion. Although the stroke of this actuator is about 5% of its original length of the wire, the author had to optimize the scissor mechanism parameters (number of scissors, arm length and hydraulic cylinder position) so the small stroke could be used as well as the actuation force is within the required level. The mechanism was defined as a 5-level scissor lift mechanism which needs a maximum force below 250 N and an actuator stroke of 22 cm.

This project aims the development of a control system for an oil-hydraulic actuator (a cylinder) installed on a scissor lift, to control the lift table's vertical speed, aiming to keep it constant along the motion. This system is composed by the conveyor lift, a set of entry conveyors and one exit conveyor, which must move boxes from the entry conveyors to the exit conveyor by positioning the conveyor lift at the right entry conveyor level. The system is controlled by a Programmable Logic Controller (PLC) and supervised locally via a Human Machine Interface (HMI) and remotely through a Webserver application. Also, in analysis, is the applicability of the Denavit-Hartenberg's (D-H) algorithm to the scissor mechanism, to obtain the equations for position, speed, and force. The Inverse kinematics equations of position were also obtained via a different method. To increment the applicability of the equations, the development of a hydraulic circuit and the selection of a position sensor for the hydraulic cylinder is also done.

An oil-hydraulic circuit was developed, with a cylinder, a proportional flow control valve with a rectifier plate for unidirectional flow control through the valve. A PLC was selected as well as other sensors and actuators, so the system can do its tasks in an automatic manner.

Materials and Methods

In a factory there is the need to move products packed in boxes, which come from 3 entry conveyors, lay at 3 different heights, which are 600 mm, 1400 mm and 2200 mm, into a single exit conveyor with a fixed height. The boxes are moved from the entry to the exit conveyor through a lift with double scissor system, actuated by a hydraulic cylinder. The lift platform has a conveyor to receive and send the boxes with 1000x400x630 mm with a maximum weight of 500 kg.

It is assumed that all the conveyors are actuated by electric motors and have photoelectric sensors to detect the presence of the boxes and the raising and lowering motion must be performed with a constant vertical speed of 150 mm/s.

Figure 1 presents the layout of the case study.

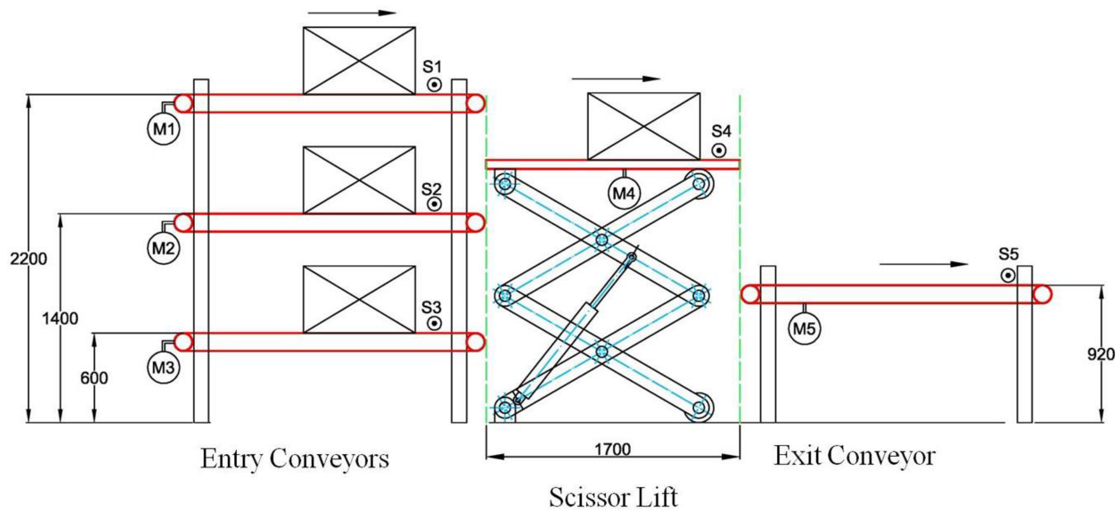

Figure 1: Layout of the case study and some of the sensors and actuators.

Kinematic equations

This section presents the direct and inverse kinematics that relate the position, speed and force of the hydraulic cylinder and the lift platform.

Position equations from Denavit-Hartenberg algorithm

The direct kinematics equations are determined via D-H algorithm and based on reference [9]. To apply the D-H algorithm, a set of 3 reference frames has been defined over the structure nodes and only using bars "AC", "CE" and the actuator, as presented on Figure 2.

The base frame x_0-y_0 is defined at the point "A" with x_0 axis aligned with the horizontal. According with the D-H algorithm, the frame x_1-y_1 is also positioned on the point "A" with the "x_1" axis aligned with the axis of the A-C bar. The frame x_2-y_2 is positioned on the point "X2" with the "x_2" axis aligned with the longitudinal axis of the cylinder. The frame x_3-y_3 is positioned on the X3 point.

The D-H parameters that define the frame position and orientation are presented on Table 1. The variables presented on bold are system variables on which θ1 and θ3 represent the rotational joints of bars "AC" over point "X1" and "CE" over point "X3", respectively. The system variable "d1" represents a prismatic joint, which is the hydraulic cylinder.

Table 1: System variables for the mechanism.

i	θ_i	α_i	a_i	d_i
1	θ1	0	$af1$	0
2	θ2	0	$\boldsymbol{d1}$	0
3	θ3	0	$\frac{lb}{2}+af2$	0

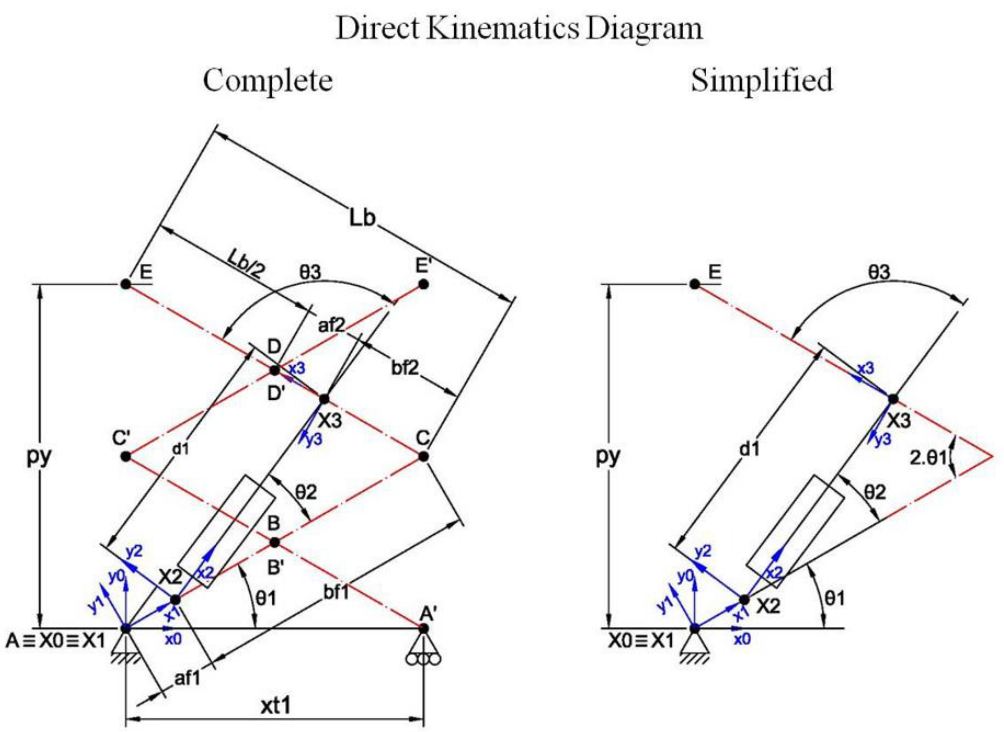

Figure 2: Complete and simplified diagram of the scissor lift.

By assembling all transformation matrix and multiplying them, the global transformation matrix, ${}^{0}A_3$ that relates the point "E" (lift platform) with the point "X1" (base reference frame) is obtained and presented in equation (1). To better legibility, a writing simplification in trigonometric terms is implemented, where C_{123} represents the cosine of the θ1 θ2 and θ3 angles' sum and S_{123} the sine of the three angles' sum.

$$ {}^{0}A_3 = \begin{bmatrix} C_{123} & -S_{123} & 0 & \left(\frac{lb}{2}+af2\right)\cdot C_{123}+d1\cdot C_{12}+af1\cdot C_1 \\ S_{123} & C_{123} & 0 & \left(\frac{lb}{2}+af2\right)\cdot C_{123}+d1\cdot S_{12}+af1\cdot S_1 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (1) $$

The matrix cells 1,4 and 2,4 represent the position equations of point "E" for the "x" and "y" direction respectively, both in relation to the base reference frame and will be represented by "p_x" and "p_y", respectively.

The final expressions of the position equation require some considerations. The angle θ3 is dependent on θ1 and θ2, as presented on (2).

$$ \theta 3 = \pi - 2\cdot\theta 1 - \theta 2 \qquad (2) $$

An expansion based on a cosine sum is applied to the "p_x" equation as well as an expansion from a sine sum can be applied to the "p_y" equation. These expansions are presented on equations ((3) and (4).

$$ C_{12} = \cos(\theta 1)\cdot\cos(\theta 2) - \sin(\theta 1)\cdot\sin(\theta 2) \qquad (3) $$

$$ S_{12} = \sin(\theta 1)\cdot\cos(\theta 2) + \cos(\theta 1)\cdot\sin(\theta 2) \qquad (4) $$

Analyzing the triangle formed from the points X2, X3 and C, and applying the sine and cosine laws to the θ2 angle, the equations (5) and (6) are obtained.

$$v_y(t) = \frac{0.22.\sqrt{2}.d1(t)}{\sqrt{-2.25+4.(d1(t))^2}} + \frac{0.11.\sqrt{2}.\left(2.25-4(d1(t)^2\right).(d1(t))}{\sqrt{-2.25+4.(d1(t))^2)^3}}$$

$$\sin(\theta 2) = \frac{bf2}{d1} \cdot \sin(2 \cdot \theta 1) \qquad (5)$$

$$\cos(\theta 2) = \frac{bf1^2 + d1^2 - bf2^2}{2 \cdot d1 \cdot bf1} \qquad (6)$$

Given the fact the point "E" is always aligned vertically with the base reference frame and the mechanical system is vertically symmetric, the equation of "p_x" is always zero, regardless of the hydraulic cylinder's length. Therefore, the angle θ1 has a valid domain characterized by the solution of the equation px=0 and presented on equation (7).

Where:

$$\theta 1 = \arctan\left(\frac{-lb - 2.af2 + \frac{bf1^2 + d1^2 - bf2^2}{bf1} + 2.af1}{d1 \cdot \sqrt{4 - \frac{\left(bf1^2 + d1^2 - bf2^2\right)^2}{bf1}}}\right) \qquad (7)$$

$$bf1 = lb - af1 \qquad (8)$$

$$bf2 = \frac{lb}{2} - af2 \qquad (9)$$

The equations that compute the point "E" position from the base reference frame, obtained via D-H algorithm and based on the previous assumptions, are presented on equations (10) and (11).

$$px = 0 \qquad (10)$$

$$py = 1/2 \cdot \left(\sqrt{2}\left(lb^2 4.lb.af1 + 4.lb.af2 - 8.af1.af2 + 4.af1^2 - 4.d1^2 + 4.af2^2\right) \cdot lb\right) / \left(\left(-lb^2 + 4.lb.af1 + 4.lb.af2 + 8.af2.af1 - 4.af2^2 - 4.af1^2\right)^{1/2} \cdot \sqrt{(2.af2 - lb)} \cdot \sqrt{(-lb + af1)}\right) \qquad (11)$$

Validation of position equations

To validate the generic equations, the scissor lift's dimensions presented on Figure 1 are needed. Table 2 presents the dimensions of the proposed scissor lift: the bar length (lb), distance from eye that fix the cylinder body to the point "A" (af1) and the distance from cylinder rod-end eye to center of the bar C-E.

