Original Research Article

Study and Design of a Mobile Hydraulic Lift Table with Single Scissors

Abstract

The “BENIN TERMINAL” machines are maintained by company technicians who encounter difficulties in the operations carried out on these machines. In fact, it is about improving the procedure for installing and removing engines and gearboxes for BENIN TERMINAL mobile vehicles through the study and design of a single scissor lift table. It has a lifting capacity of 2 tons, a width of 900 mm, a total length of 1900 mm and a maximum height of 2000 mm using a hydraulic system. It works with 12 V batteries, simple and height-adjustable workspace. The lifting table also allows certain operations requiring an average height to be carried out.This device was developed for the BENIN TERMINAL garage in order to reduce the efforts and risks encountered by technicians when installing or removing engines and gearboxes on certain machines, in particular heavy machines such as Reach Stackers and 16ton forklifts.

Keywords: assembly, gearboxes, sizing, lifting table, maintenance.

1. Introduction

The port is the set of land areas, sea or river waters, infrastructures and superstructures meeting the physical and organizational conditions allowing the reception of ships so that they can shelter there, dock there, carry out operations there, embarkation and disembarkation of goods or passengers, stock up there or carry out repairs [1]. The port represents an important pillar in the economy of the countries that have it because it represents the transit point for goods imported and exported in the country. The autonomous port of Cotonou is no exception to this reality. It represents the lifeblood of the Beninese economy with 90% of foreign trade, 45% of tax revenue and up to 60% of the country's GDP [2]. The operations carried out there are for the most part loading and unloading operations of containerized goods. These operations are carried out using modern port equipment such as gantry cranes, RTGs, mobile cranes, forklifts, port tractors.Given the strong customer demand, this equipment is in high demand. It therefore very often happens to be confronted with mechanical breakdowns on this equipment. At the Autonomous Port of Cotonou, these failures often occur on port tractors and mobile cranes. In some cases, they even lead to the replacement of certain elements such as jacks, motors and gearboxes.As part of the change of engines and gearboxes on mobile stacker cranes commonly called PPM, the procedure followed presents some drawbacks such as the immobilization of several technicians, the long duration of the operation, the associated risks. load, the engine and gearbox alignment problem. To overcome these problems, solutions exist. Several researchers have worked on the design and construction of lifting tables. Among these, we have: ZEDDAM Mohamed Aymane [3], SALEM J.T Mohamed [4], SOUAI L.W Lasaad [5], Amal Belghazi [6] and GAUTHIER Maxime [7]. Models of lifting tables have also been produced in universities such as CITE SCOLAIRE SAINT EXUPERY [8] and Lycée Claude LEBOIS [9]. The models designed by them are dedicated to loads of up to one ton. They cannot therefore be used as part of the procedure for installing the engines in the port because the load to be lifted is 2 tonnes. Furthermore, these models do not have electric autonomy and therefore can only be used in places where there is access to electric power. In the event that models corresponding to the desired technical characteristics are also encountered, these have prices of around 5 million FCFA, which constitutes a heavy financial burden for the company. Faced with these difficulties, we found it useful in the context of this work to study, design and produce a single-scissor hydraulic lifting table coupled to accumulators to give it energy autonomy.

2. Materials and methods

2.1 Block diagram of the lifting table to be produced

Figure 1 gives an overview of the lifting table to be produced. This figure was made with SOLIDWORKS 2017 software.

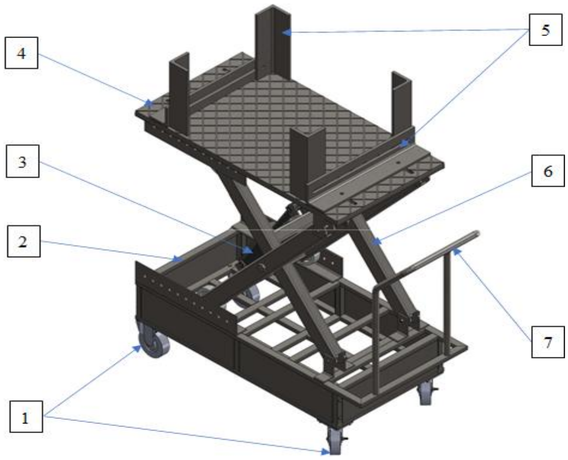


Figure 1: Overview of the *lifting* table

Caption: (1): casters; (2): base or frame; (3): hydraulic cylinder; (4): platform; (5): clamping angles; (6): scissors; (7): guide arm.
The main components of the lifting table are: the platform, the scissors and the base.

