**International Journal of Mechanical Engineering and Technology (IJMET)**
Volume 9, Issue 10, October 2018, pp. 84–101, Article ID: IJMET_09_10_008
Available online at http://www.iaeme.com/ijmet/issues.asp?JType=IJMET&VType=9&IType=10
ISSN Print: 0976-6340 and ISSN Online: 0976-6359



# DEVELOPMENT AND PERFORMANCE ANALYSIS OF HORIZONTAL WASTE PAPER BALING MACHINE

**Igbinigie Osamwonyi, I. P. Okokpujie* AND J. O. Dirisu**
Department of Mechanical Engineering, Covenant University,
P.M.B 1023, Ota, Ogun State, Nigeria
*Corresponding Author

**D. I. Igbinowmahia**
Department of Mechanical Engineering, University of Benin, Benin City Edo State, Nigeria

**K. O. Okokpujie**
Department of Electrical and Information Engineering,
Covenant University, Ota Ogun State, Nigeria

**ABSTRACT**

*In an effort to mitigate the problem of increasing amount of generated waste papers going to dumpsites for burning directly causing global warning. Horizontal waste paper baling machine was designed, Manufactured and tested using high local content and readily available materials in such a way that, maintenance will not be expensive to ensure sustainability and reliability. The principal parameters of the design included the maximum force (10 KN) of the compression, the distance the piston rod has to move before compressing the waste paper (piston stroke, 609mm), cycled time of 9 minutes and the system pressure. The major components of the waste paper baling machine designed includes the frame, electric motor, coupling, hydraulic pump, pressure hose, directional control valve, filter, pressure gauge, thermometer, tank, and hydraulic cylinder. The machine was tested for performance with a waste paper such as white papers of A4 size used for Photostat and printing upon and found to be able to bale the waste paper into a bale size of 400mm × 400mm × 700mm in 6 seconds and weight of 15kg, the efficiency of the machine was 80%, using the hydraulic power system. The bales produced by this device will aids in simple management, storage and transportation and will enhance the recycle industry.*

**Keywords:** Waste paper, baling machine, hydraulic pump, hydraulic cylinder, machining.

**Cite this Article:** Igbinigie Osamwonyi, I. P. Okokpujie, J. O. Dirisu, D. I. Igbinowmahia and K. O. Okokpujie, Development and Performance Analysis of Horizontal Waste Paper Baling Machine, International Journal of Mechanical Engineering and Technology, 9(10), 2018, pp. 84–101.
http://www.iaeme.com/IJMET/issues.asp?JType=IJMET&VType=9&IType=10

## 1. INTRODUCTION

Manufacturing industries are facing several problems in storing and handling Waste, because a large amount of space is required to store this Waste paper. In order to conquer these problems, the Waste can be compressed and stored in a block form. Today, all the modern manufacturing enterprises are trying to develop best optimized reduced weight and cost effective products that meet the intended design functionally and reliably.

Waste is defined as any material that is considered to be of no further use to the owner and is hence, discarded [1]. Paper is a thin material mainly used for writing, printing, and drawing upon and for packaging. It is produced by pressing together moist fibres, typically cellulose pulp derived from wood, and drying them into flexible sheets [2]. Honestly, any business in Nigeria generates a mountain of paper, from manufacturers, offices, schools and even retail locations like supermarkets and bookshop. This mountain of paper is also something that can be heavily recycled. Increasing pollution regulations make incinerating paper waste an issue, so it increases the need to produce easy movable and transportable paper waste. So baling waste paper is the best way to handle this massive amount of waste [3].

A baling machine is one that compresses waste paper materials into compact defined shapes, usually cubes or cuboids, making it easier to store, handle and transport for recycling. Waste paper baling machine would definitely improve waste paper recycling which has a positive effect on the growth of the economy [4, 26, 34-35].

Hydraulic ram is a mechanical actuator that is used to give a unidirectional force through a unidirectional stroke. Hydraulic actuators are the end result of Pascal’s law. This is where the hydraulic energy is converted back to mechanical energy. This can be done through the use of a hydraulic cylinder which converts hydraulic energy into linear motion and work. Hydraulic Cylinder that was chosen was good for the compressing of waste papers which is the double acting type. Cylinders are applied based on force and speed of actuation. Cylinders are sensitive to force media viscosities and performed it tasks effectively and efficiently [5]

Today’s hydraulic fluids serve multiple purposes. The major function of a hydraulic fluid is to provide energy transmission through the system which enables work and motion to be accomplished. Hydraulic fluids are also responsible for lubrication, heat transfer and contamination control. Selecting a lubricant, viscosity, viscosity index improver, anti-wear agent, anti-foam agent, seal compatibility, base stock and the additive package was considered. Three common varieties of hydraulic fluids found on the market today are petroleum-based, water-based and synthetics. Since the fluid power industry is standardized on mineral oil based hydraulic fluids. So, Mineral-based fluids will be used for this waste papers baling machine since it offers a low-cost, high-quality, readily available selection. Mineral oil and vegetable based hydraulic fluids are less sensitive to high system operating temperatures. Pump normally operates at 92ºC [6]. The ISO viscosity rating system (cst at 40ºC and 100ºC) is recommended Since, is compatible with the hydraulic components. Viscosity is 39 cst at 40ºC and 5.9 cst at 100ºC. Mineral oil as used for hydraulics systems has a density of approx. 850 $kg/m^3$ (ISO 1998)

The hydraulic pump transmits mechanical energy into hydraulic energy. This is done by the movement of fluid which is the transmission medium. All hydraulic pumps work on the same principle, which is to displace fluid volume against a resistant load or pressure. There are several types of hydraulic pumps including gear, vane and piston. The gear pump which is the external gear pump will be the right choice. A gear pump produces flow by carrying fluid in between the teeth of two meshing gears. One gear is driven by the drive shaft and turns the idler gear. The chambers formed between adjacent gear teeth are enclosed by the pump housing and side plates. A partial vacuum is created at the pump inlet as the gear teeth unmeshed. Fluid flows in to fill the space and is carried around the outside of the gears. As the teeth mesh again at the outlet end, the fluid is forced out. Volumetric efficiencies of gear pumps run as high as 93% under optimum conditions. External-gear pumps are comparatively immune to contaminants in the oil, which will increase wear rates and lower efficiency that is the reason the pump must be taking suction form a reservoir through a filter, but sudden seizure and failure are not likely to occur. In most catalogs Gear pumps are rated at 1,200RPM to 1800RPM [6].

