Design of Threshing Cylinder for a Combine Paddy Harvester (KUBOTA)

WINT WINT THET[1], SOE KYAW[2], MYINT MYINT SEIN[3]
[1, 2, 3] *Department of Mechanical Engineering, Pyay Technology University*

***Abstract*-This paper intends to describe the threshing cylinder design for combining paddy harvester (25hp). This paper discusses about the threshing cylinder of combining paddy harvester with axial flow type produced in the Industrial Zone at Pyay. The threshing cylinder is attracted with threshing discs, bars and teeth. The threshing cylinder is constructed with mild steel and threshing teeth are made with medium carbon steel. The speed of the threshing shaft is 540 rpm at power supplied 8 kW. The weight of threshing cylinder, power, pulley design, belt design, shaft diameter, threshing torque, torsional moment, bearing design, key and critical speed are calculated in design consideration.**

***Indexed Terms*-Axial flow, Critical speed, Power, Shaft diameter**

I. INTRODUCTION

Rice is one of important crop in the world, including Myanmar.Myanmar is an agricultural country, and agriculture sector is the backbone of its economy.Agriculture sector contributes 34% of GDP, 15.4% of total export earnings, and employs 61.2% of the labor force. Myanmar is considered an agrarian economy with a contribution of 34% of area and 12 million hectares are agricultural land. Paddy is one of the mainly cereal crop. Production of paddy was 32.8 million tons. Most farmers can not adopt advanced reaper, thresher with power, combine harvester, transport vehicles or trucks. For rice production process, threshing is an integral part of postharvest activities. In many developing countries, threshing is carried out manually by farmers that lead to low quality of paddy rice and grain loss. When the rice production increases, consequently the manual threshing becomes arduous. In general, the agricultural mechanization level in Myanmar is not high.

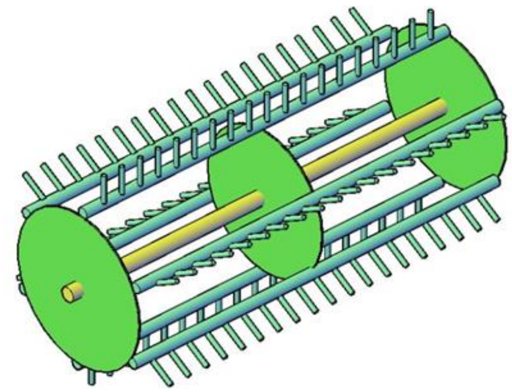

Fig. 1. Threshing cylinder

In order to mechanize this process, two main types of stationary threshing machines have been developed. They are "through-flow" type combine harvester and "axial-flow" type combine harvester. The mechanical threshing operation becomes a customary practice now. The effect of cylinder speed on threshing performance is highly significant all machine settings. Power consumption and broken grains increase and unthreshed grains decrease with the increase in cylinder speed. Though the unthreshed grainlosses decrease, but the total grain losses increase with the increase in cylinder speed [9].

II. POWER DELIVERED BY SHAFT

A. Power delivers by a shaft to Threshing Cylinder

Power is given by the shaft to the threshing cylinder is [9],

$$\text{Power} = F \times \omega \times r \tag{1}$$

Where,

F- weight of threshing cylinder (N)
ω - angular velocity (rad/sec)
r- threshing radius (m)

B. Weight of Threshing Cylinder

Weight of threshing cylinder is,

$$F = W_{discs} + W_{bars} + W_{teeth} \tag{2}$$

Where,

W_{discs} -weight of threshing discs (N)

W_{bars} -weight of threshing bars (N)

W_{teeth} -weight of threshing teeth (N)

C. Calculation the Weights of bars, discs and teeth

Weight of threshing bar is,

$$W_{bars} = m \times g \quad (3)$$

Where,

m- mass of threshing bar (kg)

g- acceleration due to gravity (m/s²)

Weight of threshing discs is,

$$W_{discs} = (W_1 \times n)+(W_2 \times n) \quad (4)$$

Where,

W_1- weight of outside discs (N)

W_2 - weight of inside discs (N)

n - number of discs

Weight of threshing teeth is,

$$W_{teeth} = m \times g \quad (5)$$

Where,

m- mass of threshing teeth (kg)

g- acceleration due to gravity (m/s²)

D. Determination of Threshing Torque

The torque is given by [9],

$$T = F \text{ x } r \quad (6)$$

Where,

F- weight of threshing cylinder (N)

r - threshing radius (m)

E. Force Required to Thresh Paddy

The threshing bars, which are attached to the shaft contained in the threshing cylinder, rotate with the shaft, giving rise to centrifugal force.