2.2 Description of the different components of the lifting table

2.2.1 The platform

It is made up of rectangular profiles that form its structure, angles for guiding the scissor rollers in translation and fixing them, and a corrugated sheet covering its surface. Its role is to support the load and to allow it to be kept in balance. Figure 2 gives an overview of the platform.

Figure 2: Platform

2.2.2 The scissors

The scissors (figure 3) are made up of rectangular and square tubes forming the very structure of the scissors, of equal angles used for fixing the jack. The rollers are mounted on the scissors by means of pins. The scissors work on the principle of the lever: the entry and exit of the cylinder rod generates the closing and opening of the scissors; which results in the ascent and descent of the platform.

Figure 3: Scissors

2.2.3 The base

Its structure is made up of rectangular profiles and square profiles. As in the case of the platform, it also consists of angles for guiding the rollers of the scissors in translation and fixing them. Its role is to support the entire device and to serve as a repository for the components of the engine. Figure 4 gives an overview of the base.

Figure 4: Base or frame

2.3 Principle of operation

Figure 5 shows the 2D kinematic diagram of the lifting table. It traces the principle of operation of the latter. Indeed, the operation of the lifting table is mainly based on the principle of the lever. The action on the control box turns the motor which in its rotation turns the pump. The pump delivers the oil from the reservoir to the double-acting cylinder through the hydraulic (flexible) fittings. The cylinder rod extends or retracts, respectively closing and opening the scissors. The closing and opening movements of the scissors are transmitted to the platform through the guide rollers. The platform goes up and down; we speak of vertical displacement of the platform.

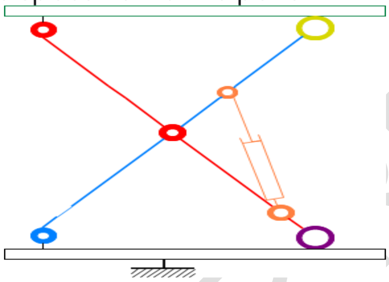


Figure 5: 2D kinematic diagram of the lifting table

2.4 Modelization

This part is devoted to the inventory of the mathematical models relating to the various components of the lifting table. These models reflect the physical phenomena that govern the operation of components. We started with the hydraulic dimensioning then the mechanical dimensioning and finally the electrical dimensioning of the lifting table.

2.4.1 Hydraulic sizing

It mainly concerns the sizing of the cylinder and the mini-plant.

2.4.1.1 Sizing of the cylinder

The main purpose here is to determine the force of the cylinder. For this, we have isolated the different components of the device. The static study on each of them led us to obtain a maximum force Fv from the formula:

$$F_v = \frac{l.sin(\theta')(\|\vec{D}\|+\|\vec{C}\|)}{BM.sin(\varphi+\phi)}(1)$$

where $\phi = 90° - \theta'$

With:

- l: half the length of an arm;
- θ': angle between a scissor arm and the vertical;
- $\vec{D}$: the force exerted by the roller on the platform;
- $\vec{C}$: the force exerted by one arm of the scissors on the base;
- BM: the distance between the axis of articulation of the scissors and the point of application of the force $\vec{F_v}$;
- φ: angle between the direction of the force $\vec{F_v}$ and the horizontal;
- Φ:angle between the axis of the section constituting the arm and the horizontal.

1.1.1.1. Sizing of the mini-plant

The mini-power plant is mainly composed of a pump, an electric motor and a tank.

- ❖ The pump

For the choice of the pump, we needed the admissible flow, the admissible pressure and the displacement of the pump. The maximum values are obtained in the low position.

