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Model Design and Development of a Telescopic Palm Fruit Harvester

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Abstract

This research paper presents a comprehensive conceptual design approach for the development of a telescopic machine system, which is portable and will provide a safe method of harvesting palm fruits. For this machine system development, the material for each component of the machine system was first selected, the boom length, maximum boom angle, force and stroke length of each hydraulic cylinder, the hydraulic pump pressure, base weight, permissible weight of the cutting system and power required were then calculated in the design analysis. Furthermore, from the calculated parameters, the model of the system was created using SolidWorks engineering software, the model was developed and tested. The result shows that the cutting time of the system for one bunch of palm fruit was longer when compared to conventional systems. It was concluded that though the machine is maintenance friendly and portable, further improvements in its design are necessary so as to develop a system that will give desirable economic output at a shorter time.

Keywords

Design Model, Telescopic, Palm Fruit Harvester, Boom, Mast, Hydraulic Cylinder, Solidworks

1. Introduction

Globally, the African oil palm (*Elaeis guineesis*) (Figure 1 and Figure 2) which is a native plant of West Africa is a prime source of vegetable oil and fat, contributing about 40% of all vegetable oil available in today's markets. On a global scale, it is regarded as the most important vegetable oil crop in the world, the yearly consumption of palm oil extracted from fruit of the oil palm is about 45.3 million tons. The current demand for palm oil is increasing on time bases due to increasing population, economic development as well as many uses in non-food

Figure 1. A oil palm tree.

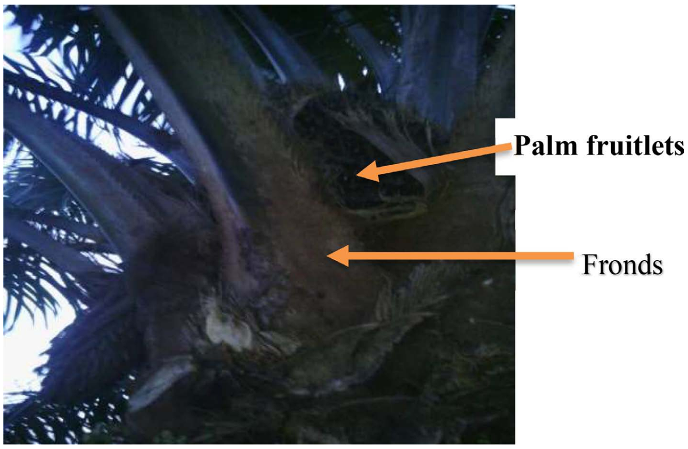

Figure 2. Fruitlets and fronds of an oil palm.

sector, this calls for more research in palm oil production sector. With the trend of increasing demand, cultivation of oil palm has expanded more than any other crop in the past ten years [1] [2] [3]. It is anticipated that over the next 4 decades, the palm oil industry will provide significant economic impact towards the society [2].

In Nigeria, which is considered a developing country and also among the largest producers, oil palm is an economic plant because it serves as a major source of income to individuals and foreign exchange to the economy [4]. As an effort to drive sustainable development of oil palm industry in the area of food security, there is a great need to expand Nigeria's production capacity in order to take back its position of global leaders in palm oil export [5] [6] [7].

However, the oil palm industry faces challenges, which is harvesting of the palm fruit (fresh fruit bunch, FFB) at maturity. Harvesting the palm fruit is the most important but onerous task because it is labor intensive and it is mostly done manually by human power which is quite difficult and risky.

From Malaysia Palm Oil Council (MPOC) information series, labour shortage is a perennial problem, the lack of an adequate and timely labour supply have been felt by the plantation industry. The increasing demand for the limited supply of labour occasioned by the sharp increase in area planted with the oil palm has also resulted in increasing labour cost. Based on the present trend, it will get to a point when it may no longer be profitable for the palm plantation sector to accommodate further increase in labour cost, this puts the country in a disadvantage position in the face of global competition [8] [9]. Because of this, palm oil industry is currently focused on developing sustainable harvesting method.

The method for harvesting palm fruits depends on the height [10]. Locally, cutlass or chisel is used to harvest palm fruit bunch within arm-reach. Tall trees above nine metres in height are harvested traditionally using the single rope-and-cutlass (SRC) or double-rope-and cutlass (DRC) method. In this method, the farmer climbs the tree with the aid of rope tied around the tree and his waist. Once within arm-reach to the fruit bunch, the farmer uses a cutlass to cut the fronds and bunches. This method is commonly used because it is faster, however it is unsafe and energy consuming. Another conventional manual method used for harvesting palm fruit above nine metres high was the bamboo pole and knife (BPK) method. In this method, a sharp curved blade (Malaysian knife) is attached to the end of a bamboo pole. The farmer stands on the ground and raises the pole to harvest the fruit bunch, this requires skill, time and energy to ensure an effective cutting operation. Report has shown that this method requires the application of a force of about 18,048 Newton for harvesting the most mature frond [11] [12]. Also, ergonomic analysis has shown low back and upper limb pains for farmers due to the force posture and repetitive movement during operation [13] [14]. There is an urgent need to increase research on mechanization of palm fruit harvesting process as a means of solving the problem of labour shortage, cost, physical work stress and increase palm oil production for industry [15]. Many attempts have been made to reduce the drudgery of the harvesting of palm fruits by developing and testing different machines. A support mechanism platform was designed and developed for easy adaptation of the motorized cutter. This reduced harvesting time by 100% when compared with the traditional method [16]. A three wheeled trailer platform with multi-stage hydraulically operated boom system harvester was developed and tested [17]. A prototype telescopic raising platform was fabricated. The machine consists of a three stage hydraulic telescopic cylinders and an axial sliding bucket [18]. The limitations to the use of platform harvesters were that it requires going up and down which makes the process so tough. In addition, the risk of falling during operation specifically in places with uneven ground level. A prototype motorize cutter called Cantas™ has been designed, fabricated and used for palm lower than 5 m height. The machine consists of a cutter head and a 2-stroke petrol engine. Its average productivity was 14 tons per day [19] [20] [21]. A chisel type motorized hand held mechanical cutter called ckat™ was developed. The harvesting machine consists of a cutting head, a pole and a 2-stroke petrol engine attached to its end. It is 1.5 m long and weigh 5.0 kg. Its average harvesting productivity was about 160 FFB per hour as compared with the manual harvesting of 110 FFB per hour [22]. An environmentally friendly motorized cutter powered by battery was developed, the trial test showed an average harvesting productivity of about 6 Tons FFB per day [23]. The limitations to this hand held mechanical harvesters were that of weight and vibration, which poses health problem to the farmer. A hydraulic and collector machine wheel type harvester was developed. The boom of the machine attains a height of 10 m, the arm has cutter blade member and a fruit catcher mechanism which is operated by the hydraulic cylinder. The average productivity was 4 to 6 tons FFB per day [24] [25]. The limitation to this machine was its speed as well as its limited productivity. To overcome most of this limitation, focus should be on developing harvesters with the following qualities: lightweight, safety, wide maneuverability, high productivity, stability, good travelling speed and reduced labour force [26]. With the level of advancement in mechanical design in this era, machine system prototypes are easily developed at cheaper cost through modeling of the physical system. In this study, materials for each component of the machine was selected, the design analysis for the components covered the boom length, force and stroke length of each cylinder, the hydraulic pump pressure, base weight, weight of the cutting system and power. The model was created with SolidWorks software, developed and finally tested. The aim of this work is to develop a model design of a telescopic palm fruit harvester which is productive, portable, cost effective and capable of providing safe working environment to enable workers perform this task in an efficient and timely manner.

