Development of Power Weeder for 3 Row-Planted Paddy

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Abstract **–Weeds are the major reason for the significant yield reduction problems in rice cultivation of Sri Lanka. Power weeders have been introduced to rice cultivation as an alternative solution for the controversial herbicide application. The effectiveness of existing power weeders is also low due to the associated rotary mechanism. Besides, "Asakura wooden clog" has been identified as an appropriate weeding mechanism, which could be developed as a power weeder. Therefore, this research was aimed to introduce an appropriate lowland power weeder, especially for medium and large-scale rice farmers in Sri Lanka using the weeding mechanism of manual "Asakura wooden clog". comprehensive design calculations, fabrications, series of performance tests and modifications were carried out to achieve this goal. Final version of prototype consists of power source, frame and separate mechanisms for power transmission, weed burying, turning/row changing, floating, manipulation and controlling which are facilitated to bare the activated load, burying the weeds, achieve the required tractive power, speed and machine control in road and field manipulation. Besides, it attained satisfactory field performances; 0.03 ha h^{-1} effective field capacity, 83.25% field efficiency, 80% weeding efficiency, 6.34% damaged plants, 580 performance index. Further, calculated cost for weeding, fuel consumption, labour and power requirement were 38.355 USD ha^{-1}, 0.503 L h^{-1}, 33 man-h ha^{-1} and 0.319 kW, respectively. Further, no ergonomic or mechanical defects were reported during the performance testing. Thus, this prototype has a potential to develop as an appropriate machinery for weeding processes in medium and large-scale, row planted rice cultivations in Sri Lanka.**

Keywords **- Asakura wooden clog, burial type weeder, design farm machinery, performance test, 3 rows paddy, weed control**

I. INTRODUCTION

Weed is one of the most important agricultural problems in rice cultivation and its competitive nature causes serious negative effects in rice production and considerable marketed losses in rice yield in the range from 10 – 50% to 50 – 90 % [1,2]. Similarly, weeding is a labor-intensive agricultural unit operation in rice cultivation and it accounts for about 25 % of total labour requirement [3] and 15.3 – 23.7% of the total farm power requirement [4]. This apparent rice yield and power loss is due to unrestricted weed competition and subsequently required huge labor forces which are unbearable and it affects defectively to the rice production in Sri Lanka. Consequently, introduction of appropriate weeding machineries has become an imperative prerequisite to enhance the rice production.

Chemical methods of weed control had gained recognition among medium and large-scale rice farmers in Sri Lanka, over other existing methods which are laborious, arduous, time consuming, leading to higher cost of production and therefore, specially limited to small scale farming. Nevertheless, excessive utilization of these agro-chemicals, leads to negative impact on the environment and human health. As a result, a new trend of minimizing the agro-chemical usage has been encouraged among farmers and agricultural policy makers. Correspondingly, some hazardous herbicides and fertilizers were banded recently in Sri Lanka to lessen the potential health impact. However, there should be an appropriate alternative approach to control weeds in rice farming with minimum environmental effect.

Because of these reasons as well as concern over the environmental degradation and growing demand for organically produced food, mechanical method of weed control is imperative [5,6]. It is very effective [7], eliminates drudgery [8] and also keeps the soil surface loosen for better soil aeration and water holding capacity [9,10] which lead to increase the potential yield of rice [2].

Therefore, Department of Agriculture (DOA), Sri Lanka, is promoting the usage of mechanical power weeders for medium and large-scale paddy farmers as an alternative approach to chemical weed control. Similarly, it has been considered as a solution of weed control for mechanically transplanted paddy fields by introducing mechanical power transplanters. As a result of that, several power weeders have been imported and distributed among paddy farmers. Most of them consist of rotary action and they have not much been popular among Sri Lankan farmers.

Presently there are many types of mechanical weeders available from simple to complex and motorized weeders that use three main techniques; (a) burying weeds, (b) cutting weeds and (c) uprooting weeds [8]. Machineries for cutting and uprooting weeds are readily available but weed burial machineries are very rear [11]. "Asakura wooden clog" is a lowland weeder which uses burial weed controlling mechanism [12].

Therefore, this study was to design, development, fabrication of an appropriate burial type lowland power weeder especially for the medium and large-scale rice farmers in Sri Lanka and test its field performances. Further, it was hypothesized that this would be achieved by developing the weeding mechanism of manual "Asakura wooden clog".

Results of this study showing the pros and cons of new design and test performances were used to make recommendation and required modifications. Hence this output would be beneficial for farmers, researchers, farm machinery producers and policy makers in various approaches.

II. MATERIAL AND METHODS

The design process was started out by listing out the design goals and choosing the working mechanism by analyzing and discussing several working models.

The design requirements for other components of the power weeder were discussed; including the frame, the weeding mechanism and power transmission system. The first prototype was fabricated and tested. Revision and modifications were done to the design. Ultimately, final version was developed, discussed and fabricated.

The machine design, development, prototype fabrication, and performance testing were done at the Engineering Workshop and the Research Unit of Faculty of Agriculture, Rajarata University of Sri Lanka, Puliyankulama, Anuradhapura. AutoCAD 2014 software was used to prepare the engineering drawings.

2.1. Development of Conceptual Design

The main target of this research project was to mechanize the present day used manual "Asakura wooden clog" as a power weeder. Hence, the main achievement of this design was to represent the walking pattern of human beings by mechanical means. This special mechanical movement was demonstrated by two working models consisting of single and double crank shafts, respectively. By studying and analyzing kinematics and their movement pattern, double crank shaft model was identified for the development (Fig.:1).

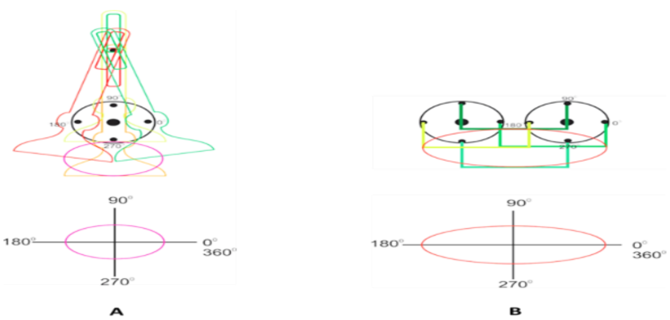


Fig. 1 Block diagrams and kinematics of working models; A - single crank shaft and B - double crank shaft

2.2. Design Parameters

Following design parameters were Considered; machine weight (<60 kg), plant spacing (inter row; 30 cm,), overlapping percentage of the weeding clog (100%), weed burying depth (5 cm), time of field application (3 - 5 WAT/Weeks After Transplant; [12]), puddle resistance (218.07 kN m^{-2}; [13]) ground clearance (27 cm), forward speed of the machine (0.20 kmh^{-1}; [14,15]), number of rows to be weeded simultaneously (3; [15]), and number of operators (1).

2.2.1. Design Specifications:

Design specifications were; (a) easy maneuvering with good tractive mechanism hence, the weeder can overcome the mobility problems in puddled fields as well as the road transportation; (b) floating mechanism to bare the dead load in muddy fields; (c) consist with good turning and row changing mechanism; In the road transportation and raw changing in the field, operator has to operate the weeder as a wheel-barrow (second class lever) using handle and the front wheel.

Hence, the machine should be light weight, low cost and appropriate for the one operator manipulation. It should have flexibility on various filed sizes and field shape irregularities, machine performances should be satisfactory and appropriate for medium scale rice farms in Sri Lanka. Moreover, it should provide safety to users and the product should last a long duration.

2.3. Machine Descriptions

Basically, power weeder consists of power source (engine), frame, power transmission mechanism, weeding mechanism, turning and row changing mechanism, floating mechanism, manipulation and controlling mechanism. (Fig. 2).