The admissible flow is obtained by applying a margin of 20% to the normal flow; let $q_{v_{adm}} = 1{,}2 \times Q(2)$. The allowable pressure was calculated with a margin of 15%; let $p_{adm} = 1{,}15 \times p(3)$. As for the displacement, it is given by the relation $C = \frac{1000 \times Q}{N}$[10]. *(4)*

With:

- Q: the admissible thrust flow rate in l/min,
- C: the displacement of the pump in cm^3/tr,
- N: the rotation speed set at 1500 rpm by the manufacturer.
- ❖ Engine

The electric motor is determined by its power $P_m = \frac{p_{adm} \cdot q_{v_{adm}}}{600.\eta_T}$[10] (5)

With:

- P_m : the power in KW;
- P_{adm}: the admissible pressure in bar;
- $q_{v_{adm}}$: the admissible volume flow in l / min;
- $\eta_T = 0{,}90$: the total efficiency of the motor.
- ❖ The reservoir

As for the tank, it is chosen in relation to its volume. By applying a safety factor of 2, the volume of the reservoir gives [11]:

$V = 2 \times q_{v_{adm}}(6)$

1.1.2. Mechanical sizing

We dimensioned *the characteristics of the profiles constituting the structure of the platform, the scissors and the structure of the base.*

1.1.2.1. The platform

Figure 6 shows the forces applied to the structure of the platform.

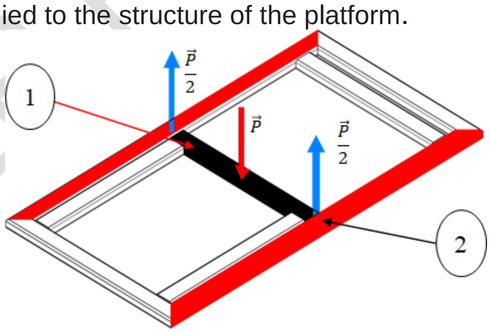


Figure 6: Forces applied to the platform structure

This involves first dimensioning the central cross member 1 and then the side profile 2. We then decided on the profile to be used for the entire structure of the platform.

By isolating the central cross member, we obtain Figure 7.

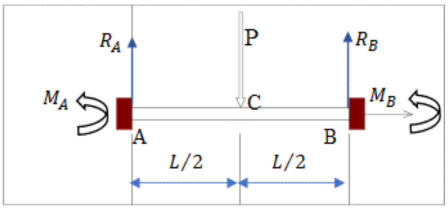


Figure 7: Stresses on the central cross member

The system being hyperstatic, we proceeded by the method by integration. We have found $M_A = -M_B = \frac{PL}{8}$ (7) and $M_{max} = M_{\left(\frac{L}{2}\right)} = \frac{PL}{8}$(8)

By applying the condition of resistance to bending, one finds: $W \geq \frac{M_{max} \cdot s}{R_e}$(9).

With:
W: flexural modulus;
s: the safety coefficient;
M_{max}: the maximum bending moment;
R_e: the elastic limit of the material.
The stresses on the side profile are shown in figure 8.

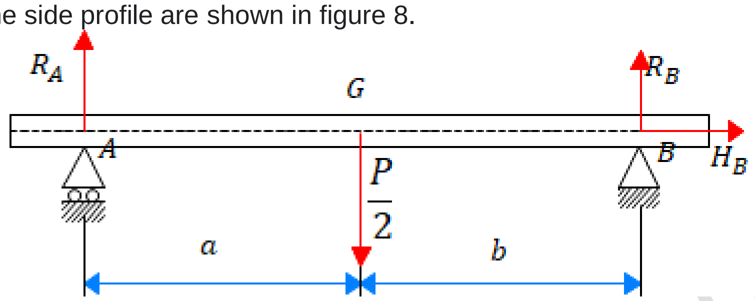


Figure 8: Stresses on the side profile

The system is isostatic. We find $W \geq \frac{Mf_{max}}{R_{pe}}$*(10)*

1.1.2.2. The scissors

This involves sizing the arms and the pull and push profiles. The arms are subjected to a compound stress as shown in figure 9.

