

Design and Fabrication of a Manually Operated Rice Transplanter for Local Farmers

Oluwole James Adeyemi¹, Ayodeji David Jakande² and Kamardeen Olajide Abdulrahman³

^{1,2,3}Department of Mechanical Engineering, University of Ilorin, Nigeria

¹jameswa2005@gmail.com; ²jakandeayodeji@gmail.com

³abdulrahman.ko@unilorin.edu.ng

Abstract

Most local farmers who cultivate rice cannot afford the high cost of purchase of imported rice transplanter. The farmers mainly depend on manual hand transplanting method to transplant rice seedlings. This method is less efficient, and highly stressful for the local farmers to depend on to meet the growing demand of food in Nigeria. This work was based on the design and fabrication of a manually operated rice transplanter for local farmers using locally available materials. The design was modeled using Solidworks software (2019) and four bar linkage mechanism was used to achieve this project. The design was easy to fabricate, operate and maintain. It requires less labour stress and reduce or eliminate backache experienced by farmers during rice transplanting process compared to the traditional method. The cost of the manually operated four row transplanter was about ₦100,000, which is more affordable when compared with the imported transplanters that cost about ₦1,000,000 for a four-row motorized transplanter and ₦160,000 for a two row manually operated transplanter

Keywords

farm mechanization, frame, Nigeria, rice seedlings, rice transplanter, tensile stress.

1. Introduction

Agricultural sector plays a significant role in the development of a nation's economy like China, India, and Thailand. (IRRI et al, 2016). Agriculture contribution to Nigeria's Gross Domestic Product (GDP) was 25.13% in 2018 (National Bureau of Statistics, 2019). Although, agriculture significance is huge in the development of a nation's economy and a source of livelihood

for many Nigerians, yet the country is still operating in a mono economic structure; relying mostly on the oil sector for source of revenue. The neglect of the agricultural sector by the federal government, has contributed to the poor growth of Nigeria's economy over the years (Jelilov & Ozden, 2017). Rice is a cereal grain, scientifically known as *Oryza sativa*. It is one of the most popular foods in Nigeria and in the world at large. It is also known as the agricultural commodity with the third-highest worldwide production annually with about 500 million tons currently, after sugarcane and maize (International Rice Research Institute (IRRI), 2020; United States Department of Agriculture, 2021). Over one billion people in the world are involved in rice cultivation, and over three and a half billion people worldwide depend on rice as a main source of food (Molla, 2020). Due to the world reliance on rice, it is essential to implement efficient methods to increase rice production as much as possible. Many countries have experimented with new technologies to obtain improved results (H. B. Patel et al., 2018).

The design and development of rice planting machine which was developed by Patel U. et al., (2017) is shown in Figure 1. in other to support farmers by the conversion of manual process of cultivating rice to mechanical process and therefore solving some problems relating with high cost of labor, and time-consuming process. The simple mechanism designed was also eco-friendly. The machine was designed to plant two rows simultaneously.

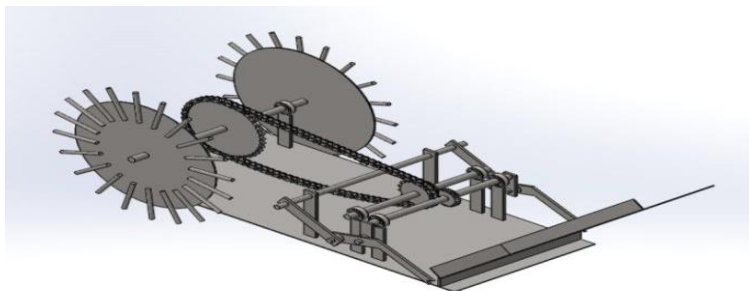


Figure 1: Isometric view of Rice Seedlings Planting Machine (Patel U. et al., 2017)

Saiyed et al., (2018) study focused on design, analysis and fabrication of an automatic rice transplanting machine for small scale Indian rice cultivators as

shown in Figure 2. Link mechanism was used to simplify cultivation process and make their work easier and accurate; it also reduced cost of labour and reduced the working time as well as increases the comfort level in farming. Although the working process of the machine designed was satisfactory, its cost of purchase and maintenance for local farmers was still very high and this remain a serious problem to most local farmers as cost of importation of such machine becomes high and not easily affordable by the local rice farmers.

