

Design and Fabrication of Prototype Telescopic Raising Platform for Harvesting Oil Palm Fresh Fruit Bunches

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Abstract

Oil palm has become the world's number one fruit crop because of its unparalleled productivity. Harvesting plays an important role in the oil palm productivity, yield and cost economics. But the harvesting of oil palm has become difficult due to gradual cutbacks in labour availability. Hence an attempt has been made to design and fabricate a prototype telescopic raising platform for harvesting oil palm fresh fruit bunches at college of Agricultural Engineering, Bapatla, India. The design details of the prototype are described in detail in the paper. Telescopic lifting mechanism, sliding mechanism and rope drive mechanisms were designed to lift and take the platform near fresh fruit bunches at crown of the tree. The diameter and speeds of the first, second and third cylinders were 12.065 cm, 9.525 cm, 7.000 cm and 0.0779 m/s, 0.0421 m/s, 0.02624 m/s respectively. The minimum deflection of the sliding mechanism was 2.81 cm and rope drive needs 0.08 hp to operate, which is less than the power availability. All the designed mechanisms worked well at the prototype, fabrication of the machine at actual scales will be a good approach towards the mechanization of the oil palm.

Keywords: Fresh Fruit Bunches, Harvesting, Oil Palm, Raising Platform, Rotating Mechanism, Sliding Mechanism, Stability

1. Introduction

Oil Palm (*Elaeis guineensis* Jacq) is a tree without branches but with many wide leaves at its top. It has become the world's number one fruit crop because of its unparalleled productivity¹. This is the highest oil yielding plant among perennial oil yielding crops, producing palm oil and palm kernel oil. It is a humid tropical palm which thrives well where annual temperature range is 29°–33°C (maximum) and 22°–24° C (minimum) with an evenly distributed rainfall of 2,500–4,000 mm, relative humidity more than 80% and not less than 5 h sunshine/day². It can be grown up to 900 m above mean sea-level. Planting is preferably done at the onset of rains during June-July. The information about the oil palm is given below.

Oil palm starts yielding at the age of 2 – 3 years from planting. It can grow up to a height of 40 ft in 25–40 years. The harvesting oil palm at small ages can be done with chisels. Harvesting is feasible with sickle and pole arrangement up to a height of 8 ft³. Beyond this height the harvester feels difficult to harvest the FFB due to uneven balance of sickle with pole and difficult in finding the stem of the bunch. The economic growing period is 25 to 40 years⁴. The up routing of trees at this age is uneconomical.

To alleviate the problem of manual harvesting of oil palm fresh fruit bunches and difficulties involved in this, an attempt has been made in this project to design the prototype telescopic raising platform for harvesting oil palm fresh fruit bunches.

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2. Methodology

The plantation of the trees in the field is in such a way that one tree is in the middle of the two trees (scattered) in the opposite row (Figure 1). A prototype telescopic raising platform was fabricated for harvesting oil palm fresh fruit bunches. The machine consists of three stage hydraulic telescopic cylinders, axial sliding bucket, that can be operated by rack and pinion mechanism and this bucket can be rotated by rope drive mechanism around 360 degrees.

The working operation of the machine will be as follows:

Figure 1 shows that the operator first move in the way A, harvests the first palm by adjusting the hydraulic cylinder up to the height they needed and by operating the sliding mechanism (rack and pinion mechanism) after harvesting has been done for the first palm, he returns back from the first palm by operating the sliding mechanism, then tractor moves forward and stops near the second palm in the adjacent row, then the operator operates the rope drive to turn 180 degrees again the harvester operates the sliding mechanism to perform harvesting operation for second palm. This way after completing the all palms in the first and second rows he will neglect passing the tractor in the B way and directly enters in to the way C. This could be possible because the shape of the bucket is semi-circular, its perimeter is more than the diameter of the tree. The exception of working of machine in the alternative rows increases the efficiency of the machine.

