OPEN ACCESS
Saudi Journal of Engineering and Technology
Abbreviated Key Title: Saudi J Eng Technol
ISSN 2415-6272 (Print) |ISSN 2415-6264 (Online)
Scholars Middle East Publishers, Dubai, United Arab Emirates
Journal homepage: https://saudijournals.com

Original Research Article

Development of a Mobile Hydraulic Lifting Machine

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DOI: 10.36348/sjet.2024.v09i06.003 | **Received:** 01.05.2024 | **Accepted:** 08.06.2024 | **Published:** 28.06.2024

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Abstract

This paper develops a mobile hydraulic lifting machine for lifting heavy automobile engines in the central workshop of Olusegun Agagu University of Science and Technology, Okitipupa. It was fabricated using locally sourced materials for economic viability and affordability. Full welding was applied to all parts to avoid failure during lifting operation and avert accidents or injuries. The maximum designed load of the machine is 500 kg, and the minimum force produced by the hydraulic cylinder is 22.22 KN. The pressure generated by the Ram force with a piston area of 8.495 x 10^{-3} m^2 was 2.616 MPa and the effort required to overcome the force produced by the cylinder was 2103 N. The maximum circumferential stress and the allowable bending stress of the cylinder were 54.413 x 106 N/m^2 and 108.8 MPa which are far below the AISC recommended, this shows the beam is safe to lift without failure. The maximum shear force and the bending moment of the cantilever were found to be 17222 N and 6200 Nm respectively, and the maximum shear stress developed at the pivots were 17.54 MPa and 22.63 MPa respectively, which are far below the allowable stress. This shows that the developed machine is safe to use without failure.

Keywords: Hydraulic lifting, Allowable stress, Automobile engines, Bending moment, Cantilever.

1. INTRODUCTION

A hydraulic lifting machine is a device that uses hydraulic fluid to raise heavy loads, such as automobile engines. It is a device used to provide mechanical advantage, allowing for the transportation of weights that would be too heavy for a single person to handle [1]. Loading and unloading cargo, moving building materials, and putting together heavy machinery are just some of the many prevalent uses of hydraulic lifting machines in the transportation, construction, and industrial sectors. The development of mobile hydraulic lifting machines has been a crucial factor in increasing productivity and efficiency in various industries [2]. These machines are capable of lifting and moving heavy loads with ease, making them a popular choice for tasks that require heavy lifting. The evolution of mobile hydraulic lifting machines dates back to the early 20th century when the concept of using hydraulic power for lifting was first introduced. Over the years, mobile hydraulic lifting machines have undergone significant changes in design, functionality, and performance [3].

In recent years, with rapid development and innovation in technology, these machines have become more sophisticated, allowing for greater precision, speed, and control. They are now widely used in construction, manufacturing, logistics, and transportation [4]. The application of hydraulic systems in lifting machines has become increasingly popular because of their efficiency, reliability, and safety. These machines are used to lift heavy loads, such as building materials, machinery, and equipment, and move them to different locations with advancements in technology leading to the creation of more efficient and powerful machines. Integrating hydraulic systems into lifting machines has allowed for greater precision, control, and safety in lifting and moving heavy loads [5].

Hydraulic systems offer high power density, which means that they can generate a lot of force from a small amount of space. They are also highly responsive, which makes them well-suited for use in machines that require precise control. One of the key challenges in the development of mobile hydraulic engine lifting machines is to ensure that they are safe and reliable. These machines often work in hazardous environments, such as construction sites, where there is a risk of injury or damage to property [6]. Overall, the study of mobile hydraulic lifting machines is a multidisciplinary field that draws on principles from Mechanical engineering, hydraulic systems, Safety engineering, and more. As these machines continue to play a critical role in many industries, ongoing research and development will be essential to ensure their continued safety and efficiency.