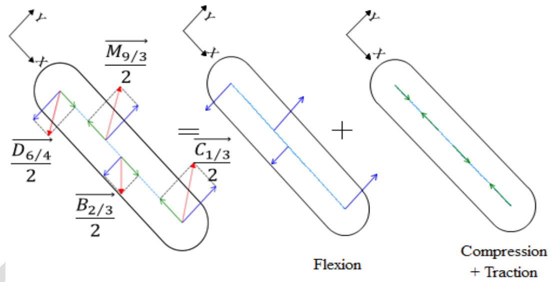


Figure 9: Stresses on the scissors

Considering the bending stresses, the resistance condition is similar to that used for the profiles of the platform structure.

Let $W \geq \frac{Mf_{max}}{R_{pe}}$(11)

For the verification in traction of the selected profile, we applied the condition of tensile strength which gives:

$\frac{N_{max}}{S} \leq R_{pe}$(12)

where$N_{max} = \left\| -\left(\frac{F_v}{2} cos[180 - (\varphi + \emptyset)]\right) \right\|$ is the maximum normal force and $S = 2\,343{,}75\ mm^2$is the section of the profile.

As for the stresses on the cylinder fixing profiles, they are identical to that of the central cross member of the platform structure.

1.1.2.3. The base

The structure of the base is shown in figure 10.

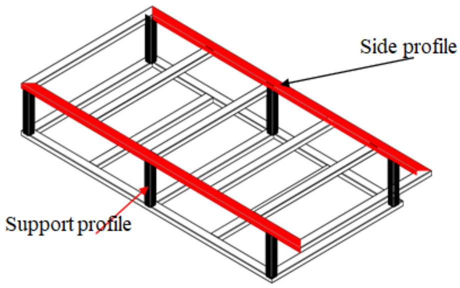


Figure 10: Base structure

This involves sizing first the upper frame and then the support profiles. The sizing of the top frame is the same as the sizing of the side profile.
The stresses on the latter are shown in figure 11.

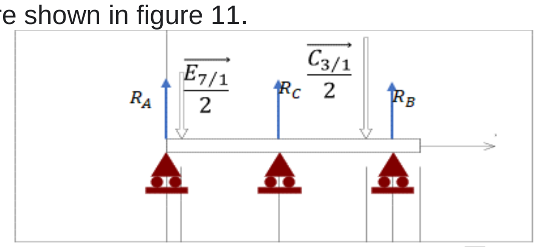


Figure 11: Stresses on the side profile

The system is hyperstatic of degree 1. Proceeding by the superposition method, we obtain

$$R_C = \frac{(E_{7/1}.{b_1}^3 + C_{3/1}.{b_2}^3)}{2.l^3} \quad (13)$$

which allowed us to bring the system back to the isostatic system indicated in figure 12.

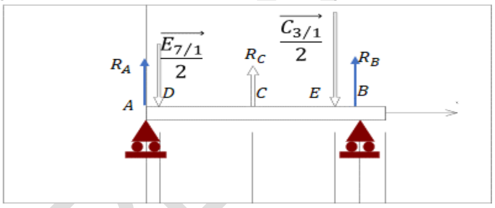


Figure 12: Isostatic system

The resistance condition applied to the latter is identical to that of the scissor arm; let

$$W \geq \frac{Mf_{max}}{R_{pe}} \quad (14)$$

As for the support profiles, they are stressed in tension or in compression. The most requested profile is the one in B. The resistance condition gave:

$$a \geq \sqrt{\frac{R_B.s}{R_e}} \quad (15)$$

Where a is the side of the square profile constituting the support profiles.

1.1.3. Electrical sizing

It mainly concerns the choice of batteries and charger.

❖ Choice of batteries

The choice of batteries is made by knowing their capacity. The capacity of a battery is given by the expression: $C = \frac{P.t}{U}$(16)

With:

- C: capacity (in Ampere hour);
- P: power (in Watt) of the mini-plant;
- t: battery life (in hours);
- U: battery voltage.

The battery life has been fixed at 2 hours.

❖ Choice of charger

To choose the charger, we need to know the voltage and current needed to charge our batteries for a defined time. The voltage is given by the characteristics of the battery to be charged. As for the current, it is given by I = C / t (17) where C is the battery capacity and t the charging time. The charging time has been set at 08 hours.