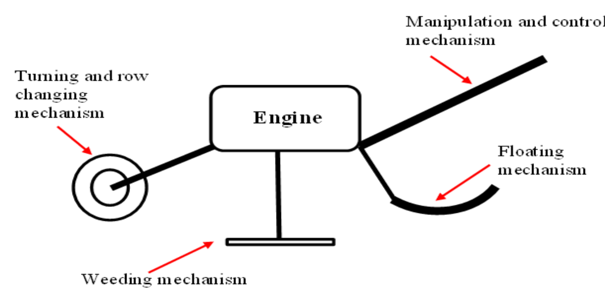


Fig. 2 Block diagram of proposed model

2.3.1. Power Source (Engine):

Hand priming start, light weight, 4-stroke, air cool, petrol engine with enough rated power was selected by considering all practical limitations and technical specifications. However, a gear box is also cupelled to achieve required speed reduction.

2.3.2. Frame:

The frame was designed to maintain the proper relative positions of the units and parts mounted on it over a long period of time and service under all the working conditions such as field manipulation and road transportation with enough strength.

2.3.3. Power Transmission Mechanism:

Power transmission mechanism was designed to provide enough tractive power by reducing the speed. The power from engine is transmitted to the gear box through belt drive which use as the overload prevention mechanism and clutch system by using a tension pulley. Then, a chain drive is linked to actuate and power is transmitted from gear box to the crank shafts (Nos. 04) which are mounted on bearings. When crank shaft rotates, weeding clogs which are attached to the crank pins are driven. Then, weeder moves forward and the weeds are buried simultaneously (Fig. 3).

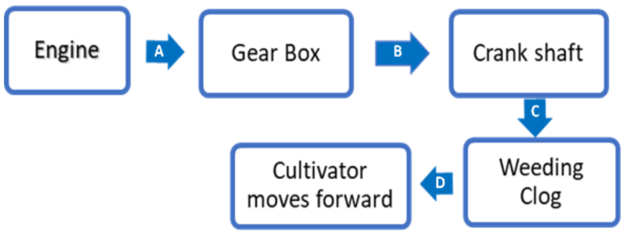


Fig. 3 Power cycle of operation; A – belt drive consists with clutch system, B – chain drive, C – direct drive

2.3.4. Weeding Mechanism:

The weeding clogs are used to bury the weeds and develop the tractive force. It is consisted with three weeding clogs as same as modified manual "Asakura wooden clog" [12] to bare the dead load of this proposed design in puddled fields.

2.3.5. Turning and Row Changing Mechanism:

In order to ensure continuous weeding, it is necessary to shift the machine from one set of rows to another (turning in field) without damaging the rice plants. This mechanism was invented effectively using the front wheel. In addition, the front wheel is used as a ground wheel in road transportation and as a floating device during field manipulation. Therefore, wider a rubber tire with steel hub fitted to a fork was proposed as the turning and the row changing mechanism of this machine.

2.3.6. Floating Mechanism:

The rear skidders were designed as floating mechanism, as it helps to float the machine in puddled rice fields without sinking. In addition, the front wheel and weeding clogs are also partially act as floating devices. Height adjustment of rear skidders has been achieved by using a simple arrangement with hand grip.

2.3.7. Manipulation and Controlling Mechanism:

The handle is used to manipulate and control the machine easily by the operator. Therefore, an arrangement was also made to adjust the height and the angle of the handle as per the need and posture of the operator. The handle was designed to make easy to lift and guide the machine. Furthermore, throttle controller and hand clutch were also fixed to the handle.

2.4. Design Description

Instead of the road transportation, this weeder has been designed for forward moving and turning (row changing) in muddy field conditions. In forward operation also two phases could be identified;(i) when the crank is at the dead centre and (ii) when the crank is at an angle (assume 90º). At the phase i, the load activated through the weeding clog should be adequate to penetrate the surface layer and in equilibrium with the soil penetration resistance (PR) of a soil layer at 5 cm (Fig. 4). At phase ii, no activated load through weeding clogs, but front wheel and rear skidder reaction should be enough to bare the dead load of the weeder (Fig. 5).

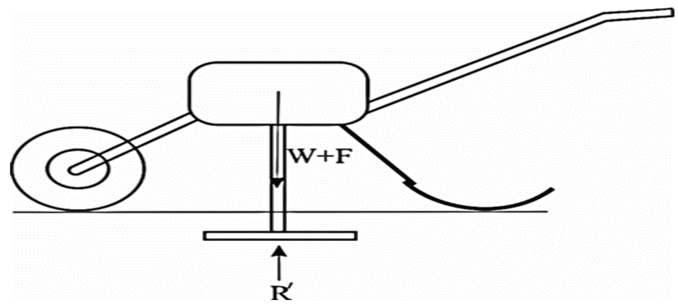


Fig. 4 Force system diagram of prototype (when crank shaft is at dead centre in field manipulation) R' – soil reaction of 5 cm depth soil layer, W – dead load of the weeder and F – force excreted by the prime mover - Impact load

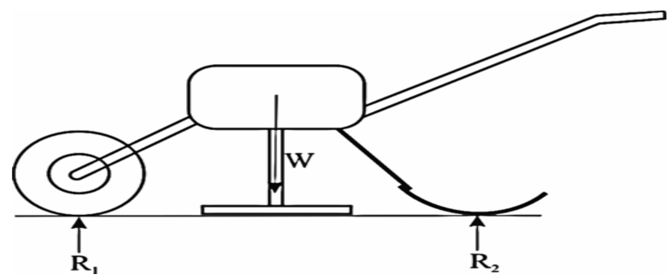


Fig. 5 Force system diagram of prototype (when the crank shaft is at an angle (90º) in filed manipulation) R_1 – soil reaction through front wheel, R_2 – soil reaction through rear skidders and W – dead load of the weeder

Besides, the tractive force of the design is developed by the weeding clog due to the shear strength of the puddled soil. In row changing or turning in field manipulation and road transportation, the dead load of the weeder is partially activated on the front wheel. Hence, front wheel should be wider enough to float the machine on the surface layer with its PR (Fig. 6).

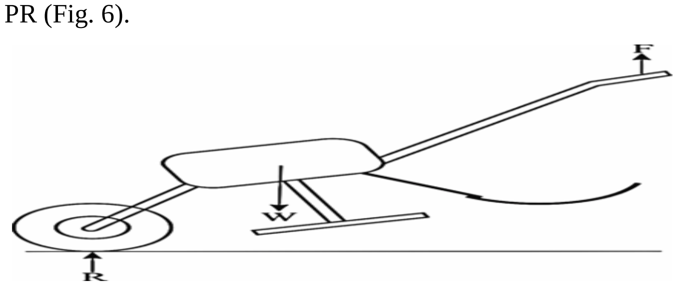


Fig. 6 Force system diagram of prototype (row changing/turning or road transportation) R – soil reaction in surface layer, W – dead load of weeder and F – operator's effort

2.4.1. Designing of Weeding Clogs:

The width of the weeding clog may directly proportional to the weeding efficiency and the plant damage percentage. Hence, it has become a critical value in weeder designing. The overall width of the clog was set as 20 cm (67% of total inter-row space). Weed burying cross strips were fabricated with 30 mm distance [12].

For the length determination of the clog, machine forward distance per one rotation of crank shaft was calculated by Equation 1.

S = θr -- *(Eq.1)*

where;

S = Arc length
θ = Soil contact angle in radians (Assume as 90º)
r = Web length of the crank shaft (5 cm)
S = π/2 × 5 = 7.854 cm

Therefore, machine moving distance during one rotation of crankshaft = 7.854 × 2 = 15.708 cm

For achieving 100 % overlapping percentage, clog length should be doubled as moving distance

Therefore, required clog length = 15.708 × 2

= 31.416 cm ≈ 30 cm

Ultimately, 30 × 20 cm clogs with 30 mm cross strips were suggested. This weeder has three weeding clogs. During field operation, the centre clog and two side clogs are contacting the puddled soil, rotationally. Therefore, it was required to design the centre clog with doubled contact area as side clogs to achieve smooth field operation. However, weeding clogs should penetrate through the surface layer of the puddle soil and retain an equilibrium with PR of sub surface layer (5 cm).

Considering the material availability, their strength and soil penetration ability, MS rods (Ø = 3.7 mm and 8 mm) and 20 mm MS flat iron (5 mm and 10 mm thick) were used for fabrication of weeding strips and the frame of the weeding clogs, respectively (Fig. 7).