In the design of hydraulic lifting systems lifting mechanical transmission control, many researchers have studied it and achieved good results. In the study of [7], a lifting machine was fabricated with a bearing capacity of 15 tons, a stroke of 10 metres, and a relatively high speed. Guo *et al.*, [8], studied the characteristics of different transmission forms in hydraulic hybrid power transmission from the aspects of transmission efficiency and speed change characteristics. They found out that adopting a new transmission form has a wider speed ratio range and higher transmission efficiency. According to the report of [9], the problems of inconvenient operation, unstable lifting speed, and large power loss in the use of hydraulic lifting machines were overcome. Wang and Chan, [10], introduced the system composition, main performance parameters of the hydraulic bolt lifting system, working principle, and its installation technology is analyzed to determine the optimal machinery transmission control scheme and analyzes the output characteristics of the scheme.

2. MATERIALS AND METHODS

2.1 Materials

The selection of the materials used in the construction of the hydraulic lifting machine was done based on the designed weight, structural resilience towards various types of forces, torsional rigidity, a factor of safety under the application of various loads, market availability, and cost constraints. The major components of the hydraulic lifting crane are base plate, vertical column, horizontal arm, lift cylinder, piston, hand lever, hook and wheels. Mild steel was used for the construction of major components of the hydraulic crane because of its machinability, strength, toughness, ductility, and hardness, and easily available and affordable.

2.2 Methods

2.2.1 Design of the Beam

The hydraulic lifting machine was designed to lift a maximum load of 500 kg. The beam is considered a cantilever and the structural analysis of the beam is shown in Figure 1 and the free body diagram (FBD) is shown in Figure 2. It was designed based on the following parameters:

i. Beam length is 1.6 m
ii. The point of application of lift is 0.36 m
iii. Maximum height is reached is 1.83 m

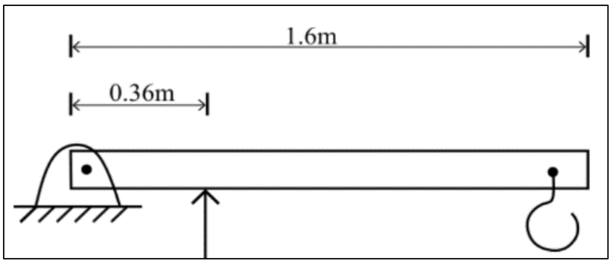

Figure 1: Structural Analysis of the Beam

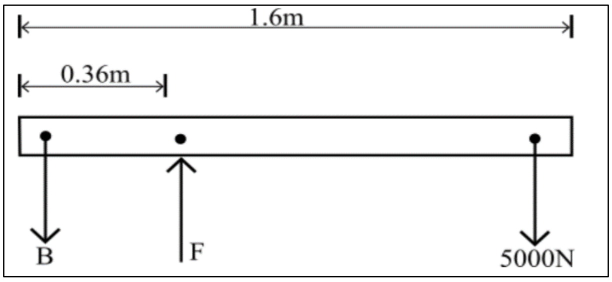

Figure 2: FBD of the Beam

Figure 3 shows a stress analysis of the beam when used to lift a maximum load 500 kg. Applying the principle of the moment, by taking moment about B, the minimum force produced at the hydraulic cylinder is 22.222 KN and the reaction R_p at the pivot P is 17.222 KN as illustrated by the structural analysis in Figure 4.