The kinematics equation validations were done by the comparison of the "p_y" position, obtained from the analytical model, with the vertical height measured on the scissor lift 2D model. The results presented in Figure 3 show the position equation returns a good accuracy.

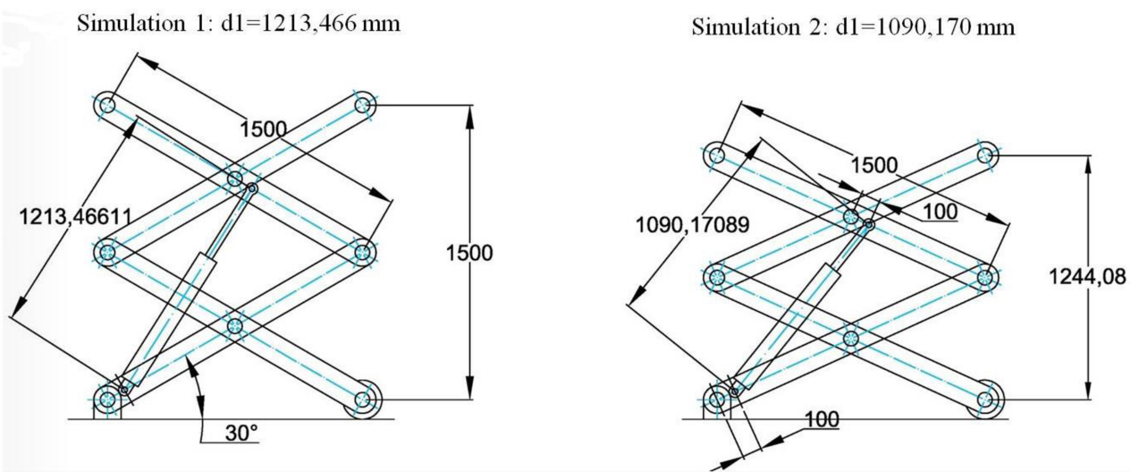

Figure 3: Results comparison between the 2d model and the analytical model (height measurement on top).

Table 2: Dimensions of the scissor lift.

lb	af1	af2
[m]	[m]	[m]
1,5	0,1	0,1

Speed equations by D-H algorithm

The D-H algorithm allows obtaining the generic speed equations on the direct kinematics through the product of the Jacobian Matrix with the system variables. The Jacobian matrix is obtained by the position equations partial derivatives, in order to each system variable and according with the type of joint: prismatic or rotational [9] and presented in equation (12).

$$J=\begin{bmatrix} -\left(\frac{lb}{2}+af2\right)\cdot S_{123}-d1\cdot S_{12}-af1\cdot S_1 & C_{12} & -\left(\frac{lb}{2}+af2\right)*S_{123} \\ \left(\frac{lb}{2}+af2\right)\cdot C_{123}+d1\cdot C_{12}-af1\cdot C_1 & S_{12} & \left(\frac{lb}{2}+af2\right)*C_{123} \\ 0 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & 0 \\ 1 & 0 & 1 \end{bmatrix} \quad (12)$$

The scissor lift linear and rotational speed of point "E" in relation to the base frame, are described by the equations (13) - (18).

$$v_x\left[(lb/2+af2)\cdot S_{123}-d1\cdot S_{12}-af1\cdot S_1\right]\cdot\omega1+v2\cdot C_{12}+\left[-(lb/2+af2)*S_{123}\right]\cdot\omega3 \quad (13)$$

$$v_x\left[\left(\frac{lb}{2}+af2\right)\cdot C_{123}-d1\cdot C_{12}-af1\cdot C_1\right]\cdot\omega1+v2\cdot S_{12}+\left[-\left(\frac{lb}{2}+af2\right)*C_{123}\right]\cdot\omega3 \quad (14)$$

$$vz=0 \quad (15)$$

$$\omega x=0 \quad (16)$$

$$\omega y=0 \quad (17)$$

$$\omega z=\omega1+\omega3 \quad (18)$$

The equations (13)-(18) allow to compute the "v_x" and "v_y" speed, since the model is a fully actuated system, i.e., has actuators applied in all joints. However, those axes are not actuated, and their motion is dependent on the hydraulic cylinder linear motion.

Applying the solution of θ1 for the "p_x" motion restriction and a cosine law over the angles of θ2 and θ3, and assuming the cylinder has a uniform motion, neglecting the acceleration, a solution for the equations of "v_x" and "v_y" can be reached. Due to the size of the mathematical solutions, the equations are presented without the θ1 + θ2 angle's solution in (19) and (20). Equation (21) is the hydraulic cylinder motion equation.

$$v_x=\frac{0,055.\left(-0,95.S_1-d1(t).S_{12}\right).d1(t)}{\sqrt{4-1.21\left(-\left(d1(t)\right)^2+2.38\right)^2}}+0.05.C_{12}-\frac{1.7.S_1\left(-0.07-\frac{0.04.\left(1.54-d1(t)^2\right)^2}{\left(d1(t)\right)^2}\right)}{\sqrt{4-\frac{2.36\left(1.54\left(d1(t)\right)^2\right)^2}{\left(d1(t)\right)^2}}} \quad (19)$$

$$v_y=\frac{0,055.\left(-0,75.C_1-d1(t).C_{12}\right).d1(t)}{\sqrt{4-1.21\left(-\left(d1(t)\right)^2+2.38\right)^2}}+0.05.S_{12}-\frac{1.7.C_1\left(-0.07-\frac{0.04.\left(1.54-d1(t)^2\right)^2}{\left(d1(t)\right)^2}\right)}{\sqrt{4-\frac{2.36\left(1.54\left(d1(t)\right)^2\right)^2}{\left(d1(t)\right)^2}}} \quad (20)$$

$$d1(t)=0,791+0,05.t \quad (21)$$

Speed equations by position derivative

To compare the D-H algorithm solutions with other methodologies, the velocity equations were obtained from the derivative of the position equation with respect to time. To use this method, it is assumed the cylinder moves uniformly on time, without acceleration, as written on (21). To simplify the reading, the equations (22) and (23) were written considering the values presented on Table 1.

$$v_x(t)=\frac{dpx}{dt}=0 \quad (22)$$

$$v_y(t)=\frac{0.22.\sqrt{2}.d1(t)}{\sqrt{-2.25+4.(d1(t))^2}}+\frac{0.11.\sqrt{2}.\left(2.25-4(d1(t)^2\right).(d1(t))}{\sqrt{-2.25+4.(d1(t))^2)^3}} \quad (23)$$

Validat

The results from both methods, D-H algorithm and derivative, were plotted and are presented from Figure 4 to Figure 7 and are compared by the results obtained on the 2D simulation model done in Working Model. The results obtained from this simulator are used as a reference due to the ability of this simulator to model the scissor lift and get the position, speed, and acceleration of a point or even an object in a time scale.

Figure 4 shows the horizontal velocity vxobtained from the D-H algorithm, presented in equation (19) and Figure 5 shows the results calculated by the derivative of the horizontal position, presented in equation (22). Looking at Figure 4, it is clearly noticeable that the horizontal velocity obtained via D-H algorithm did not produce a consistent result with the simulator, even applying the horizontal constrain motion to the point "E".

A similar approach was performed to compare both methods for the vertical velocity v_y. Analyzing the Figure 7 it can be observed that the derivative method presents results near to the results obtained from the lift model rather than the D-H Algorithm presented on Figure 6, which presents values slightly higher, up to 0,02 m/s, than the simulation results.

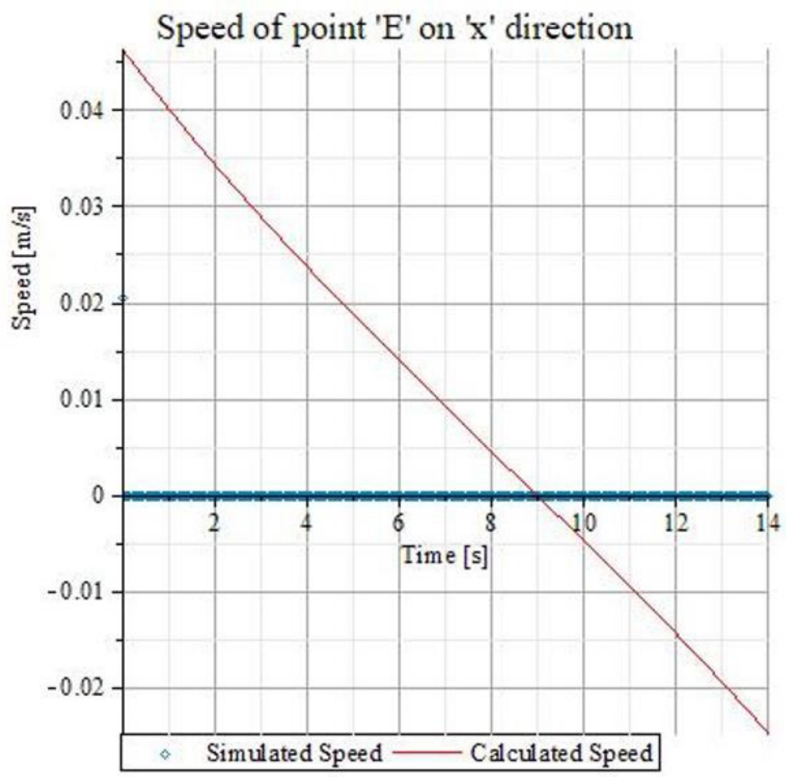

Figure 4: Comparison between simulated and calculated horizontal speed by d-h algorithm.

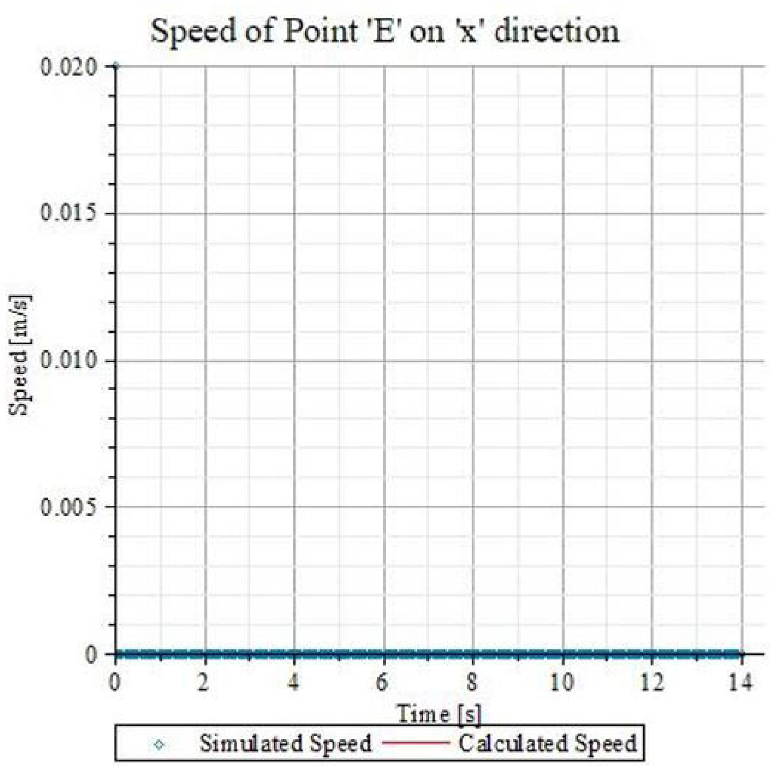

Figure 5: Comparison between simulated and calculated horizontal speed by derivative of position.