2. Methodology

2.1. Material Selection

For the material selection, the strength, machinability, ductility and hardness for each component of the machine were given prior consideration. The boom, mast, the base plate of the machine will be subjected to high load, hence mild steel (low carbon steel: EN3B) material was chosen because of its hardness and strength. Also, cold rolled steel (AISI 1020) was used for the hydraulic cylinders because weight saving and durability is a major consign so as to reduce fatigue on the farmer during movement of the machine to the farm.

2.2. Physical Model of Harvester

The physical model was created using SolidWorks CAD, based on the calculated dimensions from the design analysis. **Figure 3** represents the physical model of this harvester, the major components are the boom, mast, hydraulic cylinder, cutting system and the base. Some of the components were designed to be telescopic in operation such that the segments of these component slides over each other, making it possible for the whole component to be longer or shorter during operation.

Boom:

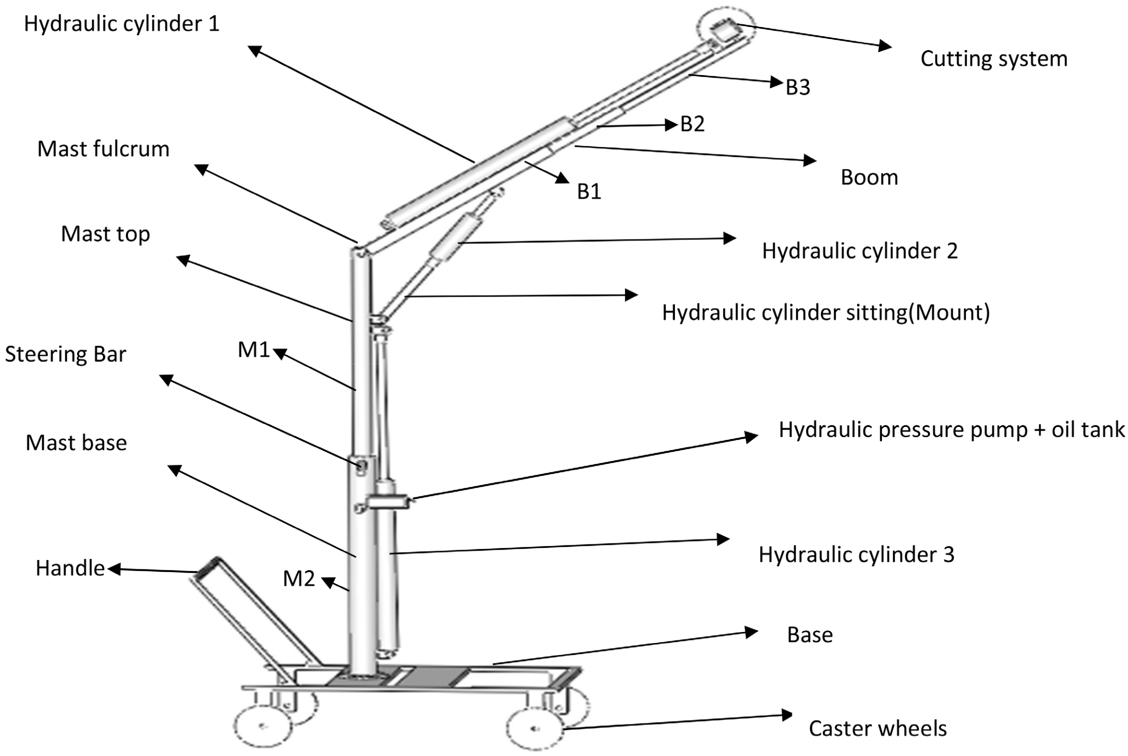

Figure 3. Physical model of the machine.

This is the steel arm and a recognizable part of the machine responsible for lifting the cutting system to the height of the palm fruit. It was designed to be telescopic and consist of three hollow square tube segments fitted one inside the other. The three segments are segment 1 (B1), segment 2 (B2) and segment 3 (B3). The dimensions selected for the square tube Segments 1, 2 and 3 are 50 × 50 × 3 mm, 40 × 40 × 3 mm and 30 × 30 × 3 mm successively. The boom was designed to be telescopic so as to occupy less space for easy storage when the machine is collapsed.

Mast

It is also telescopic and consists of two cylindrical steel hollow pipe materials fitted one inside the other. The two segments are the top (M1) and the base (M2) segments. A top segment round tube of 70 mm diameter with 3 mm thickness was selected. The top segment has a fulcrum at its top end for which serve as a bracket for the boom, as well as a welded bracket on it for hydraulic cylinder 2 sitting (mount). The base segment (M_2) round tube has a diameter of 100 mm and 3 mm thickness. Provision was also made on the base segment where the steering bar was fixed to allow for steering the boom during cutting operation. The oil tank and the hydraulic pump were equally mounted on the base segment of the mast. There is also a welded bracket on the base segment for hydraulic cylinder 3 for its sliding operation.

