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Design Fabrication and Testing Of a Motorized Paper Perforation Machine

Matthew Sunday Abolarin[1], Olugboji Oluwafemi Ayodeji[2], Jiya Jonathan Yisa[3], Ngoffia Raphael Idoo[4] Ajani Clement Kehinde[5]

Mechanical Engineering Department, Federal University of Technology, Minna Niger State, Nigeria.

olugbojioluwafemi@yahoo.com[2]
jojileg@yahoo.com[3]
clemajan@gmail.com[5]

Abstract: *This is concern with design and fabrication of a powered paper perforating machine with efficiency and less cost for bindery department in printing industries. It is a machine that will compete favorably with the manually operated ones and reduce human efforts with minimum time consumption. It perforates up to 15 sheets of paper at each stroke and is powered by a horse power electric motor whose rotational motion is converted to a reciprocating motion in the perforation pin by the principle of eccentricity*.

Keywords: Perforation, eccentricity, punch, belt, angle of contact.

1. INTRODUCTION

The perforation machine is basically designed for use in bindery department of printing industry to make a line of small holes in a piece of paper for binder inserts, calendar pages, tab and register cut sheets made of cardboard and plastic paper for wire bound products. A powered paper-perforating machine is a semi-automatic machine, which replaces the use of human effort by a small size electric motor, it makes the perforating process simple and affordable with an incredibly fast setup [1]. The machine was designed having in mind an alternative to replace the high cost automatic perforating machine and high human fatigue with manually foot- operated perforation machine. Separating the sheets is the most expensive part of fully automatic machine but this has been eliminated by the operator separating the sheets after perforation which makes the process simple and economical.

2. DESIGN ANALYSIS

For effective performance of the machine physical components, built stresses on the components are analyzed to evaluate the strength, durability, fatigue and other mechanical properties.

2.1. Frame/body analysis

2.1.1 Side Channel "U" (Assumed stretch).

$$Area = 2pcs(L \times B)m^2 \quad (1)$$

$$Volume = 2(L \times B \times t)m^3 \quad (2)$$

Where, L = length of channel/plate, B = width of channel/plate t = thickness of channel/plate.

2.1.2 Wooden Surface Plat Form

$$Volume = 2pcs(L \times B \times t)m^3 \quad (3)$$

2.1.3 Pedal (foot match)

This is placed at angle 45^0 to each other.

$$Volume = [L_1 + L_2] \times [B_1 + B_2] \times [t_1 + t_2]m^3 \quad (4)$$

2.1.4 Motor Seating Base

The Volume can be obtained from the relationship below:

$$Volume = (L \times B \times t)m^3 \quad (5)$$

2.1.5 Leg Support Stand

This consists of 4 channels of same size/dimension, the volume can be obtained from the relationship below: $Volume = 4(L \times B \times t)m^3$ (6)

2.1.6 Leg Brazing Rods

This consist of two round pipe of different sizes Area and Volume can be obtained from the relationship below

$$Area(A_{1/2}) = \left[\frac{\pi d_1^2}{4} + \frac{\pi d_2^2}{4}\right]m^2, Volume(V_{1/2}) = \left[\frac{\pi d_1^2}{4} \times L_1\right] + \left[\frac{\pi d_2^2}{4} \times L_2\right]m^3 \quad (7)$$

Where A_1, A_2, V_1, V_2, d_1, d_2 are the Area, Volume, and diameter of the two pipes respectively.

2.2 Crank Mechanism

This consist of the following parts, reciprocating rod, crank bearing housing or cam, rod housing pipe, housing pin/weight.

2.2.1 Reciprocating Rod

The volume of reciprocating rod can be obtained from the relationship below Where d = diameter of rod and, L = length of rod

$$Volume(V_{1/2}) = \left[\frac{\pi d^2}{4} \times L\right]m^3 \quad (8)$$

2.2.2 Crank Bearing Housing/Cam

$$Volume = \left[\frac{1}{3}r^2ht\right] - \left[\frac{\pi d^2 \times t}{4}\right]m^3 \quad (9)$$

Where r = radius of inner circle, h = height of cam t = thickness of cam, d = diameter of inner circle

(a) Reciprocating rod housing pipe

$$Volume = \left[\frac{\pi(D^2 - d^2)L}{4}\right]m^3 \quad (10)$$

(b) Pin Housing/Weight

It consists of three weights. Up Holding Pin Casing/Down Perforating Casing [2].