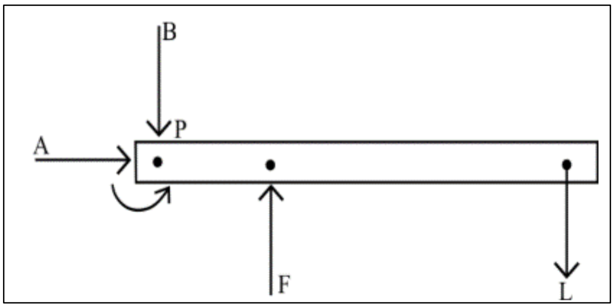

Figure 3: Stress Analysis of the Beam

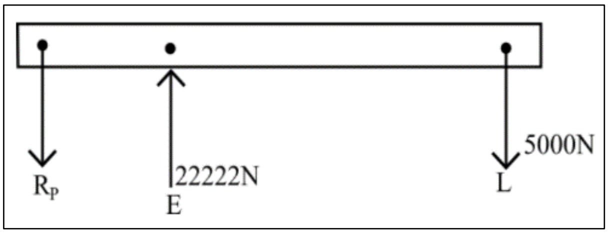

Figure 4: Beam Structural Analysis

The summary of the shear force and bending moment results is shown in Table 1 and the resulting shear force and bending moment diagram are shown in Figure 5. The maximum shear force and maximum bending moment that occurred in the arm are 17222 N and 6200 Nm respectively.

Table 1: Summary of Shear Force and Bending Moment Results

Length of Arm	0	0.36	0.36	1.6	1.6
Shear Force	17222	17222	5000	5000	0
Bending Moment	0	6200	6200	0	0

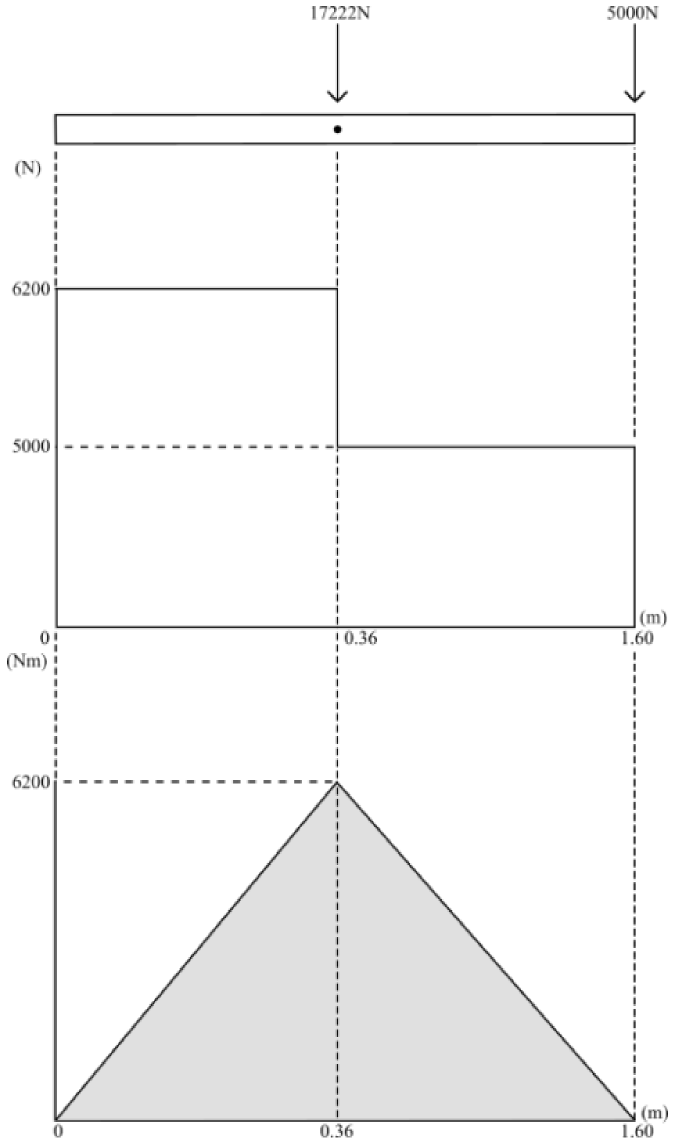

Figure 5: Shear Force and Bending Moment Diagram of the Beam

The cross-sectional area of the Arm is of hollow square section symmetrical about the centre, hence the neutral axis at the centre as shown in Figure 6.