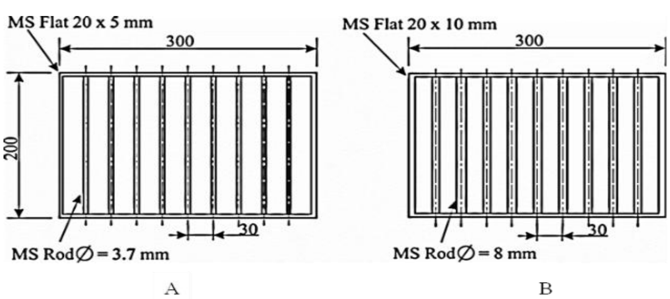


Fig. 7 Plan view of weeding clog; A - side clogs and B - centre clogs

Calculated soil contact area of proposed side and centre clogs are 11,227 mm^2 and 22,560 mm^2, respectively which is approximately doubled in contact area in centre clog. The legs of weeding units were fabricated with 20 mm × 20 mm (2 mm) MS box iron and MS bushes (25 mm OD) which are used to attach to the crank pin (Fig. 8).

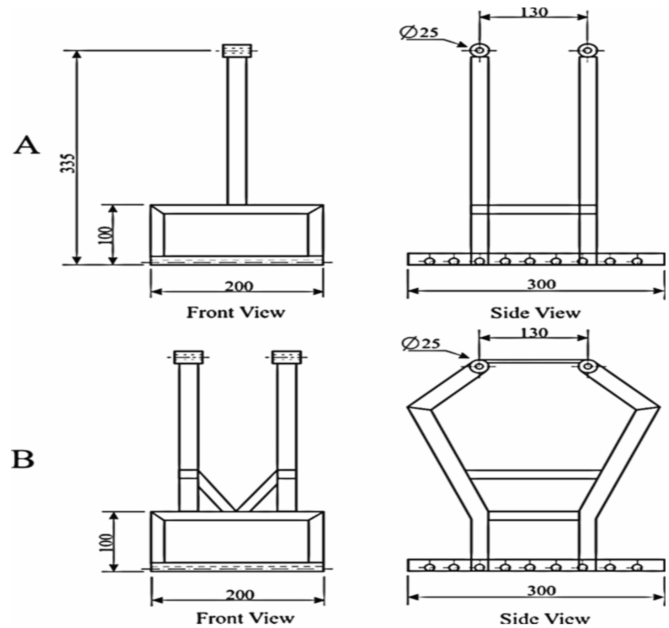


Fig. 8 Front and side views of weeding clogs A - side clog and B - centre clog

2.4.2. Power Requirement:

The highest power requirement (impact load) was observed, when the crank shaft was at dead position, in forward field manipulation (Fig. 4). Considering the equilibrium of the system;

ΣFy↑+ = 0
R' – (W+F) = 0
F = R' – W

As considering average puddle resistance of sub surface (5 cm depth) at 3 -5 WAT of rice fields is 218.07 kN m^{-2} [13] and the foot area of centre clog is 22,560 mm^2;

R' = Foot area × PR
= 22,560 × 10-6 × 218.07 × 103
= 4,919.659 N

Assuming dead load of the machine is 60 kgf (588.6 N ≈ 600 N)

F = 4,919.659 – 600
= 4,319.659 N

The required force from the prime mover (impact load) is 4.32 kN. Assuming, this force directly and equally activated on crank shafts (Nos. 04), the torque on the crank was calculated by Equation 2.

TOR = Fd -- *(Eq. 2)*

where;

TOR = Torque on crank shaft
F = Impact load (4,319.659/4 = 1,079.915 N)
d = Crank web length/Displacement

Weed burying depth of this weeder was considered as 5 cm. Hence crank web leg length was taken as 5 cm.

TOR = 1,079.915 × 0.05 = 53.996 Nm

The power of the crank shaft was calculated by Equation 3 [15].

P = TOR × 2π N/60 ---------------------------------- *(Eq. 3)*

where,

P = Power of crank shaft (W)
TOR = Torque of crank shaft (Nm)
N = rpm

Appropriate forward speed of this weeder was considered as 0.2 kmh^{-1}. Hence, forward theoretical distance per one rotation of crank shaft (d_1);

d_1 = 2 × web length
= 2 × 0.05 = 0.1 m

If the travel reduction is 50 %, actual distance per one rotation of crank shaft (d_2);

$d_2 = d_1$ × 50 %
= 0.05 m

Speed 0.2 kmh^{-1} = 0.2 ×103 m 60 min^{-1}

= 3.333 $m.min^{-1}$

Therefore, Required speed of crank shaft = speed/d_2
= 3.333/0.05 = 66.67 rpm
From Equation 3, P = 53.996 × 2π × 66.67/60
= 376.982 W
For crank shafts Nos. 04; P = 376.982 × 4
= 1,507.928 W = 1.508 kW

By adding 4.5% power excess, presently available single cylinder, 4-stroke, air cooled, petrol engine with rated power (1,800 rpm) 1.57 kW (2.1 hp) was selected. Further, spare parts availability, maintenance facility and after sale service were also considered.

2.4.3. Designing of Power Transmission System:

This design was based on the rated speed of selected engine (1800 rpm) and required output speed of crank-shaft (66.67 rpm). The total speed reduction of the transmission system was calculated as per the Equation 4.

Total speed reduction = $\frac{Engine\ rated\ speed}{Final\ drive\ speed}$ ---------*(Eq.4)* [16]

$= \frac{1800}{66.67} = 27{:}1$

Hence, this power transmission system, which included a belt drive (A), gear box, and chain drive (B) was designed to get total speed reduction of 27:1 (Fig. 3).

2.4.4. Selection of Gear Box:

Considering the market availability, price, required space (shape) and the speed reduction, a parallel line gear box with 1:20 velocity ratio was selected. Moreover, the gear box gives 1:10 velocity ratio as its highest out-put. It may be useful for high-speed operation condition in the field such as 0.4 km h^{-1}. If the lower gear ratio is used to achieve the required speed of crank shaft (66.67 rpm), input speed of the gear box should be; 66.67 × 20 = 1,333.4 rpm.

2.4.5. Designing of the Belt Drive: With referring the range of power transferring, a V shape (B type) belt and pulley system were proposed to reduce the engine rated speed (1,800 rpm) to input speed of the gear box (1,333.4 rpm). The velocity ratio of this belt and pulley system was calculated using Equation 5.

Velocity ratio = $\frac{N_2}{N_1} = \frac{d_1}{d_2}$ ---------------------------- *(Eq. 5)* [16]

where;

d_1 = Diameter of the driving pulley (cm)
d_2 = Diameter of the driven pulley (cm))
N_1 = Revolution speed of the driving pulley (rpm)
N_2 = Revolution speed of the driven pulley (rpm)

Therefore, velocity ratio = 1333.4/1800 = 0.74

The size of the driven pulley was determined by Saverin's formula (Equation 6);

d = (525 ~ 630) × $\left(\frac{P}{\omega}\right)^{\frac{1}{3}}$------------------------------ *(Eq.6)* [17]
where;

d = Diameter of the driving pulley (mm)
P = Transmitted power (kW)
ω = Angular velocity (rad s^{-1})

With, $\omega = \frac{2\pi \times 1800}{60}$ = 188.495 rad s^{-1}

d = 525 × $\left(\frac{1.57}{188.495}\right)^{\frac{1}{3}}$= 106.42 mm ≈ 100 mm (nearest standard size)

Subsequently, driving pulley diameter and the velocity ratio (VR) are 100 mm and 0.74, respectively; size of the driven pulley was calculated using Equation 5. Further, 40 mm Ø, tension pulley was used in the clutch system.

VR = $\frac{d_1}{d_2}$, 0.74 = 100/d_2

d_2 = 135.135 mm ≈ 140 mm (nearest standard size)

Considering the standard pulley size, Ø = 100 mm & 140 mm cast iron (CI) pulleys and B Type V belt was selected to for proposed design.

2.4.6. Designing of Chain Drive:

The required speed reduction was achieved from the belt drive and selected gear box; thus, the chain drive was not used for speed reduction. Therefore, velocity ratio (VR) of chain drive was taken as one (01). A roller chain was proposed and number of teeth on smaller sprocket for 1 VR is 31 [16]. Hence, same size sprockets (31 T) were selected for gear box out and the crank shafts.