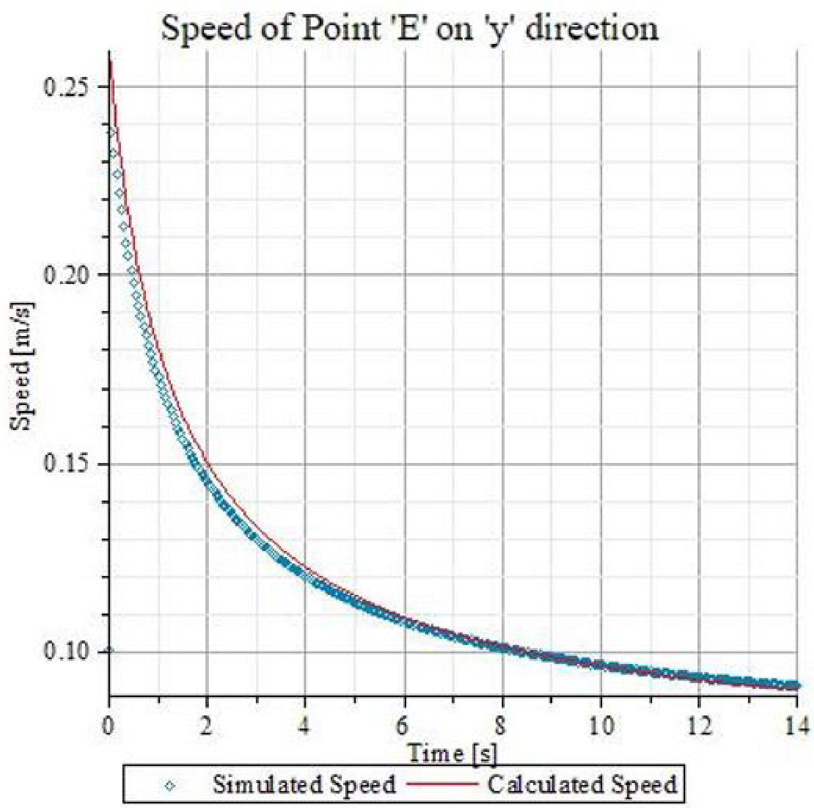

Figure 6: Comparison between simulated and calculated vertical speed by D-H algorithm.

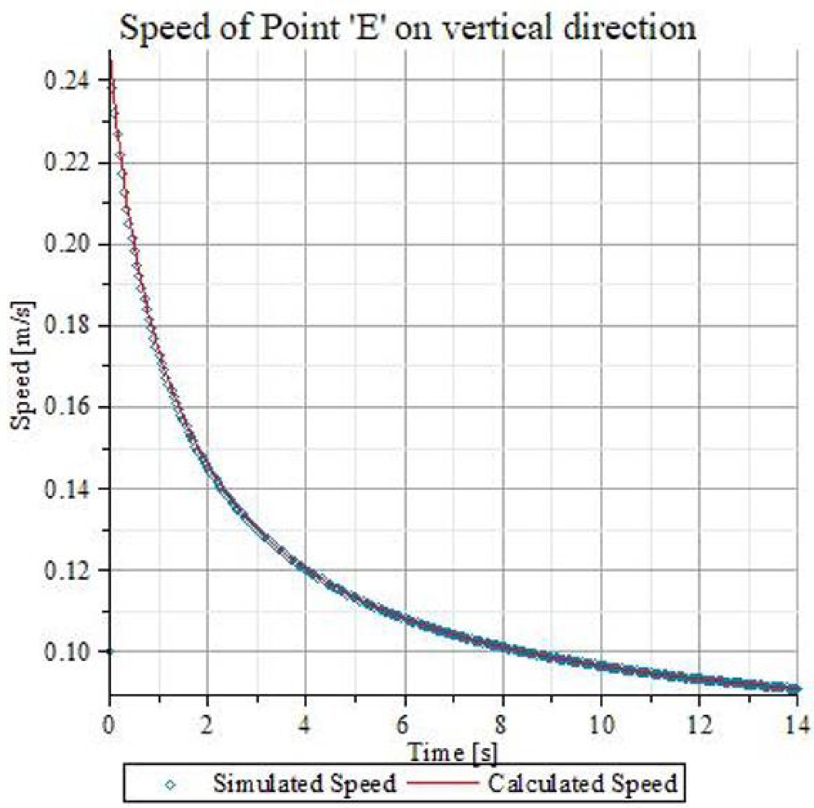

Figure 7: Comparison between simulated and calculated vertical speed by derivative of position.

Inverse kinematics

The inverse kinematics allows to obtain an equation that computes the length of the hydraulic cylinder “d1” for a certain known state of the vertical position “p_y” (Figure 8). This work follows the approach presented in reference [10].

The analysis of the diagram depicted in Figure 8 leads to the application of a cosine law to the angle (2.θ1+θ2) in the triangle X1-X2-X3, as presented in equation (24) and a cosine law to the angle (π/2 - θ1), as presented in equation (25).

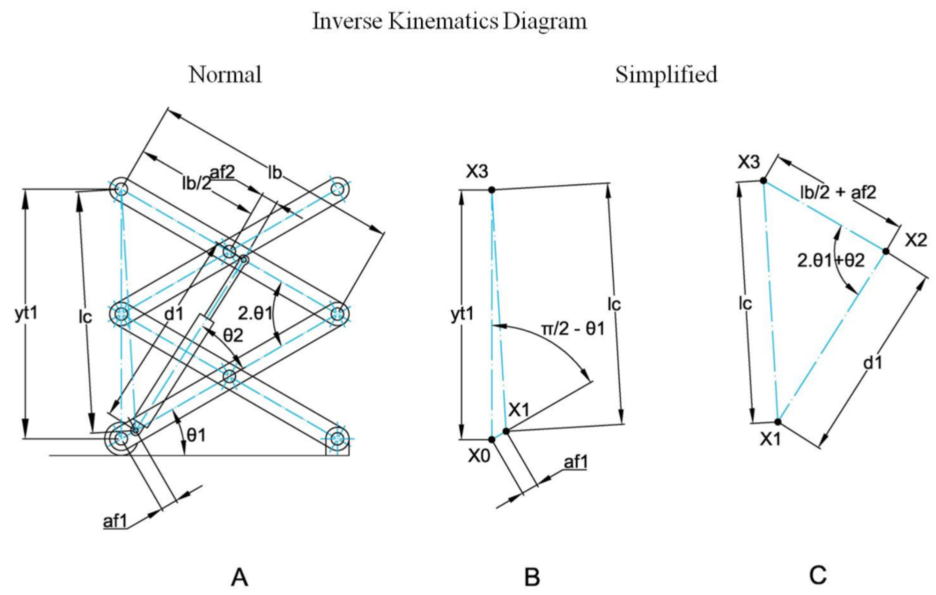

Figure 8: Normal (a) and simplified (b and c) diagram for inverse kinematics.

$$lc^2 = d1^2 + \left(\frac{lb}{2} + af2\right)^2 - 2.d1.\left(\frac{lb}{2} + af2\right)\cdot\cos(2\theta1 - \theta2) \quad (24)$$

$$lc^2 = af1^2 + yt1^2 - 2.af1.yt1\cdot\cos\left(\frac{\pi}{2} + \theta1\right) \quad (25)$$

Equaling both equations to the variable “lc²”, a generic solution for the inverse kinematics is obtained, as following:

$$d1^2 + \left(\frac{lb}{2} + af2\right)^2 - 2.d1.\left(\frac{lb}{2} + af2\right)\cdot\cos(2\theta1 - \theta2) = \mathrm{af1^2 yt1^2\text{-}2.af1.yt1.cos}\left(\frac{\pi}{2} - \theta1\right) \quad (26)$$

By applying trigonometric relations, we get the generic expression (27).

$$\begin{aligned} &+2\cdot d1^2 lb + 2lb\cdot bf2^2 + 4\cdot \mathrm{d}1^2\cdot af2 \\ &+4\cdot af2\cdot bf2^2 - 2\cdot lb\cdot bf1^2 \\ &-4\cdot af2\cdot bf1^2 + 4\cdot lb\cdot af2\bullet bf2) \end{aligned} = \frac{af1^2\cdot lb + yt1^2\cdot lb - af1\cdot yt1^2}{lb} \quad (27)$$

Solving the expression with relation to “d1”, we get two solutions: one positive and one negative. By applying the constants presented on Table 2, the inverse kinematics position equation is the positive solution presented on (28).

$$d1 = 0{,}11(1)\cdot\sqrt{2}\cdot\sqrt{22{,}781250 + 16{,}380\cdot yt1^2} \quad (28)$$

Validation of inverse kinematics

The inverse kinematics position equation was validated by comparing the results of two points with distinct vertical height “p_y”, with the same points measured from the CAD drawing presented on Table 3.

Table 3: Simulation points obtained in CAD drawing.

Point	yt1[m]	d1[m]
1	1,5	1,213
2	1,244	1,090

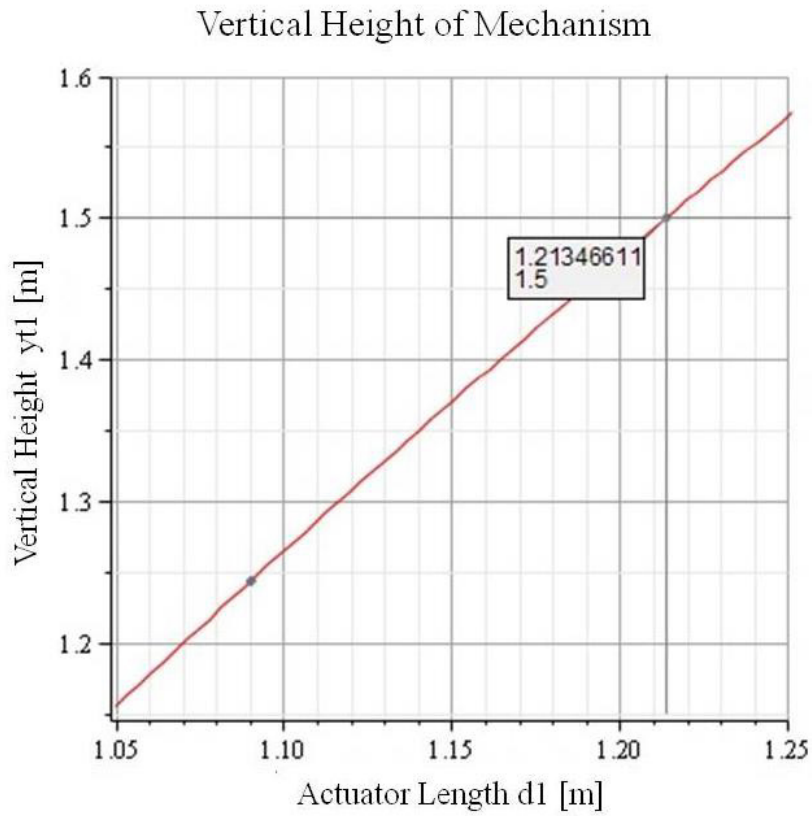

Figure 9: Simulation 1 of inverse kinematics for height of 1,5m.