Hydraulic cylinders

These are the mechanical actuators that gives linear back and forth movement to the boom and the mast. It consists of a cylinder barrel, a piston which is connected to a piston rod. The machine has three hydraulic cylinders: hydraulic cylinder I extend or retracts the boom, hydraulic cylinder 2 is for the boom inclination while hydraulic cylinder 3 is for extension and retraction of the mast. The hydraulic cylinders operate with the aid of hydraulic fluid which is pressurized by a hydraulic pump. These cylinders on the mast and boom are all screwed to brackets which are welded to the segments, thereby making it easier to detach these cylinders from its segment when necessary.

Cutting system

The harvesting(cutting) process of the fruit bunch is done by a fast, rotational motion of a circular saw blade attached to the boom. The rotary saw cutting was employed because it is fast in cutting and the tool changing procedure is easier [27]. A 210 mm diameter steel combination blade was used because it is suitable for cutting thick branches. For this design, a cutting system with minimum weight was considered to prevent failure of the boom, a battery powered system was used because of ease to recharge, lighter weight and distance to electric power supply to the farm.

The base

The other components of this machine are supported by the base. This machine base was designed to provide stability of the harvester during operation. It has a provision where the battery is kept, it also has a handle and caster wheels for easy movement of the machine.

2.3. Design Analysis

The design analysis covers development of mathematical models for components of this machine. The areas are:

Boom Length

When this machine is in operation, the length of the boom required to meet the target tree height can be determined mathematically using Pythagoras theorem, since it can be represented with lines as shown in **Figure 4**.

The boom length (c) was determined with the given Equation (1)

$$c^2 = a^2 + b^2$$

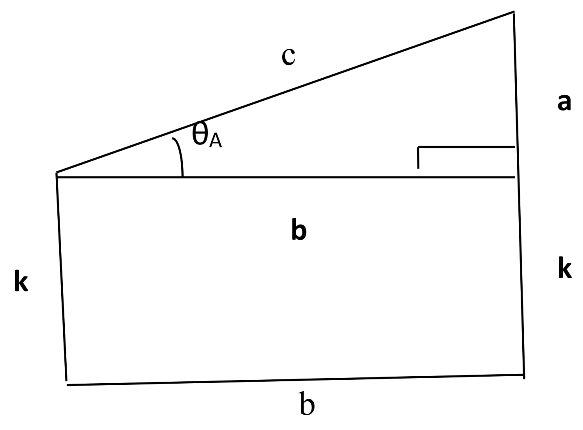

Figure 4. Line diagram representation of the machine and tree.

$$c = \sqrt{a^2 + b^2} \tag{1}$$

where, k is the Mast height, b is the horizontal distance between the tree and the machine, a is the height of the tree top above the mast and $a + k$ is the Boom height (Fruit bunch height) [28].

Maximum Boom Angle

From **Figure 4**, the maximum boom angle (θ_A) of the machine can be determined using mathematical trigonometry as shown in Equation (2),

$$\sin\theta_A = \frac{a}{c}$$

Therefore,

$$\theta_A = \sin^{-1}\left(\frac{a}{c}\right) \tag{2}$$

Stroke Length of hydraulic cylinder 2

The inclination cylinder of the machine can be represented with a diagram as shown in **Figure 5**.

From **Figure 5**, the length of hydraulic cylinder 2 in its instroke position with its mount (V) was first determined with the given Equation (3):

$$V^2 = h^2 + L^2$$

$$V = \sqrt{h^2 + L^2} \tag{3}$$

where, L ($\approx L_2$)—the distance between the welded bracket on boom segment 1 and the mast fulcrum, h—the distance between the welded bracket of top mast to the mast fulcrum,

When the cylinder extends to the maximum boom angle, we have the diagram in **Figure 6**.

From **Figure 6**, the total inclination height (Z) was then determined using trigonometry (Cosine rule) as show in Equation (5).

$$Z^2 = h^2 + L^2 - 2 \cdot h \cdot L \cdot \cos(90 + \theta_A)$$

$$Z = \sqrt{h^2 + L^2 - 2 \cdot h \cdot L \cdot \cos(90 + \theta_A)} \tag{5}$$

Also, the outstroke length of cylinder 2 (J) was determined using Equation (6)

$$Z = V + J$$

$$J = Z - V \tag{6}$$

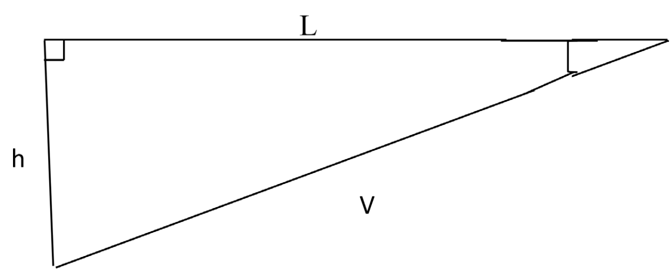

Figure 5. Section of the top mast, boom segment 1 and cylinder 2 at Instroke position.