Design power = Rated power × Service factor (K_s)

The service factor (K_s) is the product of various factors such as load factor (K_1;1.5 for heavy shock load), lubrication factor (K_2;1 for drop lubrication) and rating factor (K_3;1 for 8 hours per day) [16].

Therefore, K_s = 1.5 ×1 × 1= 1.5
The design power = 1.57 × 1.5 = 2.355 kW

The corresponding sprocket speed is 66.67 rpm, thus the power transmitted for chain is 1.18 kW per strand. Consequently, a chain No.10 with two strands can be used to transmit the power. According to the Indian Standards (IS: 2403-1991), No. 10 chain consists of following specifications; pitch 15.88 mm, roller diameter 10.16 mm, minimum width of roller 9.65 mm and breaking load (W_b) 45 kN. Pitch circle diameter (d) of the sprocket was determined by Equation 7.

$$d = p \operatorname{cosec}\left[\frac{180}{T}\right] \text{ --------------------------------------- } \textit{(Eq.7)} \text{ [16]}$$

where;

d = Pitch circle diameter (mm)
p = Pitch
T = No. of teeth

$$d = 15.88 \operatorname{cosec}\left[\frac{180}{31}\right] = 15.88 \times 9.895 = 157 \text{ mm}$$

Therefore, pitch circle diameter of the sprocket is 157 mm. Then pitch line velocity of the sprocket was calculated by Equation 8.

$$v = \frac{\pi d N}{60} \text{ --- } \textit{(Eq.8} \text{ [16]}$$

where;

v = Velocity (m s^{-1})
d = Circular pitch diameter (m)
N = Rotation speed of the sprocket (rpm)

$$v = \frac{\pi \times 0.157 \times 66.67}{60} = 0.548 \text{ m s}^{-1}$$

Subsequently, the pitch line velocity of the sprocket is 0.548 m s^{-1}; thus, load on the chain (W) was determined by Equation 9.

$$W = \frac{Rated\ \ power}{Pitch\ line\ \ velocity} \text{ ------------------------------------- } \textit{(Eq.9)}$$

$$W = \frac{1.57}{0.548} = 2.865 \text{ kN} = 2865 \text{ N}$$

Factor of safety (n) was calculated by dividing the breaking load (Wb) to the load on the chain (W).

Hence, n = 45,000/2,865 = 15.707 ≈ 16

The safety factor should be higher than 14 in chain drive with 20 – 25 mm pitch and 1200 rpm sprocket speed [16]. Therefore, this safety factor is adequate. The minimum centre distance between sprockets should be 30 to 50 times the pitch. It was taken as 30 times pitch.

Therefore, centre distance between the sprockets

= 30 × p = 30 × 15.88 = 476.4 mm ≈ 476 mm

In order to accommodate initial sag in the chain, the value of centre distance was reduced by 2 to 5 mm.

Therefore, correct centre distance (x) = 476 – 4

= 472 mm

The number of chain links and length of the chain (L) was determined by the Equation 10 and 11, respectively.

$$K = \frac{T_1+T_2}{2} + \frac{2x}{p} + \left[\frac{T_1-T_2}{2\,x}\right]^2 \frac{p}{x} \text{ -------------------------- } \textit{(Eq.10)}$$

$$L = K.p \text{ --- } \textit{(Eq.11)}$$

where;

K = Number of chain links
$T_{1\&2}$ = Number of teeth in sprockets
x = Corrected centre distance

p = Pitch

$$K = \frac{31+31}{2} + \frac{2 \times 472}{15.88} + \left[\frac{31-31}{2 \times 472}\right]^2 \times \frac{15.88}{472}$$

K = 31 + 59.446 + 0 = 90.446

L = 90.446 × 15.88 = 1,436.282 mm = 1.436 m

Hence, the required chain length was 1.436 m with 90 links [16].

2.4.7. Designing of Crank Shaft:

The crankshaft is a principal member of power transmission system, which is used to convert rotary motion of the sprockets of chain drive into reciprocating motion through the legs of weeding clogs. This weeder has multi-throw, side (overhang) crank shafts (Nos. 04) which are composed of following parts; (i) crank pin, (ii) crank web and (iii) shaft. The crank pin is used to connect the legs of weeding clogs through a bush. The shaft rotates in clamp bearings which are mounted on the frame. The crank web connects the crank pin and the shaft. However, these should be strong enough to resist fluctuating and shock force and rigid enough to keep the deflection and distortion within permissible limits. Two crank positions were considered; (i) when the crank is at the dead centre at which the bending moment is maximum and (ii) when the crank is at angle at which the twisting moment is maximum.

(i) Designing of Crank Shaft when the Crank is at the Dead Centre: Crankshaft at dead centre with its loads and distance of their application is shown in Fig. 9.

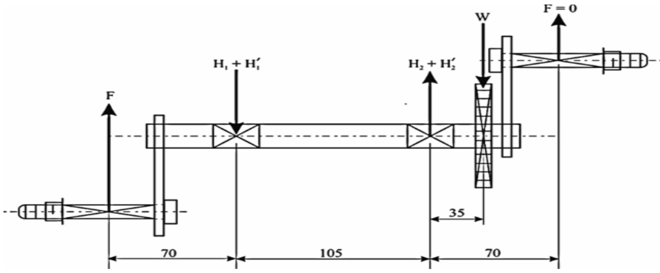


Fig. 9 Forces activated on the crank shaft $F_{1\&2}$ – impact load, $H_{1\&2}$ – reactions on bearings due to impact load, $H'_{1\&2}$ – reactions on bearings due to chain drive and W – load on the chain drive.

Due to the impact load, which is acting vertically, there will be two vertical reactions H_1 and H_2 at bearing 1 and 2, respectively. As in Equation 2, the impact load (F_1) is calculated as 1,079.915 N ≈ 1.08 kN and F_2 is equal to 0 N since, the weeding clog is not touching the ground. Besides it was assumed that, no weight of the crank shaft;

$$H_1 = \frac{F\ (70+105)}{105} = \frac{1.08 \times 175}{105} = 1.8 \text{ kN}$$

$$H_2 = \frac{F \times 70}{105} = \frac{1.08 \times 70}{105} = 0.72 \text{ kN}$$

Due to the downward acting load on the chain drive, there will be two vertical reactions H'_1 and H'_2 at bearings 1 and 2, respectively. As given in Equation 9, the load on the chain drive (W) was calculated as 2.865 kN.

$$H'_1 = \frac{W \times 35}{105} = \frac{2.865 \times 35}{105} = 0.955 \text{ kN;}$$

$$H'_2 = \frac{W \times (105+35)}{105} = \frac{2.865 \times 140}{105} = 3.82 \text{ kN [16]}$$

(a) Design of Crankpin:

Diameter of the crank pin in the bearing was calculated by Equation 12.

$F = d_c.l_c.P_b$. -- *(Eq.12)* [16]

where;

d_c = Diameter of the crankpin (mm)

l_c = Length of the crankpin = (0.6 ~ 1.5) d_c

= 0.8 d_c

P_b = Permissible stress (Assume as 5 N mm^{-2})

F = Impact load (1.08 kN)

$1.08 \times 10^3 = d_c \times 0.8 d_c \times 5 = 4(d_c)^2$

d_c = 16.432 mm ≈ 16 mm

$l_c = 0.8\ d_c = 0.8 \times 16 = 12.8$ mm ≈ 13 mm

Accordingly, the crank pins were fabricated with dimensions of 16 mm Ø and 13 mm L, respectively. However, it was extended with 20 mm thread (16M × 2). Bending moment (M), section modules (Z) and bending stress induced at the crank pin were calculated by Equations 13 – 15, respectively.

$M = ¾\ Fl_c$ --- *(Eq.13)*

$M = ¾ \times 1.08 \times 10^3 \times 13 = 10.53 \times 10^3$ N mm

$Z = \frac{\pi}{32} (d_c)^3$ --- *(Eq.14)*

$Z = \frac{\pi}{32} (16)^3 = 402.123$ Nmm^3

Therefore,

bending stress induced (σ_b)= $\frac{M}{Z}$ -----------------*(Eq.15)*

$$= \frac{10.53\ x\ 10^3}{402.123} = 26.186 \text{ N mm}^{-2} \text{ or MPa}$$

Subsequently, the induced bending stress was within the permissible limit of 60 MPa, design of crankpin is safe [16].