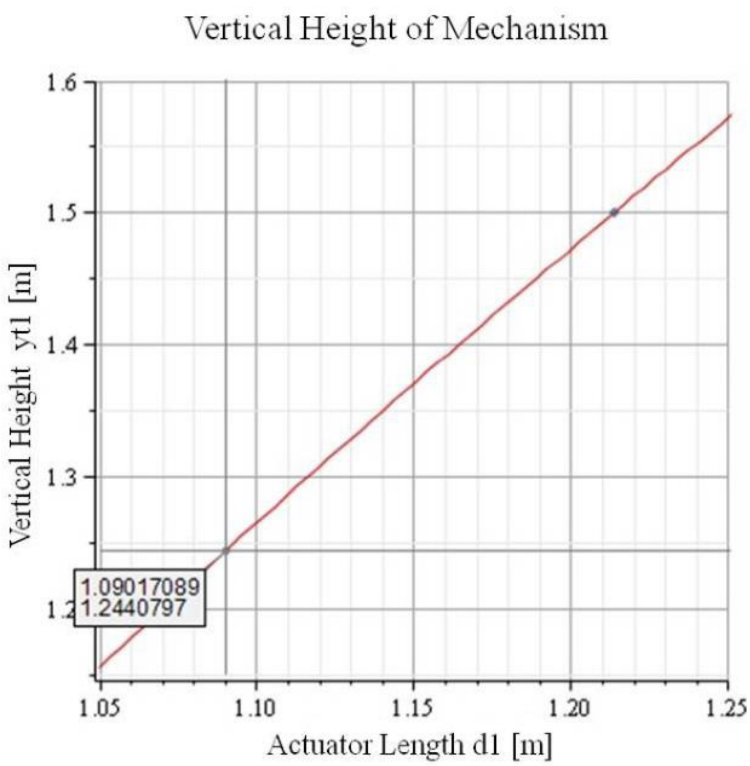

Figure 10: Simulation 2 of inverse kinematics for height of 1,244m.

Comparing the points obtained from the CAD drawing with the obtained from the graphs depicted in Figure 9 and Figure 10, it is possible to conclude that equation (28) presents accurate results.

Force equation obtained via D-H Algorithm

The D-H algorithm can be used to obtain the force of the robot arm actuators. This method was used to achieve to the equation that computes the force needed on the scissor lift's hydraulic cylinder [10].

To obtain the force and torque associated to the system's variables, the Jacobian Transpose must be multiplied by the external force vector "F" applied on the end-effector, that is, the vertical force "FE_y" applied at point "E", presented in equation (29).

$$\tau = {}^{0}J^{T} \cdot {}^{0}F \qquad (29)$$

The result is presented in equation (30), where T1 and T3 represent the torques on systems variables θ1 and θ3, and F1 represents the force on the hydraulic cylinder.

$$\tau = \begin{bmatrix} T1 \\ F1 \\ T2 \end{bmatrix}$$

$$= \begin{bmatrix} -\left(-\left(\frac{lb}{2} + af2\right) \cdot C_1 + d1 \cdot C_{12} + af1 \cdot C_1\right) \cdot F_{EY} - S_{12} \cdot F_{EY} \\ \left(\frac{lb}{2} + af2\right) \cdot C_1 \cdot F_{EY} \end{bmatrix} \qquad (30)$$

Although this solution could be valid for a serial type of manipulator robot, the result is not suitable for a scissor lift. It was possible to obtain mathematical expressions for the torques T1 and T3, these joints are free and don't have an actuator on them. Also, its movement relies on the movement of the hydraulic cylinder. So, this method was discarded because it was not possible to find any additional trigonometric or mathematical method that allows obtaining a valid equation for the force of the hydraulic cylinder.

Force equation obtained via energy conservation

An alternative method was chosen to calculate the hydraulic cylinder force based on the Principle of the Conservation of Energy. If the friction forces are neglected, the amount of energy that a system receives and sends is equal. Therefore, this principle can be applied to the scissor lift mechanism where the hydraulic cylinder receives energy from a hydraulic pump and converts it to mechanical energy with the mechanism to lift the load. The presented method was thoroughly developed by Spackman [11].

Considering the scissor lift presented on Figure 11, as a block (system) with a constant and evenly distributed mass, the total potential energy can be calculated by the equation (31).

$$E = \lim_{\Delta m_i \to 0} \sum g \cdot \Delta m_i \cdot y_i \qquad (31)$$

Since the length and width of the block are constant, the mass variation is given by the equation (32).

$$\Delta m_i = \rho \cdot g \cdot u \cdot w \cdot \Delta y_i \qquad (32)$$

Considering that the weight is evenly distributed, the work done per height unit is presented in equation (33)

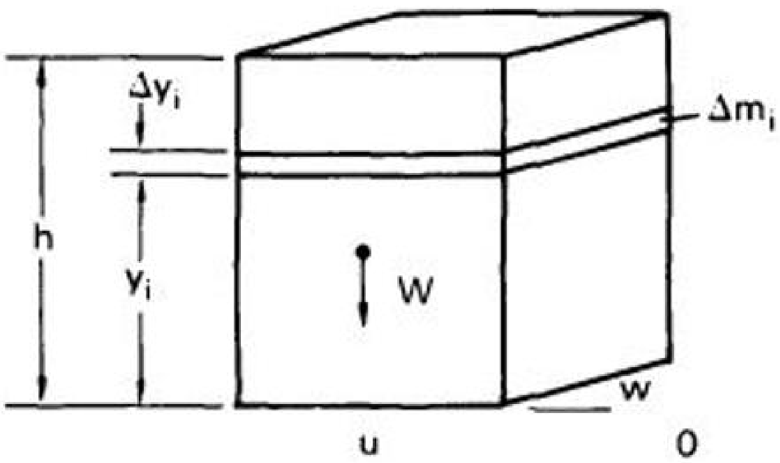

Figure 11: Simplified block model proposed by spackman [11].

$$\rho \cdot g \cdot u \cdot w = \frac{W}{h} \quad (33)$$

The potential energy is calculated as follows.

$$E = \frac{W}{h} \cdot \int_0^h y \cdot dy = \frac{W \cdot h}{2} \quad (34)$$

Again, considering the block mass keeps constant, varying only the vertical height, the work done between two different heights is presented below.

$$Wa = \frac{W \cdot (h_2 - h_1)}{2} \quad (35)$$

Figure 12 presents a generic model for a scissor lift with four scissors. The force vectors "H_{xo} " and "H_{y0}" represent external forces applied on the scissor lift on a horizontal and vertical direction, respectively. The force vectors "B_x" and "B_y" represents the tare mass of the scissor lift in the horizontal and vertical directions, respectively.

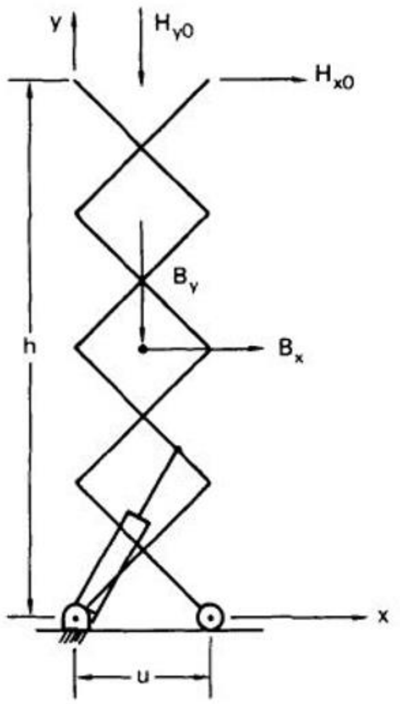

Figure 12: Model of a generic scissor lift from spackman [11].

Analyzing the elevation movement and applying the Principle of the Conservation of Energy, the generic relation can be obtained by the equation (36).

$$-\left(H_{y0} + \frac{B_y}{2}\right) \cdot (h - h_0) + \left(H_{x0} + \frac{B_x}{2}\right) \cdot (u - u_0) + \int_{l_0}^{l} F \cdot dr = 0 \quad (36)$$

Integrating the equation (36) with relation to the length of the oil-hydraulic cylinder, it is obtained the expression of the cylinder's force as following.

$$-\left(H_{y0} + \frac{B_y}{2}\right) \cdot \frac{dh}{dl} + \left(H_{x0} + \frac{B_x}{2}\right) \cdot \frac{dh}{dl} + F = 0 \quad (37)$$

The dh/dl derivative represents the height variation with the cylinder's length variation, i.e., the derivative of the vertical position "p_y" to the oil-hydraulic cylinder length "d1". The du/dl relation can be obtained via a trigonometric relation between the length of the arm "d", the bottom width "u" and the vertical height "h", as presented as following.

$$\frac{du}{dl} = -\frac{\tan(\theta)}{n} \cdot \frac{dh}{dl} \quad (38)$$

Applying this theory set, the generic expression for the oil-hydraulic cylinder's force of the scissor lift is presented as following:

$$F = \left(H_{y0} + \frac{B_y}{2}\right) \cdot \frac{dpy}{dd1} \quad (39)$$

The equation to compute the cylinder force is presented on (40), where is considered that the system dimensions presented on Table 2; the scissor lift has 2000 N of mass and an applied vertical force of 5000 N; the cylinder has an uniform motion without acceleration; it is assumed that the minimum length is 801 mm, instead of the 791 mm, due to difficulty in adjusting minimum length of the cylinder on the 2D simulator.

$$F = \frac{3335{,}622 \cdot \sqrt{8} \cdot (0{,}801 + 0{,}05 \cdot t)}{\sqrt{(0{,}051 + 0{,}05 \cdot t) \cdot (1{,}551 + 0{,}05 \cdot t)}} \quad (40)$$

Validation of the oil-hydraulic cylinder force equations

As mentioned before, the D-H Algorithm for obtaining the force equation was discarded. The equation obtained via Principle of the Conservation of Energy will be validated by comparing the oil-hydraulic cylinder force results returned by the simulator with those obtained from the equation (40).

Figure 13 presents the comparison between the simulated results and the mathematical equation of the cylinder force. Although the results are very similar, the force results obtained from the 2D simulator are slightly lower than the analytical results, with an error varying from 253 N when t=12 s to 612 N when t=0 s This happened due to difficulties in modeling the cylinder in the 2D simulator, especially when defining the minimum length in the closed position.

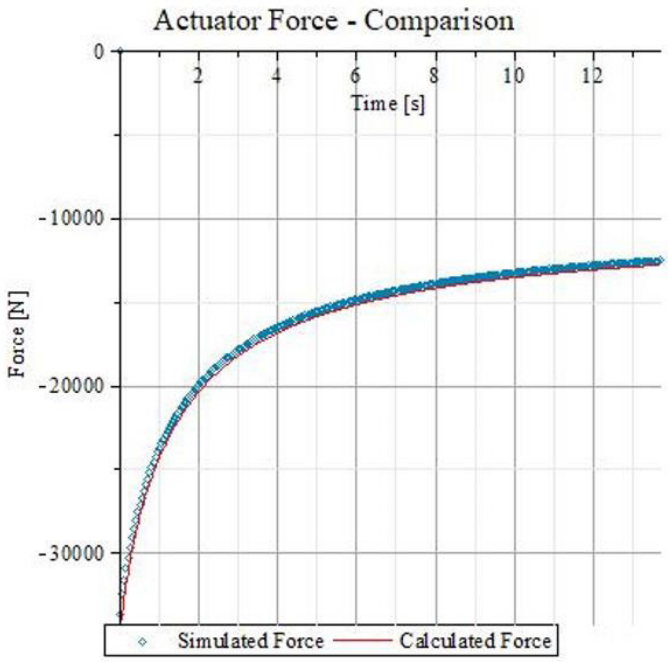

Figure 13: Comparison graph between simulation and analytical results of force equation.