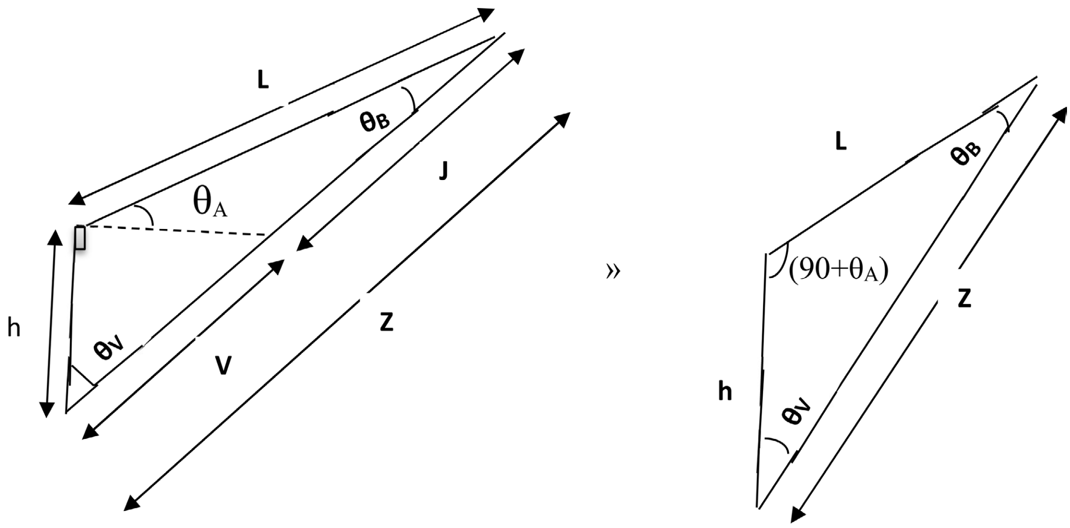

Figure 6. Cylinder 2 extension line representation diagram.

Also, applying Sine rule, cylinder 2 minimum inclination angles (θ_B & θ_V) for top and bottom reached was determined as shown in Equations (7) and (8)

$$\frac{Z}{\sin(90+\theta_A)} = \frac{h}{\sin\theta_B} = \frac{L}{\sin\theta_V}$$

$$\theta_B = \sin^{-1}\frac{h\sin(90+\theta_A)}{Z} \tag{7}$$

$$\theta_V = \sin^{-1}\frac{L\sin(90+\theta_A)}{Z} \tag{8}$$

***Force of hydraulic cylinder* 2 (F_{C2})**

The force of cylinder 2 required to elevate the boom to the maximum angle (F_{C2}) was determined using the free body diagram of the boom and cylinder 2 at its maximum outstroke position as shown in Figure 7.

According to Newton's second law of motion for a linear mechanical system, the sum of external forces (f) acting on a rigid body is equal to the product of the mass (m) and acceleration (a) of the rigid body.

$$\sum F_{\text{External}} = ma \tag{8}$$

D' Alembert's law also relates the sum of all forces acting on the object as shown in Equation (9):

$$\sum F_{ALL} = 0 \tag{9}$$

Applying these two laws, the equations of motion for the system are as follow

$$\sum F_{ALL} = 0$$

$$W_{B1}\cos\theta_A + W_{C1}\cos\theta_A + W_{B2}\cos\theta_A + W_{B3}\cos\theta_A + W_{EM}\cos\theta_A$$
$$-F_{RA} - F_{C2}\cos(90-\theta_A) = 0$$

Taking Moment about point A, we have Equation (10) as shown

$$\sum M_A = 0$$

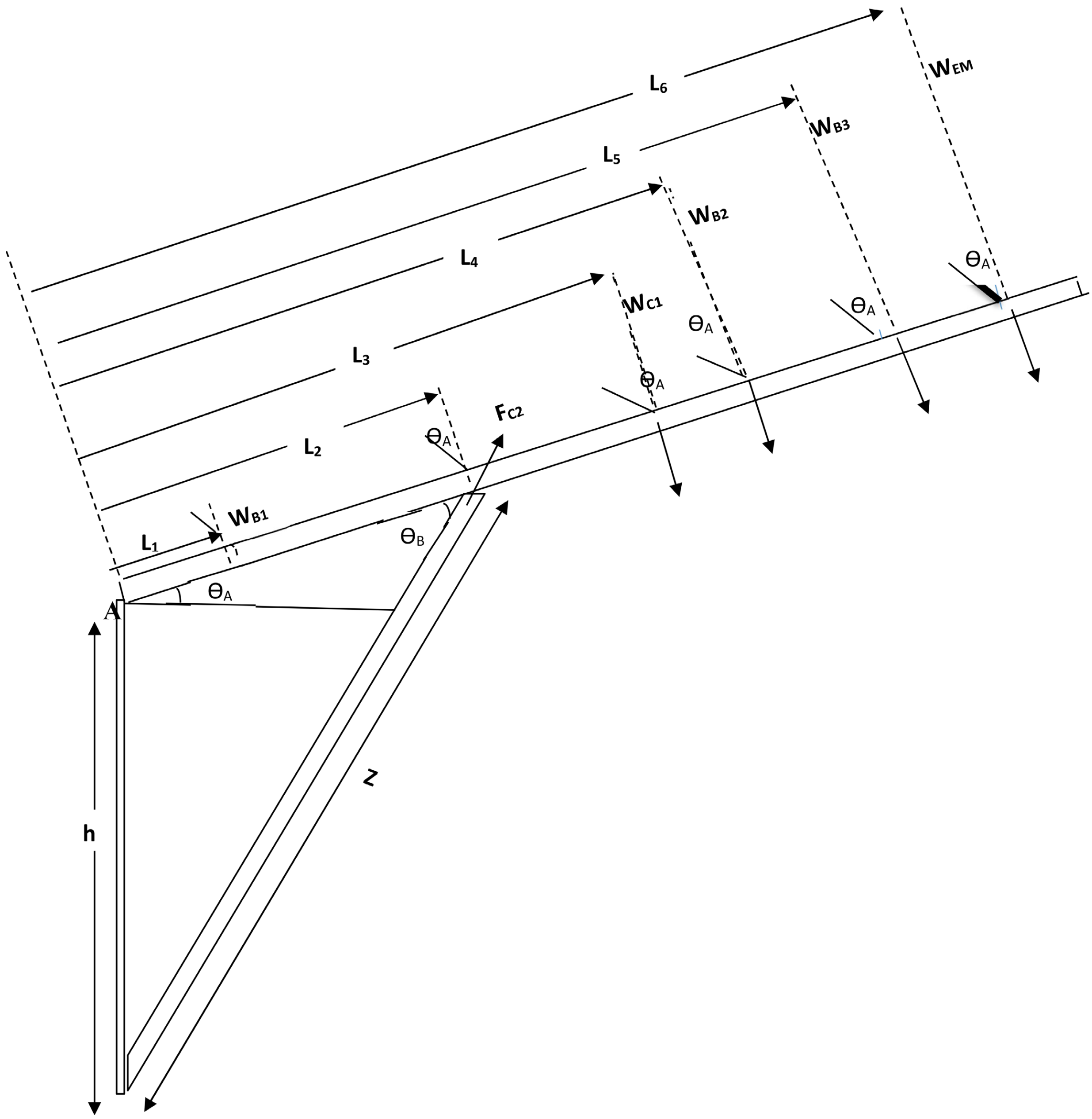

Figure 7. Free body diagram of the boom and cylinder 2.

