*International Journal of Engineering Science Invention*
*ISSN (Online): 2319 – 6734, ISSN (Print): 2319 – 6726*
*www.ijesi.org Volume 2 Issue7 ǁ July. 2013 ǁ PP.111-116*

# Design and Development of Automatic Hedge Dressing Machine

Arinola B. Ajayi
*Dept. of Mechanical Engineering, Faculty of Engineering, University of Lagos. Lagos. Nigeria.*

***ABSTRACT:*** *This paper reports the design and development of an automatic hedge dressing machine. Automatic hedge dressing machine was designed, developed and tested for reliability and safety. Shear scissors and in some cases cutlasses has been the traditional tools used in dressing hedges and surrounding shrubs in the sub Saharan Africa and third world countries in general. This method of hedge dressing is tedious and labor intensive. The machine presented in this paper is an innovative way of achieving in one day what a worker could have achieved manually in six days. The machine consists of an electric motor, the housing, the coupling and the cutting blade. The cutting blade, the coupler and the electric motor shaft were designed for safety and reliability. The cutting blade does the shearing of the shrubs while the power for shearing is transferred from the electric motor to the blade through the coupler. The machine testing done with near matured shrubs were impressive.*

***KEYWORDS:*** *Design, Hedge dressing machine, Shearing, Horticultural Trimmer.*

## I. INTRODUCTION

Man has always been striving for continuous improvement in his living condition. Hedge dressing machines has been in use in some developed countries especially USA and there are different types in use as well as reported in the literature [1 – 4] but this subject area is not exhaustive. There are different types of hedge dressing machines; there is reciprocating shear (bar knife) type; this is an horticultural trimmer with short reciprocating knives which is driven between closely spaced guards by an electric motor mounted at the inner end of the bar. The rate of work possible with this trimmer is much higher than is possible with shears or hand hedge tools but since it has unbalanced forces which makes it unsuitable for operation at high speed, therefore, its relatively low forward speed, makes it less efficient. There are other types of hedge dressing machine such as rotary cutting blade consisting of renewable knives attached to the corners of a rectangular steel plate; a shield covers the rotor to provide protection from flying debris. Another type has many rotating blades moving against stationary bar. This type employs two opposing shearing elements which meet and pass over each other with little or no clearance between them. If the shrub is mature, it can stop the machine from working which may lead to short-circuiting of the electric motor thus permanently damaging it. The objective of this paper is to design and develop an automatic hedge dressing machine with balanced forces, high forward speed and cannot be stopped by mature shrubs thus posing no threat to the electric motor.

## II. THE DESIGN AND DEVELOPMENT

The machine consist of three main units namely, the electric motor, the housing and the blade. The electric motor is ¼ hp, 6,000 rpm and single phase motor. The motor is put inside the housing at the motor section. The electric motor output shaft passes through a 10mm hole drilled at the base of the motor housing into the blade section. The electric motor is secured to the housing by four bolts at the bottom of the motor housing. From the blade section, the coupling and the blade were secured to the motor shaft with the aid of bolts.

### 2.1 Machine Description

Automatic Hedge Dressing Machine (AHDM), Figure 1, is an electrically powered machine employed for dressing and beautification of hedges and shrubs. The machine consists essentially of an electric motor that powers the rotating blade. The electric motor transmits power to the blade through the coupling shaft. This machine is designed to convert the electric power into rotational motion through electric motor which whirls a straight sharpened blade at a speed of between 4,000 and 6,000 rpm. The shearing off of the hedge is achieved by impact momentum. The machine consists of two indivisible sections i.e. the electric motor housing and the blade housing. The electric motor housing houses the electric motor which is the power house of the system. Two handles are fixed to the main body for easy carriage. The electric motor is secured firmly in the housing through four bolts. The output shaft from the engine is couple to the blade with a rigid flange coupling. The region under the blade housing where the blade rotates is the mowing zone.