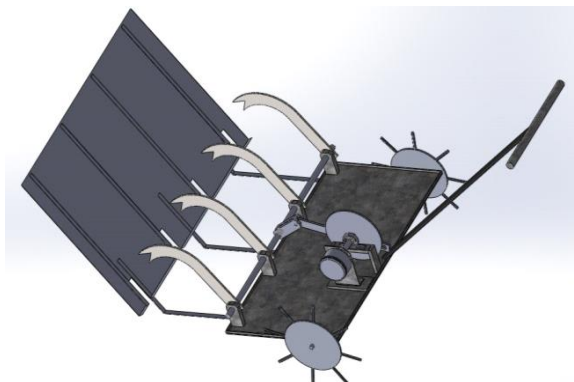


Figure 2: A 3D Isometric view of automatic rice seedlings transplanting machine (Saiyed et al., 2018).

The demand of rice crops has continued to increase without an inadequate production and supply of locally produced rice of about 4.0 million metric tons in 2018, which has led the country to depend heavily on rice importation of over 3 million tons annually, equivalent to over US\$480 million in infrequent foreign exchange in order to meet up with the rising demand of about 7 million tons of the commodity annually (BBC, 2019; BBC News, 2019; Kamai et al., 2020). The use of traditional practice for planting rice seedlings has not yielded much result in solving this problem. The traditional method is tedious and causes farmers lot of body pains and has also affected their product output (Ayedun & Adeniyi, 2019).

In the bid to solve these problems, it is necessary for farmers to be introduced to the new strategies in solving these problems through farm mechanization as highlighted by Haider et al.(2019). In this research study, rice seedlings transplanter was designed and manufactured with locally available materials to make it affordable for local farmers, when compared to the motorized

transplanter and imported manually operated two rows transplanter which are ₦1,000,000 and ₦160,000 as shown in Figure 3A and 3B respectively and to also minimize the amount of seeds used in planting and reduce the cost of rice cultivation. The modeling of the transplanter was achieved with Solidworks software (2019).



Figure 3A: Manually
Operated Two Rows Rice
Seedling Transplanter
(www.jiji.ng).



Figure 3B: Motorised Four
Rows Rice Seedling
Transplanter (www.jiji.ng).

2. Materials and Method

2.1 Material Selection

The materials used were selected based on design requirements, (mechanical properties of the material, availability, cost, ease of construction, etc.). The selected materials used are shown in Figure 4. The machine was designed and constructed to transplant rice seedlings for domestic and commercial consumption. The machine consists of the ground wheel, chain and sprockets, four bar links, tray, frame, shaft, handle, and planting finger.

Name of the component	Material size (mm)	Material
Frame	38x38x2.5(thickness) And 3.8 mm sheet metal.	Mild steel angle bar and sheet metal.
Shaft	25 mm	Mild steel rod.
Handle	Length- 895, pipe diameter-20.	Mild steel circular pipe.
Ground wheel	Diameter- 18 mm	Mild-steel rod.
Four-bar links	40 mm x 6.5 mm (thickness).	Mild steel flat bar.
Large sprocket	No of teeth-42 and diameter-190	Medium carbon steel.
Small sprocket	No of teeth-14 and diameter-78	Medium carbon steel.
Pintel chain	No of links-102	Malleable links.
Pillow bearing	Diameter-25	Bronze.
Tray	2 mm thickness	Galvanized Steel sheet.
Bolt and nuts	10 mm x 20 mm 13 mm x 30 mm, 17 mm x 50 mm, 17 mm x 70 mm, 19 mm x 70 mm.	Cast iron.
Washers	17 mm	Cast iron.
Total weight	Approximation 70 kg	