(b) Designing of Bearing: Thickness of the crank web (t) and length of the bearing (l_b) was expressed as the fraction of crankpin diameter (d_c) as shown in Equation 16 and 17, respectively [16].

$t = (0.45 \sim 0.75)\ d_c = 0.6\ d_c$ ----------------------- (*Eq.16)*
$l_b = (1.5 \sim 2)\ d_c = 1.7\ d_c$ ------------------------------ *(Eq.17)*
$t = 0.6 \times 16 = 9.6$ mm ≈ 10 mm
$l_b = 1.7 \times 16 = 27.2$ ≈ 27 mm

Accordingly, the thickness of the crank web and length of bearings were 10 mm and 27 mm, respectively. The bending moment (M) at the centre of bearing was obtained using Equation 18 and 19.

$M = F\ (0.75\ l_c + t + 0.5\ l_b)$ ---------------------------- *(Eq.18)*

$M = \frac{\pi}{32} (d_b)^3 \sigma_b$ --- *(Eq.19)*

where; d_b = Diameter of the bearing

Taking σ_b = 60 MPa or Nmm^2,

$1.08 \times 10^3\ (0.75 \times 13 + 10 + 0.5\ x\ 27) = \frac{\pi}{32} (d_b)^3 \times 60$

d_b = 18.268 mm ≈ 19 mm

Considering the availability, No. 204 (Clamp type), radial ball bearings (19 mm Ø and 25 mm L) were used to fix crankshaft on the frame.

(c) Designing of Crank Web: When the crank is at dead centre, the crank web is subjected to a bending moment (σ_b) and to a direct compressive stress (σ_d). The bending moment (M) and section modules (Z) on crank web were calculated by Equation 20 and 21, respectively [16].

$M = F\ (0.75\ l_c + 0.5t)$ --------------------------------- *(Eq.20)*
$Z = \frac{1}{6} \times w.t^2$ --- *(Eq.21)*

where; w = Width of the web.

Therefore, $M = 1.08 \times 10^3\ (0.75 \times 13 + 0.5 \times 10)$
= 15,930 N mm
$Z = \frac{1}{6} \times w.10^2 = 16.667$ w mm^3

As in Equation 15, $\sigma_b = \frac{M}{Z} = \frac{15{,}930}{16.667\ w} = \frac{0.956 \times 10^3}{w}$

Direct compressive stress (σ_d) is calculated by Equation 22.

$\sigma_d = \frac{F}{w.t}$ -- *(Eq.22)*

Therefore, $\sigma_d = \frac{1.08 \times 10^3}{w \times 10} = \frac{108}{w}$ N mm^{-2}

Total stress on the crank web (σ_T) = $\sigma_b + \sigma_d$

$= \frac{0.956 \times 10^3}{w} + \frac{108}{w} = \frac{1.064 \times 10^3}{w}$ N mm^{-2}

The total stress should not exceed the permissible limit (60 MPa or N mm^{-2}), taking 50% of it (safety factor as 02),

Therefore, $30 = \frac{1.064 \times 10^3}{w}$

w = 35.467 mm ≈ 38 mm

A length of 38 mm MS flat iron with 10 mm thickness was selected to prepare the crank web. Further, the weed burying depth (5 cm), was considered in length determination of crank web.

(d) Designing of Shaft/Main Journal: Main journal of the crank shaft was fabricated from 19 mm shafting to be compatible with selected No. 204 (Clamp type) radial ball bearings. Each main journal has a sprocket in central side for power transmission. While, the desired row spacing of the plant was 30 cm, the journal length was determined as 215 mm for facilitating easy movements of weeding clogs.

(ii) Designing of Crank Shaft when the Crank is at an Angle of Maximum Twisting Moment: Assuming same impact load (F) acting on crank shaft, in dead position is 1.08 kN and 30º maximum twisting angle (θ) [17], The angle of inclination of the leg of weeding clog with the line of stroke (φ) was calculated by Equation 23.

$Sin\ \varphi = \frac{\sin\theta}{\frac{l}{r}}$ -- *(Eq.23)* [16]

where;

l = Length of weeding clogs leg (mm)

r = Crank radius (50 mm)

$Sin\ \varphi = \frac{\sin 30°}{\frac{335}{50}} = 7.463 \times 10^{-2}$

$\varphi = Sin^{-1}(7.463 \times 10^{-2}) = 4.28^{\circ}$

There by, the thrust to the leg of weeding clog (F_Q) was calculated by Equation 24.

$F_Q = \frac{F}{Cos\ \varphi}$ --- *(Eq.24)* [16]

$F_Q = \frac{1.08}{Cos\ 4.28} = 1.083$ kN

The thrust to the leg (F_Q) of weeding clog is almost same to impact load (F).

Tangential force acting on the crank (F_T)

$= F_Q\ Sin\ (\theta + \varphi) = 1.083\ Sin\ (30^{\circ} + 4.28^{\circ}) = 0.61$ kN

Radial force acting on the crank (F_R)

$= F_Q\ Cos\ (\theta + \varphi)$

$= 1.083\ Cos\ (30^{\circ} + 4.28^{\circ}) = 0.895$ kN

Due to the tangential force (F_T), there were two reactions at the bearings 1 and 2, follows;

$H_{T1} = \frac{F_T\ (70+105)}{105} = \frac{0.61\ (70+105)}{105} = 1.017$ kN

$H_{T2} = \frac{F_T \times 70}{105} = \frac{0.61 \times 70}{105} = 0.407$ kN

Due to the radial force (F_R), there were two reactions at the bearings 1 and 2, as follows;

$H_{R1} = \frac{F_R\ (70+105)}{105} = \frac{0.895\ (70+105)}{105} = 1.492$ kN;

$H_{R2} = \frac{F_R \times 70}{105} = \frac{0.895 \times 70}{105} = 0.597$ kN

(a) Designing of Crank Web: The bending moment and stress due to tangential and radial force was determined by Equations 25 – 28, respectively.

Bending moment due to tangential force (M_{bT}),

$M_{bT} = F_T \left[r - \frac{d_b}{2}\right]$ ----------------------------------- *(Eq.25)* [16]

Where;

F_T = Tangential force acting on the crank (kN)
r = Crank radius (50 mm)
d_b = Diameter of the bearing (mm)
$M_{bT} = 0.61 \left[50 - \frac{19}{2}\right] = 24.705$ kN mm

Bending stress due to the tangential force,

$\sigma_{bT} = \frac{M_{bT}}{Z} = \frac{6M_{bT}}{t.w^2}$, ($Z = \frac{1}{6} \times t.w^2$) ---------------- *(Eq.26)* [16]

$\sigma_{bT} = \frac{6 \times 24.705 \times 10^3}{10 \times 38^2} = 10.265$ N mm^{-2} or MPa

Bending moment due to radial force,

$M_{bR} = F_R\ (0.75\ l_c + 0.5\ t)$ ---------------------------- *(Eq.27)* [16]

= 0.895 (0.75 × 13 + 0.5 × 10) = 13.201 kN mm

Bending stress due to the radial force,

$\sigma_{bR} = \frac{M_{bR}}{Z} = \frac{6\ M_{bR}}{w.t^2}$, ($Z = \frac{1}{6} \times w.t^2$) ----------------- *(Eq.28)* [16]

$= \frac{6 \times 13.201 \times 10^3}{38 \times 10^2} = 20.844$ N mm^{-2} or MPa

Then, the direct compressive stress was calculated by Equation 29.

$\sigma_d = \frac{F_R}{w.t}$ -- *(Eq.29)* [16]

$= \frac{0.895 \times 10^3}{38 \times 10} = 2.355$ N mm^{-2} or MPa

Therefore, total compressive stress (σ_c)

$$= \sigma_{bT} + \sigma_{bR} + \sigma_d$$

= 10.265 + 20.844 + 2.335 = 33.444 MPa

The twisting moment due to tangential force (T) and shear stress (τ) were calculated by Equations 30 and 31, respectively.