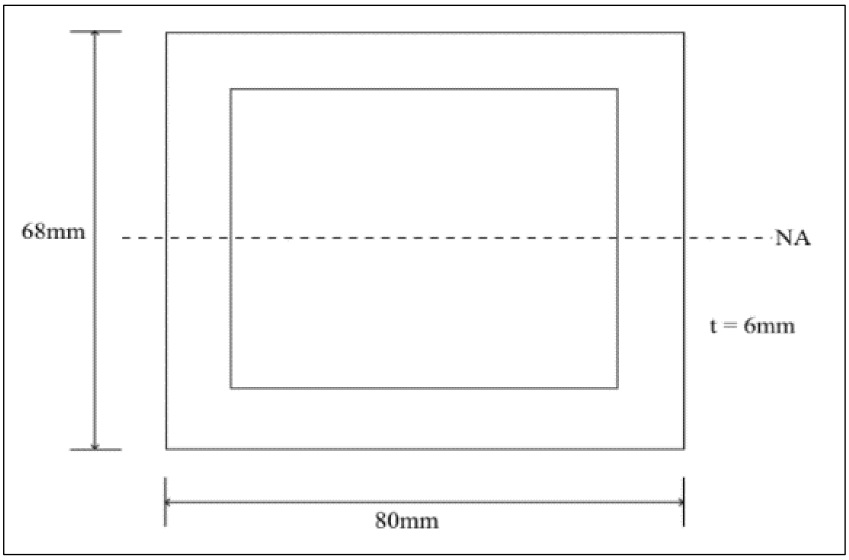

Figure 6: Cross-Sectional Area of the Arm

The moment of inertia of the hollow square section is given in Equation (1).

$$I = \frac{BH^3}{12} - \frac{bh^3}{12} \quad \text{.................. (1)}$$

Where: B is the outer breadth of the Arm, b is the inner breadth of the Arm, H is the outer height of the Arm, h is the inner height of the Arm.

At maximum loading of 500 kg, the arm is subjected to bending stress given in Equation (2)

$$\sigma = \frac{mc}{I} \quad \text{........................... (2)}$$

Where:

m is the maximum bending moment
c is the distance from the neutral axis (NA)
I is the moment of inertia of X section of the arm.

The bearing pin at the pivot P is subjected to double shear. The shearing area of the pivot is estimated using Equation 3 and the maximum shear stress developed at this pivot is estimated using Equation 4.

$$P_A = \frac{2\pi D^2}{4} \quad \text{........................ (3)}$$

$$M_s = \frac{SF}{P_A} \quad \text{.......................... (4)}$$

Where:

P_A is the shearing area
D is the pin diameter
M_S is the maximum shear stress

2.2.2 Design of Piston

A mild steel pipe of inner diameter 4 inches was bored on the lathe machine and honed on a honing machine to 104 mm. With a length of 35 cm, the piston was cast on an aluminum alloy and machined to 104 mm. in order to meet AISC standard, the following parameters were chosen for the design of the piston:

i. Stroke of the actuating cylinder is 28 cm
ii. Diameter of Ram is 40 cm
iii. Piston diameter is 140 mm

The schematic diagram of the actuating cylinder is shown in Figure 7.

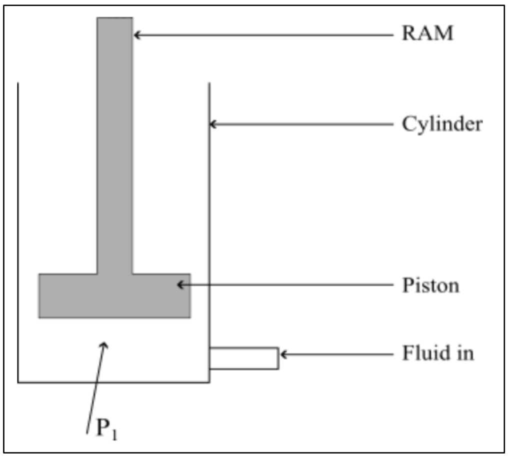

Figure 7: Actuating Cylinder

The piston area is estimated using Equation 5 and pressure generated by the effect of the ram force is calculated using Equation 6.

$$A = \frac{\pi D^2}{4} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (5)$$

$$P_1 = \frac{F}{A} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (6)$$

Where:

A is the cross-sectional area of the piston;
D is the ram diameter;
P_1 is the pressure generated by effect of ram;
F is the Ram force.