Figure 4: The components of the designed transplanter and materials selected

2.2 Design Analysis and Calculation

2.2.1 Calculating the Power Developed by the Operator

The required power (HP) of useful work done by an operator on the machine is given as (Khan et al., 2015):

$$HP = 0.35 - 0.092 \log t \dots \dots \dots (1)$$

Where the operation time in minutes is t. The power developed by the operator when considered to work for an average of 3 hours was 0.143 hp (Khan et al., 2015).

2.2.2 The Determination of Velocity Ratio of the Chain Drive

The selected number of teeth for the pinion sprocket (smaller sprocket) and the larger sprocket are 14 and 42 respectively. The velocity ratio (V.R) of the chain as given in equation (2) (KHURMI & GUPTA, 2005).

$$V.R = \frac{N_1}{N_2} = \frac{T_2}{T_1} \dots \dots \dots (2)$$

Where, speed of rotation of smaller sprocket in (rev/min) is N_1 , speed of rotation of larger sprocket (rev/min) is N_2 , number of teeth of the smaller sprocket is T_1 , and number of teeth of the larger sprocket is T_2 .

2.2.3 Determination of the Design Power of the Chain Drive

The design power of the chain drive (P_D) is the product of rated power obtained from equation (1) and the service factor K_S

$$P_D = H.P \times K_S \dots \dots \dots . (3)$$

Where, Service factor K_S is the product of load factors (K_1), lubrication factor (K_2), and rating factor (K_3). (KHURMI & GUPTA, 2005)

$$K_S = K_1 \times K_2 \times K_3 \dots \dots \dots (2)$$

Where, H.P = 105.105 W, variable load with mild stock (K_1) = 1.25, periodic lubrication (K_2) = 1.5, continuous service (K_3) = 1.5.

2.2.4 Calculation of Speed of Driving sprocket (N_1)

Speed of driving sprocket (N_2) was given by

$$N_2 = \frac{(w \times 60)}{2\pi} \dots \dots \dots (5)$$

$$\omega = \frac{v_{avg}}{r} \quad \dots \dots \dots (3)$$

Where, ω is angular velocity in rad/s, v_{avg} is the average walking speed of a man in m/s and r is the radius of the wheel in m. The moving velocity taken for the operator was 2.5 Km/h or 0.7 m/s obtained from Pal et al. (2018). The wheel diameter (D_w) considered was 580 mm.

2.2.5 Determination of the speed of Pinion Sprocket (N_1)

The pinion sprocket is used as the driven sprocket to transmit the power from the chain drive to the planting mechanism. From equation (2) the speed of the pinion sprocket (N_1) is 69.15 rpm, which is close to standard speed of 100 rpm, the power transmitted for chain (No 06) according to international standard of 0.25 kW since the design power of 0.295 kW was obtained from equation (3). Therefore, two chains of No 06 can be employed to transmit the needed power with one strand each producing the total power of 0.5 kW. Therefore, the pitch (p), the roller diameter (d_r), the width of roller (w_r) and the breaking load (W_B) was determined as 9.525 mm, 6.35 mm, 5.72 mm, 8.9 kN/chain respectively according to international standard (KHURMI & GUPTA, 2005).

2.2.6 Determination of the Pitch Circle Diameter of the Pinion Sprocket

The pitch circle diameter (d_{p1}) of the pinion sprocket was obtained with the equation from Bhat et al. (2017):

$$d_{p1} = \frac{p}{\sin\left[\frac{180}{T_1}\right]} \quad \dots \dots \dots (4)$$

