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DETERMINATION OF SHEAR FORCES, REACTIONS AND BENDING MOMENTS ON THE DRIVING SHAFT OF A PADDLE-THROWN FORAGE CHOPPER

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ABSTRACT: *As forage chopper operates, there is a combined bending and torsional stresses acting on the driving shaft of the cutting and paddling mechanisms. Therefore, the objectives of this work were to determine the shear forces, reactions and bending moments on the driving shaft of a multi- crop paddle-thrown forage chopper. The diameter of the shaft was designed using the mathematical equations as given by Khurmi and Gupta and selected from the preferred and the not preferred diameters as prescribed by the American Society of Mechanical Engineering, New York, NY, USA. The shear forces and bending moment diagrams were drawn using Microsoft simple shapes diagram. The materials used were from mild steel. A factor of safety of 1.5 was used. More so, the shaft was carrying cutters and paddles to chop and paddle out forages from a forage chopper. The result of the design gave a shaft in series of diameters 35, 75 and 35mm respectively. The shear forces are S.FA = 26.63N, S.FC = -4.57N, S.FD = -25.74N, and S.FB = -25.74N at points A, C, D, and B respectively when the shaft is carrying the cutting and the paddling mechanisms with bending moments of MA = 0, MC = 4.26Nm, MD = 25.74× 0.13 = 3.34Nm, MB = 0. Where; MA, MC, MD and MB are bending moments at points A, C, D and B respectively. In addition, the shear forces and bending moment when the shaft is under its self-load are RA = 170N, RB= 170 and BMA = BMB = 0, BMX = 29.32Nm, at point A, B and X respectively. The shaft was able to withstand the strength and rigidity required for the cutting of forages and paddling of such out of the machine's drum.*

KEYWORDS: Shear Forces, Bending Moments, Driving Shaft, Forage chopper

INTRODUCTION

A shaft is a rotating machine element, which is use to transmit power from one place to another. The power is delivered to the shaft by some tangential force and the resultant torque set up within the shaft permits the power to be transferred to various machines linked up to the shaft Khurmi and Gupta (2005). Pulleys and gears are used to transfer power from one shaft to another. The shafts are usually cylindrical, but may be square or cross-shaped in section. They are solid in cross-section but sometimes, hollow shafts are also use Khurmi and Gupta (2005). Shaft could be divided into transmission shafts that transmit power between the source and the machines absorbing power. The counter shafts, line shafts, over-head shafts and all agricultural machines shafts. Since these shafts carry machine parts such as pulleys, gears etc., therefore they are subjected to bending in addition to twisting. While, machine shafts are those shafts which form an integral part of the machine itself. The crankshaft is an example of machine shaft.

Gbaboet. al., (2013) used a shaft of mass 7.28kg and length of 0.8m to transmits power from an electric motor of 3hp to dehull and separate locust beans. On the average, they reported a dehulling and separating efficiencies of 63.33% and 55.04% respectively. Ojha and Michael

(2009) reported that R4R449 machine used for chopping forage can be classified as Cylinder cutting head and Flywheel cutting head machine. The cylinder type of head was mostly used on power chaff cutters. However, these are connected to a driving shaft. It is therefore, necessary to determine the shear forces, reactions and bending moments on the driving shaft of a multi- crop paddle-thrown forage chopper.

MATERIALS AND METHODS

Material Selection

For the design of the driving shaft of the machine the followings were considered;

a. Availability of materials

b. Durability of Material

c. Cost of materials

d. Maintenance cost and

e. Ease of construction in order to achieve the desire objectives

Shaft Design

Design of shafts of tractable materials based on strength is controlled by maximum shear theory.

Shafting is usually subjected to torsion, bending and axial load. Moreover, the diameter of a hollow shaft can be determined by applying the formula as given by Khurmi and Gupta (2005) :

$$T = \frac{\pi}{16} \times \tau \, (D_0)^3 \, (1-K^4) \quad \text{... (1)}$$

Where;

T = torsional moment or torque = 353.85Nm, $\pi = \frac{22}{7}$, D_O = external diameter, m

$$K = \frac{D_1}{D_2} = 1/2 \ \ldots \ (2)$$

Where;

