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Hydraulically operated palm oil loader system design as fresh fruit bunch collector

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ABSTRACT

Palm oil is one of the main agricultural contributors to Malaysia's Gross Domestic Product (GDP). Although commercial palm oil plantations have flourished in Malaysia for decades, there is still room for improvement especially in minimizing manpower load and developing efficient machinery to improve the various processes involved in the palm oil industry from estate to the mills. This paper presents a modified towable backhoe to assist in the Fresh Fruit Bunch (FFB) collecting process as an effort to save time. The main objective of this study is to design a machine that can be used by smallholders of oil palm in order to help speed up the work process. With a boom extension and an innovative grabber, the pressure needed to operate the hydraulic system was analyzed to ensure the towable backhoe functioned effectively and fulfilled its objectives. From the calculations done, the maximum pressure that the system can supply is 31.28 MPa which is higher than the 11 MPa minimum required for the hydraulic system to operate.

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1. INTRODUCTION

Palm oil industry in Malaysia is one of main contributors of agricultural income in Malaysia. According to the Malaysian Department of Statistics, in 2016, the production of fresh fruit bunch (FFB) from the palm oil industry contributed RM38.5 billion to the overall Gross Domestic Product (GDP) of Malaysia [1]. The palm oil industry consists of the processes of planting, farm management, harvesting and processing before the final product, palm oil, is produced at the mill. The management and harvesting processes have been identified as requiring more manpower compared to the other processes. Harnessing the latest technology and through the use of creative innovations, some of the manual work has been automated and less manpower is needed.

According to Shuib et al., [2], FFB collection was placed fourth in terms of total workforce required in the palm oil cultivation process in Malaysia in 2008. The study also highlighted some innovations that have been developed in order to assist in the harvesting and collecting of fresh fruit bunch before delivery to the mills. These included using a mechanical approach and modification of the terrain. As reported by Pebrian et al., [3], the energy expenditure per man worker is 48.49 kcal/hour/man during the holing process stage before FFB is transported to the mill.

As reported by the Malaysian Palm Oil Board (MPOB), various innovations designed specifically to improvise and minimize manpower usage in harvesting and collecting FFB before transportion to the palm oil mill. This was highlighted by Azwan and Setiawan [4-5], who listed some of the innovations to reduce manpower in palm oil processing activities. This has been supported by Shuib and Muhammad [2, 6-7] who stated that from 1987 to 2017, there were 45 innovations purpose-built for palm oil activities registered with the MPOB. Involvement from higher learning institutions also contributed to the development of new technology for the palm oil industry such as by Wan Ishak [8] who developed the Oil Palm Harvester Robot at the Universiti Putra Malaysia. Darius and his team [3] who developed the 4 wheel drive prime mover along with an attachment to reduce manpower usage in palm oil estate activities. This paper focuses on the design and fabrication of a hydraulically operated palm oil loader for FFB collecting process with the main objective of reducing the manpower required for FFB collecting activities.

In order to improve the FFB collecting process, an innovation using a hydraulic grabber on a mini excavator was developed. The objective of the development of this product was to minimize the workforce used and to reduce the manual processes in FFB collecting especially in palm oil plantations located in peat soil areas. The machine will focus on the collection of harvested FFB near the collector truck terrain. By employing a fluid power system, the FFB collector has been able to minimize manpower needs during the collection process by towing the FFB collector at the back of the collector truck. Apart from that, the towable backhoe design was focused on the ability to operate on peat soil. Peat soil contains high amounts of water which causes problems when activities are carried out on top of this type of soil. This is due to the soil's low shear strength and high permeability as described by Ramesh and Adnan [9-10]. The higher water content in peat soil causes the soil to become soft, thus making it problematic when applying a heavy load on top or within it. Several factors were taken into consideration during the design process of a hydraulic grabber for FFB collection. These include an innovative design of towable backhoe with a grabber mechanism for collecting FFB and towed by a truck to the next process.

2. RESEARCH METHOD

Design specifications for the hydraulic grabber were made related to the capabilities of the product and its technical specifications. For this study, the technical specifications of agriculture towable backhoes before modification, and the requirements to fulfil the main objective of this project were identified. Requirements of the project were identified based on a survey and site visit to the towable backhoes involved in the harvesting and collecting of FFB, and from literature review [11-14]. The results obtained were filtered and the requirements from the survey were then prioritized as shown in Table 1. The specifications and capabilities of the towable backhoe are listed below in Table 2.