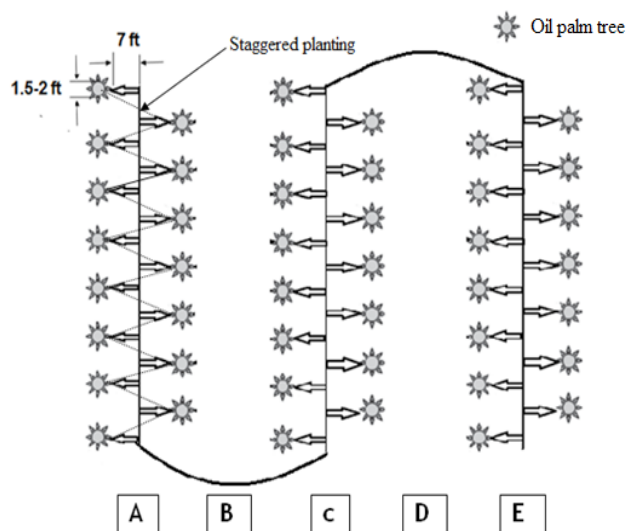


Figure 1. Working operation layout.

2.1 Design of Hydraulic Telescopic Cylinder⁵

Since the machine should be raised up by hydraulic power, the design of hydraulic cylinder is very important. Since the machine should rise up to a height of 12 m, it is better to select three stage hydraulic cylinders. Maximum pressure is required to extend the smaller diameter cylinder as compared to other cylinders. So by taking the system pressure as 6 kg/cm² (tractor hydraulic system pressure) the diameter of the third cylinder can be calculated by using the following formula:

$$\text{Pressure} = \frac{\text{force}}{\text{area}} \quad (1)$$

The remaining two cylinder diameters were selected from the manufacturers chart.

2.2 Design of Rope Drive Mechanism⁶

In this machine, the rope drive is used for rotation of the bucket about 360°. This adds the additional advantage that the operator can harvest oil palm fresh fruit bunches of two rows of trees in a single trip of vehicle. This increases the field capacity of the machine. The power required to drive the rope mechanism was founded out by following equation

$$\text{Power required to turn the bucket} = (T_1 - T_2) V \quad (2)$$

T_1 = tension in the tight side

T_2 = tension in the slack side

$$\text{Then the velocity of the pulley sheave } V_2 = \frac{\pi d_2 N_2}{60}$$

2.3 Weight Transfer, Reactions of Front and Rear Wheels⁷

The front and rear wheel reactions can be found out by using following equations.

$$R_1 = \frac{WX_1}{X_2} - \frac{Y_1 P \cos \alpha + SP \sin \alpha}{X_2} \quad (3)$$

$$R_2 = \frac{W(X_2 - X_1)}{X_2} + \frac{P(Y_1 \cos \alpha + S \sin \alpha)}{X_2} \quad (4)$$

R_1 = reaction at the front wheels.

R_2 = reaction at the rear wheels.

S = the distance from the rear axle to the hitch point of the tractor = 300 mm.

Y_1 = the distance from the point of hitch to the ground = 540 mm.

X_1 = distance of the centre of gravity from the rear wheels = 595 mm.

X_2 = Wheel base = 1785 mm.

W = total weight of the machine = 1900 kg.

2.4 Lateral Stability

2.4.1 Static Condition

In order to ensure stability to the tractor in static condition, the wheel torque should be greater than the torque required to turn the bucket.

$$W.r > F.x \quad (5)$$

Then only it will overcome lateral over turning on the static condition.

If $W \times r$ is less than $F \times x$, the tractor will overturn and to overcome the situation, dead load should be added to the tractor wheel.

Where

F = Load acting on the bucket.

x = Distance from the center of the cylinder to the bucket.

W = Load acting on the tractor rear wheel.

r = radius of the rear wheel.