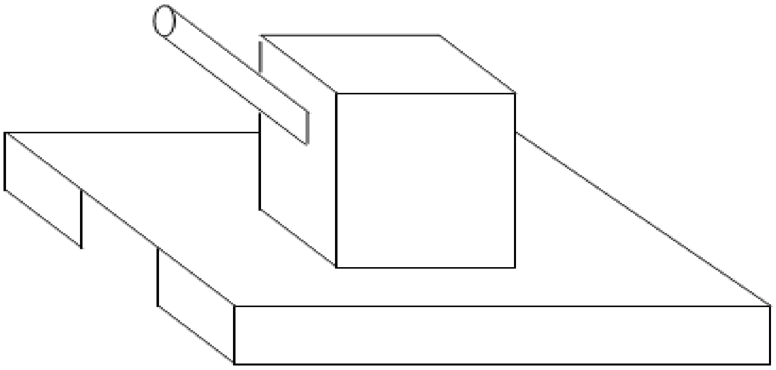

Figure 1: The outer view automatic hedge dressing machine.

***The cutting blade***: The cutting blade is made from 400mm long high carbon steel (spring steel of about 0.9 – 1.0% carbon) with a cross section of 50mm by 2mm. After obtaining the required dimensions, a length of 11mm was ground from the opposite diagonal edges of the blade until it was 0.5mm blunt. Two holes of 6mm were drilled with drilling machine, the centre of these holes were 24 mm apart and 12 mm from the centre of the cutting blade. The cutting blade is attached to the electric motor shaft through a mild steel coupling. The cutting blade is subjected to tensile stress and bending moment. There is probability of the blade striking a matured plant or hard object, therefore, the cutting edge is not sharpened like a knife edge to reduce the risk of damage.

***The coupling***: The coupling is made of mild steel fabricated from a solid cylindrical mild steel of length 70mm and diameter 40mm. It links the output shaft from the motor to the cutting blade. After the required outward dimensions were obtained, a hole of 8mm diameter was drilled to a depth of 20mm from the hub end. Another hole of 5mm was drilled along the hub, 8mm from the hub end. This hole was tapped with 6mm tap. Two holes of 5mm were drilled with drilling machine and tapped with 6mm tap. The centres of these holes were 24 mm apart and 12 mm from the centre of the coupler.

***The housing***: The housing is designed to be strong and durable. It is made from flat metal sheet. The motor section of the housing is 130mm x 130mm x 190mm while the skirt i.e. the blade section is 240mm x 240mm x 30mm. The required dimensions were marked out on the galvanized steel sheet. In the motor section, holes of 4mm were deliberately drilled on the sides for ventilation to air cool the electric motor by natural convection. The joints of the housing were secured by riveting. An 80mm x 60mm outlet was made on one side of the blade section to allow debris to be gotten rid of. Two handles were riveted to the motor housing side half way for handling. It is made of hollow steel pipes.

***The electric Motor:*** The electric motor is a 220v/250v, 1-phase, 0.25hp (187W), 6,000rpm electric motor was used.

## III. DESIGN CALCULATIONS

### 3.1 Design of the Electric Motor Shaft

The stresses acting on the shaft are torsional shear stress due to the rotational velocity and tensile stress which is small and can be neglected

Using Soderberg equation for a solid shaft with dynamic loading,

$$d_0^3 = \frac{32n}{\pi} + \left(\frac{M_a}{S_e}\right)^2 + \left(\frac{T_m}{S_y}\right)^2 \qquad (1)$$

Where $T_m$ = Torsional Moment (Nm)

$M_a$ = Bending Moment (Nm)

$d_o$ = Outside diameter of the shaft (Nm)

$S_e$ = Modified endurance limit of the shaft (Nm)

$S_y$ = Yield strength of the shaft (Nm)

$n$ = factor of safety

*Ma = 0*, since there is no bending load on the shaft. Hence equation (1) will become

$$d_0^3 = \frac{32n}{\pi} + \frac{T_m}{S_y} \qquad (2)$$

But $T_{max} = \frac{S_y}{2n}$

$$\therefore d_0^3 = \frac{16T_m}{nT_{max}} \quad (3)$$

$$T_{max} = \frac{16T_m}{d_o^3} \quad (4)$$

For a treated shaft to be considered safe, $T_m \leq 40\ MN/m^2$. The output shaft from the motor is treated and the torsional moment can be obtained from the power relation thus:

$$Power,\ p = \omega T_m$$

$$= \frac{2\pi N}{60} \times T_m \quad (5)$$

$p = 0.25\ hp = 187\ w$
N = 6,000 rpm

$$T_m = \frac{60p}{2\pi N} = 0.2976\ Nm$$

Recall from equation 4,

$$T_{max} = \frac{16T_m}{\pi d_o^3}$$

$$d_o = 0.008\ m \quad (6)$$

$$\therefore T_{max} = \frac{16 \times 0.2976}{\pi(0.008)^3} = 2959898\ N/m^2$$

$$= 2.96\ MN/m^2 \quad (7)$$

Since $2.96\ MN/m^2 < 40\ MN/m^2$, the shaft is safe. Hence, shaft diameter of 8mm is acceptable.