Centrifugal Force at r_{max},

$$F= m \times \omega^2 \times r_{max} \quad (7)$$

Centrifugal Force at r_{min},

$$F = m \times \omega^2 \times r_{min} \quad (8)$$

$$\omega = \frac{2\pi N}{60} \quad (9)$$

Where,

F-centrifugal force (N)

m- mass of threshing bars (kg)

ω-angular velocity (rad/sec)

r- radius of the arm of the threshing bar (m)

N- speed of the shaft (rpm)

III. BELT DESIGN

A. Power transmitted by belt

A belt provides a convenient means of transferring power from one shaft to another. Belts are frequently necessary to reduce the higher rotational speeds of electric motors to lower values required by mechanical equipments [6]. The procedure for selecting a V-belt drive is dependent on the motor horse power and the speed (rpm) rating. V-belts are rated from class A to E. It is always based on weaker pulley and the procedure is as follows.

The velocity of belt is [6],

$$V = \frac{\pi \times D \times N}{60} \quad (10)$$

Where,

D - diameter of weaker pulley (m)

N - rpm of weaker pulley (rpm)

Area of V-belt is,

$$A = \frac{1}{2} \times (a+b) \times h \quad (11)$$

Length of belt,

$$L = 2C + \frac{\pi}{2}(D_1 + D_2) + \frac{(D_1 - D_2)}{4C} \quad (12)$$

Mass of pelt per length,

$$m = \rho \times A \quad (13)$$

Where,

m - mass of belt per length (kg)

ρ -density of rubber belt (kg/m³)

Centrifugal tension (T_c),

$$T_c = m \times V^2 \quad (14)$$

Maximum tension in the belt (T_{max}),

$$T_{max} = \sigma \times A \quad (15)$$

Where,

σ - Stress of rubber belt (N/m²)

Belt tension ratio is [6],

$$\frac{T_1 - T_c}{T_2 - T_c} = \exp^{\mu\alpha / \sin\frac{\theta}{2}} \quad (16)$$

Where,
T_1 -tension in the tight side (N)
T_2 -tension in the slack side (N)
α -contact angle (degree)
μ -coefficient of friction
θ -groove angle (degree)

$$\sin\beta = \frac{D_2 - D_1}{2C} \quad (17)$$

$$\alpha = 180 - 2\beta \quad (18)$$

Where,
C - distance between pulleys (m)
D_1 - radius of smaller pulley (m)
D_2 - radius of larger pulley (m)
β - angle of wrap (degree)

The power transmitted by belt,

$$P = (T_1 - T_2)\, V \quad (19)$$

Design Power is given by,

$$P_{design} = P \times SF \quad (20)$$

Where,
SF-service factor

B. Number of belts

The number of belts required for a given power transmission is [6],

$$n = \frac{P_{design}}{P_{rate} \times K_\theta \times K_L} \quad (21)$$

Where,
P_{rate} - Rated power (kW)
K_θ - Arc of contact factor
K_L - Length correction factor

C. Pulley Groove Dimensions

The groove dimension for pulleys has been from standardized size depending on the type of class of belt intended for use. Classes of belts, pitch diameters of the pulley, and the groove has been selected from B type [1].

IV. SHAFT DESIGN FOR THRESHING CYLINDER

Design of the shaft, based on strength is controlled by the maximum bending and shear theory. Shafting is usually subjected to bending and torsion. Shaft diameter is mainly considered from the weights and forces acting on the shaft [4].

In this shaft design, shaft diameter is calculated from the ASME code equation. Distributed load on shaft from threshing cylinder and concentrated load from pulley weights are considered as the forces acting on the shaft. The following equations are based on shaft design calculation for bending moment, torsion and shear stress.

A. Determination of pulleys weight

The weight of pulley can be determined as follows.