Table 1. The Requirements Specified by the Towable Backhoes based on the Survey and Site Visit

Criteria	Description
Load	Able to grab and move a minimum of 10 FFB at a time
Rotation	Able to collect FFB on the right and left side of the truck path
Height	Able to lift up to 2.5m height from the ground
Terrain	Able to operate on peat soil and can be towed by a truck.

Table 2. Specifications of the Hydraulic System and Power Source for A Hydraulically Operated Palm Oil Loader

Parameter	Description
Power source	ICE
Model	JF270
Engine capacity	270 cc
Maximum Output Power of ICE (kW)	5.2
Maximum RPM	3600
Pump Type	External Gear Pump
Volumetric displacement, V_D	0.00000277m^3
Maximum pumping pressure	31.28 MPa@3600RPM
Calculated Minimum Pressure requirement	11 MPa
Excessive Pressure (%)	64.8 @ 3600RPM
Piston Diameter, D_P	68 mm
Rod Diameter, D_R	35 mm
Maximum angle θ_4 for boom cylinder (Cylinder 1) in retraction stroke	15°
Maximum angle θ_3 for arm 2 cylinder (Cylinder 3) in retraction stroke	14.5°
Pitching angle of Arm 1, θ_1 during lifting.	45°
Turning angle of stick	149.5°
Swing angle of boom	140°
Weight FFB (10 bunches)	200kg
Weight of arm 1, m_1	69.5kg
Weight of arm 2, m_2	50kg

For this innovation, a mathematical model was used to analyze the capabilities of the grabber with a maximum load, and the maximum load that could be retained by each of the cylinders. Once the maximum capabilities were determined for each cylinder, then the fabrication could take place. For the solutions and mathematical models used in the design and fabrication process as refer to [15-18].

3. RESULTS AND ANALYSIS

From the design requirements and technical specifications listed, the design of a hydraulic powered system FFB Grabber was developed. The schematic diagram of the hydraulic system for the hydraulic grabber has been designed as a reference model before the development of the mathematical models [19-21]. The schematic diagram that was designed is shown in Figure 1.

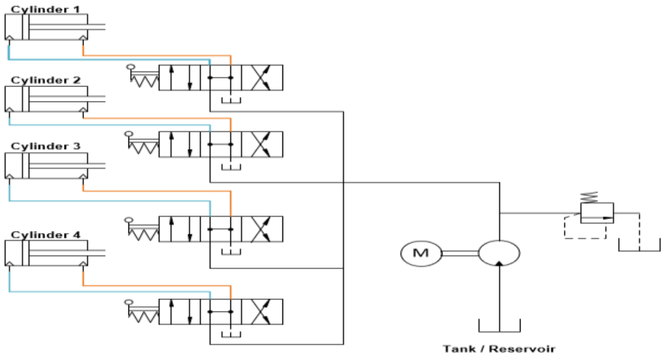

Figure 1. The schematic diagram of a hydraulic grabber for FFB collection

Figure 1 depicts a total of 4 cylinders in which each cylinder has its own function. Each cylinder is connected to the valve with a manual spring level as a controller for the hydraulic system and is connected to the tank located within the chassis of the towable backhoe. The lever position and function of each lever together with the cylinders' action are listed in Table 3 below. There are two levers with Lever No. 1 controlling the grabber and lower arm function while Lever No. 2 controls the upper arm and arm swing.

Table 3. Operations and functions of the Hydraulically Operated Palm Oil Loader

Operation	Boom Cylinder	Stick Cylinder	Grabber Cylinder	Boom Swing Cylinder
Lowering both arm 1 & 2	Piston in expansion and hold pressure	Piston in expansion and hold pressure	hold pressure	No pressure
Opening grabber	hold pressure	hold pressure	Piston in Expansion until widely open	No pressure
Closing grabber	hold pressure	hold pressure	Piston extraction until end base and hold pressure	No pressure
Lifting both arm 1 & 2 at higher than open cabin of lorry	extraction and hold pressure	extraction and hold pressure	hold pressure	No pressure
Swing both arm 1 & 2 to the right at upper of cabin	hold pressure	hold pressure	hold pressure	Piston in Expansion
Swing both arm 1 & 2 to the left at upper of cabin	hold pressure	hold pressure	hold pressure	Piston in retraction