$T = F_T (0.75\ l_c + 0.5t)$ -------------------------------- *(Eq.30)*

= 0.61 (0.75 × 13 + 0.5 × 10) = 8.998 kN mm

$\tau = \frac{T}{Z_p} = \frac{4.5\ T}{w.t^2}$, ($Z = \frac{w.t^2}{4.5}$) ------------------------------ *(Eq.31)*

$= \frac{4.5 \times 8.998 \times 10^3}{38\ x\ 10^2} = 10.656$ MPa

The total maximum stress (σ_{max}) was calculated by Equation 32.

$\sigma_{max} = \frac{\sigma_c}{2} + \frac{1}{2}\sqrt{(\sigma_c)^2 + 4\tau^2}$ ---------------------- *(Eq.32)*

$= \frac{33.444}{2} + \frac{1}{2}\sqrt{33.444^2 + 4\ x\ 10.656^2}$

= 36.551 MPa

Accordingly, the calculated (σ_{max}) was less than the permissible value of 60 MPa, hence the design is safe [16].

(b) Designing of Shaft at the Junction of Crank: Bending moment (M) and twisting moment (T) were determined by Equations 33 and 34, respectively [16].

$M = F_Q (0.75\ l_c + t)$ ----------------------------------- *(Eq.33)*
= 1.083 (0.75 × 13 + 10) = 21.389 kN mm
$T = F_T.\ r$ --- *(Eq.34)*
= 0.61 × 50 = 30.5 kN mm

Therefore, equivalent twisting force (T_e) = $\sqrt{M^2 + T^2}$

$= \sqrt{21.389^2 + 30.5^2} = 37.252$ kN mm

Besides, $T_e = \frac{\pi}{16} (d_{sl})^3\ \tau = 37.252 \times 10^3$

Considering, the diameter of shaft at the junction of crank (d_{sl}) = d_b = 19 mm
$\frac{\pi}{16} (19)^3\ \tau = 37.252 \times 10^3$
$\tau = 27.66$ MPa

Subsequently, the induced shear stress was less than permissible limit of 40 MPa; hence, the design is safe [16].

Based on the above calculations and design parameters/requirements, the resultant crankshaft is shown in Fig. 10.

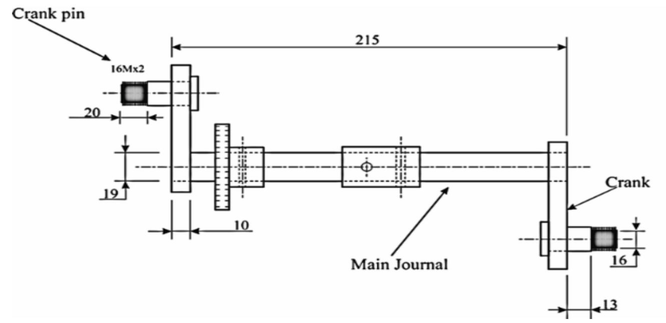


Fig. 10 Crank shaft

2.4.8. Designing of Front Wheel: In row changing or road operation, the frame is acting as second-class lever and the highest load through the front wheel is achieved. Hence, it was required to design the front wheel wide enough to bare the resulting load or to be an equilibrium with the PR of surface layer (Fig. 6).

Considering the equilibrium of the system;

$\Sigma F_y \uparrow^+ = 0$
R + F – W = 0

Considering, the maximum effort for male worker (F) is 218 N [18]. and the dead load of the machine (W) is 60 kgf (588.6 N ≈ 600 N), Reaction by surface soil (R);

R + 218 – 600 = 0
R = 382 N

Hence, there should be enough soil contact area to bear this dead load (382 N) through the surface puddle resistance. The ground clearance of the design is basically depending on the diameter of the front wheel; hence it was taken as 25 cm. Assuming soil contact angle of wheel is 15^0, the length of the soil contact area (S) could be calculated by equation 1.

S = θr
$S = \frac{\pi}{12}$ x 12.5 = 3.272 cm

If, the width of the tyre is d; the contact area of soil (A) could be calculated by Equation 35.

A = 0.78Sd -- *(Eq.35)*
= 0.78 × 3.272 × d

Considering surface puddle resistance at 3 - 5 WAT of rice fields was 218.07 kN m^{-2} [13],

R = Soil contact area × PR
$382 = 0.78 \times 3.272 \times d \times 10^{-4} \times 218.07 \times 10^3$
d = 6.864 cm ≈ 7 cm

Leaving 15% for the soil variability, a rubber pneumatic tyre with 8 cm width was selected as front wheel. Furthermore, its floating and non-shrinking ability in field operation was highly considered. Additionally, its open "V" lug design helps easy operation in road and off-road conditions. Considering the market availability, 4.10/3.50 – 4 rubber tyre with metal hub (Ø = 25 cm) was selected.

Moreover, an adjustable (fork length and angle) inclining fork was designed using heavy duty Galvanized Iron tubes (3 mm) (Ø = 22 & 25 mm). These adjusters help to use this machine in diverse soil strength conditions (Fig. 11).

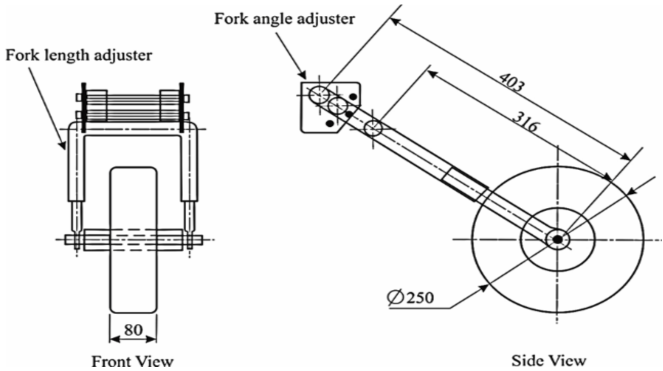


Fig. 11 Designed front wheel

2.4.9. Designing of Handle: The handle was designed considering the ergonomics, anthropology of human and safety aspects, the distance between two handle bars was determined as 550 mm for easiness of turning the machine. Distance from operator to machine was determined by considering easy controlling ability. Further, height and direction (horizontal and vertical) adjusters were designed to suit this machine for different operators and postures (Fig. 12).

Handle of the power weeder was fabricated by heavy duty GI tubes (Ø = 22 mm, 2mm) and the base of the handle was by Mild Steel (3 mm) sheets. The base of the handle was directly fixed to the gearbox by using nuts and bolts. The control unit consisted of accelerator and clutch leaver, which are fixed on the right- and left-hand grip of the handle, respectively. Furthermore, this hand clutch leaver consists of self-locking device for continuing clutch engagement.

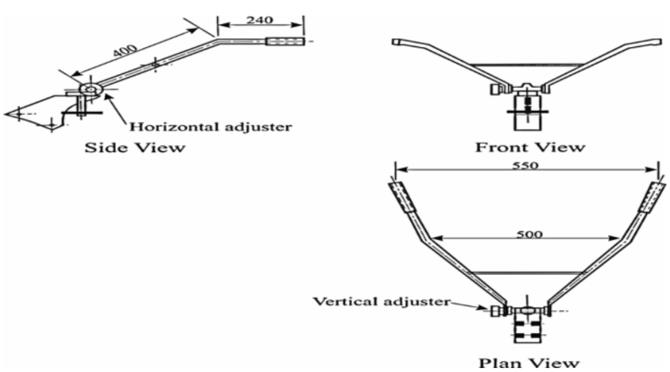


Fig. 12 Designed handle

2.4.10. Designing of Rear Skidder: When the crank shaft at angle position (90º) weeding clogs are not touching the ground and the total dead load is balanced by the reactions of front wheel and the rear skidders, respectively (Fig. 5). Assuming the weeding clogs are attached at the centre of the frame and the distance to the centre as *l*;

With considering the equilibrium of the frame

$$\sum \mu_0 = 0$$
$$R_2\, l - R_1\, l = 0$$
$$R_1 = R_2$$

Considering the length of the soil contact area (S) of front wheel, width (d) and surface puddle resistance at 3 - 5 WAT of rice fields are 3.272 cm, 8 cm and 218.07 kN m^{-2}, respectively.

$$R_1 = \text{Soil contact area x PR}$$
$$= 0.78 \times 3.272 \times 10^{-2} \times 8 \times 10^{-2} \times 218.07 \times 10^{3}$$
$$= 445.24 \text{ N}$$

Therefore, a rear skidder was required to design to create 445.24 N soil reaction (R_2).