2.2.3. Design of the Master Cylinder

A steel pipe of diameter 30 mm was machined and precisely honed to 32 mm to form the master cylinder as shown in Figure 8.

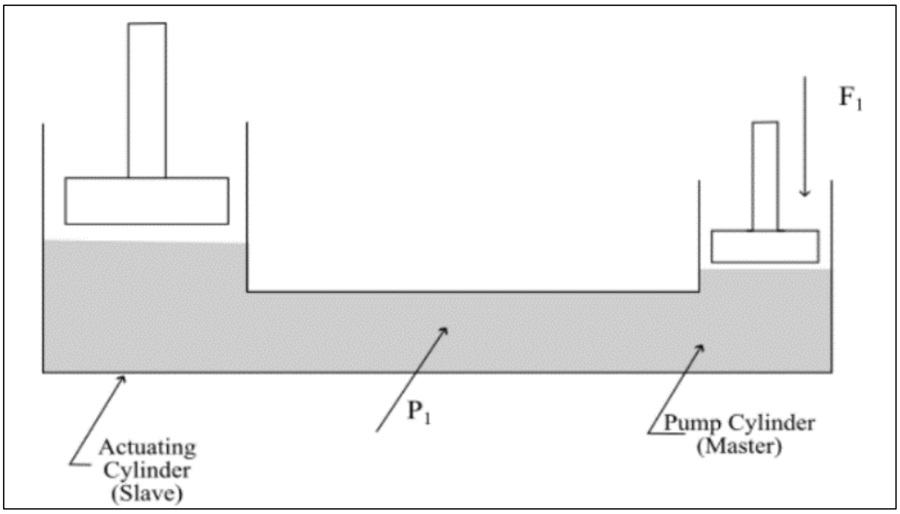

Figure 8: Schematic diagram of Master Cylinder

The area of the master cylinder is estimated using Equation (7), and the force developed by the master cylinder is given by Equation (8).

$$A_m = \frac{\pi D^2}{4} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (7)$$

$$F_1 = P_1 \times A_m \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (8)$$

Where:

P_1 is the pressure generated by effect of ram;
A_m is the area of master cylinder;
F_1 is the force developed by the master cylinder

The cantilever arm of the pump was designed to have mechanical advantage (MA) of 10 as shown in Figure 9. Taking moment about the pivot, the effort required is calculated to be 210.38 N or 21kgf.

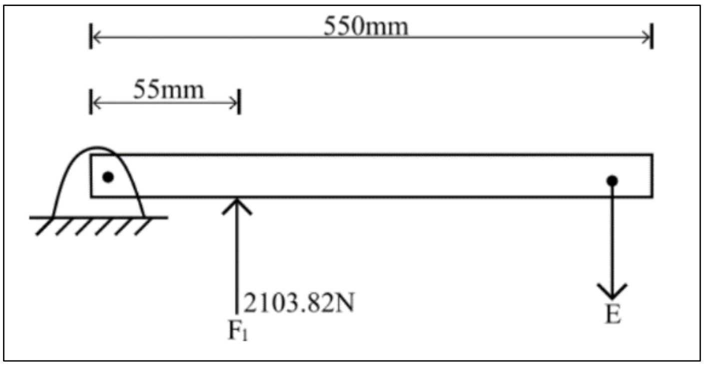

Figure 9: Cantilever Arm

The single acting cylinder can be assumed to be a thin-walled cylinder, since the thickness is about 1/10th of diameter, the circumferential stress, (σ_c) the cylinder is subjected to is given by Equation (9)

$$\sigma_c = \frac{pd}{2t} \ldots\ldots\ldots\ldots\ldots\ldots (9)$$

Where:

p is the maximum pressure inside the cylinder;
d is the diameter of the cylinder;
t is the thickness of cylinder

The allowable bending stress of the cylinder is calculated using equation (10), assuming factor of safety (k) of 2.

$$\sigma_b = \sigma_c \times k \text{ .... (10)}$$

The flow arrangement of the hydraulic lifting machine is shown in Figure (10).