A centrifugal pump used rotating impeller to create flow by adding energy to the fluid. Centrifugal pumps are commonly used to move liquids through pipes [7]. The fluid enters the pump impeller along or near to the rotating axis and is accelerated by the impeller, flowing vertically outwards into a diffuser or volute chamber (casing), from where it exits into the downstream piping [8]. A shaft is a rotating machine element which is used to transmit power from one place to another [9-11]. The shaft is the connection between impeller and driver unit which is in most cases is an electric motor but can also be a gas turbine.

Machining is the process of removing precise amount of material from the work piece to attain the desired shape. Machining is one of the important aspects during developing the paper bailing machine. The different component has to be determined during the machining operation in other to avoid the corrosion, vibration, and to produce good quality component. [12-33].

Furthermore, the accomplishments of various researcher's efforts related to the current study have been reviewed in the previous subsections of this work. The aim of this project is to develop horizontal waste paper baling machine with high local content, which will reduce cost of transportation of waste product, global warning and will also enhance the recycle industry.

## 2. MATERIAL AND METHODS

Detail design of the system: design parameters for sizing a hydraulic circuit

The required force for compressing average volume of $0.05m^3$ of papers ranging from 8KN to 10KN; taking the upper limit of force and the following design specification material must be known [10]

Maximum force required, $F_1$=10KN, Total stroke required, L = 609mm, Total cylinder cycle time, T =10sec.

Maximum pressure allowed, P=2.400N/$mm^2$

### 2.1. Design of the hydraulic ram

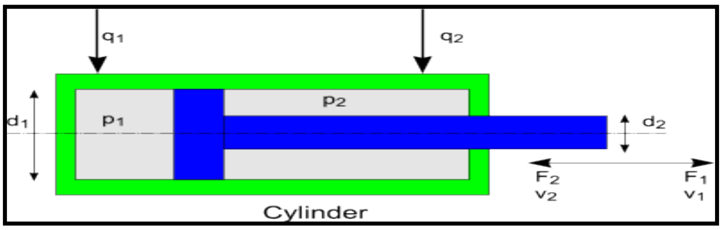



**Figure 1** Hydraulic Cylinders

$d_1$ = Diameter of cylinder (mm), $F_1$= Force in extending piston (N), $d_2$ = Diameter of piston rod (mm) $F_2$ = Force in retracting piston (N), $P_1$ = Pressure on face of piston (mm), $V_1$ = Velocity extending (m/s)

$P_2$ = Pressure on rod side of piston (mm), $V_2$ = Velocity retracting (m/s), $q_1$= Fluid flow to the face of piston ($m^3$/s), $q_2$ = Fluid flow to the rod side ($m^3$/s)

Sizing hydraulic ram in Figure 1 at least 1.1 safety factors was used for load balance or work force to get full force and/or nominal force buildup time. Minimum Cylinder Bore is given by,

$$d_1 = \sqrt[2]{\frac{xF_1 \times \frac{1}{P}}{0.7854}} \quad 1$$

Where, $0.7854d_1^2 = \frac{\pi d_1^2}{4}$

$d_1$ = Diameter of cylinder, $F_1$= Force in extending piston is 10000N, P = Maximum pressure allowed is 2.400N/$mm^2$, x = safety factor is 1.1

Minimum cylinder bore $= \sqrt[2]{\frac{1.1 \times 10000 \times \frac{1}{2.400}}{0.7854}}$

$d_1$ = 76.4mm bore diameter.

The piston rod diameter is normally given as half of the diameter of ram that is 50mm to 150mm

Therefore, Piston rod diameter, $d_2 = \frac{d_1}{2}$

Where, $d_1$ = Diameter of cylinder is 76.4mm, $d_2$ = Piston rod diameter $d_2 = \frac{76.4}{2}$ = 38.2mm

Therefore, Extraction piston area is given by,

$$A_1 = \frac{\pi d_1^2}{4} \quad 2$$

Where, $d_1$ = Diameter of cylinder is 76.4mm = $\frac{\pi \times 76.4^2}{4}$ = 4585.9$mm^2$ = 0.004585$m^2$

$$\text{Piston rod area is given by, } A_2 = \frac{\pi d_2^2}{4} \quad 3$$

Where, $d_2$ = Piston rod diameter is 38.2mm = $\frac{\pi \times 38.2^2}{4}$ = 1146.2$mm^2$ = 0.001146$m^2$

## 2.2. Extension fluid pressure of hydraulic cylinder

The Pressure generated inside the hydraulic cylinder is given as

$$\text{Fluid pressure, } P_1 = \frac{F_1}{A_1} \quad 4$$

Where, $F_1$ = Force in extending piston is 10000N, $A_1$ = Extension piston area is 4585.9$mm^2$, $P_1$ = Extension fluid pressure, Extension fluid pressure = $\frac{10000}{4585.9}$ = 2.181N/$mm^2$ = 316.2Psi

## 2.3. Cylinder volume capacity

Volume displacement of a hydraulic cylinder can be calculated as

$$\text{Volume, } q = (A_1 - A_2)mm2 \times L(mm) \quad 5$$

Where, q = Volume displacement, $A_1$ = Extension piston area is 4585.9$mm^2$, $A_2$ = Piston rod area is 1146.2$mm^2$

L = Stroke is 609mm

Therefore, q = (4585.9-1146.2) $mm^2$ × 609mm = 2094777$mm^3$ = 0.002094$m^3$

## 2.4. Cylinder flow rate

Stroke is 609mm (24'') then, 1218mm is for cylinder extends and retracts.

Total Flow rate is given by, $Q = \frac{A_1 \times L}{T}$ 6

Where, Q = total flow rate, $A_1$ = Extension piston area is 4585.9$mm^2$ ≡ 0.004585$m^2$, L = Stroke for cylinder extends and retracts is 1218mm, T = cycle time is 10seconds

Therefore, $= \frac{0.004585 \times 1.218}{10}$ Q = 0.0006$m^3$/s

## 2.5. Cylinder velocity

For travel time of 10sec for a distance of 609mm (24''), Cylinder Velocity is given as

$= \frac{Q\ (\mathrm{m3/s})}{(A_1 - A_2)(sq-m)}$ 7

Where, Q = total flow rate is 0.0006$m^3$/s, $A_1$ = Extension piston area is 4585.9$mm^2$ ≡ 0.004585$m^2$, $A_2$ = Piston rod area is 1146.2$mm^2$ ≡ 0.001146$m^2$, V = cylinder velocity

Therefore, $= \frac{0.0006}{0.004585 - 0.001146}$ = 0.17m/s

## 2.6. Extended velocity of the hydraulic cylinder

Forward velocity is given by

$V_1 = \frac{Q}{A_1}$ 8

Where, $V_1$ = forward velocity, Q = total flow rate is 0.0006$m^3$/s, $A_1$ = Extension piston area is 4585.9$mm^2$ ≡ 0.004585$m^2$, Therefore, $= \frac{0.0006}{0.004585}$ = 0.13m/s