$$W_{B1}L_1\cos\theta_A + W_{C1}L_3\cos\theta_A + W_{B2}L_4\cos\theta_A + W_{B3}L_5\cos\theta_A$$
$$+W_{EM}L_6\cos\theta_A - F_{C2}L_2\cos(90-\theta_B) = 0$$

$$F_{C2} = (L_1W_1 + L_3W_{C1} + L_4W_{B2} + L_5W_{B3} + L_6W_{EM})\cos\theta_A / \cos(90-\theta_B) \quad (10)$$

Where, W_{B1}—Weight of the boom 1; W_{c1}—Weight of hydraulic cylinder1; W_{B2}—Weight of boom section 2; W_{B3}—Weight of boom section 3; W_{EM}—Weight of the electric motor (cutter).

***Force of hydraulic cylinder* 1 (F_{C1})**

The force of the hydraulic cylinder 1 (F_{C1}) required to extend the boom was determined from the free body diagram Figure 8.

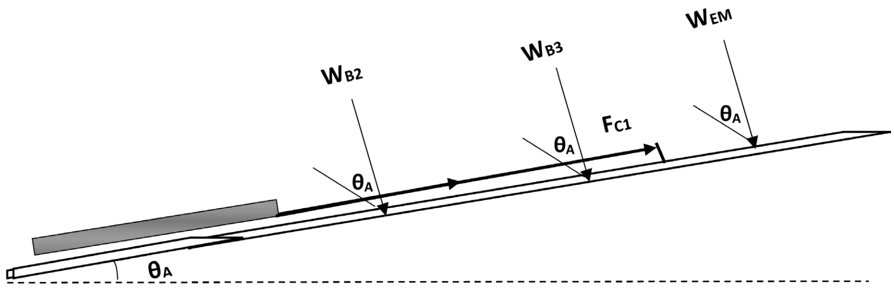

Figure 8. Free body diagram of the boom and hydraulic cylinder 1.

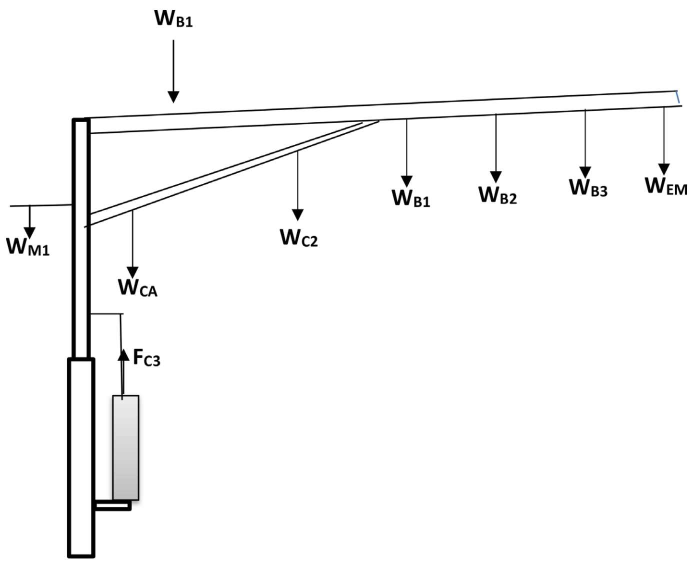

Figure 9. Free body diagram for hydraulic cylinder 3 force analysis.

Resolving the forces, summation of forces about the horizontal *i.e.* $\sum F_H = 0$

$$F_{C1} = W_{B2}\sin\theta_A + W_{B3}\sin\theta_A + W_{EM}\sin\theta_A \quad (11)$$

***Force of Hydraulic Cylinder* 3 (F_{C3})**

From the free body diagram of Figure 9, the force of hydraulic cylinder 3 (F_{C3}) required was determined from Equation (12).

Summation of forces about the vertical *i.e.* $\sum F_V = 0$

$$F_{C3} = W_{M1} + W_{CA} + W_{B1} + W_{C1} + W_{C2} + W_{B2} + W_{B3} + W_{EM} \quad (12)$$

where, W_{CA}—Weight of hydraulic cylinder 2 mount, W_{M1}—Weight of top segment of the mast,

W_{C2}—Weight of hydraulic cylinder 2, F_{C3}—Force of hydraulic cylinder 3 to lift the sections [29].

Hydraulic Pump Pressure for the System

Since the three hydraulic cylinders (1-3) will be working successively during operation, the hydraulic pump pressure specification required for the system will be based on the hydraulic cylinder with the maximum force (F_{max}) required for its outstroke, this is in assumption that the area of the piston for each hydraulic cylinders are equal.

The pressure of the hydraulic pump is determined with the given Equation

(13)

$$P = \frac{F_{\max}}{A} = \frac{4F_{\max}}{\pi D_p^2} \tag{13}$$

where, $F_{\max}$—Maximum hydraulic cylinder force; A—Area of hydraulic cylinder piston; D_p—diameter of piston [30].

Oil tank capacity **(** V_T**)**

The volume of hydraulic oil required for the machine operation (*i.e.* the hydraulic oil tank capacity) was determined from the given Equation (14):

$$V_T = \frac{\pi D_P^2}{4} L_{S1} + \frac{\pi D_P^2}{4} L_{S2} + \frac{\pi D_P^2}{4} L_{s3}$$

$$V_T = \frac{\pi D_P^2}{4}\left(L_{S1} + L_{S2} + L_{S3}\right) \tag{14}$$

where, D_P—diameter of piston of hydraulic cylinder, L_{S1}—Stroke length of hydraulic cylinder 1,

L_{S2}—Stroke length of hydraulic cylinder 2, L_{S3}—Stroke length of hydraulic cylinder 3,

Furthermore, given that the length and width of the tank is assumed, the height required for the fabrication of the tank was determined using Equation (15)

$$H_T = \frac{V_T}{L_T W_T} \tag{15}$$

where, L_T—Tank length, W_T—Tank width, H_T—tank height.