2.2.7 Calculating the Pitch Line Velocity of the Pinion Sprocket

The pitch line velocity of the pinion (v_{p1}) was obtained as (KHURMI & GUPTA, 2005):

$$v_{p1} = \frac{\pi d_{p1} N_1}{60} \quad \dots \dots \dots (5)$$

2.2.8 Calculating load on the chain (W_c)

The load on the chain was obtained as (Bhat et al., 2017):

$$W_c = \frac{\text{Rated power (HP)}}{\text{Pitch line velocity}(v_{p1})} \dots \dots \dots (6)$$

$$W_c = 678.097 \text{ N}$$

$$\therefore \text{factor of safety of the chain drive} = \frac{W_B}{W_c} \dots \dots \dots (7)$$

The load on the chain and the factor of safety of the chain from eqn. (9) and eqn. (10) respectively was obtained as 678.097 N and 26.2499. For a good design, the lowest centre distance between a small and a large sprockets should be 30 to 50 times the pitch value (KHURMI & GUPTA, 2005). Therefore, centre distance between the sprockets (x) was taken as 30 multiplied by pitch to obtain the value of 285.75 mm apart. In order to take care of initial sag in the chain, the centre distance value could be reduced within 2-5 mm (Bhat et al., 2017). Therefore, the centre distance obtained was 282 mm when reduced by 3.75.

2.2.9 Determination of number of chain links (K_l)

The number of chain links (K_l) was obtained from eqn. (11) as given by Khurmi and Gupta, (2005).

$$K_l = \frac{T_1 + T_2}{2} + \frac{2x}{p} + \left[\frac{T_1 - T_2}{2\pi} \right]^2 \times \frac{p}{x} \dots \dots \dots (8)$$

The length of chain was determined from eqn. (12)

$$L = K_l \times p \dots \dots \dots (9)$$

2.3 Design of Shaft

2.3.1 Calculating the Pulling Force (F)

Based on the work of Pal et al (2018) the moving velocity of the operator is 2.5 Km/h which is equal to 0.7 m/s. Horse Power (HP) obtained from eqn. (1) is the same as power transmitted (P). The transmitted power (P) by a chain drive is (Khan et al., 2015; Pal et al., 2018)

$$P = \frac{F_{pull} \times v_{avg}}{75} \dots \dots \dots (10)$$

Where, F_{pull} = The pull force and v_{avg} = The speed of the machine which is the same as the operating speed of the machine is obtain as follows:

$$F_{pull} = \frac{P \times 75}{v_{avg}} \dots \dots \dots (11)$$

$$F_{pull} = 15.32 \text{ Kgf}$$

$$F_{pull} = 150.3 \text{ N}$$

2.3.2 Calculating the Forces acting On the Wheel Shaft

Assuming the mass of rice seedlings to be transplanted as 10 kg and the mass of the frame and other elements of the planter apart from the wheel and the wheel shaft as 60 kg. Therefore, total mass acting on the wheel (m_f) = 70 kg
Total force acting on the wheel shaft was computed using eqn. (15)

$$F_t = m_f \times g \dots \dots \dots (12)$$

Where, acceleration due to gravity (g) is 9.81 m/s

$$F_t = 686.7 \text{ N}$$

2.3.3 Determination of shaft diameter (d)

The acting forces on the shaft were determined by considering the total load (F_t) of 686.7 and the pulling force (F_{pull}) of 150.3 N. according to maximum shear stress theory (Bhowmik et al., 2016).

$$d^3 = \frac{16}{\pi \tau_{max}} T_e \dots \dots \dots (13)$$

$$T_e = \sqrt{(K_m M_G)^2 + (K_t T)^2} \dots \dots \dots (14)$$

Where, T_e is equivalent twisting moment in Nmm, d is the diameter of the shaft in mm, τ_{max} is the maximum shear stress in N/mm^2 , M is the maximum

bending moment in Nmm, T is the maximum torsional moment in Nmm, K_m is Combined shock and fatigue factor for bending moment, and K_t is Combined shock and fatigue factor for twisting moment. Considering the equation of drive machine in order to obtain the torque (T) acting on the shaft (Bhowmik et al., 2018);

$$P = \frac{2\pi N_2 T}{60} \quad \dots\dots\dots (15)$$

From eqn. (18), making the torque (T) the subject of the formula to obtain eqn. (19)

$$T = \frac{P \times 60}{2\pi N_2} \quad \dots\dots\dots (16)$$

Where, T is torque acting the shaft wheel, P is power Transmitted, and N_1 is Speed of driving wheel at revolution per minute. P is equal to the Rated power (HP) which is 105.105 W.