## 2.7. Extension force of the hydraulic cylinder

Force during extension is given by

$F_1 = P_1 \times A_1$ 9

Where, $F_1$ = Force in extending piston, $A_1$ = Extension piston area is 4585.9$mm^2$, $P_1$ = Extension fluid pressure is 2.181N/$mm^2$ Therefore, = 2.181×4585 =10000N

## 2.8. Retraction force of the hydraulic cylinder

Force during retraction is given by

$F_2 = P_1 \times (A_2 - A_1)$ 10

Where, $F_2$ = Force in extracting piston, $A_1$ = Extension piston area is 4585.9$mm^2$, $A_2$ = Piston rod area is 1146.2$mm^2$ , $P_1$ = Extension fluid pressure is 2.181N/$mm^2$

Therefore, = 2.181× (4585-1146) = 7504N

## 2.9. Retraction fluid pressure of hydraulic cylinder

Hydraulic pressure during the retracting stroke is

Fluid pressure, $P_2 = \frac{F_2}{(A_1 - A_2)}$ 11

Where, $F_2$ = Force in retracting piston is 7504N, $A_1$ = Extension piston area is 4585.9mm$^2$, $A_2$ = Piston rod area is 1146.2mm$^2$, $P_2$ = Retraction fluid pressure. Therefore, $= \frac{7504}{4585.9-1146.2}$ = 2.182N/mm$^2$ = 316.2Psi

## 2.10. Extension hydraulic cylinder power

Cylinder KW power during the extending stroke

$$\text{Power} = F_1 \times V_1 \qquad 12$$

Where, $F_1$ = Force in extending piston is 10000N, $V_1$ = forward velocity is 0.13m/s, $P_e$ = extension cylinder power. Therefore, = 10000×0.13 = 1300W

## 2.11. Retraction hydraulic cylinder power

Cylinder KW power during the retracting stroke

$$\text{Power} = F_2 \times V_2 \qquad 13$$

Where, $F_2$ = Force in retracting piston is 7504N, $V_2$ = Return velocity is 0.17m/s, $P_r$ = retraction cylinder power

Therefore, = 7504×0.17 = 1276W

## 2.12. Hydraulic oil pump-displacement

Pump displacement needed for m$^3$/s of output flow is

$$= \frac{Q,(\frac{m3}{s})}{N(rps)} \qquad 14$$

In most catalogs Gear and Vane pumps are rated at 1,800RPM (30RPS) while Piston pumps are rated at 1200RPM,(Tutle,2012) so 1800RPM(30rps) was chosen for the pump

Where, Q = total flow rate is 0.0006m/s$^2$, N = pump speed is 30rps, $q_d$ = pump displacement

Therefore,

$q_d = \frac{0.0006}{30}$ = 0.00002 cubic meter per revolution

## 2.13. Hydraulic oil pumps-horsepower required

Horsepower required by hydraulic pumps. Formula for calculating horsepower required by a hydraulic pump

Can be expressed as

$$P_w = QP \qquad 15$$

Where, $P_w$ = pump rating in watt (W), Q = required pump capacity is 0.0006m$^3$/s, $P_1$ = required pressure 2.181N/mm$^2$≡2.181N/m$^2$. Therefore, Pw = 0.0006×2181000 = 1308.6watt = 1.75hp

## 2.14. Efficiency

Note that the equation above can be used for a ***100%*** efficient pump - which in the practical life is never true. An overall efficiency of μ ≤ ***93%*** ≤ μ is common. (Tutle, 2012).

The equation 3.17 can be modified to

$$P_w = QP / (\mu / 100) \qquad 16$$

Where, μ = overall efficiency (%) and 93% overall efficiency was chosen for the design.

Therefore, $P_w = \frac{0.0006 \times 2181000}{0.93}$ = 1407W =1.89hp

## 2.15. Motor power selection

Hydraulic pump rating =1407W

Since the machine designed for Small and medium scale enterprise, a single phase electric motor is desirable to power the pump. Assuming power transmission of 95%, power rating of electric motor is therefore

1.89/0.95 =1.99HP (2.01HP≡1500W was selected)

## 2.16. Hydraulic hose design: Pressure Lines

Good practice demands for a maximum velocity of 4.6m/s in a pressure line up to 500 Psi.

Therefore,

$$A = Q/V \quad (17)$$

Where, Q = Total flow rate is 0.0006m$^3$/s, V = maximum velocity in a pressure line up 500Psi is 4.6m/s

D = internal diameter hose in the pressure lines, A = Area of the pressure hose

Therefore, = 0.0006/4.6, A = 130.4mm$^2$

$$A = \frac{\pi D^2}{4} \quad (18)$$

Where, A = Area of the pressure hose is 130.04mm$^2$, 130.4 = $\frac{\pi D^2}{4}$ D = 13mm

Therefore, a minimum of 13mm internal diameter hose is needed for the pressure lines.

## 2.17. Suction/return lines

For the pump suction and return lines, the maximum velocity by practice is 1.2m/s for pressure up to 500Psi

The diameter of the pipe is given by,

$$D = \sqrt{\frac{4Q}{\pi V}} \quad (19)$$

Where, Q = Total flow rate is 0.0006m$^3$/s, V = maximum velocity in a suction and return line is 1.2m/s

D = internal diameter hose in the suction and return lines. Therefore,

$= \sqrt{\frac{4 \times 0.0006}{\pi \times 1.2}}$ = 25mm

A minimum of 25mm internal diameter pipe is therefore used for the suction and return pipes.

## 2.18. Hydraulic reservoir design

The major consideration in designing the reservoir is the volume. The minimum volume of the reservoir should equal volume of the whole fluid in the system, plus volume in the reservoir that will allow for sludge and dirt.

$$Volume\ of\ fluid\ in\ cylinder = \frac{\pi D^2}{4} \times L \quad (20)$$

Where, D = maximum bore diameter is 76.2mm, L = Stroke of cylinder is 609.6mm, V = volume of fluid in cylinder. Therefore, V= $\frac{\pi 76.2^2}{4} \times 609.6$, s = 2094777mm$^3$ = 0.002094m$^3$

Length of each of the pressure hose = 0.7m, Diameter of hoses = 0.011m, Volume of the three hose = 3 × $\frac{\pi \times 0.011^2}{4}$ × 0.7 = 0.0001995m$^2$

Diameter of each of the suction and return hose = 25mm = 0.025m

Volume of both hoses = 2× $\frac{\pi \times 0.025^2}{4}$ × 0.7 = 0.0006873m$^3$

Total volume of the fluid in the system = 0.002094 + 0.0001995 + 0.0006873 = 0.002980m$^3$

Often over sizing the reservoir will allow more fluid dwell time to allow cooling through the process of thermal radiation. Giving 40$^0/_0$ allowance for other joints and space in the reservoir:

$\frac{40}{100} \times 0.002980 = 0.001192\text{m}^3$ , Total volume of reservoir = 0.001192 + 0.002980 = 0.004172m$^3$

## 2.19. STRUCTURAL MEMBERS: Machine Frame

Material = Structural steel

Design stress. $\sigma = \frac{yield\ point\ stress}{safety\ factor}$ 21

Where, yield point stress for structural steel is 250N/mm$^2$

Safety factor = 2

Therefore, $\sigma = \frac{250}{2} = 125\text{N/mm}^2$

Maximum load on frame = 10000N

Four members carrying this load equally, each will bear 2500N.