(1) Volume of the uploading pin

$$Volume = [L \times B \times t] + \left[\frac{\pi d^2}{4}\right]_{pin} \times L_{pin} \times np$$

$$[LBt]\left[\frac{\pi d^2 \in p}{4}\right]_{pin} \quad (11)$$

Where L = length of bar, d = diameter of pin, B = breadth t = thickness of bar

(2) Volume of down perforating casing

$$Volume = [L \times B \times t] - \left[\frac{\pi d^2 L}{4} \times n_p\right]_{pinholes} \quad (12)$$

Where n_p = Number of pin, d = diameter of pin L = length of pin

(3) Volume of top weight

$$Volume = (L \times B \times t)m^3 \quad (13)$$

Where L = length of the weight, B = breadth t = thickness

(c) Volume of hollow square channel

$$AssumestretchedVolume = (L \times B \times t)m^3 \quad (14)$$

(d) Length of stroke

$$Lengthofstroke = \frac{1}{2}z\pi r \quad (15)$$

2.3 Transmission on belt

2.3.1 Power Transmission on Belt

The power transmitted is given by the relationship below

$Power = Torque \times w$ [3]

$$P = M_t \times w \quad (16)$$

Where M_t = Torque, and

$$w = \frac{2\pi n M_t}{60} m/s \quad (17)$$

Where N = speed

but $M_t = (T_1 - T_2)$ R (18)

Where T_1 and T_2 are belt tension on tight and slack sides.

R = radius of pulley

Substituting equation 18 into 17, we obtain

$$Power = \frac{2\pi N(T_1 - T_2)R}{60} w \quad (19)$$

2.3.2 Ratio of Belt

This is given by the relationship

$$\frac{T_1}{T_2} = R_b, \qquad T_1 = R_b T_2 \quad (20)$$

Where T_1 = Tension of tight side, T_2 = Tension of slack side

R_b = Belt ratio

Substituting equation (20) into (16), we obtain

$$T_2 = \frac{P}{Rw}(R_b - 1) \quad (21)$$

But $w = \frac{2\pi N}{60}$

Substituting the above into equation (21), we obtain,

$$T_2 = \frac{P}{R\left(\frac{2\pi N}{60}\right)(R_b - 1)} \quad (22)$$

Hence,

$$T_1 = R_b \frac{P}{R\left(\frac{2\pi N}{60}\right)(R_b - 1)} \quad (23)$$

Where P = power transmission, R = radius of shaft pulley

R_b = belt ratio, N = speed of shaft

2.3.3 Force Transmitted

The force transmitted is given by relationship,

$$T_1 - T_2 = \frac{P \times 60}{2\pi r N} = F_2 \text{ [3]} \quad (24)$$

Where $T_1 - T_2$ = force transmitted, P = power, N = speed

r = radius of pulley

2.3.4 Angle of Contact

This is the angle made between two different diameter of pulley and is given by the relationship below.

$$\emptyset = \frac{2.3 log\frac{T_1}{T_2}}{\mu} \quad [4] \quad (25)$$

Where μ = slip of belt, $\emptyset$ = angle of contact, $\frac{T_1}{T_2}$ = ratio of belt

2.3.5 Component Forces

The component forces acting on the shaft member can be obtained from the relationships

Total tension $T_T = (T_1 + T_2)$ (26)

Vertical component = $T_T \cos\theta$ (27)

Horizontal Component = $T_T \sin\theta$ (28)

Resultant = $\sqrt{V^2 + h^2}$ (29)

2.3.6 Slip of Belt

$$\mu = 0.54 - \frac{42.6}{152.6 + v} \quad (30)$$

Where θ = angle of contact, T_1 and T_2 are values of tension on tight and slack side respectively.