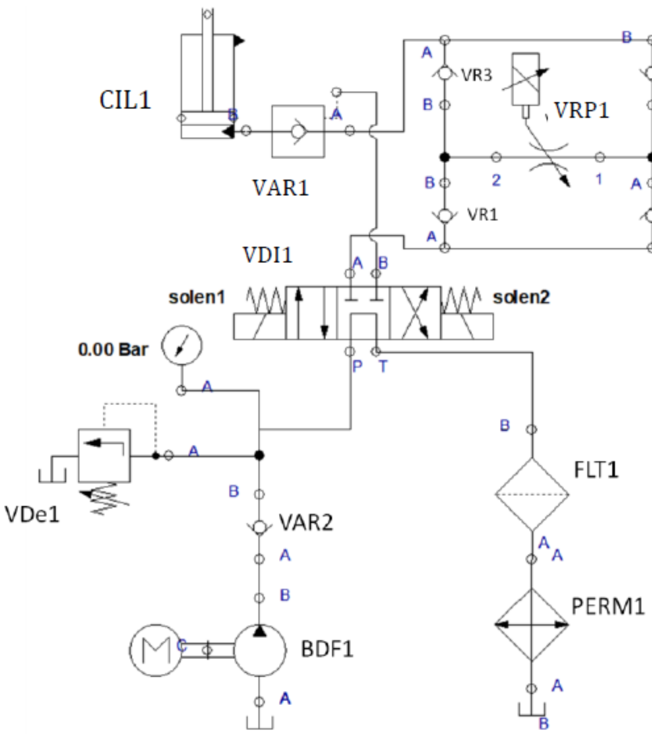

Figure 14: Oil-hydraulic circuit.

Therefore, the mathematical model presented in equation (40) is valid to obtain the oil-hydraulic cylinder force.

Hydraulic Circuit and Main Components

Hydraulic circuit

The scissor lift is actuated by an oil-hydraulic cylinder, which converts the hydraulic energy into an applied force to the structure. To control the oil-hydraulic cylinder it is projected the hydraulic circuit depicted in Figure 14.

The concept of the circuit is to control and adjust the cylinder speed to a certain value, by the adjustment done to the oil flow that enters or exits the cylinder's chamber, depending on the cylinder length.

To control the scissor lift motion, the cylinder (CIL1) is a double effect oil-hydraulic cylinder, mostly based on ISO 6020-2 specifications, and with a magnetostrictive position transducer to measure the stroke position of the cylinder. Because the scissor lift can do the lowering motion of the platform just with the gravity effect, the cylinder works as a single effect on the positive chamber, and the negative chamber will remain vented with a special filter.

To stop the cylinder motion in a certain instant in time, a check valve (VAR1) retains the oil on the positive chamber of the cylinder.

The cylinder speed is controlled by a proportional flow control valve (VRP1) with a sandwich rectifier plate, controlling the oil flow in the positive cylinder chamber in both raising and lowering motion. This valve allows a maximum flow of 18 L/min and it is controlled by a dedicated signal conditioner.

The directional valve (VDI1) 4/3 has two 8 W solenoids, one on each side, with automatic centering in case of power failure.

All components except the cylinder were selected from Parker Hannifin©, for the sake of compatibility.

Oil-hydraulic cylinder

The calculations presented on the section of Kinematic Equations, were done to determine the valid mathematical models for position, speed, and force of the oil-hydraulic cylinder. On this section, the oil-hydraulic cylinder will be dimensioned, having in mind the cylinders minimum and maximum length, to develop a feasible cylinder from the manufacturing point of view. To assist the dimensioning, the cylinder is modeled in a 3D CAD software along with the magnetostrictive transducer.

The dimensions "af1" and "af2" (Figure 2) play a key role on dimensioning the hydraulic cylinder and in its orientation with relation to the scissor lift's structure, influencing the maximum force applied by the cylinder.

The optimum values of "af1" and "af2" are calculated by retrieving dimensions from the 3D model with the cylinder in the retracted position, from the Figure 15.

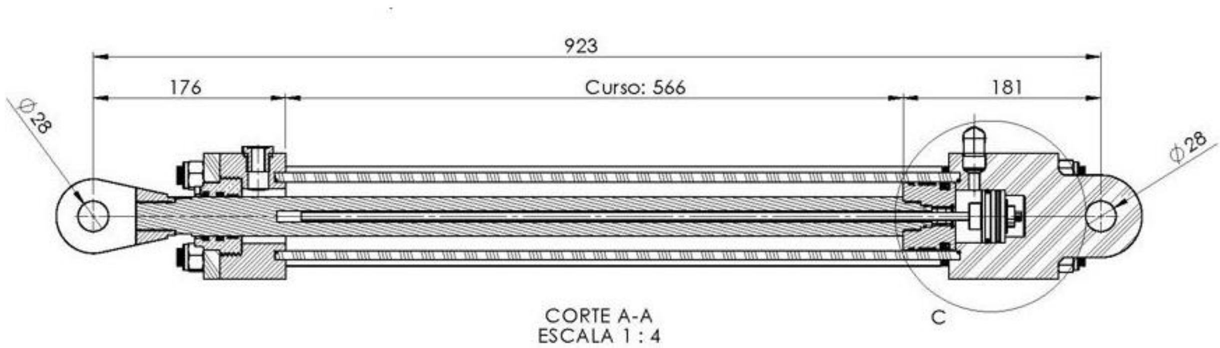

Figure 15: Cross section view of the hydraulic cylinder.

A Cosine Law was applied on the angle 2.θ1 and it is presented on equation (41). To apply this equation the minimum and maximum limits of the angle θ1, re to be considered on the calculation, which are limited by the vertical height of the scissor lift, as presented on equations (42) and (43).

$$d1=\sqrt{\left(\frac{lb}{2}-af2\right)^2+\left(lb-af1\right)^2-2*\left(lb-af1\right)\left(\frac{lb}{2}-af2\right)\cdot\cos(2\theta 1)} \qquad (41)$$

$$\theta 1_{\min}=\arcsin\left(\frac{yt1_{\min}}{2\cdot lb}\right) \qquad (42)$$

$$\theta 1_{\max}=\arcsin\left(\frac{yt1_{\max}}{2\cdot lb}\right) \qquad (43)$$

The constant "k1" is defined as the sum of the dimensions 176 mm (on the left) and 181 mm (on the right). Also, "k1" results in the difference between the minimum length and the stroke, as presented on equation (44). Because the stroke results in the difference between the maximum and minimum length of the cylinder, the final "k1" equation is presented on equation (45).

$$k1=d1_{\min}-stroke=357mm \qquad (44)$$

$$k1=2*d1_{\min}-d1_{\max} \qquad (45)$$

The equation of "k1" with "af1" and "af2" as variables, results in the inclined plane presented on Figure 16.

To narrow the solutions, a plane defined at k1=0, 357 m is defined. The intersection between the two planes results in a linear equation as presented on Figure 17. This graph presents the ideal pair of values for "af1" and "af2", respectively.

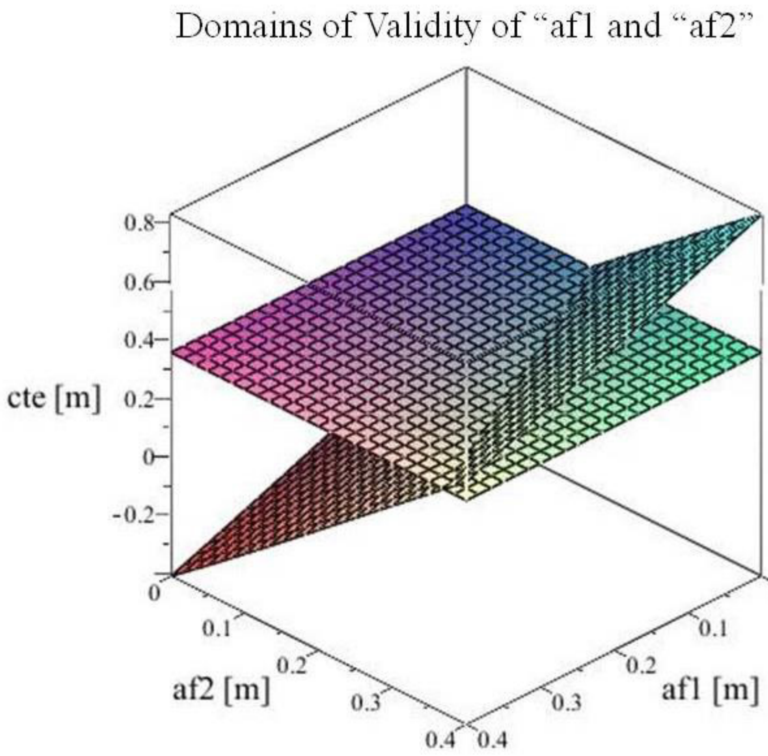

Figure 16: 3d graph for determining the linear domain of solutions for af1 and af2.

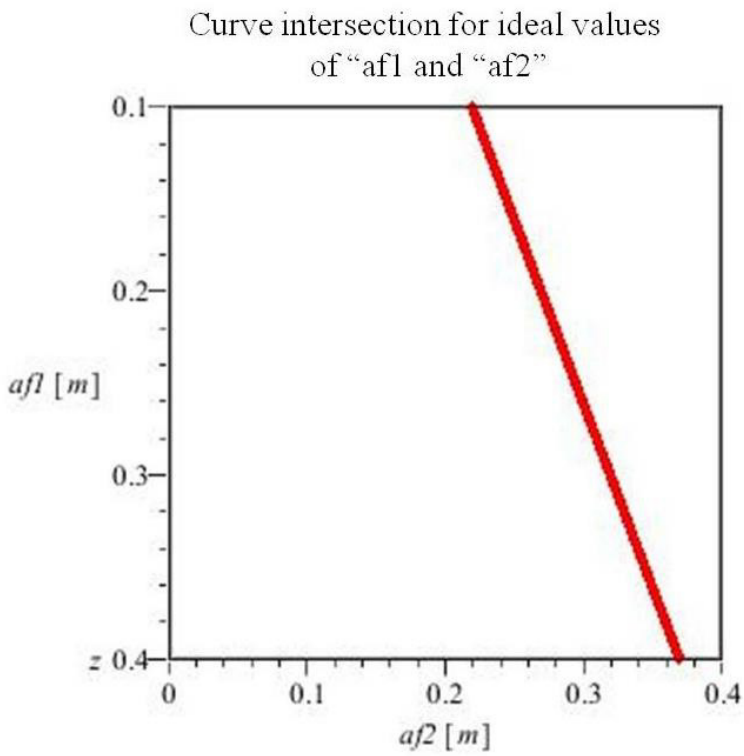

Figure 17: Plane intersection for ideal values of "AF1" and "AF2".

The force that the cylinder needs to apply in the scissor lift is influenced by these dimensions. By using the force equation (40), but with "af1" and "af2" as variables, a set of 3 mathematical simulations were done with it, obtaining the respective force values presented on Table 4. The values chosen for "af1" and "af2" were 0,10 m and 0,22 m, corresponding to F_{max} = 47750 N and F_{min} = 15291 N, that are required when the cylinder is full retract and full extended, respectively. These force values were the smallest values obtained in the simulations.

Table 4: Simulation results for the cylinder dimensions and force.

Simulation	1	2	3
af1 [m]	0,10	0,14	0,22
af2 [m]	0,22	0,24	0,28
$d1_{min}$ [m]	0,900	0,878	0,835
$d1_{max}$ [m]	1,440	1,398	1,313
δd1 [m]	0,540	0,520	0,478
$d1_{max}$ / $d1_{min}$	1,60	1,59	1,57
F_{act} ($d1_{min}$) [N]	47750	49873	54715
F_{act} ($d1_{max}$) [N]	15291	15877	17195

With these values for "af1" and "af2", the new position, velocity and force equations were written, based on the algebraic form of each equation. Each equation is presented from (46) to (49).

$$d1(t) = 0{,}111 \cdot \sqrt{2} \cdot \sqrt{30{,}654 + 13{,}356 \cdot yt1^2} \qquad (46)$$

$$yt1 = 0{,}4 + 0{,}15t \qquad (47)$$

$$v1(t) = \frac{dd1}{dt} = \frac{0{,}055 \cdot \sqrt{2} \cdot (1.603 + 0{,}601 \cdot t)}{\sqrt{34{,}654 + 13{,}356 (0{,}4 + 0{,}15t)^2}} \qquad (48)$$

$$F_{act}(t) = -\frac{1915{,}401 \cdot \sqrt{30{,}654 + 1{,}603t + 0.301t^2}}{\sqrt{0{,}053 + 0{,}040t + 0{,}0074t^2}} \qquad (49)$$

The oil-hydraulic cylinder was dimensioned to buckling as per standard recommendations and the Ø36 mm rod was selected.