## 2.20. Stress on Members

The stress on each member is given by

$\sigma = \frac{F}{A}$ 22

Where, F = force each member will bear is 2500N, $\sigma$ = Design stress is 125N/mm$^2$, A = area of the structural-steel. Therefore, $A = \frac{F}{\sigma} = \frac{2{,}500}{125} = 20\text{mm}^2$

Since angle bars are readily available, it was chosen from commercially available sizes.

Therefore, the properties of structural- steel Equal legs of $50 \times 50\ \times 4mm$

Angle with an approximate area of 480mm$^2$ was selected

## 2.21. Elongation Check

Elongation for a total base plate length of the machine, L = 1640mm

$\sigma = \frac{FL}{AE}$ 23

Where, F = force each member will bear is 2500N, E = modulus of Elasticity is 200GPa = 200kN/mm$^2$ for steel

L = a total base plate length of the machine is 1640mm, A = approximate area of the structural-steel is 480mm$^2$

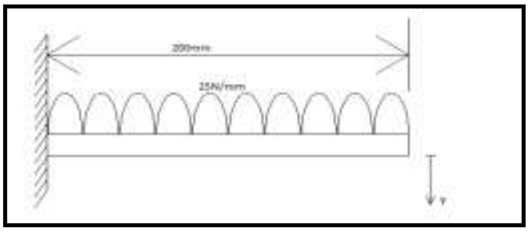
Therefore, $\sigma = \frac{2500 \times 1640}{480 \times 200 \times 10^3}$ = 0.0427mm

This is acceptable considering the kind of machine.

## 2.22. Ram Head

The ram head is loaded as shown in figure 3.9. Half of it can then be modeled as a cantilever with a uniformly distributed loaded.

Fig.2. uniformly distributed loaded of the ram head with a central point load



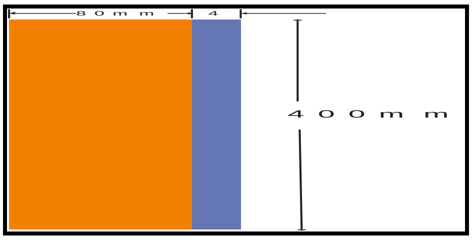




**Figure 2** Cantilever with uniformly distributed loaded of the ram head. **Figure 3** Beam of timber and steel (composite section)

The deflection ***y*** of the end of the cantilever of length L carrying a uniformly distributed load ω, is given by

$$y = \frac{wl^4}{8EI} \quad 24$$

*Where*, w **=** load per length on the beam, **E =** young's modulus, I = moment of inertia,

$$I = \frac{bt^3}{12} = \frac{wl^4}{8Ey}$$

$$t = \sqrt[3]{\frac{3wl^4}{2Eyb}} \quad 25$$

Where, Design for a deflection of 0.5mm, since the span ranging from 20mm to 500mm (Rajput, 2008)

and W =$F_1$ =10KN, L = half of the length of the ram head is 200mm, E = 200GPa = 200N/mm$^2$ for steel

$$B = 400mm. \text{ Therefore, } t = \sqrt[3]{\frac{3 \times 10000 \times 200^4}{2 \times 200000 \times 0.5 \times 400}} \; t = 84mm$$

Due to weight considerations of available materials, 4mm plate is used in Conjunction with wood and it is brazed as a composite beam of span 400mm consists of a timber section 80mm wide and 400mm deep with a steel plate 400mm deep and 4mm thick fixed with timber in Figure 3. The composite beam is subjected to a point load of 10kN at middle of the beam. The deflection of the beam under the load can be calculated as; From the geometry of the composite beam, the center of gravity of the beam section coincides with the center of gravity of the timber section. Therefore, flexural rigidity for the timber section about its center of gravity,

$EI_{(timber)}$ (Rajput, 2008) 26

Where, E is modulus of elasticity of timber is 10Gpa or 10kN/mm$^2$, I is moment of inertia of timber

Therefore,

$= (10\times10^3) \times [\frac{80\times400^3}{12}]$N-mm$^2$ = 4267×10$^9$ N-mm$^2$

Similarly,

$EI_{(steel)}$ 27

Where, E is modulus of elasticity of steel is 200Gpa or 200kN/mm$^2$, I is moment of inertia of steel

Therefore,

$= (200\times10^3) \times [\frac{4\times400^3}{12}]$N-mm$^2$ = $4267\times10^9$ N-mm$^2$

Therefore, total flexural rigidity for the composite section about its centre of gravity

$\sum EI = E_1I_1+E_2I_2$ 28

$\sum EI = EI_{(timber)} + EI_{(steel)}$

$\sum EI = 4267\times10^9 + 4267\times10^9 = 8534\times10^9$ N-mm$^2$

Deflection at the centre of the beam,

$y_c = \frac{wl^3}{48\sum EI}$ 29

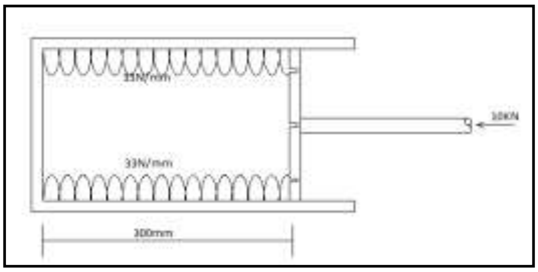
Where, W= maximum force applied is 10000N, L = length of the ram head is 400mm

$\sum EI$ = total flexural rigidity for the composite section of the ram head is $8534\times10^9$ N-mm$^2$

Therefore,

$y_c = \frac{10000\times400^3}{48\times8534\times10^9}$ = 0.0015mm. Therefore, the design is safe