3.1. Calculations on the Capacities of the Cylinders

Table 2 shows the specifications and maximum achieved by the pressurized hydraulic pump in the palm oil loader during the lifting of both arms 1 and 2 at a height higher than the open cabin of a lorry and during the swinging of both arms 1 and 2 to the left or right above the cabin. This study only considered the effects of arm 1 and arm 2 movements with the existing load to reach the maximum operational figures of the hydraulically operated palm oil loader to pick and place the FFB into the cabin as shown in Figure 2. Assigning the stick Cylinder as Cylinder 3 in Figure 2, then F3 as maximum load during the piston retraction stroke as the function of axis x'. This assumes the origin O2 has a pin joint for a complete free body diagram of arm 2 in the schematic. The parameter of force is depicted in Table 4.

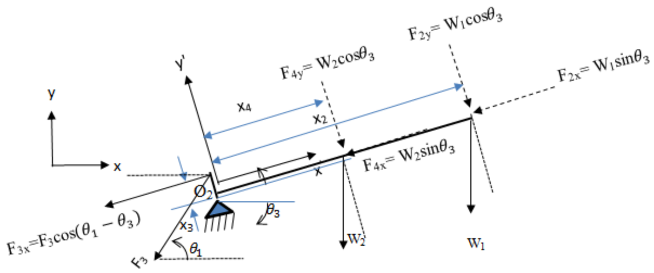

Figure 2. Schematic diagram for load acting on arm 2 with the pin joint at origin O2

Table 4. Parameter and Value of Force Acting on Arm 2 (Stick Arm)

Parameter	Value
Mass of arm 2, m_2	50kg
Mass of grabber, m_g	50kg
Mass of FFB (10 bunches), m_{ffb10}	200kg
$W_1 = (m_g + m_{ffb10})g$	2452.5N
$W_2 = m_2 g$	490.5N
x_2	2.06m
x_3	0.215m
$x_4 = x_2/2$	1.03m
θ_3 as fixed value	14.5°
θ_1 as variable value.	45°, 50°, 60°

By using moment equilibrium at Z axis in (1), when normalized to x'y' axis [12]:

$$\sum M_{01}:\ F_{3x}x_3 = F_{2y}x_2 + F_{4y}x_4 \tag{1}$$

We get the normal force acting on piston at cylinder 3, F3 as follows,

$$F3 = \frac{(W_1\cos\theta_3)x_2 + (W_2\cos\theta_3)x_4}{x_3\cos(\theta_1-\theta_3)} \tag{2}$$

To determine the minimum pressure for the retraction stroke for Cylinder 3; if the designed diameter size of piston, DP, and rod, DR, are fixed at 68mm and 35mm respectively.

$$\text{PRET3} = \frac{F_3}{A_P - A_R} \tag{3}$$

$$= \frac{F_3}{\frac{\pi}{4}(0.068^2 - 0.035^2)m^2} = \frac{F_3}{0.00266957m^2}$$

If θ_3=14.5° is fixed, then (3) becomes the formula for the minimum pressure required for the retraction stroke for Cylinder 3. Table 5 depicts the results for the different variables of pitching angle θ_1=45°, 50°, 60°. The calculations were performed using Microsoft Excel to compute the respective results using a fixed input and different variables to determine the angle effects in the design of the pumping effort, and to determine the minimum pressure required to accomplish the overall operations as depicted in Table 1. The value of normal force and pressure retraction stroke increased in the range of 29043.24N to 35702.17N and 10.88 MPa to 13.37MPa respectively.

Table 5. Results for Different Variables of Pitching Angle Θ_1=45°, 50°, 60°

Parameters	Pitching angle of Arm 1 (Boom), θ_1		
	45°	50°	60°
Normal force acting on piston at cylinder 3, F_3(N)	29043.24	30737.81	35702.17
Equivalent mass, m_e (kg)	2,960.55	3,133.3	3,639
Effective area, $A_P - A_R$ (m^2)	0.00266957	0.00266957	0.00266957
Pressure, P_{RET} (MPa)	10.88	11.51	13.37

Cylinder 1 as boom Cylinder shown in Figure 1, then F1 is maximum load during piston retraction as the function of axis x' as shown in the free body diagram of arm 1 and 2 as shown in Figure 3. Origin O1 has a pin joint for overall reacting force on the free body diagram of arms. The Cylinder 3 was in a pressurized condition to hold the arm 2 as a solid frame. The parameters of forces are depicted in Table 6.