2.4.2 Dynamic Condition (Steady State Turning)

It is most common that the overturning of the tractor in dynamic condition. In order to overcome this overturning, the tractor should travel within suitable speed. The speed generally depends on the C. G. (Z_{cg}) of the tractor.

This can be calculated by the following equation⁸.

C.G. = (weight of the man $\times$ its distance from the Ground level) + (weight of the telescopic Cylinder $\times$ its distance from the Ground level) + (weight of the sliding bar $\times$ its distance from the Ground level) $\div$ (weight of the man + weight of the telescopic Cylinder + weight of the sliding bar).

Z_{cg} = C.G. of the tractor + displacement of C.G. due to machine weight.

The maximum speed with which the tractor can move safely without overturning could be calculated by⁹

$$u_s = \sqrt{\frac{gAR}{Z_{cg} \cos \gamma}} \quad (6)$$

Where g = acceleration due to gravity,

γ = angle between the assumed force $\frac{m_t u^2}{R}$ and the tipping plane.

$$\gamma = \tan^{-1} \left(\frac{Y_1}{L} \right)$$

$$A = \sqrt{\left(x_{cg} - EL / B \right)^2 + \left(y_{cg} - Ey_1 / B \right)^2}$$

L = wheel base.

A = component of R_1 .

R = turning radius.

2.4.3 Deflection of Sliding Bar

The deflection of the sliding bar was obtained by considering the sliding bar as cantilever beam having load at one end. The deflection of the beam¹⁰ was given as

$$\delta = \frac{Wl^3}{EI} \quad (7)$$

Where

W = Point load acting at the end of sliding bar.

l = length of the bar.

E = modulus of elasticity (cast iron).

I = moment of inertia = $\frac{bd^3}{12}$.

3. Results and Discussions

3.1 Design Parameters of Harvesting Machine

The design parameters namely, selection of hydraulic cylinder, rope drive mechanism and deflection of sliding bar are calculated, by assuming the total and individual weights of the machine.

3.1.1 Selection of Hydraulic Telescopic Cylinder

The hydraulic cylinder was designed by considering the maximum pressure (system pressure). The area of the sleeve cylinder was calculated depending upon the load acting on the third cylinder sleeve. Top sleeve area is calculated and its nearer sleeve area is selected from the manufacturers chart, for the designed system pressure. Velocity of extension is also calculated from the discharge rate of hydraulic pump. The diameters of the first second and third sleeves were found to be 12.065cm, 9.525cm and 7 cm respectively. And the speed for first, second and third cylinders are 0.0779 m/s, 0.0421 m/s and 0.02624 m/s respectively. The closed length of telescopic cylinder was obtained as 1.69 m.

3.1.2 Design of Rope Drive Mechanism

The rope drive mechanism was designed using equations and standard specification. The power required to rotate the bucket is calculated by considering speed. Torque is calculated by considering load acting on the bucket. At this corresponding power diameter of rope sheave is selected as 12.5 cm. By considering the speed ratio diameter of bucket sheave is calculated as 4 cm. The power required to drive bucket by rope drive is 0.08 hp.

3.1.3 Deflection of Sliding Bar

Sliding bar resembles cantilever beam with point load acting at one end of the bar. The material considered is cast iron. The breadth and depth of the sliding bar is calculated for minimum deflection of 2.81 cm by trial and error method using equations given in section 2.4.3. The width and depth of sliding bar are 10 cm and 10 cm respectively.

3.2 Stability of the Designed Harvesting Machine

3.2.1 Front and Rear Wheel Reactions of the Tractor

The front wheel and rear wheel reactions were calculated by using weight transfer concept as given in section 2.3 using equation (3). The front wheel reactions were obtained as 331.19 kg by using equation (4) the rear wheel reaction obtained as 1568.80 kg. In order to prevent the overturning of tractor, the front wheel reaction should be 15% of its total weight (tractor and machine). The 15% weight was achieved by adding 88 kg weight at the front of the tractor.