## 2.23. Compression chamber walls



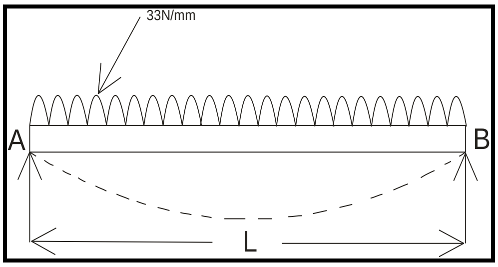




**Figure 4** Compression Chamber Walls **Figure 5** uniformly distributed load of the side wall

Designing for the worst case, when the force that will be applied has being stop, the compression force is transmitted to the lateral axis, since the force is not completely transmitted to the walls when compressing, the side walls can be modeled considering it as a simply supported beam of length and carrying a uniformly distributed load of per length as shown in Fig. 5

From the geometry of the Fig. 5 the reaction at A is same as reaction B

$R_A = R_B = \frac{wl}{2}$ 30

For the uniformly distributed load,

$W = wl$ 31

Uniformly distributed load,

$$w = \frac{10000}{300} = 33.33N/mm$$

For a simply supported beam, the maximum deflection occurs at the mid-span.

This deflection is given by

$$y_{max} = \frac{5wl^4}{384EI} \quad 32$$

$$I = \frac{5wl^4}{384Ey_{max}}$$

Where, w = uniformly distributed load is 33.3N/mm,E is modulus of elasticity of steel is 200Gpa or 200kN/mm$^2$

Putting $y_{max}$ =0.5mm, since the span ranging from 20mm to 500mm (Rajput. 2008), I = moment of inertia of side plate

Therefore,

$$I = \frac{5\times 33.33\times 300^4}{384\times 200000\times 0.5} = 35152.73\text{mm}^4$$

$$I = \frac{bt^3}{12} \quad 33$$

Therefore, $t = \sqrt[3]{\frac{12I}{b}}$

Where, I = moment of inertia of side plate is 35152.73mm$^4$, b = length of the side plate is 400mm, t = thickness of side plate

$$= \sqrt[3]{\frac{12\times 35152.73}{400}} = 10.2\text{mm}$$

## 2.24. Design of bale chamber band

The bands or crossbar provides support for the baling chamber plates and also supports the upper weight of the machine. The bar can be modeled as shown in Figure 6.

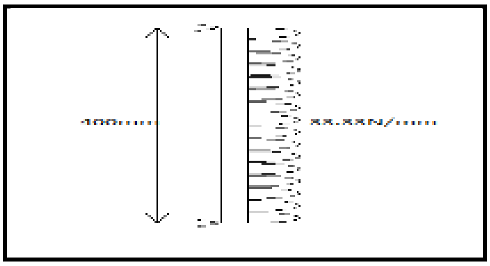


**Figure 6** Bale Chamber Band

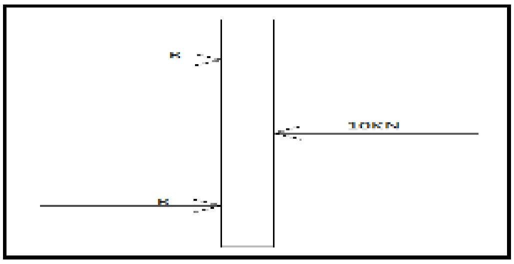


**Figure 7** Door Hinges

Recall from equation 25,

$$y_{max} = \frac{5wl^4}{384EI}$$

$$I = \frac{5wl^4}{384E_{max}} = \frac{5\times 33.33\times 400^4}{384\times 200000\times 5} = 122593.1\text{mm}^4$$

From table of angles, a 50x50x6 is selected with second moment of area of 12.8cm$^4$.

The baling chamber door will bear the same force as the ram head therefore; the same configuration is used for the door as the ram head.

## 2.25. Design of Door Hinges

The door hinges resist the force of the ram head against the papers to effect baling. The worst loading a hinge will resist is when its bears the whole force. The hinge is likely to fail due to tension. The hinge is modeled as shown in Figure 7

The door is model as a simply supported beam with a central point load

Moment of inertia of the door section

$$I = \frac{wl^3}{12} \quad 34$$

Where, b = thickness of the door is 4mm, d = depth of the door is 400mm, I = moment of inertia of the door

Therefore, $= \frac{4 \times 400^3}{12} = 21333333mm^4$

The maximum deflection of the door

$$y_c = \frac{wl^3}{48EI} \quad 35$$

Where, $y_c$ = deflection of the door at its center, w = maximum force on the door is 10000N

E = modulus of Elasticity is 200GPa = 200N/mm$^2$ for steel, I = moment of inertia of the door section is 21333333mm$^4$

Therefore,

$$\frac{10000 \times 400^3}{48 \times 200000 \times 21333333} = 0.003125mm$$

## 2.26. Pump Efficiency

During initial performance quality checked of the machine operation. Two (2) gallon of mineral oil was poured into the reservoir tank and the electrical motor was switch on. Immediately, the pump started taking suction and returning it back to the tank via the safety valve due to that lever on the directional control valve has not been activated for compression.

The directional control valve was actuated for fluid to flow to the hydraulic cylinder for extension (2.8sec) and retraction (2.2sec) of the platen, the time for the processes was 5 second.

Therefore, Recall from equation 6, for the total flow rate is

Actual total Flow rate,

$$Q = \frac{A_1 \times L}{T} \quad 36$$

Where, Actual cylinder diameter bore ($d_1$) is 58mm (2.28''). The actual hydraulic cylinder is 529mm (23'') for a stroke then, 46'' is for cylinder extends and retracts. Q = Actual total flow rate, $A_1$ = Actual Extension piston area is 2642mm$^2$ ≡ 0.002642m$^2$, L = Actual Stroke for cylinder extends and retracts is 1058mm, T = Actual cycle time is 5seconds

Therefore, $= \frac{0.002642 \times 1.058}{5}$. Actual flow rate, Q = 0.00055m$^3$/s

Recall from equation 16, for the fluid power in horsepower is

$$P_w = QP \quad 37$$

Where, $P_w$ = pump rating in watt (W), Q = required pump capacity is 0.00055m$^3$/s, P = required pressure is 2181000N/m$^2$. Therefore, $P_w$ = 0.00055×2181000 = 1199.5watt, Actual pump rating, = 1.61hp

The overall efficiency is

$$= \left(\frac{actual\ horsepower}{calculated\ horse\ power}\right) \times 100 \quad 38$$

Where, Actual pump rating, = 1.61hp, the pump power calculated is 1.75Hp, efficiency $= \frac{1.61}{1.75} \times 100 = 91.8\%$