$$T = \frac{105.105W \times 60}{2\pi \times 23.05}$$

$$T = 43.5435 \text{ Nm} = 4354.35 \text{ Nmm}$$

Considering a suddenly applied load with minor shocks only combined shock and fatigue for bending moment (K_m) selected was 2, combined shock and fatigue factor for twisting moment (K_t) selected was 1.5 and the maximum permissible shear stress (τ_{max}) used was 42 MPa (KHURMI & GUPTA, 2005). The values obtained from eqn. (18) and eqn. (19) was substituted into eqn. (17) to obtain the value of equivalent twisting moment (T_e) as 112644.0606 Nmm. The maximum bending moment (M_b) on the shaft was obtained as 57227.27 Nmm with the aid of share force and bending moment diagrams. The diameter of shaft obtained from eqn (16) was 23.9043 mm but 25 mm diameter was used as the wheel shaft based on market availability.

2.4 Design of the Frame

The frame is considered to be one of the significant structures of any appliances that give fundamental support for other components and load placed upon it.

The frame was made from mild steel angle bar of $(38 \times 38 \times 2.5)$ mm. The frame has two beams welded together with a cross beam at both ends as shown in Figure 5 and on the other end welded to a mild steel flat plate as shown in Figure 6 which support the load and the planting mechanism. The load acting on the frame chassis is approximately 686.7 N as obtained from eqn. (15).

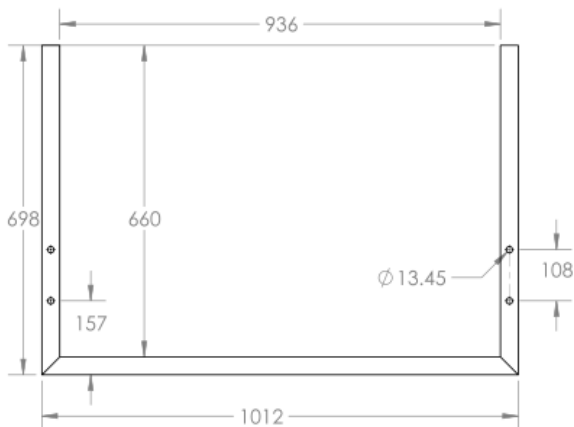


Figure 5: Frame of the rice transplanter

2.4.1 Calculation Of Stress and Strain on the Frame

The total measured length of required frame hollow square tube,
 $L = 2332 \text{ mm}$