Weight of pulley,

$$V_1 = \frac{\pi}{4} \times D_1^2 \times (e + 2f) \quad (22)$$

$$V_2 = \frac{\pi}{4} \times D_2^2 \times (2e + 2f) \quad (23)$$

$$V_3 = \frac{\pi}{4} \times D_3^2 \times (e + 2f) \quad (24)$$

$$m = \rho V \quad (25)$$

$$W_{pulley} = m \times g \quad (26)$$

Where,
D - diameter of the pulley (m)
m - mass of pulley (kg)
V - volume of pulley (m^3)
W_{pulley} - weight of pulley (N)

B. Calculation of Torsioinal Moment

Torsioinal moment acting upon the camshaft is [8],

$$M_t = \frac{9550 \times kW}{rpm} \quad (27)$$

$$M_t = (T_1 - T_2) \times r \quad (28)$$

Where,
kW- power delivered (kW)
N- revolution per minute (rpm)

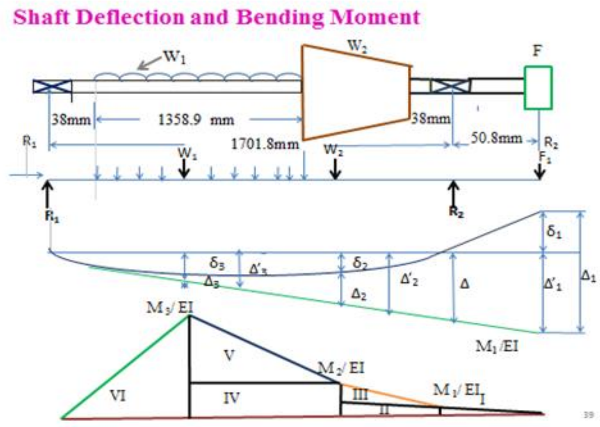

Fig. 2. Bending Moment Diagram

C. Determination of Minimum Diameter of Shaft

By using ASME code equation [5],

$$d^3 = \frac{16}{\pi S_s} \times \sqrt{(K_b M_b)^2 + (K_t M_t)^2} \quad (29)$$

Where,

d - diameter of the shaft
M_t - torsional moment (N-m)
M_b - bending moment (N-m)
K_b - combined shock and fatigue factor
K_t - combined shock and fatigue factor
S_s - allowable shear stress (MN/m²)

For shaft without key-way,

$S_s = 55MN/m^2$.

For shaft with key-way,

$S_s = 40MN/m^2$.

A shaft with key-way was used for this work. So M_t, M_b, S_s and d can be calculated and checked for design satisfactory.

D. Bending Stress

Bending load, bending stress (tension or compression) is [10]:

$$\sigma_b = \frac{M_b \times r}{I} \quad (30)$$

For circular shaft [2],

$$I = \frac{\pi d^4}{64} \quad (31)$$

Where,

σ_b - bending stress (N/m²)
I - moment of inertia (m^4)

E. Shear Stress

Shear stress is determined using [10],

$$\tau = \frac{T \times r}{J} \quad (32)$$

Where,

T- torque (N-m)
J - polar moment of inertia (m^4)

F. Angle of twist

The amount of twist permissible depends upon the particular application and is given by,

$$\theta = \frac{584 \times M_T \times L}{G \times d^4} \quad (33)$$

Where,

θ - angle of twist (degree)
L - length of shaft (m)
G - torsional modulus of rigidity (N/m^2)

V. BEARING SELECTION

Bearing must be selected based on its load carrying capacity, life expectancy and reliability. The relationship between the basic rating life, the basic dynamic rating and the bearing load are [6]:

$$L = \left(\frac{C}{P}\right)^p \quad (34)$$

Where,

L -nominal life in million of revolution
C -basic dynamic capacity (N)
P -equivalent bearing load (N)
$\left(\frac{C}{P}\right)$- loading ratio

p= 3 for ball bearing

$$L = \frac{60NL_n}{1{,}000{,}000} \quad (35)$$

L_n-nominal life in working (hr)

$$P = (X F_r + Y F_a) \quad (36)$$

Where,

X- radial load factor for the bearing
Y- axial load factor for the bearing
F_r- actual radial bearing load (N)
F_a- actual axial bearing load (N)

VI. KEY DESIGN

A. Key Dimensions

The function of a key is to prevent relative rotation of a shaft and the member to which it is connected, e.g., hub of a gear, pulley or crank [6].