## 2.27. Heat Loss from Reservoir

The machine was placed in a well-ventilated environment during testing, after the machine has been operated for 4hours. To checked for the heat loss from reservoir of the waste paper baling machine based on adequate air circulation. The following readings were taken with a thermometer: Temperature of reservoir walls before operation due to the oil in the tank is 28ºC Temperature of reservoir walls after 4hours operation is 40ºC

Temperature of air due to well ventilated environment is 29ºC

Therefore,

Heat loss from reservoir is given by,

$$Q = C\times (T_f - T_a) \times A \quad 39$$

Where, Q = heat loss (kw), C = heat loss coefficient. For a steel tank a value of C = 0.00145kw/ ºC/$m^2$ [11]

$T_f$ = Temperature of fluid is 40ºC, $T_a$ = Maximum ambient temperature 29ºC, A = Area of the tank surface exposed to a flow of air is 0.04$m^2$. Since the area of reservoir (0.2×0.2) m is 0.04$m^2$ (40000$mm^2$)

Therefore, Q = 0.00145× (40 - 29) ×0.04

Q =0.0006kw

Therefore, the heat losses from the reservoir after operation are in an optimum condition. So, a cooler in the system would not be needed.

## 2.28. Machine Efficiency-Power Unit

An electrical A.C single phase induction motor converts electrical energy into mechanical energy. This motor drives the pump to provide volume flow to withstand load pressure. The efficiency as indicated on the name plate is 0.87. For constant delivery pump Q (calculated) = 0.0006$m^3$/s and Q (actual) = 0.00055$m^3$/s Therefore, efficiency is 0.92. To conversion efficiency from electrical motor to hydraulic flow is 0.87×0.92=0.80. It means that input of 2hp from electrical motor gives an output of 1.6hp at the hydraulic pump.

Figure 8 and 9 show the dimensioned orthographic view of the machine and the complete design of the horizontal waste paper baling machine

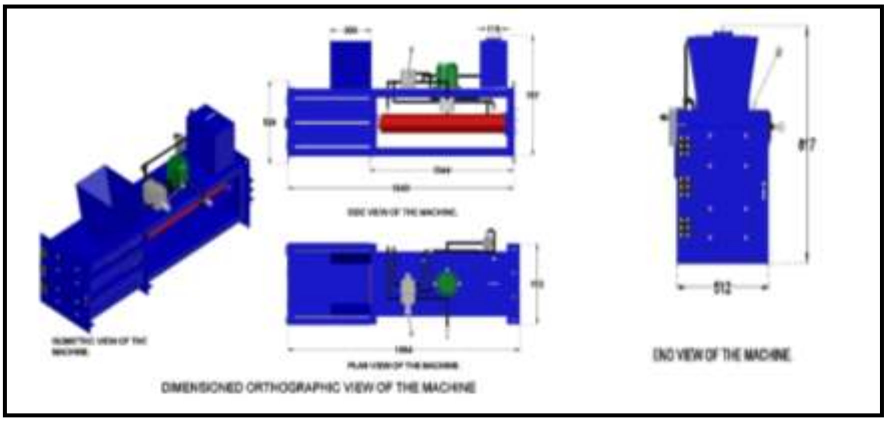


**Figure 8** The dimensioned orthographic view of the horizontal waste paper baling machine

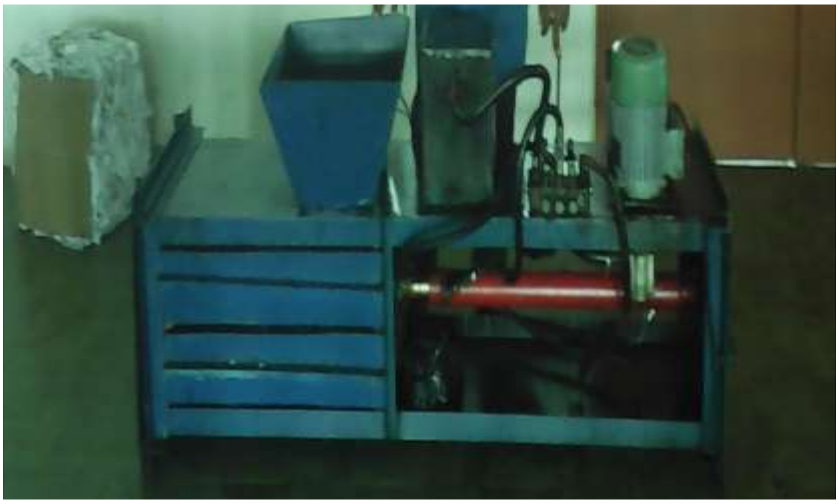


**Figure 9** The complete design of the horizontal waste paper baling machine

## Working Principle of the Machine

During operation of the waste papers baling machine is actuated by hydraulic system which has electrical single phase A.C induction motor as prime mover. Therefore, in the machine electrical energy is transformed into mechanical energy through hydraulic energy. The energy reaches the hydraulic cylinder (actuator) in the form of pressure and volume flow while transmitting power through hydraulic, the loss of energy could be due to flow losses and friction. The compression of hydraulic oil develops frictional heat which has to be controlled by radiation. During the period of manufacturing, metal sheets and angle bar were cut, machined, welded and bolted according to designed specification of machine and welded joints were checked for proper welding, the cutting process was carried out with cutting fluid that is friendly to the environment and also to avoid the metals from facing high corrosion effect [11-14]. Investigation of creep responses of selected engineering materials was also carried out using the procedure stated by Nwoke et al [15]. In the midst of manufacturing and assembling various parts of the machine, a test was done for proper functionality of all the hydraulic components (electric motor, coupling, pump, pressure hose, directional control valve, filter, pressure gauge, thermometer, tank, and hydraulic cylinder) and found to be operating within the design limits.

The manufactured and assembled machine was prepared for testing, the reservoir tank was properly cleaned and the surrounding environment was cleared of all unsafe objects.

# 3. RESULT AND DISCUSSION

Waste papers such as white papers of A4 size used for Photostat and printing upon was collected from shop complex around the University of Benin and used for testing the machine. The waste collected was manually shredded by hands to easy the baling process. The machine was started and left to run for 5 minutes unloaded, before actuating the lever on the directional control valve for baling. The shredded waste paper was fed into the baling chamber via the hopper. The baling process was started by actuating the lever on the directional control valve for hydraulic fluid to flow to the hydraulic cylinder that pushes the piston rod out along with the platen, which was compressing the waste papers fed in through the hopper. The platen was retracted back for other waste to be fed in for compression also then the process was repeated for 6, 40, 71, 90 and 117 seconds in the orders. Finally, the compressed waste papers were tied with a twine which is the baled.