μ = coefficient of friction between belt and pulley = 0.33

2.3.7 Speed Reduction

The speed of shaft can be obtained using the relationship,

$N_1 = D/d \times N_2$ (31)

Where N_2 = speed of motor, D = diameter of driven shaft

d = diameter of electric motor pulley

This speed ratio

$$\frac{N_2}{N_1} = \frac{D}{d} \quad (32)$$

2.3.8 Work Done by Belt

The work done by the belt on the pulley is given by relationship,

$w = (T_1 - T_2)$ V N/m/s (33)

Where v = velocity belt

2.3.9 Velocity of Driver

$$V = \frac{\pi N D}{60} \text{ m/s} \quad (34)$$

Where N = speed of motor, D = diameter of pulley

2.3.10 Power Transmitted by Belt

This is given by the relationship

$$P = \frac{(T_1 - T_2)V}{75} Hp \quad (35)$$

2.3.11 Centre Distance

This is the centre distance between two pulleys which is given in relationship below

$x_{min} = 0.55$ (D + d) + T, T = 8

$x_{max} = 2$(D + d) (36)

x = 0.9D (37)

Diameter of large pulley = D, Diameter of small pulley = d

Nominal thickness = T, For x/r = 0.9

2.3.12 Pitch Length of Belt

This is the total length of belt wrapped round two moving pulleys. The pitch length is given by the relationship below

$$L = 2x + \frac{n}{2}(r_2 + r_1) + \frac{(r_2 + r_1)^2}{2x} \quad [5] \quad (38)$$

Where r_1, r_2 are radius of the pulleys respectively

x = centre distance

2.4. Design of perforating pin

2.4.1 Total Force on Pin

This is the force required to be carried on the pin for perforation and is given by

Lord on pin = F_T

$F_T = F_1 + F_2$ (39)

F_1 = preload force exacted by the weight of the perforating pin housing, $F_2 = T_1 - T_2$ i.e. load from electric motor [3]

$$TotalStress = \eta_{1T} = \frac{4F_T N}{\pi d^2} \quad (40)$$

$$safetyfactor = \frac{\delta y}{\delta T} \times N \quad (41)$$

2.4.2 Force Required to Punch the Sheet

$$F_3 = \pi D \times t \times \eta_{1P} \quad (42)$$

Where D = diameter of punching pin, t = thickness of paper

η_{1P} = shear stress of paper

The total number of force needed to perforate with 28 pins × F_3

2.4.3 Work done by Punching Pin

This is the actual work done by the pin to travel through a distance.

F_4 × distance = F_4 × distance travelled by pin (43)

2.5 Shaft design

2.5.1 Volume of Shaft

To design the shaft volume, this is obtained from relationship,

$$V = \pi r^2 l \quad (44)$$

Where l = length of shaft (m), r = radius of shaft (m)

2.5.2 Mass of the Shaft

The mass of shaft is given by the relationship,

$$\rho = M/V \quad (45)$$

ρ = density of shaft = 7.8×10^3kg/m^3

V = volume of shaft

2.5.3 Diameter of the Shaft

$$d^3 = \frac{16}{\pi}\delta s \times \sqrt{(k_b m_b)^2 + (k_t + m_t)^2} \text{ [6]} \qquad (46)$$

$K_b = 2.0$, $K_t = 2.0$

Where k_b and k_t are combined shock and fatigue factors

$\delta s = 55\text{m}^4/\text{m}^2$, m_b = Bending moment (max), m_t = Torsional moment

2.6 Bearing design

2.6.1 Bearing Life

$$L_{io} = \left(\frac{10^6}{60N}\right)\left(\frac{C}{P}\right)^k \text{ [3]} \qquad (47)$$

Where L_{io} = life of bearing in millimeter evolution

k = exponential for ball bearing, N = speed of shaft

P = equivalent dynamic load

2.7 Determination of engagement spring parameters

2.7.1 Wire Diameter (d)

$$d = \frac{3\sqrt{8FD}}{\pi \delta w} \qquad (48)$$

Where F = force transmitted on shaft, D = diameter of spring δw =maximum working stress for compressive spring