Table 5: Pressure and oil flow results on the cylinder.

Nominal Pressure [bar]	Cylinder Diameter [mm]		
	50	63	80
	Hydraulic Cylinder Force [N]		
100	17609,3	27917,3	44975,6
110	19374,8	30719,0	49492,2
120	21141,9	33524,5	54016,1
170	29973,4	47542,7	76617,6
180	31739,7	50346,3	81137,8
240	42326,4	67143,5	108212,7
250	44085,0	69930,2	112700,5
Speed [m/s]	**Oil Flow [L/min]**		
0,022	2,59	4,11	6,63
0,068	8,01	12,72	20,50

The cylinder inner diameter was selected from 3 standard values for the selected rod size. It required calculating the force the cylinder does for each pressure value and for each of the standardized inner diameters. Table 5 presents the force values as well as the corresponding oil flow values. The selected inner diameter is 63 mm because requires 170 bar of pressure at maximum force, which is a value below the standard 200 bar hydraulic pumps available on the market have and it returns an oil flow compatible with the proportional flow control valve, i.e., a flow between 4,11 L/min and 12,72 L/min.

The cylinder thickness was calculated based on the theory of thick wall pressure cylinder and it was obtained a thickness of 8,5 mm, with a safety factor of 4,04.

Position transducer

The selected magnetostrictive transducer is a Temposonics model: MH-E. It is installed inside of the hydraulic cylinder, as shown in Figure 18, and has a wave guide which is installed in a hole on the cylinder rod. A simple magnet is installed on the piston [12].

When the cylinder stroke changes, the magnet changes its position in relation to the fixed wave guide, causing a deformation on the wave guide, as exemplified in Figure 19. It generates a signal on the sensor, in a form of current with a range from 4 to 20 mA, according with the rod position [12].

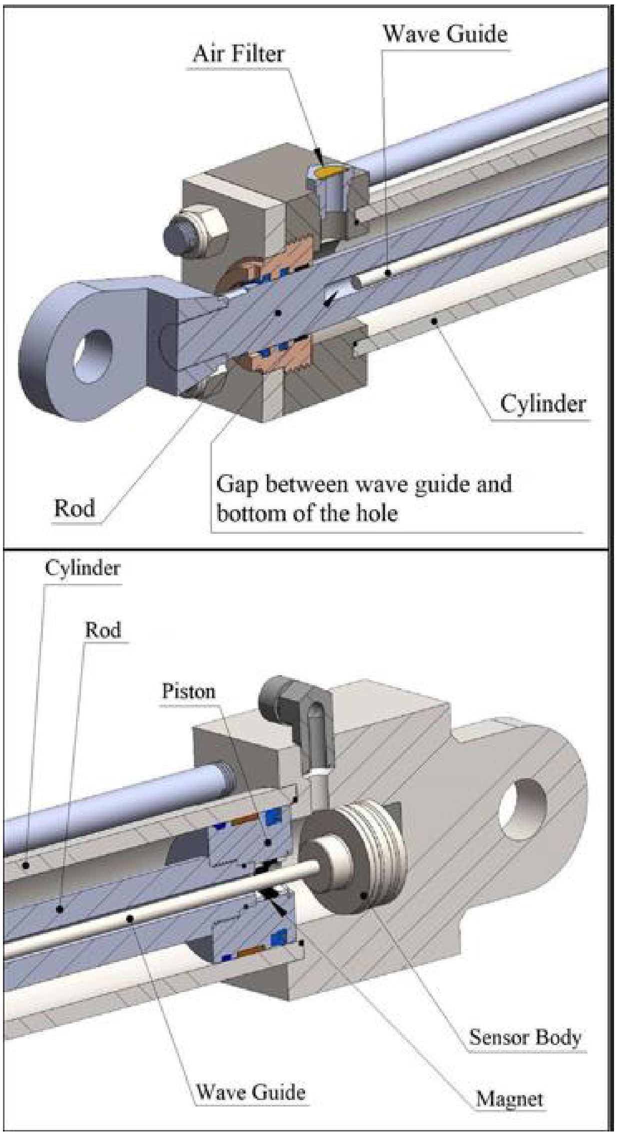

Figure 18: Detailed views on the cylinder.

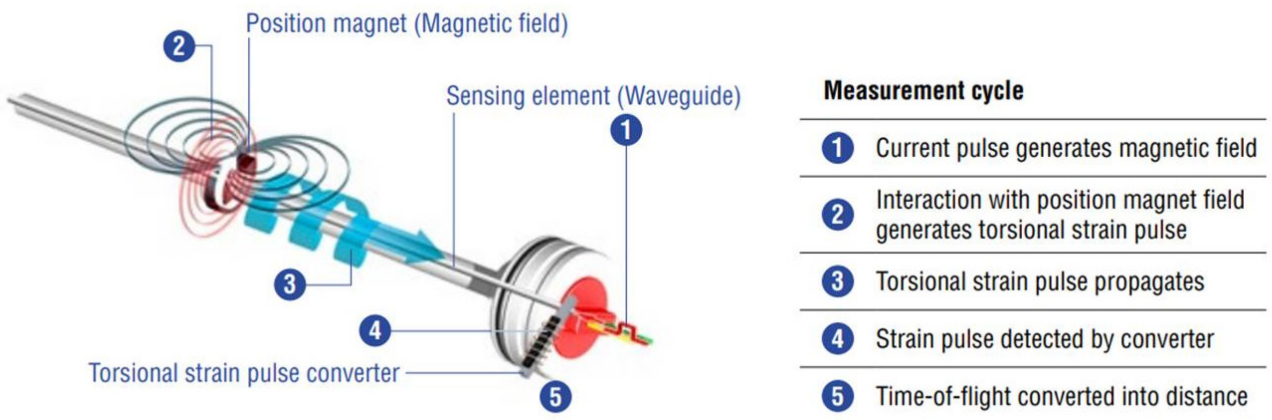

Figure 19: Work principle of the magnetostrictive sensor [12].

This sensor was selected due to the advantages of absolute positioning, not requiring calibration after first use and it's a compact solution without external components on the cylinder, minimizing the collision's risk with other components [12].

Proportional flow control valve

The cylinder speed namely the rod cylinder speed, is regulated by a proportional flow control valve, with a rectifier circuit, combining both meter in and meter out methods, through its opening rate. The proportional feature allows opening the valve to a specific value, within the nominal flow and limits specified on the valve characteristic curve.

It is selected the valve Parker DUR 18 L06 PK11, presented in Figure 20, with the NG06 size, which works with a maximum flow of 18 L/min. This valve was selected because the pressure variation does not influence the flow on the valve, due to a pressure compensation spool that keeps the pressure constant on the control hole. The pressure drop on this valve is around 3,01 bar for the maximum flow of 12,7 L/min [13].

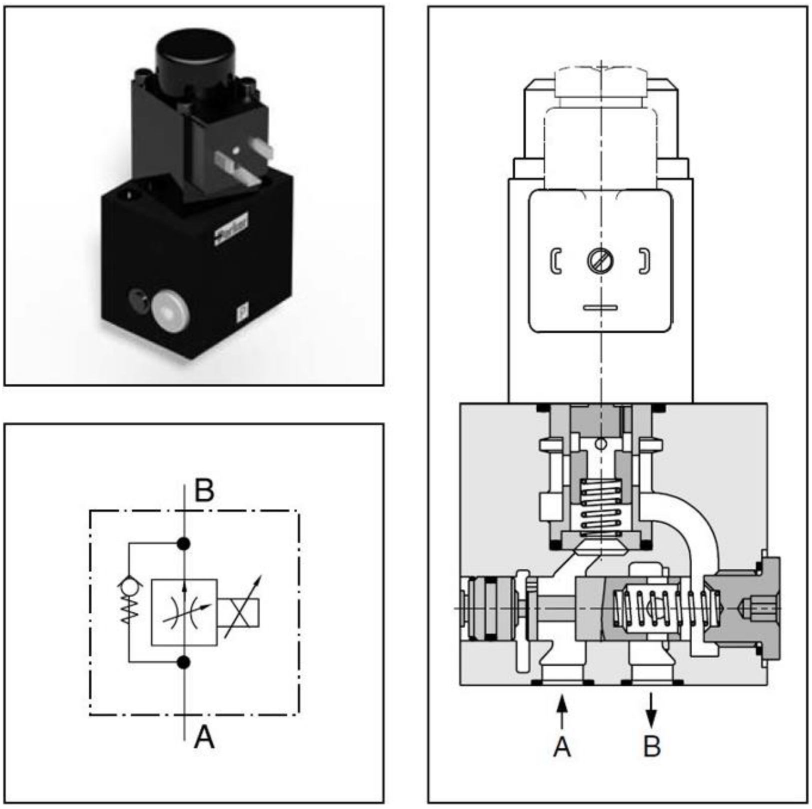

Figure 20: Flow control, proportional valve [13].

In addition, this valve is equipped with a sandwich rectifier plate that can be assembled directly onto it and enables the valve to control the oil flow incoming and outcoming from the oil-hydraulic cylinder [13].

This proportional valve requires a separate controller to regulate its input signal. The controller is a Parker PCD00A-400 and receives the output signal from a PLC and process it to the proportional valve signal range [14].

Supervision and Control System

The box transfer system with a scissor lift is controlled by a PLC, being responsible by the control of the oil-hydraulic cylinder movement and the motors conveyors.

The aim of the PLC and its programming is to control the motion of a conveyor platform with a scissor lift, to transfer boxes from three possible entry conveyors to the exit conveyor, as presented on Figure 1. In addition, it is required that the platform moves up and down with constant speed, through the speed and stroke control of an oil-hydraulic cylinder placed in the scissor lift structure.

It is considered that all process is controlled by a PLC and supervised using an HMI.

The box transfer system works under the following mode (Figure 1):

- The boxes arrive to the system through the 3 entry conveyors. Motors M1 to M3 are on when the system is on. They are off when the system is off, when it isn't detected a box through the sensors S1 to S3 or when a box is being transferred to the exit conveyor.
- Every time it is detected the presence of a box on one entry conveyor, its motor stops and the scissor lift move the conveyor platform to the respective level;
- Once the conveyor platform has reached to the desired conveyor, the motors of both conveyors are turned on to transfer the box. The entry conveyor motor continues to run, even after the box transference, stopping only when an eventual second box arrives to the corresponding sensor. The scissor lift conveyor motor stops after detecting the box in its sensor;
- The scissor lift moves, with a constant speed, to the level of the exit conveyor;
- Once the lift platform arrives to the exit conveyor, the motors of both conveyors, scissor lift and output conveyor, are turned on.
- The motor of the scissor lift conveyor stops after the box is transferred and detected by its sensor. The output conveyor motor also stops after the box has passed completely by its sensor.

This process repeats while the system is on.

PLC inputs and outputs

To control this system, a PLC Siemens S7-1214C was chosen, with an analog signal module SM 1234 and an additional signal board SB 1222, to get more digital output ports. This PLC was compared with a S7-1500 and it was chosen instead of this last one, because it is a more suitable solution for simple projects, it has a suitable resolution for this system and is a small investment when compared with a S7-1500, which has features more advanced and unnecessary to the purpose of this project.

According with the previous procedure, the boxes are detected by sensors placed in all conveyors. To perform this, it was selected a reflective type of photoelectric sensor, that is PNP sensor type.

The HMI is composed by a touch screen to supervise the system. Also, 3 buttons were added to activate the functions of start, stop, and restart the program. Table 6 presents the list of the PLC tags corresponding to the PLC inputs.

Table 6: List of the PLC tags - inputs (sensors and buttons).