### 3.2 Design of the Cutting Blade

The allowable working stress should be less than 40MN/$m^2$ to satisfy the ASME Code for commercial steel shafting. The blade is made of hot rolled, annealed steel AISI 1095,

$S_{ut} = 106\ kpsi$ and $S_y = 60\ kpsi$

The allowable working stress, $\sigma_{all} = \frac{yield\ stress}{factor\ of\ safety} = \frac{S_y}{n}$ (8)

$$S_y = \frac{6{,}000 \times 6.89}{100} = 413.4\ MN/m^2 \quad (9)$$

Taken factor of safety to be 16 [5]

$$\sigma_{all} = \frac{413.4}{16} = 25.8\ MN/m^2 \quad (10)$$

Since 25.8 *MN/m*$^2$ < 40 *MN/m*$^2$, the material is safe.

When the cutting blade is rotating at a very high speed, with angular velocity, $\omega$, the blade will appear as a rotating disc with uniform radius and thickness. The velocity of the rotating disc is given by:

$$\omega^2 = \frac{8\sigma}{(3+\upsilon)\rho r^2} \quad (11)$$

Where $\omega$ = permissible speed
$\upsilon$ = poisson ratio
$\rho$ = density of the material
$r$ = radius of the blade (radius of the disc)
$\delta$ = permissible stress

The assumptions made in employing the above relation are:

i) There is no stress concentration
ii) The thickness of the blade is negligible compared to the radius
iii) The stress is constant over the thickness of the blade.

$\omega$ = 6,000 *rpm* = 628.4 *rad/s*
$\upsilon$ = 0.3
$\rho$ = 7900 $kg/m^3$

$$r = \sqrt{\frac{8 \times \sigma_{all}}{(3+\upsilon)\rho\omega^2}} = \sqrt{\frac{8 \times 25.8 \times 10^6}{(3+0.3) \times 7900 \times 628.4^2}} \qquad (12)$$

= 0.1416 *m*

The diameter of the blade = 283 *mm*

The cutting blade length should not be more than 283 mm for it to be safe. The other dimensions are: thickness = 2 *mm*, and width = 35 *mm*

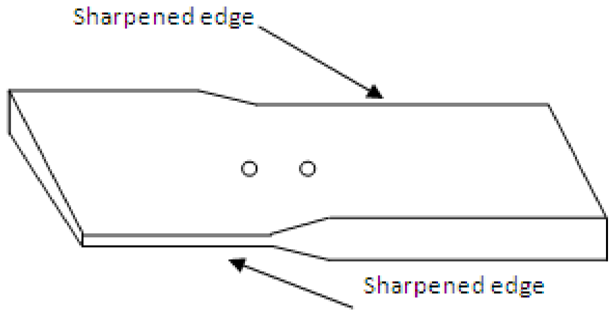


Figure 2: The cutting blade

**3. 3. The Design of the Coupling and the Bolt**

In this section, the coupling and bolt are designed.

***3.3.1 The Coupling***

The coupling attaches the cutting blade to the motor shaft. Failure of this part can result in fatality. The diameters of the hub $D_H$, and the flange, *h*, are established on the basis of proportions. The diameter of the hub is usually 1.75 – 2 times the diameter of the shaft and the diameter of the flange is $2D_H$ [6]

$D_H = 2\, d_o$ (13)
= 2 x 8 *mm* = 16 *mm* (14)
And $h = 2\, D_H$
$h$ = 32 *mm*

The thickness of the coupling is based on the bearing pressure of the bolt or the web. The torque capacity is based on the bearing is

$$T_m = S_B (d \times t) D_{bc} \times z \qquad (15)$$

Where $S_B$ = allowable bearing pressure of the bolt or web (whichever is weaker)
$d$ = bolt diameter
$D_{bc}$ = diameter of the bolt circle
$t$ = flange thickness
$z$ = number of bolts