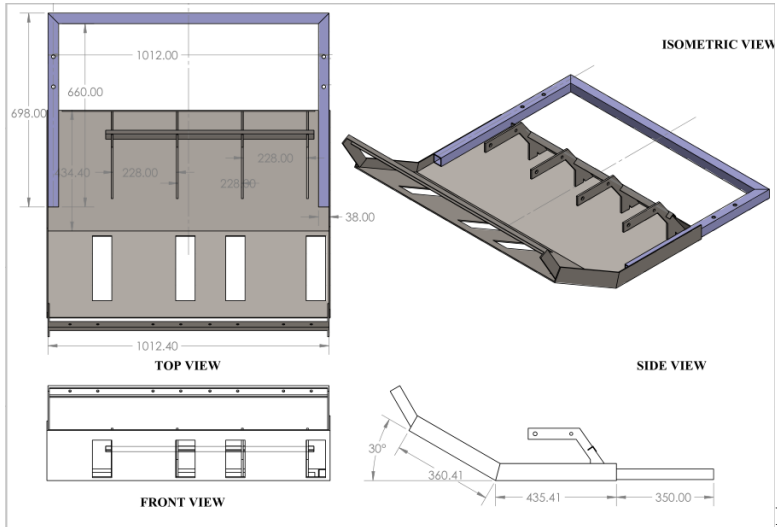
Total surface area of frame (A_s),

$$A_s = L \times (w) \dots \dots \dots (17)$$

Where, w = the width of the beam and $A_s = 2332 \times 38 = 88616 \text{ mm}^2$

Volume of the frame,

$$V = 2332 \times 38 \times 2.5 \times 2 + 2332 \times 33 \times 2.5 \times 2 - \frac{8\pi d^2 h}{4}$$



Figur

e. 6: The views of the manual rice transplanter frame.

Where, the diameter of the cut-out hole from the frame (d) is 13.45 mm to fix the pillow bearing of the shaft wheel and the thickness of the frame beam (h) is 2.5 mm. $V = 825018.3902 \text{ mm}^3$. The properties of material used,

AISI 1045 (Abdulkarim et al., 2017):

Density of the material, $\rho = 7850 \text{ kg/m}^3$

Young modulus of the material, $E = 205 \text{ GPa}$

Yield strength: $530 \times 10^6 \text{ N/m}^2$

Mass of the frame, $M_f = 6476.3944 \text{ grams}$

Total weight on the frame, $W = 70 \text{ kg} = 70 \times 9.81 = 686.7 \text{ N}$

2.4.2 Calculating the Moment of Inertia (I)

The moment of Inertia (**I**) is obtained from equation (21)

$$I = \frac{bd^3}{12} - \frac{b_1d_1^3}{12} \quad \dots \dots \dots (18)$$

Where, b is the outer width of the beam cross section, d is the outer depth of the beam cross section, b_1 is the inner width of the beam cross section and d_1 is the inner depth of the beam cross section. The moment of Inertia (I) obtained from equation (21) was 74934.5833 mm^4 . The stress and strain acting on the frame obtained using basic engineering equations were 0.0077 N/mm^2 and 3.7561×10^{-8} respectively. The Calculation of the deflection (y) of the beam considering a uniformly distributed load in the frame using equation 22 is 7.407 mm .

$$y = \frac{5w_f L^4}{384 \times EI} \quad . \quad (22)$$

3. Result and Discussion

The rice seedling transplanter was made on the basis of simple machine design knowledge to design the parts such as the shafts, chain and sprocket assembly and frame with the use of standard formulas. The use of locally available materials aids the fabrication of the transplanter with minimum cost affordable for small scale farmers.

Calculations of stress, strain, deflection of the frame and the total power required by the manually operated rice seedling transplanter was done. From the calculated results obtained, it was observed that the frame chassis will definitely withstand the design load. The calculated stress, strain and deflection on the frame using steel material AISI 1045 with yield strength of 530 MN/m^2 , gave $7.7 \times 10^{-3} \text{ N/m}^2$, 3.7561×10^{-8} and 7.407 mm respectively. Given a low strain value when compared with previous work of Abdulrahman and Yahaya (2016) with frame strain of 9.628×10^{-8} and other similar works, indicate that the chassis should be able to withstand the applied load.

The selection of material for the rice seedling transplanter was done based on its availability in the market. The materials majorly used were mild steel and galvanized iron.

4. Conclusion

The rice transplanter was designed and fabricated using available locally sourced materials and was found to be satisfactory. The selection of four bar mechanism was effective and simple to fabricate. The design calculation results showed that the chain drive will be able to withstand the design load of 678.10 N. It also revealed a deflection of 7.407 mm when the chassis was subjected to maximum load of 686.7 N. The cost of the transplanter of about ₦100, 000, which is low when compared to the imported rice seedling transplanters. Four rows of seedling could be achieved using this machine. While on the other hand, the machine is easy to operate, faster and saves local farmers the stress in transplanting rice seedlings.

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