D_1 = internal diameter, m. K was assumed to be 1:2 for easy selection of bearings. τ = shear stress = Ss (allowable) = 40MN/m^2 was considered for this design for shaft with key way. Therefore, from equation (1) the external diameter of the shaft becomes:

$$D_0^3 = \frac{16T}{\pi\tau(1-K^4)} \ldots \ (3)$$

$$D_0^3 = \frac{16\times353.85}{\pi\times44\times10^6(1-0.5)} = 8.19\times10^{-5} = \sqrt[3]{(8.19 \times 10^{-5}} = 0.043\text{m} = 43\text{mm}$$

For actual design of shaft, factor of safety must be considered and 1.5 was assumed considering that, Factor of Safety (F.S) = $\frac{allowable\ stress}{working\ stress}$ (Khurmi and Gupta, 2005)

Therefore; D_0 = 43×1.5 = 64.5mm. However, according to ASME (1995) code in table B1 in the appendix, the above diameter falls under un-preferred range and so a shaft of 75mm was considered. Since; D_0 = 75mm. Therefore, equation (2) becomes $D_1 = \frac{75}{K} = \frac{75}{2} =$ 37.5mm (internal diameter). Nevertheless, a diameter of 35mm solid shaft was choosing for proper selection of bearings.

The Bending Stress

According to Khurim and Gupta (2005) bending stress can be determined as:

$\sigma = \frac{M}{Z}$.... (4)

Where;

σ = Bending stress, N/m^2, M_x= Bending moment, Nm (29.32Nm)

$Z_{xx} = z_{yy} \frac{\pi}{32}\left(\frac{do^4 - d1^4}{do}\right)$, ... (5)

Where;

d_o and d_1 = external and internal diameter of shaft, m (d_o=0.075m, d_1= 0.035mm)

Therefore; $Z_{xx} = \frac{\pi}{32}\left(\frac{0.075^4 - 0.035^4}{0.075}\right) = 3.94 \times 10^{-05} m^3$. Substitute M_x and Z_{xx} into equation (4) we have; $\sigma = 29.32/3.94 \times 10^{-05} = 743063.24 N/m^2$

Determination of Torsional Shear Stress

Torsional shear stress is zero at the centroid axis and maximum at the outer surface.According to Khurmi and Gupta (2009), torsional stress is determined using

$\tau = Tr/J$. ... (6)

But; $J = \frac{\pi}{32}[(D_0)^4 - (D_1)^4]$, r = D_0/2 = 0.0375m. Hence;

$\tau = \frac{32Tr}{\pi\,[(D_0)^4 - (D_1)^4]}$ ... (7)

Where;

τ = Torsional stress, N/m^2, T = torsional moment, Nm, r = radius of shaft, m, J = Polar moment of area, m^4, D_0 and D_1 = external and internal diameter of shaft, m

Therefore, using equation (7),

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$$\tau = \frac{32\times353.85\times0.0375}{\pi[(0.075)^4-(0.035)^4]} = 424.62/0.0000947\ N/m^2 = 44.8MN/m^2.$$

Determination of Torsional Deflection of Shaft

The torsional deflection was determined by using the torsion equation given by Khurmi and Gupta (2005) for shafts connected in series as:

$$\theta = \frac{T}{C}\left[\frac{L1}{J1} + \frac{L2}{J2} + \frac{L3}{J3}\right] \quad \text{... (8)}$$

Where;

θ = angle twist in degrees, T = torque, Nmm, C = Modulus of rigidity for the shaft material, N/mm^2 ($G = 84KN/mm^2$), L = Length of the shaft, mm (L1 =140mm, L2 = 39mm, L =160mm), J = polar moment of inertia of the cross sectional area about the axis of rotation, But;

$$J_1 = \frac{(\pi\times d^4)}{32} = 3.142\times35^4/32 = 1.47\times10^5 mm^4$$

$$J_2 = \frac{\pi[(D_0^4)-(D_1{}^4)]}{32} = 3.142\times(75^4-35^4)/32 = 2.95\times10^6 mm^4$$, Since; $J_3 = J_1$. Therefore;

$$\theta = \frac{(353.85\times10^3)}{(84\times10^3)}\left[\frac{140}{(1.47\times10^5)} + \frac{390}{(2.95\times10^6)} + \frac{160}{(1.47\times10^5)}\right] = 0.0088^0$$

The twist angle (0.0088^0) is within the permissible amount of twist, predicted by Khurmi and Gupta (2005) which should not exceed 0.25^0 per meter length of shaft.