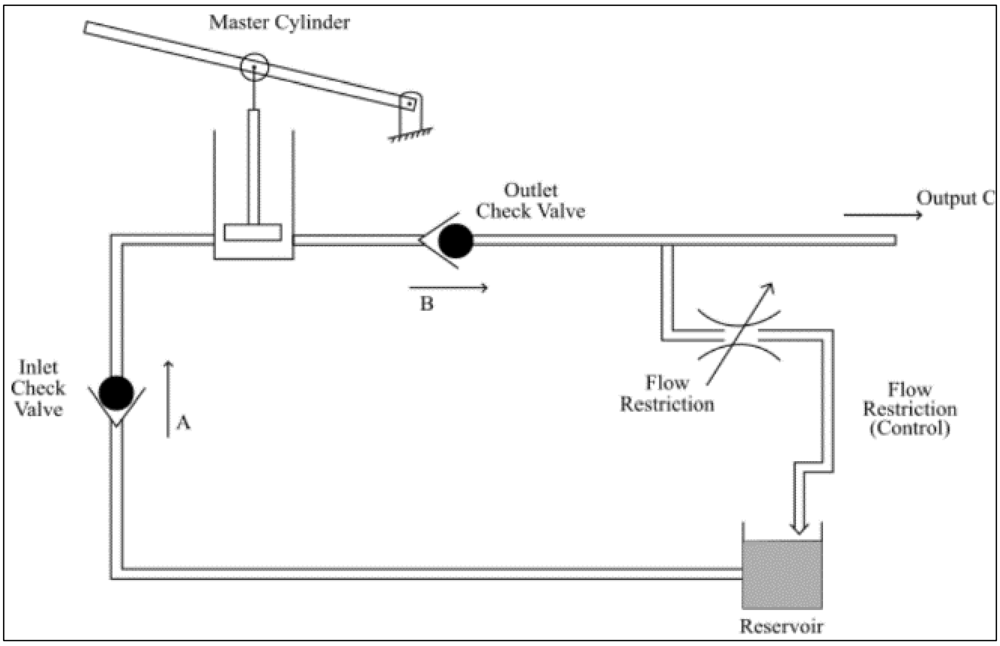

Figure 10: Flow Arrangement of Hydraulic Lifting Machine

The exploded views of the hydraulic lifting machine are shown in Figure 11.

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	Welded frame	Mild steel	1
2	pump cylinder	Mild steel	1
3	plunger	Mild steel	1
4	puller arm	Mild steel	1
5	Roller		3
6	handle	Mild steel	1
7	Fluid tube	copper tube	1
8	Crane Hook		1
9	piston		1

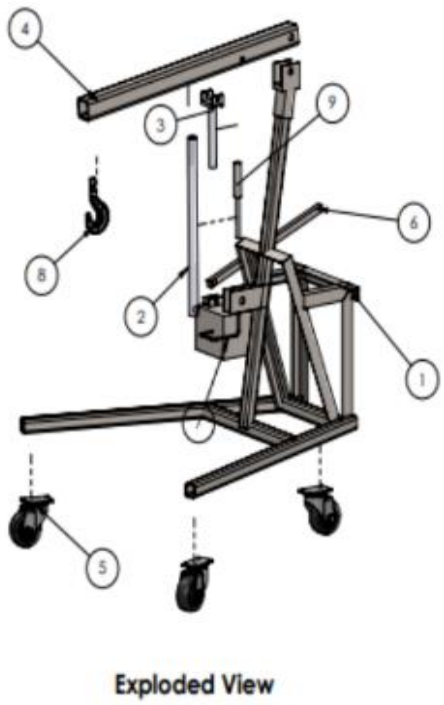

Figure 11: Exploded View of the Hydraulic Lifting Machine

The summary of the design analysis is shown in Table 2.