Thus, a good design requires the calculated volume of tank to be greater than the volume of oil

Base Weight of the Machine

The base weight (W_{BA}) that will balance the machine and prevent it from tipping over during operation was determined by applying E-crane equilibrium design principle. From **Figure 10**, we derived Equations (16)-(22) as:

$$X_1 = L_1 \cos\theta_A \tag{16}$$

$$X_3 = L_3 \cos\theta_A \tag{17}$$

$$X_4 = L_4 \cos\theta_A \tag{18}$$

$$X_5 = L_5 \cos\theta_A \tag{19}$$

$$X_6 = L_6 \cos\theta_A \tag{20}$$

$$X_7 = L_7 \sin\theta_V \tag{21}$$

$$X_8 = L_8 \sin\theta_V \tag{22}$$

From Basic crane design principle, we have Equations (23) and (24):

$$\begin{aligned}&\left(W_{M1} + W_{M2}\right)\left(L_B - \left(X_a + K_a + y\right) + W_{BA}\left(\frac{L_B}{2} - y\right)\right.\\&= W_{B1}X_1 + W_{C1}X_3 + W_{B2}X_4 + W_{B3}X_5 + W_{EM}X_6 + W_{CA}X_7 + W_{C2}X_8 + W_{RPO}M\end{aligned} \tag{23}$$

where,

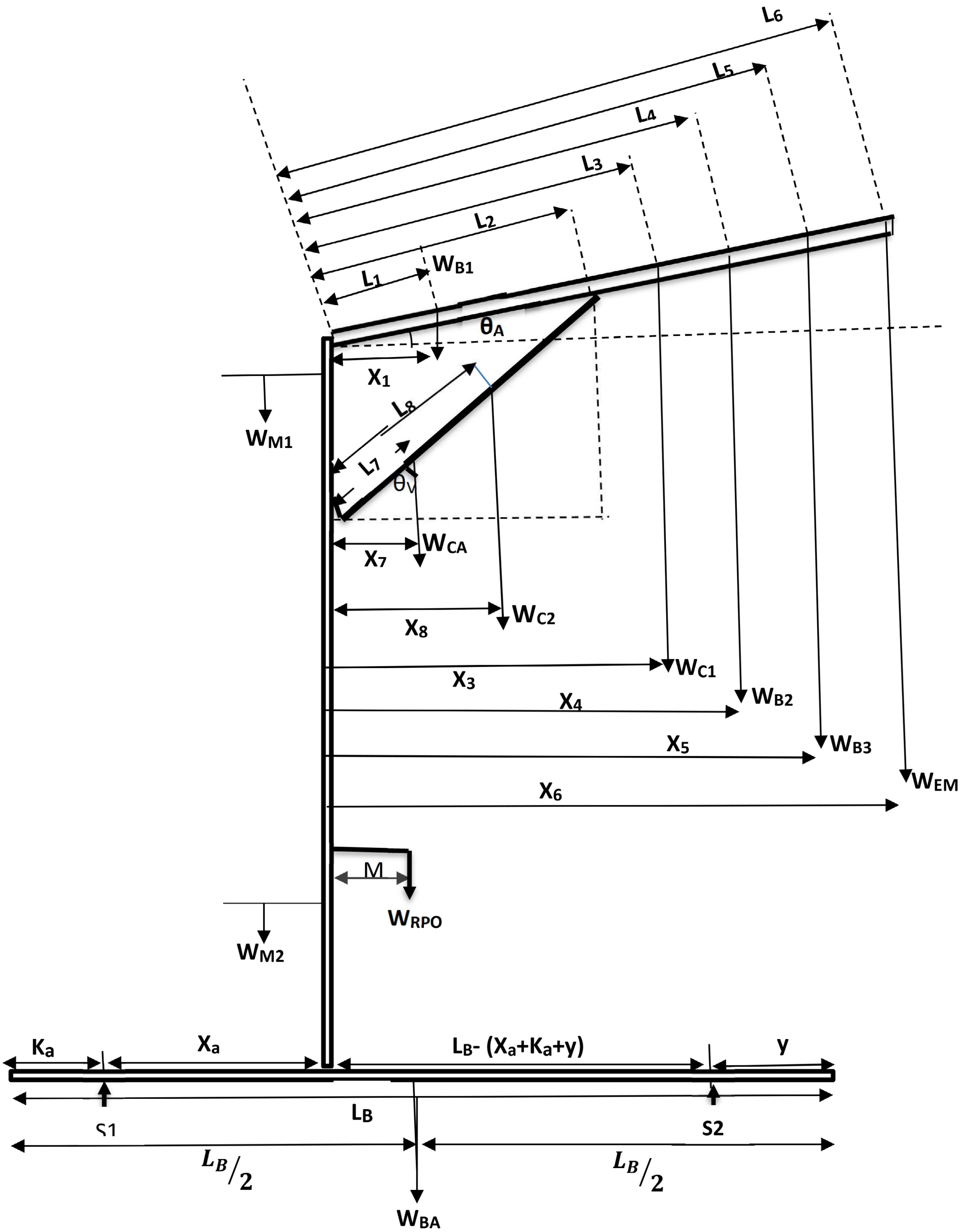

Figure 10. Free body diagram of the complete machine.

$$W_{RPO} = W_{C3} + W_P + W_O \tag{24}$$

W_P—Weight of hydraulic pump, W_{c3}—Weight of hydraulic cylinder 3; W_O—Weight of oil filled tank, L_B—Length of the base, W_{BA}—Required base weight of the machine [31] [32].

Permissible Weight of the Cutting System

The safe weight of the electric cutting mechanism on the boom can be modeled as a force impacted on a cantilever beam at one end of it as shown in **Figure 11**.