1.2. Estimated cost of building the lifting table

- The overall cost (Cg) of the machine was evaluated based on:
- the cost of the raw material (Cm);
- the cost of standard parts and accessories (Cp);
- the cost of machining the parts (Cu);
- the cost of labor (Co).

Let Cg = Cm + CP + Cu + Co (18)

2. Results and analysis

2.1. Results

The results from the application of the mathematical models of the elements composing the lifting table are listed in tables 1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11 and the estimated cost of the lifting table in table 12.

Table 1: Value of the force to be deployed by the jack

Units	Parameter
Cylinder	Force to extend F_v (N)
	$143\ 617{,}554\ N$

Table 2: Values of the characteristic parameters of the mini-plant

Units	Parameter				
Mini hydraulic power station	Pump flow rate Q(l/min)	Pump pressure P (bar)	Pump displacement C(rpm)	Engine power Pm(W)	Tank volume V(l)
	2.827	222.795	1.885	1166	5.654

Table 3: Values of the characteristic parameters of the central cross member

Units	Parameter		
Central cross member of the platform structure	Moment at embedding A M_A (N.mm)	Maximum bending moment M_{max} (N.mm)	Flexural modulus W (cm^3)
	1 814 850	1 814 850	7,67

Table 4: Parameter values of the platform side profile

Units	Parameter	
Profilé latéral de la structure de la plateforme	Maximum bending moment M_{max} (N.mm)	Flexural modulus W (cm^3)
	3 254 428,452	13,751

Table 5: Values of the characteristic parameters of the scissor arms

Units	Parameter			
Scissor arm	Maximum bending moment M_{max} (N.mm)	Flexural modulus W(cm^3)	Maximum normal effort N_{max} (N)	Profile section S(mm^2)
	25312190.499	106.952	63229.687	2343.75

Table 6: Values of the characteristic parameters of the thrust profile

Units	Parameter		
Pushprofile	Moment at embedding A M_A (N.mm)	Maximum bending moment M_{max} (N.mm)	Flexural modulus W (cm^3)
	10681414.976	10681414.976	45.132

Table 7: Values of the characteristic parameters of the traction profile

Units	Parameter		
Pushprofile	Moment at embedding A M_A (N.mm)	Maximum bending moment M_{max} (N.mm)	Flexural modulus W (cm^3)
	10514517.867	10514517.867	44.43

Table 8: Values of the characteristic parameters of the lateral profile of the base structure

Units	Parameter

Side profile of the base structure	Reaction in C R_C (N)	Maximum bending moment M_{max} (N.mm)	Flexural modulus W (cm^3)
	2604.313	836000	3.532

Table 9: Values of the characteristic parameters of the support profiles of the base structure

Units	Parameter	
Base structure support profiles	Reaction in BRB(N)	Square profile side a (mm)
	4179.902	4.202

Table 10: Values of the characteristic parameters of the battery

Units	Parameter			
Battery	Power of the mini-plant Pm (W)	Mini-power plant supply voltage U (V)	Battery life t (heure)	Battery capacity C (Ah)
	1166	24	2	300

Table 11: Values of the characteristic parameters of the charger

Unités	Paramètres			
Battery	Mini-power plant supply voltageU (V)	Battery capacityC (Ah)	Charging timet (heure)	Charger IntensityI(A)
	24	300	8	50