2.7.2 Compressive Shear Stress of spring (S_s)

$$S_S = \frac{PD_S}{0.393d^3} \qquad (49)$$

P = force on one spring cover by shaft force divided by 2

D_S = diameter of spring, d = wire diameter

2.7.3 Spring Index

$$C = \frac{P_s}{d} \qquad (50)$$

Where P_S = diameter of spring, d = wire diameter

2.7.4 The Mean Radius of Helix

$$H = \frac{Cd}{2} \qquad (51)$$

Where C = spring index, d = diameter

2.7.5 Spring Constant S_C

$$S_C = F/y \qquad (52)$$

Where F = force on shaft, y = deflection

2.7.6 Diameter of Coil P_C

$$P_C = 2RH \qquad (53)$$

Where R_H = mean radius of spring

2.7.7 Deflection of spring (y)

$$y = \frac{8FD_S^3N}{d^4G} \qquad (54)$$

Where F = w = force on spring, D_s = diameter of spring

d = diameter of wire, G = modulus of rigidity

2.8 Weld joint design

2.8.1 Lap Weld

We find that throat thickness is defined by,

$$t = S \times \sin 45^0 = 0.707S \qquad (55)$$

.'. Minimum Area of the weld or throat

Area A = throat thickness × length of weld and substituting equation (55) into the Area, we obtain;

A = t.L = 0.707S.L

If F_t is the allowable tensile stress for the weld metal, then the tensile strength of the joint for single fillet weld will be

P = Throat Area × allowable tensile stress

= 0.707S.LF_t [3] (56)

Making F_t subject, we obtain

$$F_t = \frac{P}{2\times 0.707SL} \qquad (57)$$

Where F_t = allowable tensile strength of the material ***Circular***

2.8.2 Fillet Weld Subjected to Only Bending

Consider a circular rod connected to the rigid housing by a fillet weld, Let, d = diameter acting on the perforating pin, m = bending moment acting on the perforating pin, s = size or leg of weld, t = throat of thickness, z = section modulus of the weld section is given by $\frac{\pi d^2}{4}$ [3]

We know that the bending stress is also given by

$$F_b = \frac{M}{Z} = \frac{M}{\frac{\pi t d^2}{4}} = \frac{4M}{\pi E d^2}$$ But bending moment, m = F × e

Therefore $F_b = \frac{4(F\times e)}{\pi t d^2}$

But t = 0.707S, hence Maximum Normal Stress, F_{max}

$$F_b = \frac{4}{0.707}\left[\frac{F\times e}{\pi S d^2}\right] = \frac{5.66Fe}{\pi S d^2} \qquad (58)$$

Shear stress is given by

$$F_s = \frac{F}{A} = \frac{F}{tL} = \frac{F}{0.707SD^2} \qquad (59)$$

Maximum Shear Stress

$$F_{s(max)} = \frac{1}{2}\sqrt{(F_b)^2 + 4(F_s)^2} \qquad (61)$$

3. TESTING

Testing of the machine was carried out to evaluate its performance. This was done by loading and aligning the paper to be perforated, the motor was switched ON and the pedal-engagement depressed which allow the falling of the perforating housing by a reciprocating motion through the revolution of the bigger pulley shaft via the cams displacement. At the end of the operation, the paper was spiral binded and the motor switched OFF, though the action was designed to be a continuous perforating process with more paper inserted.

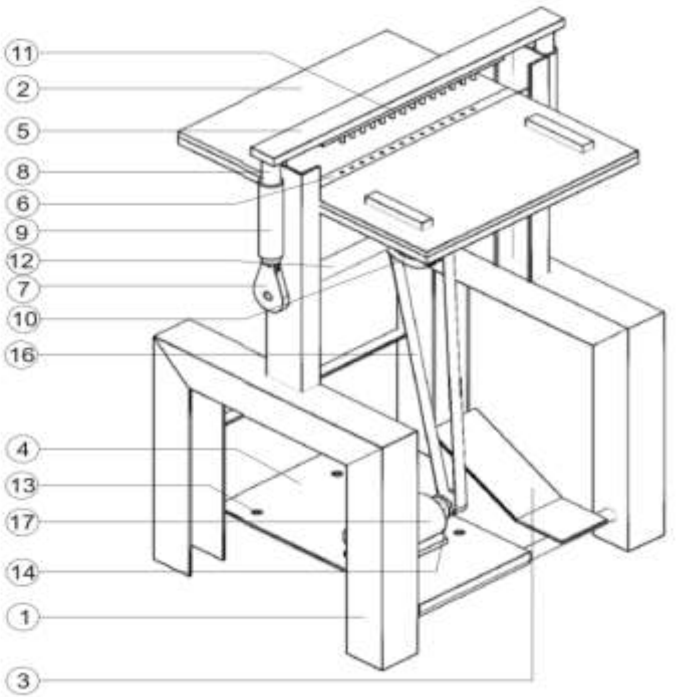

Figure 1: Isometric View of Motorized Perforating Machine.