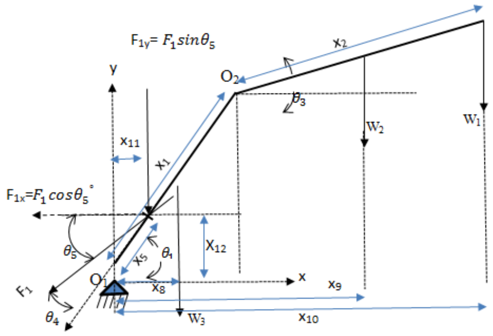

Figure 3. Schematic of free body diagram for arm 1 with the pin joint at origin O1 and O2

Table 6. Parameter and Value of Force Acting on Arm 1 (Boom Arm)

Parameters	Values
Mass of arm 1, m_1	62.5kg
Mass of cylinder 3 and cylinder 1, $m_{c3}=m_{c1}$	3.5 kg (approximately)
$M_{T1}= m_1+ m_{c3}+m_{c1}$	69.5kg
$W_3=m_{T1}g$	681.8N
x_1	1.92m
x_2	2.06m
$x_{4=}$ $x_2/2$	1.03m
x_5	0.74m
$x_6=x_1/2$	0.96m
x_7	0.13m
θ_3 as fixed value.	14.5°
θ_4 as fixed value.	15°
θ_5 as variable value.	$\theta_1 - \theta_4$
θ_1 as variable value.	45°, 50°, 60°

By using moment equilibrium Z axis in (4) at the point when all reacting forces are as shown in Figure 4, we can compute:

$$\sum M_{z2}: F_{1x}x_9 - F_{1y}x_{11} = W_1x_{10} + W_2x_9 + W_3x_8; \quad (4)$$

$$(F_1 cos\theta_5)x_9 - (F_1 sin\theta_5)x_{11} = W_1x_{10} + W_2x_9 + W_3x_8;$$

Force acting on cylinder 1 is,

$$F1=\frac{W_1x_{10}+W_2x_9+W_3x_8}{(cos\theta_5)x_9-(sin\theta_5)x_{11}} \quad (5)$$

To determine the minimum pressure for the retraction stroke for Cylinder 1, the designed diameter size of piston, DP, and rod, DR, are fixed at 68mm and 35mm respectively.

$$PRET1=\frac{F_1}{A_P-A_R} \quad (6)$$

$$=\frac{F_1}{\frac{\pi}{4}(0.068^2-0.035^2)m^2} = \frac{F_1}{0.00266957m^2}$$

If θ_4=15° is fixed, then (6) becomes the formula for minimum pressure required for the retraction stroke for Cylinder 3. Table 7 depicts the results for the different variables of pitching angle θ_1=45°, 50°, 60°. The calculations to compute the respective results using fixed input and different variables to determine the effect of angle in the design of pumping effort, with the minimum pressure required to accomplish overall operations as depicted in Table 1. According to Table 7, the value of normal force and pressure retraction stroke is in the range of 26,956.8N to 72,563.98N and 10.1 MPa to 27.18MPa respectively.

Table 7. Results for Different Variables of Pitching Angle θ_1=45°, 50°, 60°

Parameters	Pitching angle of Arm 1 (Boom), θ_1		
	45°	50°	60°
Normal force acting on piston at cylinder 1, F_1 (N)	31,050.38	72,563.98	26,956.8
Equivalent mass, m_e (kg)	3,165.18	7,396.94	2,747.9
Effective area, $A_P - A_R$ (m^2)	0.00266957	0.00266957	0.00266957
Pressure, P_{RET} (MPa)	11.63	27.18	10.1

The trend of the graphs as shown in Figure 5 confirmed the effect of angle on gravity on arm1 and arm 2. The horizontal line shows the pressure and normal force in cylinder 3 which depicts an increasing value without any gravity effects compared to the situation for cylinder 1. The weight distribution due to higher center of gravity causes the vertical structure to have better and smoother handling in terms of hydraulic pressure capability and produce lower energy consumption. While the horizontal structure with a higher theta angle need more space heavier and more energy needed in terms of hydraulic pressure requirements. However, the pressure in cylinder 1 shows the dramatic increase of the overall pressure to a maximum of 27.18MPa in order to stabilize the overall structural frame of the Hydraulically Operated Palm Oil Loader. Then it reduces to 10.1MPa with a reduction in energy consumption from less pumping effort required by the petrol-fueled Internal Combustion Engine (ICE) with 5.2kW output power at a maximum of 3600 RPM as mentioned in Table 2.