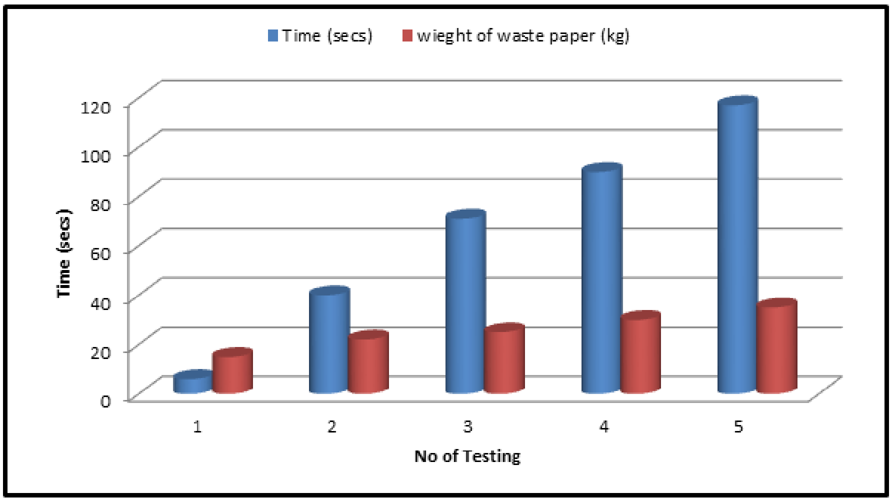



**Figure 10** graph of time (mins) vs. no of testing

Figure 10 shows the graph of time (mins) applied during the testing period with the no of testing, it can be seen that the time at which the machine compressed the first bale was really good, that shows that the performance of the machine is high the machine was tested and was found to baled volume of A4 size waste papers in four sack bags into a rectangle shape of 400mm×400mm×700mm in 6 seconds of weight of 15kg. This time was taking seriously because from the graph it shows that even when the time is increased the weight of the waste papers improved with slight changes and subsequent time as shown. There are no significant mechanical losses in the hydraulic component due to the machine efficiency that results in the hydraulic fluid heating. However, the heat losses from the reservoir which is 0.0006kw during operation are in an optimum condition. So, a cooler in the system would not be needed because of the steel material that is used for the design of the reservoir tank.

## 4. CONCLUSION

A waste paper baling machine was designed, manufactured, and tested. The machine was tested to ensure conformability to design objectives and serviceability. The machine was found to be satisfactory when baling waste papers and the tested machine bale the waste of 400mm×400mm×700mm, for 6 seconds cycle time with a system pressure of 22bar, the hydraulic cylinder is to further push the secured baled with twine out. The machine runs on a single phase 2Hp electric motor at a speed of 1800rpm and keyed straightly to the constant delivery pump of 0.00055m$^3$/s. The efficiency of the machine when tested was 80% and a bale weight is 15kg.

There are no significant mechanical losses in the hydraulic component due to the machine efficiency that results in the hydraulic fluid heating. However, the heat losses from the reservoir which is 0.0006kw during operation are in an optimum condition. So, a cooler in the system would not be needed because of the steel material that is used for the design of the reservoir tank.

## ACKNOWLEDGMENT

The authors wished to acknowledge the management of Covenant University for their part sponsorship and contribution made to the success of the completion of this research paper.

## REFERENCE

[1] Popoola, A.A., Ayangbile, O.A. and Adeleye, B.M., 2016. Assessment of solid waste management systems in Ibadan North, Oyo State using geo-spatial techniques. Ethiopian Journal of Environmental Studies and Management, 9(6), pp.666-679.

[2] Omole, F.K. and Alakinde, M.K., 2013. Managing the Unwanted Materials: The Agony of Solid Waste Management in Ibadan Metropolis, Nigeria. International Journal of Education and Research, 1(4), pp.1-12.

[3] Budynas, R.G. and Nisbett K.J. 2011. Mechanical Engineering Design. McGraw-Hill Companies Inc. 8th Edition, New York, NY

[4] Khurmi, R.S. and GUPTA, J.K. 2005. Textbook: Machine Design, Fourteenth Edition. Ram Nagar: Eurasia Publishing House (Pvt.) Ltd, New Delhi-110 055. P.474-557, 600-623, 727-758.

[5] Chandratre Varun,Rathod Pratik, Shaikh Mohammed Mohsin, Solanki Manish, Sheetal Parkar. 2017. Improvement in Waste Handling System. International Journal of Advance Engineering and Research Development Scientific. Special Issue SIEICON-2017, pp 1-3

[6] Okokpujie, I.P., Okokpujie, K., Salawu, E.Y. and Ismail, A.O., 2017. Design, Production and Testing of a Single Stage Centrifugal Pump. International Journal of Applied Engineering Research, 12(18), pp.7426-7434.

[7] Rabie, G.M., (2009) Fluid Power Engineering, McGraw Hill Inc. New York.

[8] P. Okokpujie Imhade, O. Okokpujie Kennedy, O. Ajayi Oluseyi, Azeta Joseph and N. Nwoke Obinna. (2017) "Design, Construction and Evaluation of a Cylinder Lawn Mower". Journal of Engineering and Applied Sciences, 12(5) 1254-1260.

[9] Robert R. (1999). Fluid power formulas, Hydraulics-Engineers Handbook files, New York, USA

[10] Okonkwo, U.C., Abdulraheem, A.A., Okokpujie, I.P. and Olaitan, S., 2017. Development of a Rocket Stove Using Woodash as Insulator. Journal of Engineering and Applied Sciences, 10(1), pp.1-13.

[11] Okokpujie, I.P., Okokpujie, K.O., Nwoke, O.N. and Azeta, J., 2018. DEVELOPMENT OF A 0.5 KW HORIZONTAL AXIS WIND TURBINE. Journal of Engineering and Applied Sciences, 13(8), pp.2202-2208.

[12] Okokpujie, I., Okonkwo, U. and Okwudibe, C., 2015. Cutting parameters effects on surface roughness during end milling of aluminium 6061 alloy under dry machining operation. International Journal of Science and Research, 4(7), pp.2030-2036.

[13] Okokpujie, I.P. and Okonkwo, U.C., 2015. Effects of cutting parameters on surface roughness during end milling of aluminium under minimum quantity lubrication (MQL). International Journal of Science and Research, 4(5), pp.2937-2942.

[14] Okokpujie, I.P., Salawu, E.Y., Nwoke, O.N., Okonkwo, U.C., Ohijeagbon, I.O. and Okokpujie, K., 2018. Effects of Process Parameters on Vibration Frequency in Turning Operations of Perspex Material. In Proceedings of the World Congress on Engineering (Vol. 2).