Assuming a 7.5° soil contact angle, S = 1.636 cm, curvature of the skidder (r) = 12.5 cm and surface puddle soil resistance (PR) = 218.07 kN m^2. If, width of rear skidder is d;

The contact area of soil (A) = 2sd = 1.636 × 2d

$$R_2 = A \times PR$$
$$445.24 = 1.636 \times 2d \times 10^{-4} \times 218.07 \times 10^{3}$$
$$d = 6.24 \text{ cm}$$

Including an additional 30%, 8 cm wide two rear skidders were designed by using MS sheets (Fig. 13). Moreover, heavy duty GI tube (Ø = 21 mm, 2 mm) was used to make the frame of rear skidder. MS shaft (Ø = 16 mm), GI tube collar and screwing device with self-griping handle were used to prepare the height adjuster.

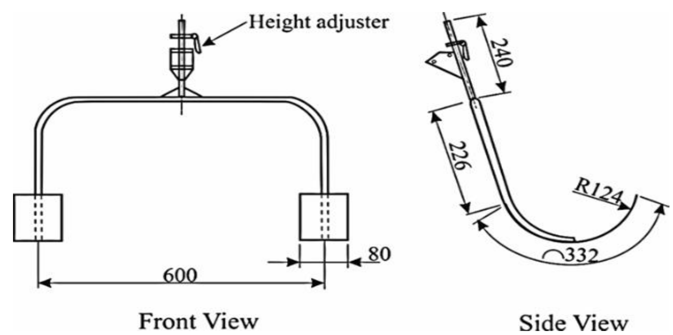


Fig. 13 Designed rear skidder

2.4.11. Designing of Frame: The frame (347 mm x 512 mm) was designed considering the dimensions and the range of movement of various components. Heavy duty iron boxes 25 mm × 25 mm (3 mm) were used to make this open frame. The front section of the frame facilitated for mounting engine, while the rear section was used for mounting the gearbox and the controlling unit (handle). Four crank shafts were mounted on the centre of the frame (Fig. 14).

Heavy duty 19 mm × 19 mm (3 mm) iron boxes were used to fabricate lower arches (Nos. 02). These lower arches were used to mount the chain adjuster for chain and sprocket power transmission system. Furthermore, they work as a guard for the gear box and power transmission system. As shown in Fig. 15, lower arches attached to the main frame using nuts and bolts.

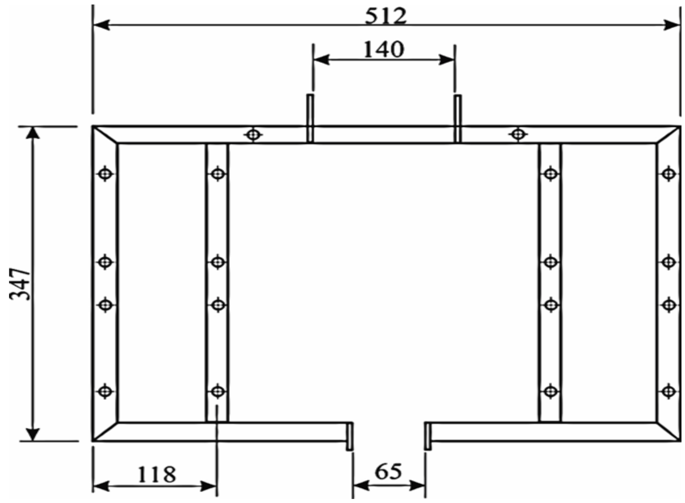


Fig. 14 The frame (plan view)

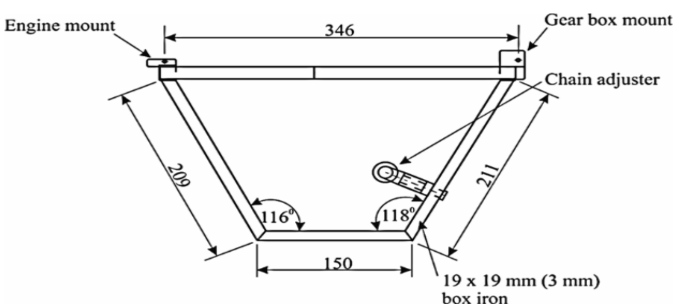
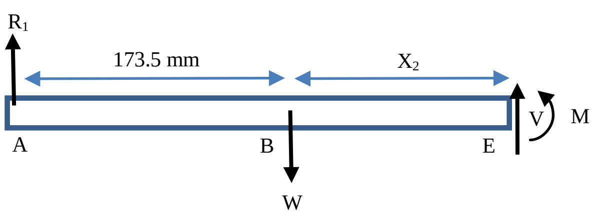


Fig. 15 Lower arch and side view of frame

Force Analysis on Frame: When crank shaft is at an angle (90º), dead load and reaction from front wheel and temporary patterning of the frame are shown in Fig. 16.



As calculated above, dead load (W) and soil reaction through the 8 cm wide front wheel (R1) was 588.6 N and 445.24 N, respectively.

Considering the equilibrium of the cantilever system of AC,

$\sum Fy \uparrow + = 0,$
$R1 - W + R3 = 0$
$R3 = 588.6 - 445.24 = 143.36\ N$
$\sum Fx \rightarrow + = 0$
$R2 = 0$

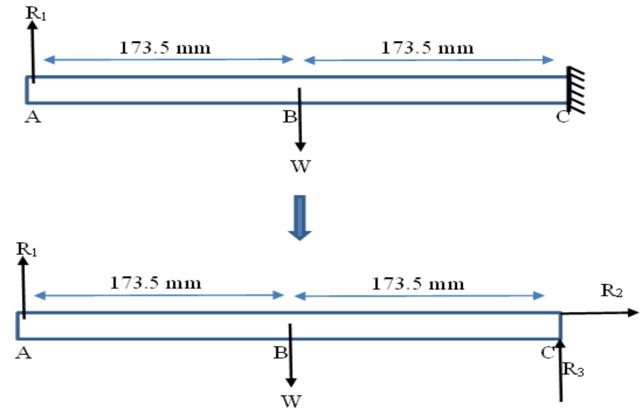
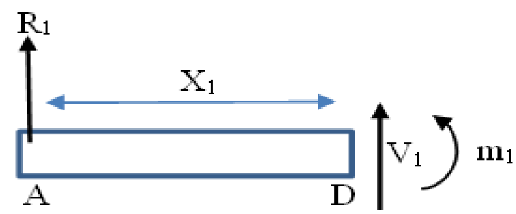


Fig.16 Force system and free body diagram of the frame; R_1 – soil reaction through the front wheel, W – dead load; 600 kgf = 588.6 N, $R_{2\ \&\ 3}$ – reactions of cantilever.

Considering the equilibrium of the cantilever AD,



$\sum F_y \uparrow^{+} = 0,$
$R_1 + V_1 = 0$
$445.24 + V_1 = 0$
$V_1 = (-)\ 445.24\ N$
$\sum \mu_0\ D = 0$
$m_1 - R_1X_1 = 0$
$m_1 = 445.24\ X_1$
if $X_1 = 0$, $m_1 = 445.24 \times 0 = 0$
if $X_1 = 173.5 \times 10^{-3}$,
$m_1 = 445.24 \times 173.5 \times 10^{-3} = 77.249\ Nm$
$\approx 77.25\ Nm$

Considering the equilibrium of the cantilever AE,

$\sum F_y \uparrow^{+} = 0,$

$R_1 - W + V_2 = 0$

$445.24 - 588.6 + V_2 = 0$

$V_2 = 143.36\ N$

$\sum \mu_0\ E^{+} = 0$

$m_2 + W\ X_2 - R_1\ (173.5 \times 10^{-3} + X_2) = 0$

$m_2 + 588.6\ X_2 - 445.24\ (173.5 \times 10^{-3} + X_2) = 0$

$m_2 + 588.6\ X_2 - 77.25 - 445.24\ X_2 = 0$

$m_2 = 77.25 - 143.36\ X_2$

if $X_2 = 0$, $m_2 = 77.25 - 143.36 \times 0 = 77.25\ Nm$

if $X_2 = 173.5 \times 10^{-3}$,

$m_2 = 77.25 - 143.36 \times 173.5 \times 10^{-3} = 52.38$ Nm

As shown in Fig. 17, shearing force and bending moment are distributing throughout the frame in a safe manner. No obligations were identified [19]. However, there was a 52.38 Nm bending moment remaining on the mechanism. Hence, heavy duty nuts and bolts (Ø = 8 mm) were used to obtain required strength.