Table 2: Summary of the Design Analysis Result

S/N	PARTICULAR	EQUATION	CALCULATED VALUE
1	Moment of Inertia (m^4)	1	1.623×10^{-6}
2	Bending Stress of the Arm (MPa)	2	152 (well below allowable stress)
3	Shearing Area of the pivot (m^2)	3	982×10^{-6}
4	Maximum Shear stress at the pivot (MPa)	4	22.63 (well below allowable stress)
5	Area of the piston (m^2)	5	8.495×10^{-3}
6	Pressure generated by effect of Ram (MPa)	6	2.616
7	Master cylinder Area (m^2)	7	8.14×10^{-4}
8	Force developed by Master cylinder (KN)	8	2.1038
9	Circumferential stress of the cylinder (MPa)	9	54.4
10	Allowable Bending stress of the cylinder (MPa)	10	108.8 (Far below AISC value)

The complete assembly of the hydraulic lifting machine is shown in Picture 1.

Picture 1: Complete Assembly of the Mobile Hydraulic Lifting Machine

3. RESULT AND DISCUSSION

The performance evaluation of the machine was carried out under loading and unloading conditions, to determine the number of strokes and the time it takes to reach the designed maximum height. The graph showing the weight against the number of strokes and lifting and lowering time is shown in Figure 13.

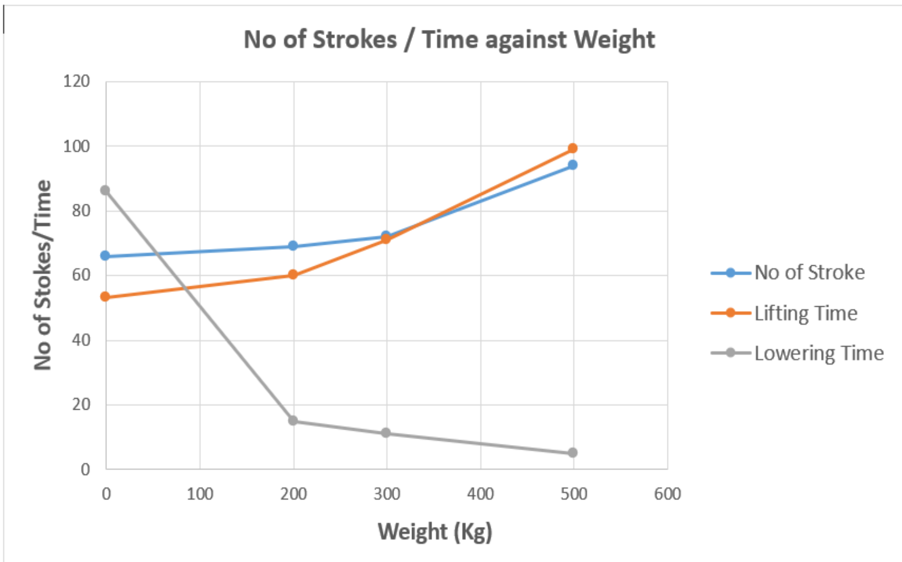

Figure 13: Graph of Weight against Number of stroke/Time

It was observed that the number of strokes increases with an increase in weight as a result of lower pressure generated by the hydraulic pump. The lifting time increases with an increase in the weight as a result of the lower force impacted on the plunger, conversely, the lowering time reduces with an increase in weight as a result of the effect of gravitational force.

4.0 CONCLUSION

A mobile hydraulic lifting machine has been developed and evaluated. The machine was used to lift a load of 500 kg to a maximum height of 2.20 m within 99 seconds. The circumferential and allowable bending stresses of the cylinder were found to be 54.3 MPa and 108.8 MPa respectively which are far below the AISC values. This indicates that the machine is safe to use.

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