As shown in **Figure 12**, the deflection of the boom due to the weight of the cutter was given in Equation (25)

$$y_{\max} = \frac{PC^3}{3 \cdot E \cdot I} \tag{25}$$

where $y_{\max}$—maximum deflection, P—maximum load, C—boom length, E—Young Modulus of Material (*i.e.* steel), I—Area moment of inertia of the boom.

For a boom which is a rectangular hollow cross sectional members, the area moment of inertia of each segment was determined using Equation (26)

$$I = \left(\frac{W_i^3 H_i}{12} - \frac{w_i^3 h_i}{12} \right) \tag{26}$$

where, W_i—External width of segment; H_i—External height of segment; w_i—Internal width of segment, h_i—Internal height of segment, i—1, 2, 3.

For a design with minimal deflection, the area moment of Inertial with the highest value was selected [33].

Since dead load deflection seldom contribute to deflection, the allowable elastic deflection for the steel member (boom) is given in Equation (27) as shown

$$y_{\text{all}} = \frac{C}{360} \tag{27}$$

where, y_{all}—Allowable deflection. [34]

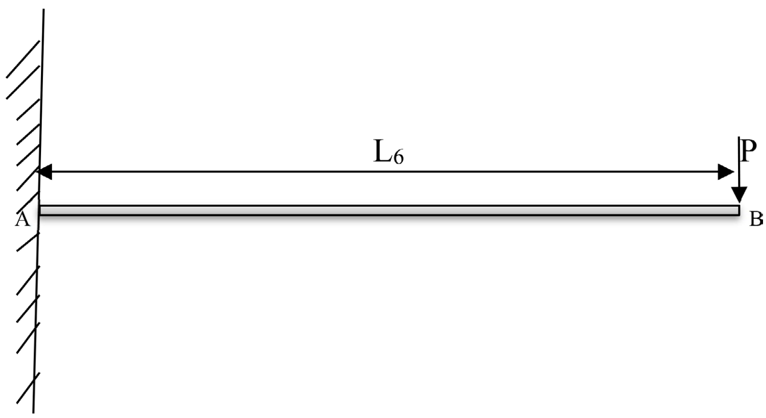

Figure 11. Free body diagram of the electric cutter on the boom.

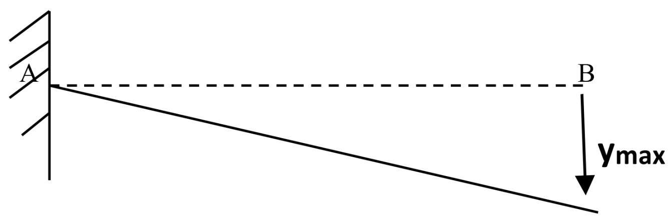

Figure 12. Deflection of the boom.

For a satisfactory deflection criterion, maximum deflection must be equal or less than the allowable deflection (*i.e.* $y_{\max} \leq y_{\text{all}}$), therefore the maximum weight (load) of the electric motor (cutter) for an allowable deflection which is the design load as derived was determined with the given Equation (28).

$$P_s = \frac{3 \cdot E \cdot I}{360C^2} \tag{28}$$

where, P_s—Permissible (Design/allowable) weight of the cutting system.

Thus for a safe design of this machine, W_{EM} should be equal to or less than P_s (*i.e.* $W_{EM} \leq P_S$) [35] [36] [37].

Power Requirement

The power (P_{WR}) requirement for the electric cutter of the machine was determined from Equation (29)

$$P_{WR} = \frac{M_a \cdot R \cdot g\left(2\pi N\right)}{60} \tag{29}$$

where, P_{WR}—Power of electric motor(watts), M_a—Mass of cutting blade(kg), R—radius of the cutting blade, N—Rotational speed in rev/minutes., g—acceleration due to gravity ≈ 9.81 m/s² [38].

In summary, the principal dimensional and mechanical parameters of the harvester are listed in **Table 1** and **Table 2**.

Table 1. Model design parameters for the harvesting machine.

Symbol	Description	Values
a	Height of the tree above the mast	1 m
k	Mast height	2.44 m
b	Horizontal distance between machine and tree	1.48 m
L	Distance from the welded bracket on boom segment 1 to the mast fulcrum	0.59 m
h	Distance from the welded bracket of the top mast to the mast fulcrum	0.315 m
W_{B1}	Weight of boom segment 1	24N
W_{C1}	Weight of hydraulic cylinder 1	42.94N
W_{B2}	Weight of boom segment 2	15.9N
W_{B3}	Weight of boom segment 3	76N
W_{EM}	Weight of electric motorised cutter	35.28N
W_{CA}	Weight of hydraulic cylinder 2 mount	17.48N
W_{M1}	Weight of top segment of the mast	37N
W_{C2}	Weight of hydraulic cylinder 2	29.82N
D_P	Diameter of piston	0.063 m
L_{S1}	Stroke length of hydraulic cylinder 1	0.62 m

Continued

L_{S2}	Stroke length of hydraulic cylinder 2	0.14 m
L_{S3}	Stroke length of hydraulic cylinder 3	0.75 m
L_T	Oil tank length	0.15 m
W_T	Oil tank width	0.10 m
W_P	Weight of hydraulic pump	24.1N
W_O	Weight of oil filled tank	60.86N
L_B	Length of the base	1.17 m
W_{M2}	Weigth of lower segment of the mast	53N
y	Distance between the wheel support 2(S2) to the right side end of the base frame	0.03 m
X_a	Distance between the wheel support 1 (S2) to the left side of th mast	0.325 m
K_a	Distance between the wheel support 1 (S2) to the left end of the base frame.	0.08 m
L_1, L_3, L_4,	Incline lenghts of W_{B1},W_{C1},W_{B2},W_{B3},W_{EM},W_{CA},W_{C2}	0.44, 0.87, 1.04,1.46,
L_5, L_6, L_7, L_8	successively from the mast fulcrum	1.755, 0.17,0.50 m
W_{C3}	Weight of hydraulic cylinder 3	43.94N
M	Distance of W_{RPO} from the mast	0.02 m
W_1, W_2, W_3	External width of boom segment 1,2,3 successively	0.05, 0.04, 0.03 m
H_1, H_2, H_3	External height of boom segment 1,2,3 successively	0.05, 0.04, 0.03 m
w_1, w_2, w_3	Internal width of boom segment 1,2,3 successively	0.044, 0.034, 0.024 m
h_1, h_2, h_3	Internal height of boom segment 1,2,3 successively	0.044, 0.034, 0.024 m
L_B	Base frame length	1.17 m
E	Young modulus of the steel material	210×10^9 N/m
M_a	Mass of cutting blade	0.496 kg
R	Radius of the cutting blade	0.09 m
N	Rotational speed	3000 rev/min.
n	Factor of safety	3
g	Acceleration due to gravity	9.81 m/s^2

Table 2. Calculated parameters values based on Table 1 definition.