3.2.2 Weight Transfer to Rear Wheels

In equation (4) the subtracted value from $\frac{WX_1}{X_2}$ is amount

of weight that will be transferred to the rear wheels. The same will add at rear wheel which was given in section. The amount of weight transferred from front to rear wheels is 302.140 kg.

3.2.3 Lateral Stability of the Tractor in Static Condition

The stability of the tractor in static condition was estimated using equation (2.5). According to this equation the torque of the wheel is more than the torque required to

rotate the bucket. Torque ($F \times x$) required for the bucket is calculated as 450 N-m obtained from equation (5). Wheel torque ($w \times r$) is calculated as 458.874 N-m. Hence the tractor mounted with harvester is safe from the lateral overturning.

3.2.4 Lateral Stability of the Tractor in Dynamic Condition

Overturning of tractor in dynamic condition is most common while turning the tractor. This can be reduced by maintaining reasonable speeds. The minimum speed in dynamic condition was calculated using section 2.4.2. The speed was calculated by using equation (6). The speed obtained as 6 kmph. Tractor will be safe from lateral overturning under dynamic condition, if the forward speed of the tractor is less than the calculated forward speed. There are certain parameters which can affect the stability are height of the center of gravity from the ground, turning radius and moment arm. Since the side bucket is mounted on the telescopic cylinder, its center of gravity moves away from the tractor. Due to the height of the telescopic cylinder, center of gravity rises upward from the ground to the height of 7012 mm. The 3D view of the raising platform is shown in Figure 2.

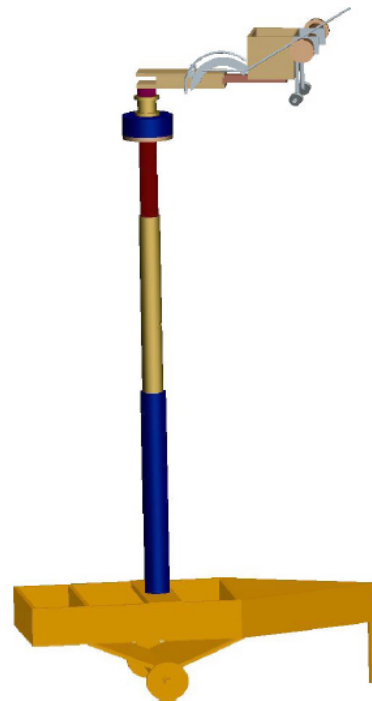


Figure 2. Telescopic raising platform.

3. Conclusions

1. The stability of the tractor along with the harvesting system was established by determining the R_1 and R_2 . The R_1 value should be 15% of the total weight of the machine plus tractor. If it does not exist the weight should be added at the front of the tractor.
2. The total weight transfer to rear wheels of tractor was calculated as 302.140 kg when the implement is attached.
3. In order to ensure stability of the tractor in static condition, it should always satisfy the $W \times x > F \times r$. The design satisfied the above condition, so there is no need of addition of weights at the rear side.
4. In order to ensure stability of the tractor in dynamic condition. The forward speed (u_s) should be within the limit. In this design it was estimated as 6 kmph. This would be the maximum speed which should not be crossed while turning the tractor.
5. The diameter and speeds of the first, second and third cylinders of the telescopic platform are 12.065 cm, 9.525 cm, 7.0 cm and 0.0779 m/s, 0.0421 m/s, 0.02624 m/s respectively.
6. The closed length of the telescopic hydraulic cylinder obtained was 1.63 m, when hydraulic cylinder is in contracted position.
7. The rope drive mechanism requires 0.08 hp. Therefore the manual operator can operate this mechanism without use of motor.
8. The minimum deflection achieved was 2.81 cm. This is the minimum deflection of the sliding bar within the safe limit of its weight of 180 kg for both operator and dead weight of the bucket.

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