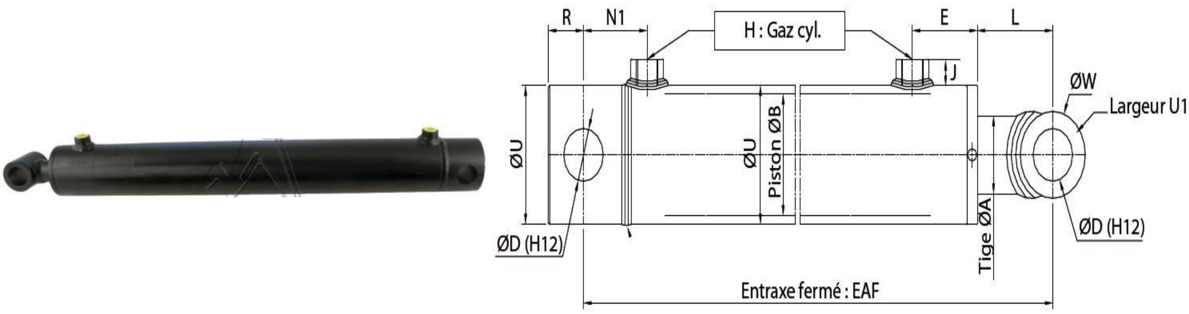
Table 12: Cost of the lifting table

Cost of raw material	958 199,9 FCFA
Machining cost	16 000 FCFA
Cost of standard parts and accessories	2 164 025,57 FCFA
Cost of labor	300,000 FCFA
Global cost	**3 138 225,47 FCFA**

3.2. Results analysis

3.2.1. The cylinder

The force to be deployed by the jack has been evaluated at approximately 144 KN (Table 1). We therefore chose a double-acting cylinder (Figure 10) with a rod diameter of 50 mm and a piston diameter of 100 mm that can deploy up to 157 KN in thrust. The selected cylinder is shown in figure 13.



a b

Figure 13: (a) Double acting hydraulic cylinder. (b) Dimensions of the cylinder

3.2.2. The mini hydraulic power station

In view of the maximum values obtained (Padm = 222.795 bar; qvadm; C= 1.885 cm3/tr; V=5.654 L et Pm=1.166 kW), we have chosen the mini- center of figure 14 in the catalog of the French group COMEO

Figure 14: Mini-hydraulic unit with control box [12]

Technical characteristics [12]:

- Motor 12V 1.8 KW 2600TPM
- Gear pump: 4.2 cm^3/rev
- Flow: 6 L / min
- Plastic tank: 8 liters
- Max pressure: 250 bars

The hydraulic circuit of the mini-plant is shown in figure 15.

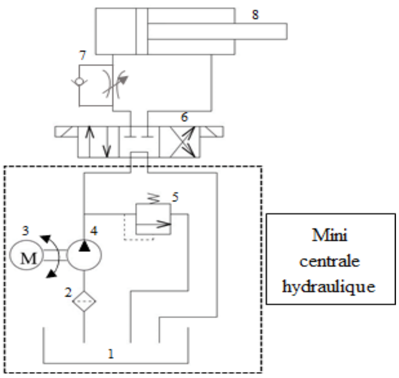


Figure 15: Hydraulic circuit of the lifting table

Caption:
1: Tank; 2: Filter; 3: Electric motor; 4: Fixed displacement pump; 5: Pressure relief valve; 6: 4/3 distributor; 7: One-way flow reducer; 8: Cylinder.

3.2.3. The platform

The flexural strength condition applied to the central cross member gave us a value of $7{,}67\ cm^3$for the flexural modulus. The value immediately above this in the Celsius catalog is $8{,}52cm^3$ which corresponds to a rectangular profile 60 x 40 x 4 [13].
For the side profile, we find $W = 13{,}751\ cm^3$. We therefore choose a rectangular profile 80 x 40 x 8 of modulus $W = 16{,}1cm^3$ [13]. We have chosen this profile for the entire platform because it is suitable for all loads.

3.2.4. The scissors

The rectangular profile 120 x 80 x 12.5 is suitable for the arms of scissors because it has a flexural modulus $W = 115cm^3$[13] or you need a profile whose flexural modulus is such that $W \geq 106{,}952\ cm^3$. This same section responds to the tensile stresses to which the arm is subjected.
As for the tension and compression profiles allowing the mounting of the jack, they have very similar bending moduli, i.e. $W = 45{,}132cm^3$we therefore chose a square profile of 80 mm and 8 mm thick having for flexural modulus$W = 47{,}3cm^3$ [13].

3.2.5. The base

For the side profile of the upper frame of the base, we have chosen a rectangular profile 50x40x3 with a flexural modulus $W = 3{,}96cm^3$[13]. This profile will be able to respond to requests, because the resistance condition is:
$W \geq 3{,}532cm^3$.
The profile chosen for the support profiles is a 40x40x3 square profile [13]. It will be able to respond to requests, because the condition on (a) is a 4.202 mm.