Tag Name	Data type	Address	Equipment
sfe_ten_01	Bool	%I0.0	Photoelectric sensor upper entry conveyor
sfe_ten_02	Bool	%I0.1	Photoelectric sensor middle entry conveyor
sfe_ten_03	Bool	%I0.2	Photoelectric sensor lower entry conveyor
sfe_tre_01	Bool	%I0.3	Photoelectric sensor scissor lift
sfe_ts_01	Bool	%I0.4	Photoelectric sensor exit conveyor
start	Bool	%I0.5	Start Button
restart	Bool	%I0.6	Restart Button
stop_pause	Bool	%I0.7	Stop button

The electric motor of each conveyor and the hydraulic pump electric motor are controlled by its own contactor.

To inform the user about the state of movement of the scissor lift, two LED lights with red and green color were installed.

The control of the lifting and lowering motion of the oil-hydraulic cylinder is done by the solenoids of the directional valve. The proportional valve is controlled by its controller, which needs an additional port to activate it.

These components are all connected to the outputs of the PLC, as presented on Table 7.

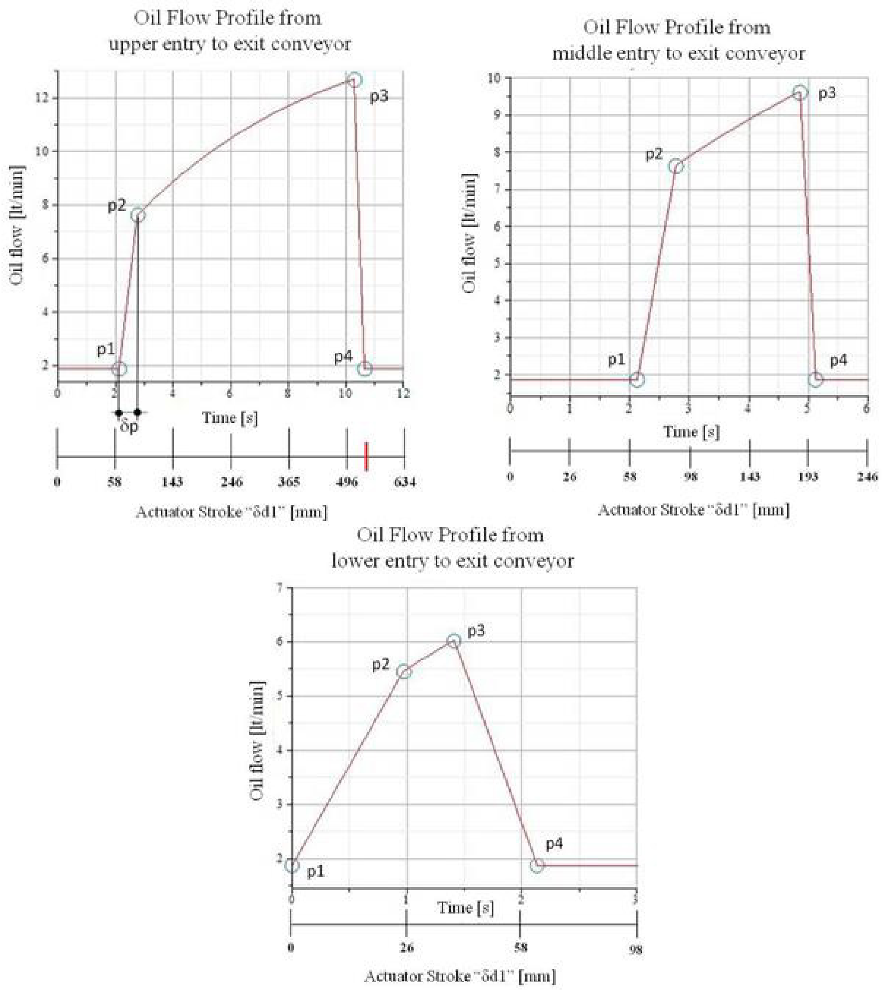

Figure 21: Oil flow profiles for each entry conveyor.

Table 7: List of the PLC tags – outputs (contactors, solenoids and lights).

Tag Name	Data type	Address	Equipment
ctm_ten_01	Bool	%Q0.0	Contactor Upper Entry Conveyor
ctm_ten_02	Bool	%Q0.1	Contactor Middle Entry Conveyor
ctm_ten_03	Bool	%Q0.2	Contactor Lower Entry Conveyor
ctm_tre_01	Bool	%Q0.3	Contactor Scissor Lift Conveyor
ctm_ts_01	Bool	%Q0.4	Contactor Exit Conveyor
ctm_bh_01	Bool	%Q0.5	Contactor Hydraulic Pump
luz_verde	Bool	%Q0.6	Green Light
luz_vermelho	Bool	%Q0.7	Red Light
vd_pb	Bool	%Q1.0	Directional Valve Solenoid "b" position
vd_pa	Bool	%Q1.1	Directional Valve Solenoid "a" position
ativar_vrcp	Bool	%Q4.0	Enable Proportional Flow Control Valve

The magnetostrictive sensor measures the position of the stroke on the oil-hydraulic cylinder and returns an analog signal in a range of 4 to 20 mA. This sensor is connected to an analog module of the PLC.

Also, the controller of the proportional flow control valve is connected to the analog module. This controller regulated the flow of the valve by receiving a tension on a range of 0 to 10 V. Table 8 presents the list of the PLC analog variables.

Table 8: List of the PLC tags - analog input and output.

Tag Name	Data Type	Address	Equipment
sme_tre_01	Integer	%IW96	Magnetostrictive Sensor
vrcp_01	Integer	%QW104	Signal Conditioner of Proportional Valve

Automatic control system

The scissor lift velocity is controlled by the PLC, through the oil flow regulation that goes on the proportional valve and, consecutively, to the oil-hydraulic cylinder. Since the inverse kinematic system is non-linear, a constant platform vertical velocity will correspond to a varying cylinder velocity. Thus, at each cylinder position corresponds to a different instantaneous velocity from the previous position one but, due to the scissor lift geometry effect, the platform will move with a constant vertical velocity.

The implemented control method consists in defining a velocity profile along the stroke of the oil-hydraulic cylinder by using the inverse kinematic velocity equation presented in equation (48).

Because the hydraulic components may have a different response time, the oil flow will be adjusted to be slow near the target values of stroke. This means, near the target heights of the scissor lift, so the system can have a suitable response time. This adjustment consists in creating a "ramp" from a linear equation inserted on the oil flow equations so the oil flow can drop, and consequently the oil-hydraulic cylinder velocity too. Equation (50) presents the oil flow ramp equation.

$$Qc_{ramp} = m \cdot \delta d1 + b \qquad (50)$$

This linear equation is defined by a slope variable "m" and an offset variable "b", which are calculated at the early beginning of the program run. The variable $\delta d1$ represents the current stroke of the cylinder. These profiles are specific for a pair of entry-exit conveyors, although the equation that defines the oil flow for constant vertical speed is the same.

The values outside of the target values (for entry and exit conveyor cylinder stroke), that is stroke below point "p1" and stroke above point "p4", have a flow that remains constant at its minimum predefined value, in this case 1,87 L/min. Figure 21 presents the three oil flow profiles.

Because the velocity and oil flow equations are written in order of the time variable, and the position transducer measures the position of the stroke, it was necessary to insert an equation that enables the calculation of the time for the specific measured stroke of the oil-hydraulic cylinder.

This was possible by inverting the position equation written in order to the time, which results from joining equation (46) with equation (47). Once known the time for that value of stroke, it is possible to get the velocity and therefore the oil flow.

PLC programming

The PLC programmed to execute the operation mode, described in Supervision and control system Section, was performed in Ladder language (LD), implementing the structured programming concept based on an Organizational Block (OB1), specific functions (FC's) and Function Blocks (FB's). Most of the processed data associated with the program, like the initial variables, as the height of the entry and exit conveyors, are stored on a Data Base (DB1).

Given the extension of the PLC programming, Figure 25 presents a programming scheme resume, that is described next.

On the first scan cycle of the program, the PLC calculates variables, like the stroke the oil-hydraulic cylinder needed to reach the height of each conveyor as well as the parameters "m" and "b" for the 3-oil flow profile functions and stores the results in DB1.

While the program is running, it is converted the measurement from the magnetostrictive transducer signal data in Binary Coded Decimal (BCD) to the corresponding decimal value in millimeters. It also converts the digital value of oil flow to send to the proportional valve in BCD format. For each case, a specific FC function is used to perform the calculations.

The OB1 block controls the conveyor's motors and the hydraulic pump through Set/Reset Instructions. It has 3 modes: start, restart and stop. On activating the start mode, the program starts, and all entry conveyor's motors start, until a box is detected. Also, the hydraulic pump starts. The stop mode pauses the program execution and the conveyor's motors stop but the pump remains on. The program can resume operation by pressing again the stop button. The restart mode stops all actuators and reinitiates the whole program.

The OB1 block also controls the sequencing of the system movements of the system. It detects from the photoelectric sensors which entry conveyors contains the box and for each case individually and activates the function FB3 associated with the conveyor. After executing that specific function, the OB1 goes back again to search for a new box to move.

The FB3 function allows controlling the box transfer from an entry conveyor, passing through the scissor lift conveyor, to the exit conveyor. This function is called 3 times on OB1, each time for each individual entry conveyor that detects a box.

When called, the function FB3 calls two additional functions. Function FC6, which controls the vertical positioning of the scissor lift conveyor, according to the target stroke necessary to reach the desired conveyor, as well as controls the speed of the oil-hydraulic cylinder through the oil flow profile function FC3. Function FC7 controls the active conveyors motors, to move the box from a source to a destination conveyor like, for example, from an entry conveyor (source) to the scissor lift conveyor (destination). These FC6 and FC7 functions are called two times inside FB3: once for the entry conveyor and once for the exit conveyor, since the movement sequence repeats itself, changing basically the target values of positioning on FC6 and the source and destination conveyors on FC7.

The automatic control of the scissor lift's velocity, previously presented, is operated by the function FC3 that defines the oil flow profile to be applied to the oil-hydraulic cylinder, according to its current stroke. The instant oil flow is calculated through a mathematical function for each stroke position, regardless of being or not on the approach zone. The mathematical function is parametric and the parameters "b" and "m" previously calculated are used on this block for each case of entry-exit conveyor. It adjusts instantaneously the oil flow into the positive chamber of the oil-hydraulic cylinder, resulting in an adjusted velocity.

Human machine interface

To supervise the system, an HMI was added, based on a Siemens KTP-700 and connected to the PLC through a Profinet cable.

The HMI has 3 screens. The "System" screen presented in Figure 22 presents the whole system and the values of oil flow, status of electrical motors and photoelectric sensors. It presents graphically the movement the scissor lift with image animation and the hydraulic cylinder stroke changes. It also presents the current cylinder stroke and vertical height of the scissor lift in a numerical form. It has 3 buttons to perform the actions presented before for the 3 modes: Start, Stop and Restart.