For a good result, the width of a key is:

$$b = \frac{d}{2} \tag{37}$$

The thickness of a key is:

$$t = \frac{d}{6} \tag{38}$$

The length of the key can be calculated as:

$$L = \frac{\pi d}{2} \tag{39}$$

B. Shear Stress in Key

$$\text{Shear area} = b \times L \tag{40}$$

$$F_s = s_s \times b \times L \tag{41}$$

$$T_s = F_s \times r \tag{42}$$

Where,

T_s - torque due to shear (N-m)

F_s - force to resist shear (N)

C. Compressive Stress in Key

$$\text{Compression area} = \frac{t \times L}{2} \tag{43}$$

$$F_c = \frac{s_c \times t \times L}{2} \tag{44}$$

$$T_c = F_c \times r \tag{45}$$

Where,

F_c - force to resist compression (N)

s_c - compression stress (N/m²)

T_c - torque due to compression (N-m)

VII. CALCULATION OF CRITICAL SPEED

All rotating shaft even in the absent of external load, deflect during rotation. It magnitude of the deflection depends upon the stiffness of the shaft and its supports, the total mass of shaft and attached parts.

The deflection is considered as a function of speed, show maximum values at so called critical speed. And it is calculated by using moment area method and Rayleigh-Ritz Equation.

$$\Delta = A_1\bar{x}_1 + A_2\bar{x}_2 + A_3\bar{x}_3 + \ldots\ldots \tag{46}$$

Where,

A_1 - area of portion I of the M/EI diagram

$\bar{x}_1$ - distance from the ordinate at point A to the centre of gravity of A_1

A_2 - area of portion I of the M/EI diagram

$\bar{x}_1$ - distance from the ordinate at point A to the centre of gravity of A_2

The critical speed of the shaft given by Rayleight-Ritz equation is [5],

$$\omega_c = \sqrt{\frac{g\sum W\delta}{\sum W\delta^2}} \tag{47}$$

Where,

ω_c - critical speed (rad/s)

δ - deflection (m)

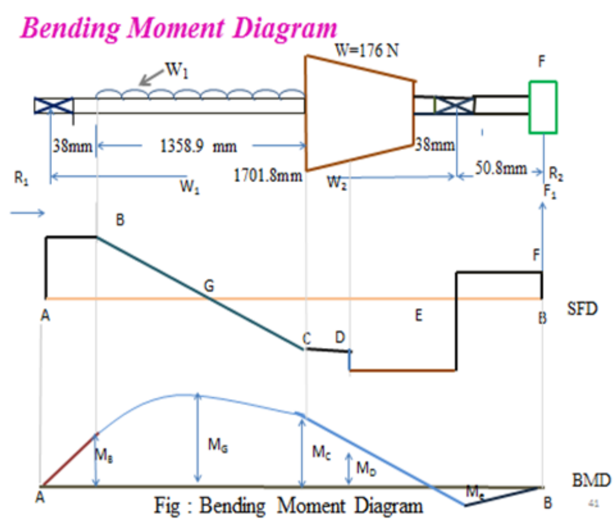

Fig.3. Shaft deflection and Bending Moment

VIII. RESULTS OF THRESHING CYLINDER DESIGN

Input data for design calculations is obtained from the Pyay Industrial Zone. They are;

TABLE I SPECIFICATIONS FOR DESIGN CALCULATION

Parameters	Symbol	Unit	Value
Diameter of bar	D_b	mm	31.75

Length of bar	L	mm	1339.
Number of bars	n	-	6
Outer diameter of outside disc	D_o	mm	472.4
Inner diameter of outside disc	D_i	mm	444.5
Length of cylinder	D_o	mm	1358
Diameter of teeth	D_i	mm	12
Thickness of discs	t	mm	4
Number of discs	n	-	3
Diameter of teeth	D_t	mm	14
Length of teeth	L	mm	86
Number of teeth	n	-	108
Speed of threshing cylinder	N	rpm	540
Service factor	SF	-	1

TABLE II RESULT TABLE FOR THRESHING CYLINDER

Parameters	Symbol	Unit	Value
Weight of threshing discs	W_{discs}	N	19
Weight of threshing teeth	W_{teeth}	N	80
Weight of threshing bars	W_{bars}	N	489.14
Weight of threshing shaft	F	N	39.744
Weight of threshing cylinder	F	N	622.22
Threshing Power	P	kW	8
Threshing Torque	T	N-m	141.67
Angular velocity	ω	rps	56.55

TABLE III RESULT TABLE FOR V-BELT DESIGN

PARAMETERS	UNIT	FOR THRESHING PULLEY	FOR BLOWER PULLEY	FOR AUGER PULLEY
POWER INPUT	KW	8	3.62	2.314
TENSION IN TIGHT SIDE	N	1137.795	334.645	274.085
TENSION IN SLACK SIDE	N	308.779	32.256	2.30
VELOCITY OF WEAKER PULLEY	M/S	20	12	8.944
LENGTH OF BELT	M	1.01	2.23	1.01
MASS OF PER LENGTH	KG	0.224	0.224	0.87
DANSITY OF RUBBER BELT	KG/M^3	1140	1140	1140
POWER RATING	N	32.236	3.62	2.319
NUMBER OF BELT	-	3	2	2