3.2.6. Battery

By setting the autonomy to 2 hours, the capacity of the batteries is 300Ah. For more performance, we chose two KAPITAL brand batteries (photo 1) with a capacity of 200 Ah each, for a total capacity of 400 Ah.

Photo 1: Batteries

Below are the technical characteristics of this battery:
– Voltage: 12V
– Capacity: 200 Ah
They are available at the BENIN TERMINAL store just like hydraulic oil and fittings.

3.2.7. The charger

As for the charger, by setting the charging time at 8h, we obtain an intensity of 50 A. Remember that the voltage at the battery terminals is 12V. We therefore chose the battery charger (SBC2150 12V-50A-2 outputs) found in the "Batteries & Solar energy" catalog (figure 16).

Figure 16: Battery charger

Technical characteristics of the charger:
– Input voltage: 100 to 240 VAC
– Charging current: 50A
– Battery capacity to charge: 150 to 800 Ah

– Battery voltage : 12 VDC
–Dimensions : 261 x 220 x 80 mm
– Weight: 3.2 kg
The electrical circuit of the mini-plant is shown in figure 17.

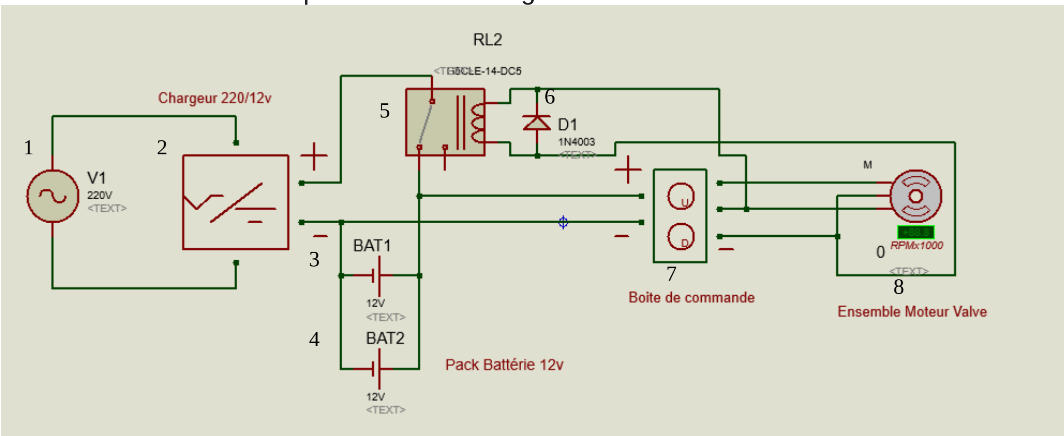


Figure 17: Electrical circuit of the lifting table

Caption:
1: Sector; 2: Charger; 3: Battery 1; 4: Battery 2; 5: Relay; 6: Diode; 7: Mini-central control unit; 8: Motor and valve assembly for the mini-control unit.

3.2.8. Actual cost of the lift table

The overall cost of the lifting table is 3,138,225.47 FCFA. By applying an increase of 10%, we obtain a realization price of 3 782 048.017 FCFA. This increase is due to the fluctuation of the cost of materials in the market and the rate of inflation.

3. Conclusion

The present work, which is part of the improvement of the procedure for installing and removing engines and gearboxes from BENIN TERMINAL mobile vehicles, has led us to the study and design of a lifting table mobile hydraulics with single scissors. This device will be of great use in improving the said procedure, because it will make it possible to overcome the long execution time of the operation, the large number of participants, the immobilization of certain machines as well as the problems associated with the operation, the alignment of the engine and the gearbox without forgetting the risks associated with the load. The safety aspect was therefore taken into account in all phases of the conceptual study. The total cost of the implementation has been estimated at 3,782,048,017 FCFA. This value is reasonable because the prices of the models offered by the manufacturers are around 4,800,000 FCFA [14].

COMPETING INTERESTS DISCLAIMER:

Authors have declared that no competing interests exist. The products used for this research are commonly and predominantly use products in our area of research and country. There is absolutely no conflict of interest between the authors and producers of the products because we do not intend to use these products as an avenue for any litigation but for the advancement of knowledge. Also, the research was not funded by the producing company rather it was funded by personal efforts of the authors.

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