[15] Orisanmi, B.O., Afolalu, S.A., Adetunji, O.R., Salawu, E.Y., Okokpujie, I.P., Abioye, A.A., Akinyemi, O.O. and Abioye, O.P., 2017. Cost of Corrosion of Metallic Products in Federal University of Agriculture, Abeokuta. International Journal of Applied Engineering Research, 12(24), pp.14141-14147.

[16] YEKINI, S.E., Okokpujie, I.P., Afolalu, S.A., Ajayi, O.O. and Azeta, J., 2018. Investigation of production output for improvement. International Journal of Mechanical and Production Engineering Research and Development, 8(1), pp.915-922.

[17] Nwoke, O.N., Okokpujie, I.P. and Ekenyem, S.C., 2017. Investigation of Creep Responses of Selected Engineering Materials. Journal of Science, Engineering Development, Environmen and Technology (JOSEDET), 7(1), pp.1-15.

[18] Ezugwu, C.A., Okonkwo, U.C., Sinebe, J.E. and Okokpujie, I.P., 2016. Stability Analysis of Model Regenerative Chatter of Milling Process Using First Order Least Square Full Discretization Method. International Journal of Mechanics and Applications, 6(3), pp.49-62.

[19] Okonkwo, U.C., Okokpujie, I.P., Sinebe, J.E. and Ezugwu, C.A., 2015. Comparative analysis of aluminium surface roughness in end-milling under dry and minimum quantity lubrication (MQL) conditions. Manufacturing Review, 2(30), pp.1-11.

[20] Okokpujie, I.P., Ajayi, O.O., Afolalu, S.A., Abioye, A.A., Salawu, E.Y. and Udo, M., 2018. Modeling and optimization of surface roughness in end milling of aluminium using least square approximation method and response surface methodology. International Journal of Mechanical Engineering and Technology (IJMET), 9(1), pp.587-600.

[21] Okonkwo Ugochukwu, C., Nwoke Obinna, N. and Okokpujie Imhade, P., 2018. Comparative Analysis of Chatter Vibration Frequency in CNC Turning of AISI 4340 Alloy Steel with Different Boundary Conditions. Journal of Covenant Engineering Technology (CJET) Vol, 1(1).

[22] Okokpujie, I.P., Ikumapayi, O.M., Okonkwo, U.C., Salawu, E.Y., Afolalu, S.A., Dirisu, J.O., Nwoke, O.N. and Ajayi, O.O., 2017. Experimental and Mathematical Modeling for Prediction of Tool Wear on the Machining of Aluminium 6061 Alloy by High Speed Steel Tools. Open Engineering, 7(1), pp.461-469.

[23] Azeta, J., Okokpujie, I.P., Salawu, E.Y. and Okokpujie, K.O., 2018. ANALYTICAL STUDY OF A ROAD TRAFFIC CONFLICT AT THE T-JUNCTION OF UNIVERSITY OF BENIN MAIN GATE. Technology, 9(8), pp.1048-1061.

[24] J. A. O. Oyekunle, J. O. Dirisu, I. P. Okokpujie and A. A. Asere, 2018. Determination of Heat Transfer Properties of Various PVC and Non-PVC Ceiling Materials Available in Nigerian Markets, International Journal of Mechanical Engineering and Technology, 9(8), , pp. 963–973.

[25] Nwoke, O.N., Okonkwo, U.C., Okafor, C.E. and Okokpujie, I.P., 2017. Evaluation of Chatter Vibration Frequency in CNC Turning of 4340 Alloy Steel Material. International Journal of Scientific & Engineering Research, 8(2), pp.487-495.

[26] Azeta, J., Okokpujie, K.O., Okokpujie, I.P., Osemwegie, O. and Chibuzor, A., 2016. A Plan for Igniting Nigeria's Industrial Revolution. International Journal of Scientific & Engineering Research, 7(11), p.489.

[27] Okokpujie, K.O., Odusami, M., Okokpujie, I.P. and Abayomi-Alli, O., 2017. A Model for Automatic Control of Home Appliances using DTMF Technique. International Journal of Scientific & Engineering Research, 8(1), pp.266-272.

[28] Ikumapayi, O.M., Okokpujie, I.P., Afolalu, S.A., Ajayi, O.O., Akilabi, E.T. and Bodunde, O.P., 2018, July. Effects of Quenchants on Impact Strength of Single-Vee Butt Welded Joint of Mild Steel. In IOP Conference Series: Materials Science and Engineering (Vol. 391, No. 1, p. 012007). IOP Publishing.

[29] Okokpujie, I.P., Ohunakin, O.S., Bolu, C.A. and Okokpujie, K.O., 2018. Experimental data-set for prediction of tool wear during turning of Al-1061 alloy by high speed steel cutting tools. Data in brief, 18, pp.1196-1203.

[30] Okokpujie, K.O., Abayomi-Alli, A., Abayomi-Alli, O., Odusami, M., Okokpujie, I.P. and Akinola, O.A., 2017. An automated energy meter reading system using GSM technology.

[31] Okokpujie, I. P., Salawu, E. Y., Nwoke, O. N., Okonkwo, U. C., Ohijeagbon, I. O., & Okokpujie, K. (2018). Effects of Process Parameters on Vibration Frequency in Turning Operations of Perspex Material. In Proceedings of the World Congress on Engineering (Vol. 2).

[32] Afolalu, S.A., Salawu, E.Y., Okokpujie, I.P., Abioye, A.A., Abioye, O.P., Udo, M., Adetunji, O.R. and Ikumapayi, O.M., 2017. Experimental analysis of the wear properties

of carburized HSS (ASTM A600) cutting tool. International Journal of Applied Engineering Research, 12(19), pp.8995-9003.

[33] Ogundimu, O., Lawal, S.A. and Okokpujie, I.P., 2018, September. Experimental study and Analysis of Variance of Material Removal Rate in High Speed Turning of AISI 304L Alloy Steel. In IOP Conference Series: Materials Science and Engineering (Vol. 413, No. 1, p. 012030). IOP Publishing.

[34] Dunmade, I., Udo, M., Akintayo, T., Oyedepo, S. and Okokpujie, I.P., 2018, September. Lifecycle Impact Assessment of an Engineering Project Management Process–a SLCA Approach. In IOP Conference Series: Materials Science and Engineering (Vol. 413, No. 1, p. 012061). IOP Publishing.

[35] Onawumi, A., Udo, M., Awoyemi, E. and Okokpujie, I.P., 2018, September. Alternate Maintainability Evaluation Technique for Steering System of Used Automobiles. In IOP Conference Series: Materials Science and Engineering (Vol. 413, No. 1, p. 012062). IOP Publishing.