RESULTS AND DISCUSSION

Reaction of Forces due to Self-Load of Shaft

The reaction of the forces RA and RB in figure 1 due to self-weight of the shaft assuming the diameter of the shaft is uniform and simply supported at A and B may be calculated as given by S.R Khurim (2005):

For uniformly distributed load;

$$R_A = R_B = \frac{wl}{2} = 0.5wl \quad \text{… (9)}$$

Where;

W= weight of shaft per unit length, N/m (W= 492.8N/m) this was calculated from the density, volume and length of the shaft and gravity (g).

L = length of shaft, m (L= 0.69m)

Therefore;

$$R_A = R_B = \frac{492.8 \times 0.6}{2}$$

$R_A = R_B = 170N$,
That is, $R_A = 170N$, $R_B = 170$
For the Bending Moment;

$B_{MA} = B_{MB} = 0$

Where;

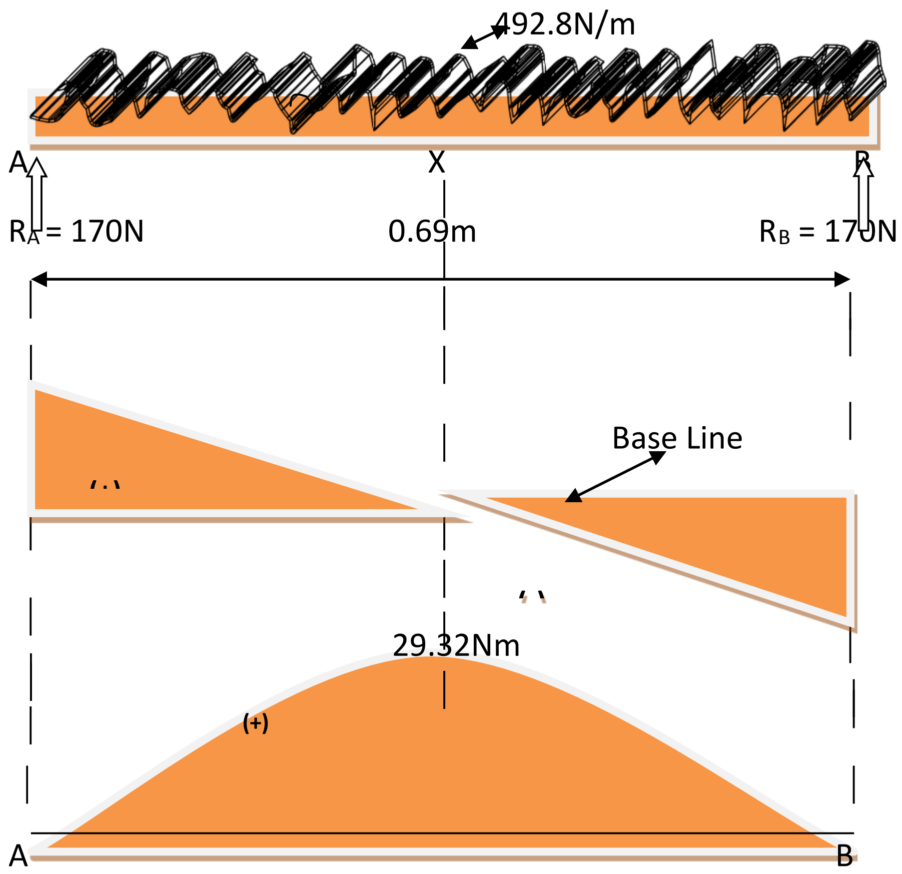
B_{MA} = Bending moment of the shaft about the support reaction A
B_{MB} = Bending moment of the shaft about the support reaction B
Therefore, the B_{MX} about point X in figure 1as given by S.R Khurim (2007) was:

$$B_{MX} = \frac{wl^2}{8}, \quad \dots (10)$$

$$B_{MX} = \frac{492.8 \times 0.69}{8}$$

$B_{MX} = 29.32Nm$

Fig. 1: Reactions, Shear Forces and Bending Moment Diagram



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Reaction of Forces due to External Loads on the Shaft

The reaction forces R_A and R_B in figure 2 due to external loads on the shaft as assumed to be concentrated loads from the cutters (12N), paddles (1.97N) and the disc plates (19.2N) carrying both the cutters and the paddles which were simply supported by bearings at point A and B as shown in figure 2. The force of reactions at R_A and R_B in Fig. 2 can be determined by taking moment about A and equating the same as:

$R_B \times 0.49 = (21.17 \times 0.36) + (31.20 \times 0.16)$

$$R_B = \frac{(7.6212+4.992)}{0.49}$$

$R_B = 25.74N$

Since, upward for must be equal to downward force,

Therefore;

$R_A + R_B = 31.20 + 21.17$ Fig.1

Therefore;

$R_A + R_B = 52.37N$

But, $R_B = 25.74N$

Therefore;

$R_A = 52.37 – 25.74$

$R_A = 26.63N$

Determination of Shear Force Diagram;

The shear force diagram was shown in fig. 2 and their values were:

$S.F_A = R_A = 26.63N$

$S.F_C = 26.63-31.20 = -4.57N$

$S.F_D = -4.57-21.17= -25.74N$

$S.F_B = -25.74N$

Where;

$S.F_A$, $S.F_C$, $S.F_D$ and $S.F_B$ are shear forces at points A, C, D and B respectively.

Determination of Bending Moment Diagram Due to External Loads on Shaft

Fig. 2 shows the bending moment diagram and her values were:

$M_A = 0$

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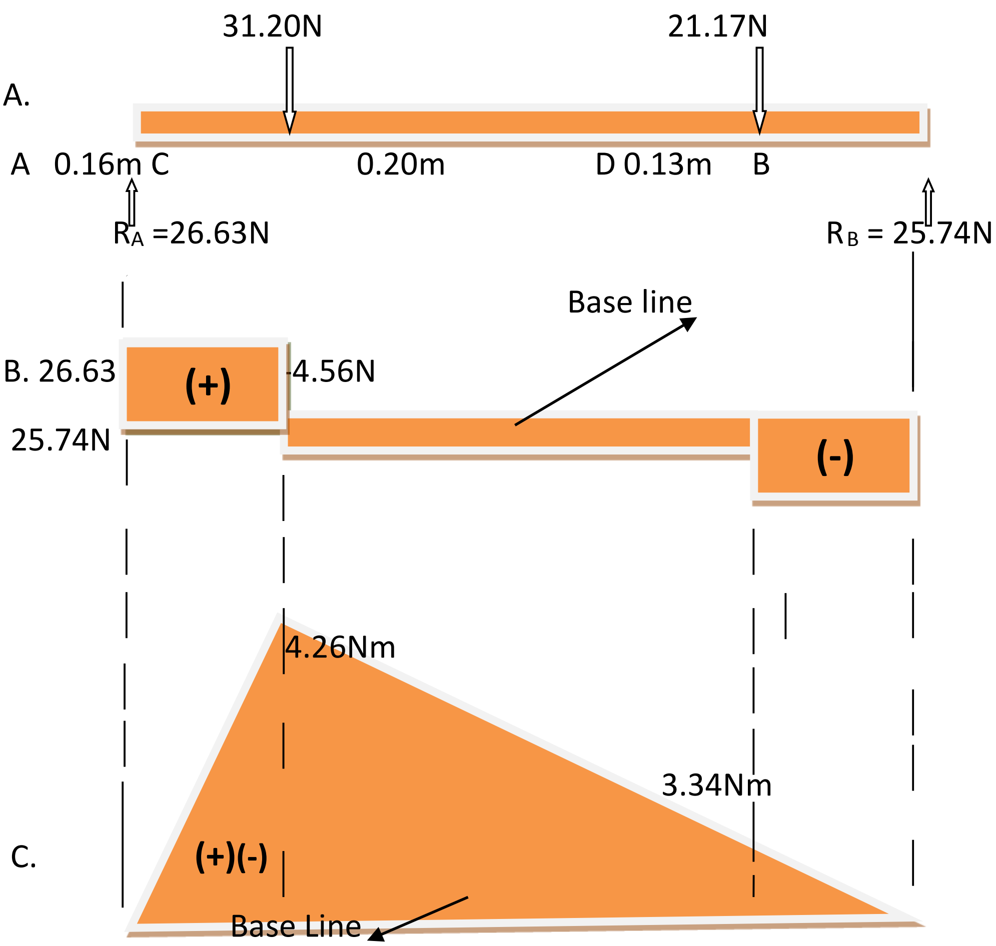
M_C = 26.63N ×0.16 = 4.26Nm

M_D = 25.74× 0.13 = 3.34Nm

M_B = 0

Where;

M_A, M_C, M_D and M_B are bending moments at points A, C, D and B respectively.



CONCLUSIONS

The shaft was able to withstand the strength and the rigidity required for cutting forages and paddling out the chopped forages from the machine's drum.

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