Table 1: part description of perforating machine.

S/N	*Parts Description.*
1	Supports
2	Platform wood
3	Foot match
4	Motor seat base
5	Pin weight
6	Die casting
7	Crank housing
8	Reciprocating rod
9	Reciprocating rod slot
10	Big pulley
11	Punch housing
12	Shaft
13	Motor sit adjustable screw
14	Small pulley
15	Bearing
16	Belt
17	Electric Motor

4. DISCUSSION OF RESULT

It was discovered during testing that the forces required to perforate different quantities of papers differs. As earlier stated, the force required to perforate say 15 paper is 51.99N. But the total force exerted by the weight and the force transmitted through the tensions ($T_1 - T_2$) was known to be 250.90N. Force left over = 250 – 51.99 = 198.91N It is therefore observed that perforation was actually made possible because the total input force was greater than the required force needed for any perforation to be carried out. Hence, perforation depends on the following: The attached weight housing, the inputted force obtained from the motor, the diameter of the shaft used [7].

5. CONCLUSION

The use of a semi-motorized perforating machine in a small scale binding department will actually go a long way to increase the output of binding within the binding department, as a result of its simple method and principles of operations. Consequently, the machine is cheaper compared to others that are manually operated and even the imported type of the same machine. The machine has drastically reduced the excessive energy used during perforating with the manual type. Thus, volumes of sheet of papers, calendars, writing memo and pads can be perforated within a lesser time. Furthermore, the incorporation of the engagement pedal mechanism is a good measure of improvement, hence it efficiency will be greatly increased if the input power of the motor is increased and again dependent on the operator.

6. RECOMMENDATION

To improve on the performance and efficiency, it is recommended that:

(1) The machine be designed alongside manually operated in other to be used when there is no power.

(2) There would be need for inter-changeable perforating pins to accommodate bigger or smaller diameter perforations.

(3)Prospective modification can be done with selection of materials with less weight putting into consideration need for easy mobility, though strength should be greatly checked.

(4)The unit comprising the electric-motor pulley and shaft may be enclosed to reduce the risk of accident.

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Author Profile

K.C. Ajani he obtained his B. Eng. at the Department of Mechanical Engineering, Federal University of Technology Minna, Nigeria in

2013. He his is presently a Graduate assistant lecturer in the same Department. He is a member of the Nigerian Institution of Mechanical Engineer NIMechE. He has interest in aerodynamics, aerospace and mechanical system designs. He presently has a number of publication. He his the corresponding author and has his photo on behalf of the other authors.

Prof. M.S. Abolarin Is a professor in the Department of Mechanical Engineering, Federal University of Technology Minna, Nigeria. He has more than 25 years of experience in teaching and research. He also works as a visiting lecturer in other reputable universities in Nigeria. He particularly interested in mechanical engineering design, production, Fabrication and machining. He is a Fellow of the Nigerian Society of Engineers FNSE, and also a member of the Nigerian Institution of Mechanical Engineer NIMechE.

Dr. O.A. Olugboji fro he got is B.Eng. and M. Eng. at the Department of Mechanical Engineering, Federal University of Technology Minna, Nigeria. He obtained his PhD at Newcastle University at the department of Mechanical and System Engineering in United Kingdom. He has published several papers in referred international journals He is presently the Head of Department at the Department of Mechanical Engineering, Federal University of Technology Minna, Nigeria. His research interest is in mechanical system designs, control, automation and with particular interest in pipeline system design. He is a member of the American society of Mechanical Engineers ASME, a fellow of the Nigerian Society of Engineer, FNSE and also a member of the Nigerian Institution of Mechanical Engineer NIMechE.

J.J. Yisa is presently an assistant lecturer at the Mechanical Federal university of technology Minna and undergoing his master's program presently in the same department. He is interested in power, plant designs and thermofluid related research area. He is a member of the Nigerian Institution of Mechanical Engineer NIMechE.

R .I. Ngoffia he is a graduate in the Department of Mechanical Engineering, Federal University of Technology Minna, Nigeria. He is a student member of the Nigerian Institution of Mechanical Engineer NIMechE.