Symbol	Description	Values
c	Boom length	1.79 m
θ_A	maximum boom angle	34˚
V	Length of hydraulic cylinder 2 at instroke	0.67 m
Z	Total inclination height	0.81 m

Continued

J	Outstroke length of hydraulic cylinder 2	0.14 m
θ_B	Hydraulic cylinder 2 top minimum inclination angle	19˚
θv	Hydraulic cylinder 2 bottom minimum inclination angle	37˚
F_{C2}	Force of hydraulic cylinder 2(≈ Fmax)	603N
F_{C1}	Force of hydraulic cylinder 1	71N
F_{C3}	Force of hydraulic cylinder 3	303N
P	Hydraulic pump pressure	1.9 bar
V_T	Oil tank capacity	4.7×10^{-3} m^3
H_T	Oil Tank height	0.31 m
W_{RPO}	Sumation of weights W_{C3},W_P,W_O	128.9N
X_1, X_3, X_4, X_5	Resolved horizontal distances of W_{B1}, W_{C1}, W_{B2}, W_{B3},	0.36, 0.72, 0.86, 1.21
X_6, X_7, X_8	W_{EM}, W_{CA}, W_{C2} successively from the mast.	1.45, 0.099, 0.287 m
W_{BA}	Base weight of machine	168.6N
I	Area moment of Inertia of the boom	2.085×10^{-7} m^4
P_S	Permissible weight of the electric cutting system	114N
P_{WR}	Power required	138w

3. Results and Discussion

In this research, the development of a model design for a portable telescopic palm fruit harvester was done for farmers in rural areas. The materials for the development were sourced locally which makes this machine to be maintenance friendly. With the required parameters obtained from the conceptual design, detailed development of a telescopic palm fruit harvester was achieved as shown in Figure 13(a) and Figure 13(b).

At the end of this design, the performance evaluation was determined as shown in Table 3 and plotted on a bar chart in Figure 14; the result shows that when the battery is fully charged, the cutting time of the system for one palm fruit bunch was 52 minutes excluding the time for positioning the machine. The result also shows an increase in cutting time for each bunch harvested which is as a result of drain in battery. In addition, some difficulties such as delayed manual hydraulic pump stroke, intense vibration generated during cutting through hard section of palm fruit bunch stalk, deformation of the boom segments as well as damage of some fruit bunch were encountered, hence, there is need for some design modifications to improve its performance output.

4. Conclusion

The mechanized palm fruit harvester presently used in Nigeria is very expensive

Figure 13. Palm fruit harvester.

Table 3. Performance evaluation.

Fruit Bunch	Cutting Time (Minutes)
1	52
2	70
3	89
4	105
5	124
6	140

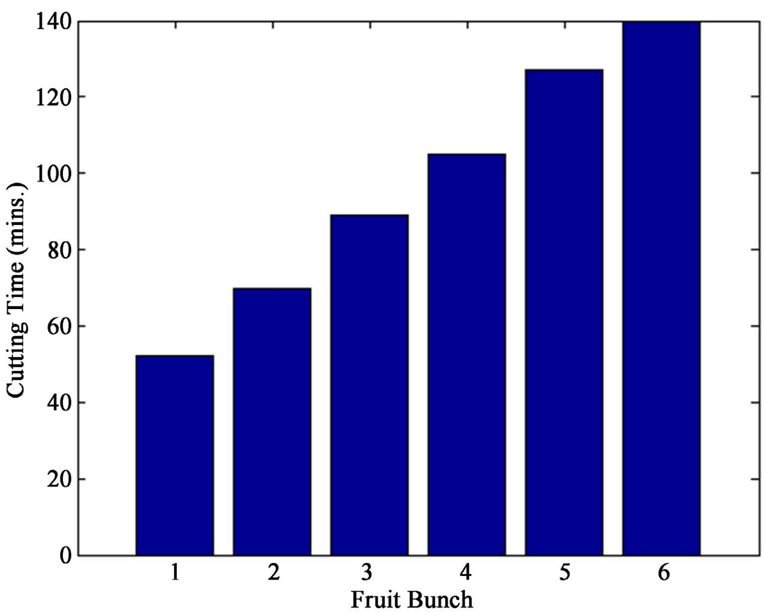

Figure 14. Plot of performance evaluation.

and not maintenance friendly. With Nigeria's unique arable landmass, a lot can be achieved with the design of this unique palm fruit harvester, there is no under estimating the need to utilize and improve on this design. Its portability, low cost, maintenance friendly and use of less power consumption cutting system put it in a class above so many and market it as a workable design research both economically and technically. Therefore, further improvement in the design of the machine and its development in the near future will bring about a remarkable reduction in labour requirement for harvesting palm fruit in rural areas. Some key areas for future design modification and improvement are: use of electromechanical hydraulic pump system to obtain speedy operation of the hydraulic cylinders, improving the stability mechanism of the machine considering ground levels, structural analysis covering static and fatigue analysis of each component of the machine to avert deformation, improving the power source since the inverter used was unable to power the cutting system for too long and finally installing magnifying glass to enable operators sight the palm fruit bunch stalk clearly from a far distance so as to eliminate the issue of damages to the fruits. Note also that the complexity of the model is such that the author was not able to show and discuss all the details of the work. The author stays away with pleasure to the disposal of the interested readers for any further discussion on the approach followed here.

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Conflicts of Interest

The author declares no conflicts of interest regarding the publication of this paper.

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