TABLE IV RESULT TABLE FOR SHAFT DESIGN

Parameters	Symbol	Unit	Value
Weight of auger pulley	F_1	N	8.78
Weight of threshing pulley	F_2	N	19
Weight of sieve pulley	F_3	N	12.65
Torsional Moment	M_t	N-m	63.78
Diameter of the shaft	d	mm	43
Bending Stress	σ_b	MN/m^2	47.01
Shear Stress	τ	MN/m^2	5.92
Angle of Twist	θ	degree	0.118

TABLE V RESULT TABLE FOR BEARING DESIGN

Parameters	Symbol	Unit	Value
Service life	L_n	hr	8000
Loading ratio	C/P	-	7.96
Equivalent bearing load	P	N	2657.85
Basic dynamic loading	C	N	21156.5
Bearing number	SKF-6280		
Inner diameter	d	mm	40
Outside diameter	D	mm	80
Width	B	mm	18

TABLE VI RESULT TABLE KEY DESIGN AND CRITICAL SPEED

Parameters	Symbol	Unit	Value
Length of key	L	mm	59.6
Width of key	w	m	9.5
Thickness of key	t	m	6.3
Shear force	F_s	kN	22.648
Compression force	F_c	kN	20.76
Torque due to shear	T_s	N-m	430.312
Torque due to compression	T_c	N-m	394.45
Critical speed	ω_c	rad/s	167.34

IX. RESULT AND DISCUSSION

In the threshing cylinder design, total cylinder weight, threshing power, threshing speed, critical speed, deflection, threshing torque and moment are calculated. In the design calculations of power transmission shafts, calculations are made by assuming that the shafts are in the state of torsion and bending only. In V-belt drive calculation, the appropriate belt type to be used and the number of belts is calculated. In bearing design and selection, the appropriate bearing numbers are calculated for each bearing. After checking the results, existing theories and calculated results are nearly the same. But there are some weak points in the design calculations of the machine. Because calculations are made by considering only load conditions on the driving shaft and other forces are not considered such as dynamic, vibration effect and so on. And also feed rate and moisture content of paddy are not considered in this research.

X. CONCLUSION

This paper is designed of threshing unit for combining paddy harvester with 347mm diameter and 840mm length cylinder from the Pyay Industrial Zone and got the specification data from it. Design calculations of these machines are necessary to decide whether they are fitted with standards or not. In this paper, existing theories and calculated results are nearly the same. So, design is satisfied.

REFERENCES

[1] Abdulkadir Baba Hassan, Matthew Sunday Abolarin.“The Design and Construction of Maize Threshing Machine,”Niger State Nigeria. 2009.

[2] PSG Tech. “Design data. Faculty of Mechanical Engineering,” Poolamedu Sathankulam Govindsamy Naidu College of Technology (PSG Tech), DPV Printers, Coimbatore, India.1989.

[3] Oni, K.C.; and Ali, M.A. “Factors influencing the threshability of maize in Nigeria.”Agricultural Mechanization in Asia, Africa and Latin America (AMA) 17(4): 39-44. 1986.

[4] Kaul,R.N.;and Egbo,C.O “Introduction to agricultural Mechanization Macmillan,” London, England, UK. pp. 128-41. 1985.

[5] Robert L. Mott, P.E, 1985. “Machine Elements in Mechanical Design, Macmillan Publishing Company,” New York.

[6] Black, P.H.; and Adams, O.E., Jr. “Machine design. 3rd ed. McGraw-Hill,”New York, NY, USA. 1968.

[7] Robert L. Mott, P.E,. “Machine Elements in Mechanical Design, Macmillan Publishing Company,”New York. 1985.

[8] International Rice Research Institute (IRRI), “Standard Paper of Throw-in Type Thresher,” Drawing and Test .1981.

[9] Lambert Henry Wilkes, M.S.,“Farm Machinery and Equipment.”McGraw-Hill Book Company, New York. 1961.

[10] Williams, W.A. “Mechanical power transmission manual.”Conover-Mast Publ., New York, NY, USA.1953.