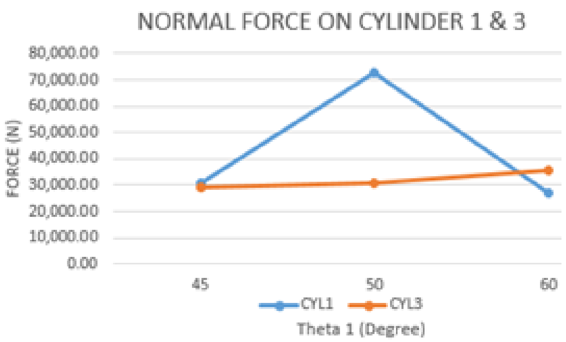

Figure 4. Normal force cylinder 1 & 3 vs theta 1

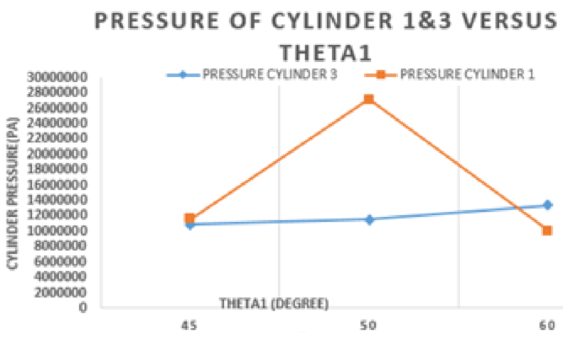

Figure 5. Pressure of cylinder 1 & 3 vs theta 1

The pump needs to support the minimum pressure requirements of the palm oil loader, and the specification of the hydraulic system and power source as detailed in Table 6. The preferred pump type is an external gear pump as it is able to provide the overall efficiency, η_O, of about 80-90% [22-25]. In order to verify the capabilities of the selected pump, the basic formulations are shown below: -

The pump volumetric displacement, VD.

$$V_D = \frac{\pi}{4}\left({D_O}^2 - {D_i}^2\right)L \quad (7)$$

$= \frac{\pi}{4}(0.068^2 - 0.035^2)0.008$ =0.00000277m3

Then, pump flow rate, QT.

QT=VDN (8)

=0.00000277 m3$\frac{3600}{60}$ = 0.000166 m3s-1

Actual flow rate, QA in (9) , volumetric efficiency, $\eta_V = 0.8$

$$\eta_V = \frac{Q_A}{Q_T} \tag{9}$$

QA=$\eta_V Q_T$ = 0.8(0.000166 m3s-1) = 0.000133 m3s-1

Power output of the Internal Combustion Engine (ICE), PO(ICE) is equal to Power input to Gear pump, Pi; hence input power of pump, PI(PUMP)= PO(ICE)=5.2kW.

Since the given power output of ICE=5.2 kW, overall efficiency of pump, η_O,

$$\eta_O = \frac{P_O}{P_I} \tag{10}$$

PO(PUM)=$\eta_O P_I$=0.8(5.2 kW) =4.16 kW=QAp , then

p = $\frac{4.16kW}{0.000133\text{ m3s}-1}$=31,278,195Pa =31.28 MPa.

Therefore, the maximum pressure produced by the selected gear pump is 31.28 MPa which is higher than the 27.18 MPa minimum required for the designed hydraulic specification as shown in Table 6. Using the selected gear pump provides excessive pressure available of about 13.1% to allow the remaining pressure to be used for cylinder 4 for a maximum angle of 140° in swinging the boom.

4. CONCLUSION

Through the mathematical method, the maximum pressure that the towable backhoe can withstand is 31.28MPa which is more than the required 27.18MPa. The variation of θ_1=45°, 50°, 60° with a fixed value of Maximum angle $\theta_4 = 15°$ for boom cylinder (Cylinder 1) and Maximum angle θ_3=14.5° for arm 2 cylinder (Cylinder 3) in retraction stroke were studied to determine the overall pumping effort used to overcome the moment effect where the fixed angle of frame is needed to optimize the minimum pressure required. The gear pump with a maximum pumping pressure of 31.28 MPa or 3600RPM is acceptable for the Hydraulically Operated Palm Oil Loader for the minimum pressure requirements for all θ_1=45°, 50°, 60°.

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