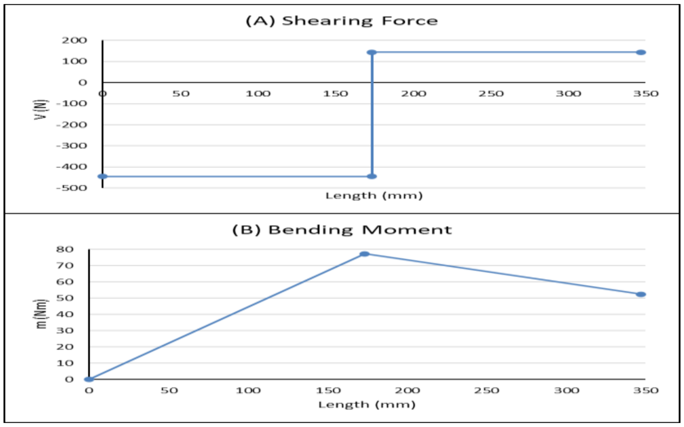


Fig. 17 Results of beam analysis; A- shear force and B - bending moment diagram

2.5. Development of Prototype

After completing the designing process, the idea has been properly put out on the form of engineering drawings, then prototype of the machine was fabricated as per the design specifications.

The fabricated prototype was tested at various speeds in different field conditions and check for the field performances. Defects of the prototype was identified and appropriate modifications were suggested to overcome those.

Besides, the manufacturing cost for a unit of the prototype including all of the material and labour cost was determined.

2.6. Performance Evaluation

These evaluations were conducted as RNAM test codes and procedures for weeders [20] on purposely selected, mechanically transplanted, well grown with Bg 352, and regular shape six paddy fields (min. 10 ×20 m^2) in North Central Province (NCP) of Sri Lanka. Five field samples from each test field were drawn by 50 × 50 cm^2 quadrant. Weeding operation was done at 3 weeks after transplanting (WAT).

Initially, machine specifications of final version of the prototype were checked. Performance indicators such as effective field capacity, field efficiency, weeding efficiency, plant damage percentage were determined using equation 36 – 38 [20]. The performance index was calculated by using equation 39 [21].

$$S = \frac{A}{T_p + T_l} \text{ -- } (Eq.\ 36)$$

$$E_f = \frac{W_e \times T_p}{W_t(T_p + T_l)} \times 100 \text{ -------------------------------- } (Eq.37)$$

$$n = \frac{W_1 - W_2}{W_1} \times 100 \text{ ----------------------------------- } (Eq.38)$$

where,

S = Effective field capacity (ha/h)
A = Area covered (ha)
Tp = Productive time (h)
Tl = Non productive time (h)
Ef = Field efficiency (%)
We = Effective working width
Wt = Theoretical working width
n = Weeding efficiency (%)
W1 = Weed count per unit area before operation
W2 = Weed count per unit area after operation

$$PI = \frac{S \times (100 - PD) \times n}{Power\ (hp)} \text{ ----------------------------- } (Eq.39)$$

where,

PI = Performance Index
S = Field capacity (ha/h)
PD = Plant damage (%)
n = Weeding efficiency (%)

In order to assess weeding costs, the fixed and variable costs were calculated [20,22,23,24,25]. Further, fuel consumption and labour requirement were also considered.

III. RESULTS AND DISCUSSION

3.1. Developed of Prototype

The final design (Plate 1), was fabricated as discussed design details and specifications.

The fabricated prototype showed satisfactory field performances. Further, it was not reported any damage or break down throughout the testing period. However, few defects were identified. It is required to apply continuous operator's effort for balancing the weeder. It may cause for the in-farm drudgery and body pains in long time operations. Further, when the front wheel is getting sunk in mud there is no mechanism to raise it by single operator.

To overcome those defects of the prototype, it was suggested to increase the width of rear skidder by 100%. Therefore, two 16 cm wide skidders with 7.5° soil contact angle and 12.5 cm curvature (r) were designed. Modified/Final version of the prototype is shown in Plate 2. Then, it showed satisfactory performances, good ergonomics and safety status in performance tests. For instance, it was not required operator's effort, only the guidance is enough for the field manipulation. If the front wheel is getting sunk, operator can push the handle downward, then front wheel will raise by loading on wider skidders as a 1st class lever.

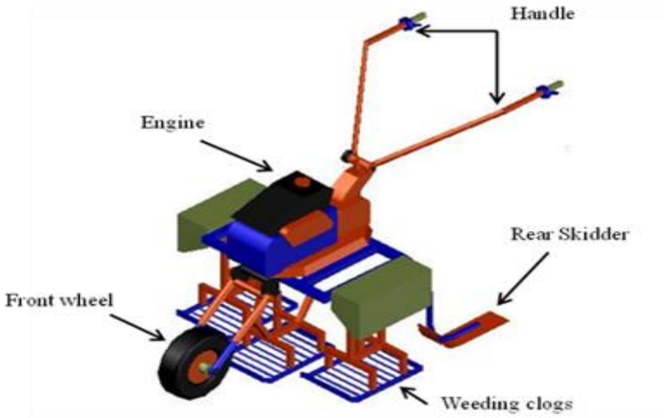


Plate 1 3D View of Final Design

Plate 2 Modified/Final Version of the Prototype

Estimated manufacturing cost for a unit of prototype including all of the material and labour cost was 362.5 USD. However, it is assumed to reduce production cost up to 225 USD in commercial production.

3.2. Machine Performances

Table 1 illustrate the machine specification of final version of the prototype.

Table 1. Machine Specifications of Final Version of the Prototype

Machine specification	**Values**
Overall dimensions (length × width × height) (cm)	150×80×78
Weight (kg)	62.9
Ground clearance (cm)	27
Weeding component width (cm)	20 × 3
Depth of cut, cm	5
Working width, cm	90
Number of rows covered in single pass	03
Type of soil working tool	Burying
Power source	1.57 kW, single cylinder, 4-stroke, air cooled, petrol engine with rated engine speed of 1800 rpm

Besides, it showed higher comparative field performances over existing weeders [21,26,27,28,29] (Table 2). Further, fuel consumption, labour and power requirement were 0.503 l/h, 33 man-h/ha and 0.319 kW, respectively. Besides, tests did not report any ergonomics defect or machine breakdown throughout the test and it was easy to operate.

Table 2. Field Performances of final version of the prototype

Field Performance	Average
Field Capacity (ha/h)	0.0308 (±0.0056)
Field Efficiency (%)	83.25 (±9.89)
Weeding Efficiency (%)	80.0 (±8.7)
Plant Damaged (%)	6.34 (± 4.91)
Performance index	580 (±152.2)
Cost of weeding (USD)	38.355(±5.25)

IV. CONCLUSIONS

Through kinematics studies, it was identified that, the model having double crank shaft as the appropriate working model to represent the required weed burying action by the mechanical means. Then, a prototype was designed and developed to fulfill various design parameters and specifications. Further, it consists of seven components; power source (engine), frame, power transmission mechanism, weeding mechanism, turning and row changing mechanism, floating mechanism, manipulation and controlling mechanism. Modified version attained satisfactory field performances, good ergonomics and safety status in performance tests. Thus, this prototype has potential to develop as an appropriate machinery for weeding processes in medium and large-scale, row planted rice cultivations in Sri Lanka. Further, it suggested conduct a comprehensive field performance evaluation and appropriate modifications.

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