The allowable bearing stress is
$S_B = 0.18\, S_{ut}$ or $0.3\, S_y$ (whichever is smaller)
$Sut$ = 69kpsi and $Sy$ = 48kpsi
$S_B$ = 0.18 x 69kpsi or $S_B$ = 0.3 x 48kpsi
= 12.42kpsi = 14.4kpsi
$S_B$ = 12.42kpsi is less
$S_B$ = 85.574 MN/m$^2$
From equation (15),

$$t = \frac{2 \times T_m}{S_B \times d \times D_{bc} \times n} = 5\,mm \qquad (16)$$

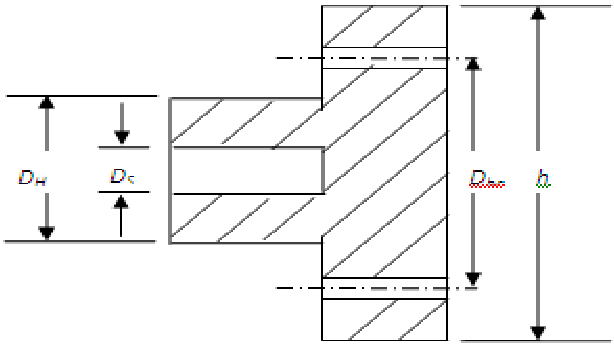

Figure 3: The coupler

ii) The bolts attach the cutting blade to the coupling flange. The analysis for the bolts can be made in one of these several ways:

a) Assuming that the bolts are finger tight and the load transferred from the coupling to the blade by a uniform shear stress in the shank of the bolt.
b) Assuming that the bolts are just finger tight and the load transferred from the coupling to the blade with a maximum shear stress in the shank of the bolt equal ¾ times the average shear stress.
c) Both the bolt and the shaft are assumed to have the same torque capacity in torsion (same material)

From (a) above,

$$K_t M_t = S_s \left( \frac{\pi d^2}{4} \right) \left( \frac{D_{bc}}{2} \right) \times z \qquad (17)$$

Where $M_t = T_m$, the maximum torque
$K_t$ = factor of safety
$S_s$ = stress for the shaft
But

$S_s = T_{max}$ from (c) above;
$z$ = 2 bolts

$$D_{bc} = D_H + \frac{h - D_H}{2} = \frac{h + D_H}{2}$$

$$= \frac{32 - 16}{2} = 24\ mm$$

Since the coupling is being designed for shock and fatigue, $Kt$ is taken to be 1.
From (17) above,
$d$ = 2.31 mm, (18)
Also, from assumption (b) above,

$$M_t = \frac{3S_s}{4} \times \left( \frac{\pi \ d^2}{4} \right) \times \left( \frac{D_{bc}}{2} \right) \times z \qquad (19)$$

From the equation (19) above,
$d = 2.67\ mm$ (20)

The bolt diameter should be 3 *mm*, which is the nearest standard bolt size. These are *M6* bolts.

## IV. CONCLUSION

The design and development of automatic hedge dressing machine has been presented. This is an improvement over what is available and can be easily manufactured in a local fabrication shop. The performance of the machine is satisfactory but there is limitation of power cord. Long extension cord will be necessary for a large garden that does not have power socket outlet already installed.

## REFERENCES

[1] Barner, R., Kepner, R. A., and Barger, E. L *Principles of Farm Machinery* (John Wiley & Sons, Inc. N.Y, 1978)
[2] Stone, A. A. and Gulvin, H. E. *Machines for Power Farming* (John Wiley & Sons, Inc. N. Y, 1977)
[3] Culpin, C. *Farm Machinery(* Crosby Lockwood & Sons, Ltd. London, 1963)
[4] Peterson, F. *Handbook of Lawn Mower Repair(* Enslow Publishers Incorporated, 1982)
[5] Kurmi, R. S. and Gupta, J. K. *Machine Design* (Eurasia Publishing House. New Delhi, 2005)
[6] Hall, A. S., Holowenko, A. and Laughlin, H. G. *Machine Design (* McGraw-Hill Publishers. USA, 1968)