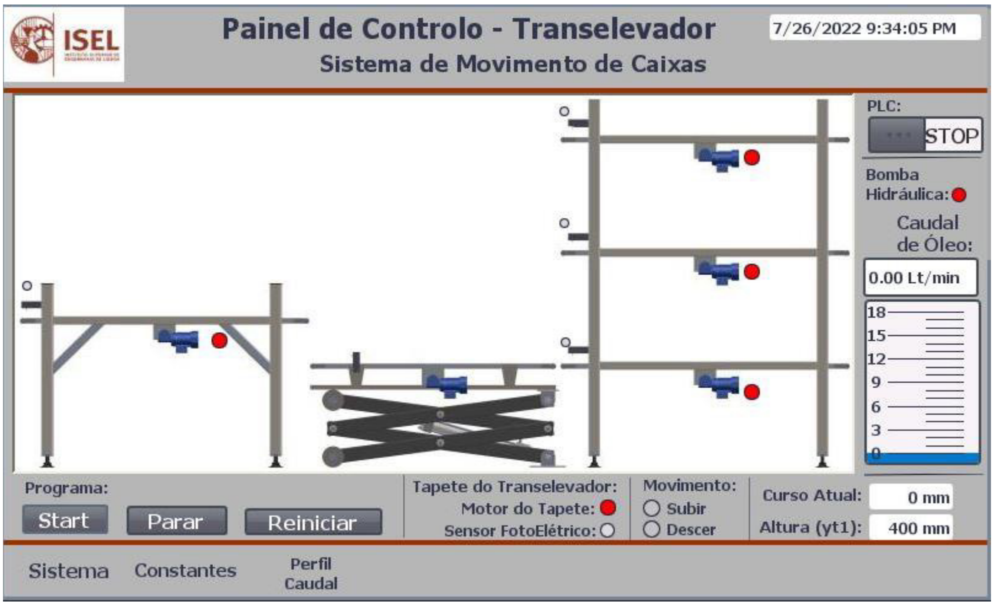

Figure 22: "System" screen from HMI.

The "Constants" screen presented on Figure 23, is an informative screen and presents the system constants, configured on the DB1, such as vertical height of each conveyor, the clearance dimensions of the scissor lift "k1" and "k2", the scissor minimum height, the diameter, and minimum length and maximum stroke of the cylinder.

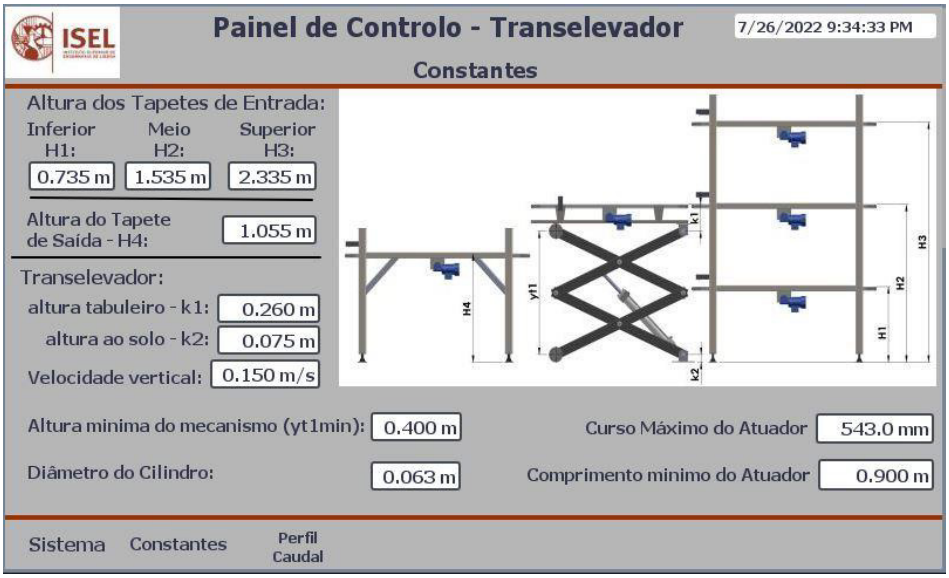

Figure 23: “Constants” screen from the HMI.

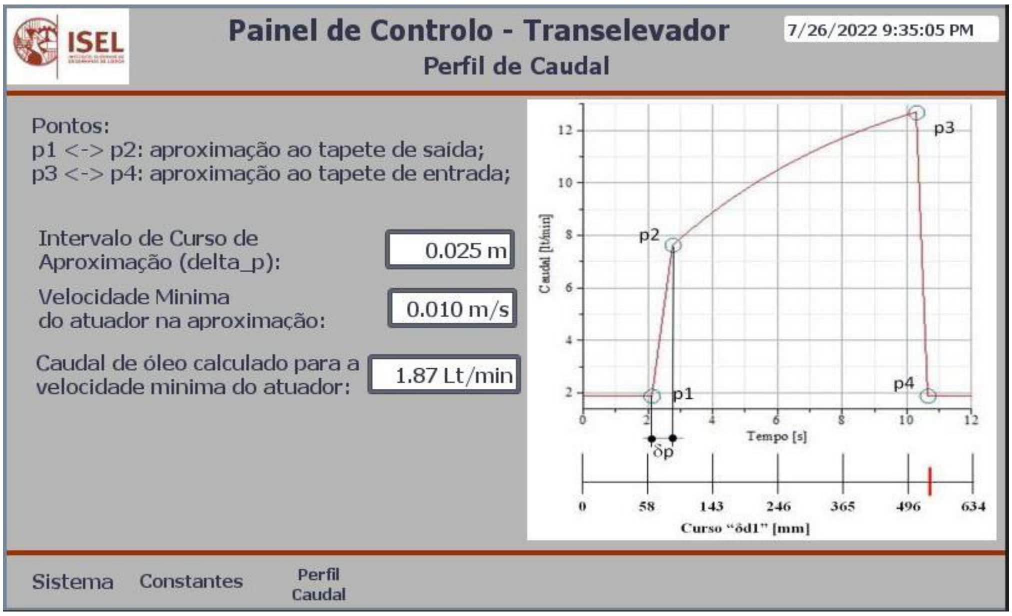

Figure 24: “Oil flow profile” screen from the HMI.

The “Oil Flow Profile” screen presented on Figure 24 is also an informative screen and presents the parameters related with the “m” and “b” parameters calculation of the profile functions.

Remote assistance

The remote assistance enables the user or maintenance technici

an to remotely supervise the process running in the PLC, using a computer connected to the internet.

This function is performed through a “Webserver” integrated in the PLC. Once the Webserver is activated in the PLC configuration, it is necessary to connect the PLC to a local network with internet connection and set an IP address to get the desired remote access. For security purposes, it is recommended to define a username and a password to access the PLC.

After that, a webpage can be developed based on HyperText Markup Language (HTML) programming to access the variables processed and stored inside the PLC.

Once the webpage defined, the user can open a web browser from his computer and access the PLC data remotely.

Conclusion

This work allowed analyzing the case study of a scissor lift working on a box movement system with conveyors, including the direct and inverse kinematics study, the proposal of a hydraulic circuit and the implementation of a PLC and its program to command the movement of the oil-hydraulic cylinder of the scissor lift.

The results obtained on the direct kinematics show the application of the D-H algorithm isn't effective in returning a velocity equation that is true to the simulation result. The same conclusion was obtained for the force equations. Regarding the velocity equations, it was necessary to derive the position equation, written relatively to the time, to get a valid equation. Regarding the force equation, it was necessary to apply the Principle of Conservation of Energy, so the result of the equation matches with the result obtained on the simulator.

By the results obtained in this part of the study, we conclude this algorithm is suitable only to kinematic structures with joints in serial displacement, like a manipulator robot, where all joints are layout in sequence and each joint is controlled by an actuator, which allows a specific position and speed, and its movement does not physically depend on the movement of other joints.

Despite these limitations, the algorithm is still useful to get the position equation of a certain point in the kinematic structure, although it requires trigonometric manipulation to reach a mathematical solution.

The inverse kinematic equations had the most use along this work rather than the direct kinematic equations. It was applied on the oil-hydraulic cylinder dimensioning process and automation sections. The need to keep the scissor lift speed constant led to the usage of the inverse kinematics equations, because once known the destination points of the scissor lift, it is therefore necessary to know the necessary stroke for the hydraulic cylinder.

The magnetostrictive sensor presents itself as a good solution to measure cylinder's stroke because it is assembled inside of the cylinder, avoiding the external assembly, and placing the sensor at risk of external contact and consequent break downs. Also, the absolute position measurement is the biggest advantage, not requiring the recalibration of the sensor after installation or maintenance operation. However, due to the long wave guide, it is necessary to drill a hole on the cylinder's rod with a substantial depth, which may be difficult to machine it with conventional tools for drilling.

The speed control system of the scissor lift by means of the PLC allows the control of the speed of the oil-hydraulic cylinder through the control of the oil flow that goes in and out of the cylinder in an instantaneous way.

The proposed solution for speed control by means of defining an initial and final ramp, on the oil flow profile, near the target values of the stroke, proved to be an alternative solution to the PID controller, avoiding the overshoot phenomenon and hydraulic circuit instability to reach the target value of stroke.

In the PLC programming, the method of using the time as an intermediate variable to obtain the value of instantaneous speed proved to be the solution to control the speed of the oil-hydraulic cylinder, associating the instantaneous stroke directly into the calculated linear speed and oil flow.

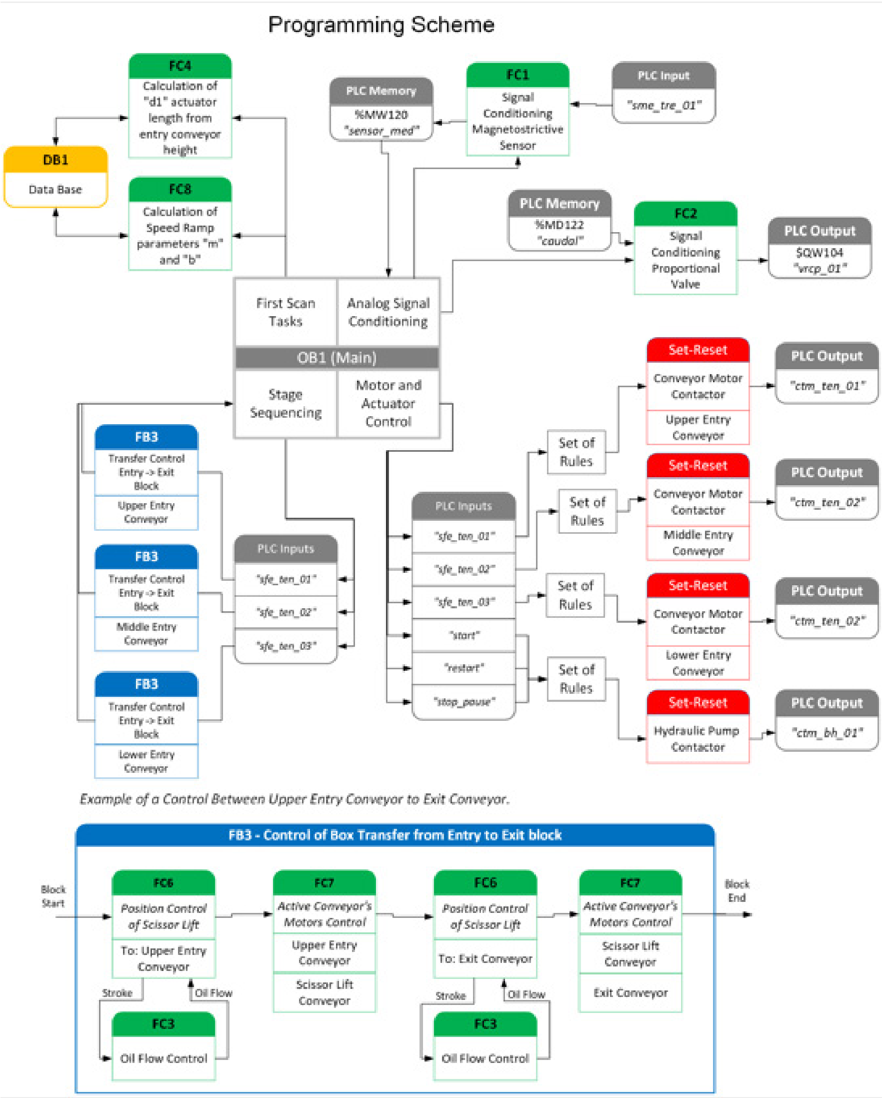

Figure 25: PLC Programming Scheme.

Acknowledgement

None.

Conflict of Interest

There is no conflict of interest.

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